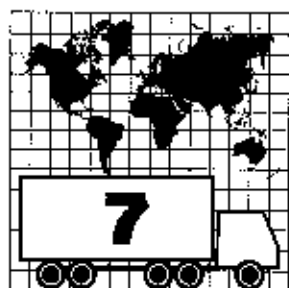


7th International Symposium
Heavy Vehicle Weights & Dimensions
proceedings



Challenges in the 21st century

PROCEEDINGS

'7th International Symposium on Heavy Vehicle Weights and Dimensions'

Challenges in the 21st century

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Auditorium Technical University, Delft, The Netherlands

The 7th International Symposium on Heavy Vehicle Weights and Dimensions is an intercontinental forum for researchers, policy makers and industry leaders in the field of freight transportation by road. The specific goal of this symposium is to bring together the worlds of vehicle technology, vehicle-infrastructure interaction, safety, regulations and policy and to listen and to discuss the challenge of economic, safe and environmental friendly transport in the 21st century.

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Preface I

Road freight transport by heavy vehicles.

In Europe 80% of the total freight transport is by road, 16% is by rail and 5% is over water. In The Netherlands these figures are considerably different: 55% is road freight, 41% is over water and 4% is by rail. Of course this is to be expected in a country with so many waterways. The need for transport of goods keeps growing as a consequence of the growing population, increased luxury, etc. The EU expects that this growth will be 29% from 2000 until 2010 and up to 42% in the year 2015.

The numbers of heavy vehicles and road freight transport movements are not the major part of the total traffic but their impact is considerably high. This applies both to the positive and to the negative aspects. Positive is the contribution to the economic development. This kind of transport is fast, efficient and almost every place is reachable. The dark side includes the effects on the environment (emissions and noise), safety and road damage through e.g. overloading. The accident risk of heavy vehicles is more or less equal to the accident risk of passenger cars and light trucks, but the consequences of the accidents involving heavy vehicles are considerably larger. The fatality risk of a heavy vehicle accident is three times higher than that of an accident involving passenger cars only. Also the damage to the infrastructure is an important issue. It can be observed in recent years that the attention from the government increases for the negative aspects of road freight transport.

The introduction of longer and heavier vehicles is currently under consideration. The acceptance of the use of these vehicles might improve the efficiency of the available roads, and is attractive from both the economic and logistic points of view. On the other hand the impacts on the environment, infrastructure and road safety have also to be considered.

Road freight transport is in fact a combination of three main sub-systems: the driver, the vehicle including its load, and the infrastructure. There is a lot of research going on to understand the driver's behaviour and to improve his/her driving performance. The heavy vehicle industry is very active in research and development on the necessary improvements and also pavement and bridge engineers spend quite some time to find solutions to improve the infrastructure.

Most of the problems associated with safety, economy and other mentioned aspects are affected by the characteristics of both infrastructure and vehicles and by the manner in which these interact. In spite of the complex interaction between infrastructure and vehicles, there has been rather limited interaction and flow of information between the infrastructure engineers and the vehicle engineers. Multi-disciplinary meetings like this '7th International Symposium on Heavy Vehicle Weights and Dimensions', where vehicle industry, transport organisations, politicians, consultants and research organisations meet, are an excellent opportunity to improve the whole road freight transport system. A variety of interesting papers on heavy vehicle performance and infrastructure interaction was produced and published in this book. May these papers be a source of information in meeting 'The challenge of the 21st Century for heavy vehicle transport' and may these papers inspire all parties to work closely together.

Ir. Gerrit. Tanis

Managing director of TNO Automotive

The Netherlands is a low-lying country in the delta of three main European rivers: the Rhine, Meuse and Scheldt rivers. More than half of the country lies below sea-level. Fifteen percent of the surface area consists of water. Half of this water is fresh, inland water: rivers, lakes, pools, canals, brooks and marshes, bearing testimony to the fact that The Netherlands is dominated by water.

This country is also one of the world's most populated nations. The highest population density is located in the so-called 'Randstad' area, which includes the cities of Amsterdam (capital), The Hague, Rotterdam and Utrecht. It is situated in the western part of the country, in typical Dutch polder landscape (land claimed from water and below sea-level) along the coast of the North Sea. On average, there are 454 inhabitants per square kilometre in The Netherlands. This figure rises to some 900 inhabitants in the Randstad.

A dense population demands an extensive infrastructure. The total road network comprises over 125,000 kilometres, of which about 2,300 kilometres constitute the main motorway network operated by Rijkswaterstaat, one of the Directorates-General of the Ministry of Transport, Public Works and Water Management.

Rijkswaterstaat is responsible for development, maintenance, upgrade and reconstruction of the Dutch main road and water network as well as the defence of the country against river and sea floods. The Road and Hydraulic Engineering Institute (DWW) is one of the advisory institutes of the Rijkswaterstaat and covers the dry as well as the wet infrastructure and plays a leading role in development and implementation of new techniques, methods and systems.

To design and maintain the infrastructure, knowledge of the loading of the infrastructure and the corrosive effects on pavements of heavy traffic is required. Here is found the relation to heavy vehicle technology, especially in the vehicle-infrastructure interaction. DWW is involved in this subject, also on international level through national and European and even global projects (DIVINE, COST-323, COST-334, WAVE, TOP-TRIAL).

The series of symposia on Heavy Vehicle Weights and Dimensions that was initiated by the DIVINE project proves an excellent platform for exchanging and sharing knowledge in the field of vehicle-infrastructure interaction, and therefore DWW took the opportunity to organise the 7th International Symposium on Heavy Vehicle Weights and Dimensions. The more so, since our institute celebrates its 75 years anniversary in 2002.

Besides, it emphasises the role of our institute in supporting initiatives that improve the interaction between existing technology fields and underlines the importance of integrated solutions for the traffic and infrastructure problems that we are facing today.

The content of the symposium deals with both technical and policy issues as expressed in the working title:

'The challenges of the 21st century'. We are facing both technical and institutional challenges in the field of transport of goods. Globalisation as well as being part of the European Union underlines the importance of coordination and cooperation between researchers and policymakers on an international level. The proceedings and the opportunity given during the conference to discuss the various subjects are a small but important step forward towards joining together the efforts for solving the problems of the transport of goods by road and reaching for optimal solutions for economy and environment.

Ir. Luuk Bosch

Head of Infrastructure Department of the Road and Hydraulic Engineering Institute
Ministry of Transport, Public Works and Water Management

Preface III

On behalf of the International Forum for Road Transport Technology, we are indebted to TNO Automotive and The Road and Hydraulic Engineering Division of The Dutch Ministry of Transport, Public Works and Water Management for hosting our seventh international symposium. What better location than The Netherlands, with leading edge road transport and intermodal practices already in place?

The Forum has been fortunate to sponsor previous successful symposia in North America, Europe and Asia Pacific, fostering important research into heavy vehicle behaviours affecting infrastructure and safety. While many issues confront the road transport industry in all our countries – with important challenges of reducing traffic accidents, congestion and emissions and maintaining a skilled workforce – it is vital to focus on the role of technology in solving some of these problems.

Perusal of the program for our seventh symposium reveals that the research is now being put into practice. In this important stage of innovation in road transport and its regulation, it is even more critical for us to come together to exchange, monitor and inspire.

It is particularly pleasing to see that landmark international scientific collaborations on infrastructure effects will be presented at this symposium, along with new scientifically-based methods of truck regulation which are becoming dynamic and targeted rather than static and monolithic.

On behalf of the International Forum for Road Transport Technology Board, I welcome all members of the Forum and colleagues from around the world to this unique gathering generously supported by transport agencies and companies of The Netherlands.

Dr. Peter Sweatman
President
International Forum for Road Transport Technology

ACKNOWLEDGEMENTS

This book could not have been published without the input of papers for the '7th International Symposium Heavy Vehicle Weights & Dimensions'. We would like to express our sincere appreciation to the authors and co-authors of the papers.

We are also particularly grateful for the support, review effort and advice of the Technical Committee:

*Dr. Reto Cantieni, Prof. dr. David Cebon, Ir. Peter de Coe, Dr.-ing. Wolf Hahn,
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John Woodroffe.*

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The European Vehicle Passive Safety Network

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IMPACT FACTORS ON MEDIUM SPAN BRIDGES DUE TO MULTIPLE VEHICLE PRESENCE

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ABSTRACT

The Dynamic Amplification Factor for Bridges is of major concern in both their design and assessment. Research to date has focused on the single truck event. However, in many bridges the critical loading case is that of multiple truck presence on the deck. To accurately determine the dynamic amplification factor it is necessary to examine the effects of multiple trucks traversing a bridge. Experiments in Slovenia were carried out to examine the dynamic amplification factor for single and two truck events. Numerical models were constructed and validated from these experiments. These models were then used to compare the dynamic amplification factors produced from both single and multiple trucks crossing the bridge at various speeds. Important conclusions are drawn for bridge design and assessment purposes.

1. INTRODUCTION

Dynamic Amplification Factors (DAF's) have a significant effect on the design and assessment of bridges and there is great potential benefit from rationalising the approach used by bridge engineers. The dynamic amplification factor is defined as the ratio of the combined response of the bridge (both static and dynamic) to the static response. Codes of practice commonly calculate amplification factors as a function of span length, but the interaction between bridge, trucks and road roughness involves many more parameters which are difficult to identify and allow for when attempting to determine the bridge response (DIVINE, 1997). Much of the research to date on this phenomenon has concentrated on the single truck event and has identified situations where 'frequency matching' can occur between the truck and the bridge (DIVINE, 1997). However, for a great many bridge the critical loading event involves more than one truck, in which case the amplification effects are substantially different. Knowledge of the response of a bridge to multiple truck crossing events is essential for an accurate calculation of the relevant amplification factor. This paper investigates the comparison of the DAF produced from a single truck and the two truck event. Experiments were conducted in Slovenia to examine the response of a simply supported bridge for both single and two truck events. The results from this were used to calibrate a finite element model (produced in MSC/NASTRAN). This model was used to examine the amplification factor of the bridge for various truck crossing scenarios.

2.1 SITE SETUP

The tests were carried out on a bridge over the Mura river in north-eastern Slovenia. The bridge is a 32 m long simply supported span and is part of a larger structure. Figure 1 shows a schematic of the bridge.

Fig. 1 – Schematic of instrumental bridge

It is of concrete slab-beam construction. It has a first natural frequency of 2.802 Hz and damping of 3%. The bridge was instrumented with 12 strain transducers. Six transducers were placed on the bridge beams underneath each lane. Each pair of transducers had a longitudinal spacing of 4m. Two axle detectors were placed in each road lane to record truck position and velocity. The vehicles used in the experiment consisted of one two axle and one three axle pre-weighted truck. The two axle truck had masses of 3460 kg and 12900 kg on each axle, while the three axle truck had a front axle mass of 6240 kg and a tandem mass of 18220 kg.

2.2 EXPERIMENTAL PROCEDURE

Experiments were conducted to examine the DAF's produced for both single and two truck events. A total of 12 passes for both the two and three axle trucks crossing the bridge were carried out. In each case the truck was the only vehicle on the bridge. These passes were carried out for the trucks travelling in various directions and at three different speed ranges (slow, average, and fast). To examine the two truck event, data was recorded for 25 passes with the two trucks on the bridge simultaneously. The two trucks were travelling in opposite directions, again at various speeds and meeting at various longitudinal positions along the bridge span. Figure 2 shows the two trucks passing each other on the instrumented span. The effect of one truck being stationary on the bridge as the other truck crossed was examined with 12 passes of the three axle truck as the two axle truck remained stationary at midspan.

Fig. 2 – Two truck event

3.1 BRIDGE MODEL

The bridge was modelled in finite elements as a 32 m long simply supported plate-beam model. The main deck was modelled using plate elements and the beams were modelled using offset beam elements. Lateral stiffeners were included, again using offset beam elements. Figure 3 shows the first two natural frequencies of the bridge. The road surface profile is modelled as a random process as described by a power spectral density function (Yang & Lin 1995). The profile is classified according to the International Standards Organisation (ISO), (Wong 1993). In this case it is taken to have an ISO value of 'good'. The stress at various longitudinal positions along the beam is output.

Fig. 3- First mode shape (2.802 Hz), second mode shape (3.681 Hz)

3.2 TRUCK MODELS

The trucks were modelled as rigid frames in finite elements. They were represented as body mass and axles masses. The axle masses can move in the vertical direction while the body mass can move in the vertical direction and rotate. The body mass in both trucks is modelled as a frame consisting of bar elements. The suspension and tyres are modelled as a spring and dashpot system. The values of stiffness and damping are taken as typical values from the literature (Kirkegaard et al. 1997, Baumgartner 1998, Lutzenberger & Baumgartner 1999, Huhtala 1999). The trucks travel in opposite directions and at a constant velocity. Figure 4 shows a schematic of the three axle truck, the two axle truck being similar.

Fig. 4 - Schematic of the three axle truck represented in MSC/NASTRAN

3.3 Finite element dynamic interaction

The modelling of vehicles and bridges was carried out with the general purpose finite element analysis package MSC/NASTRAN for Windows, which provides the capability for performing a transient dynamic response (MSC/NASTRAN). A C++ program to perform simulations of truck models crossing a bridge has been developed. The program generates NASTRAN input code for any arbitrary one-dimensional or spatial bridge and vehicle model. The dynamic interaction of the bridge and vehicle incorporates a road surface profile and is implemented using a set of auxiliary functions to enforce the compatibility conditions at the bridge/vehicle interface (Cifuentes, 1989). The road roughness can be generated from theoretical power spectral density functions or real measurements. The speed/acceleration of the vehicles, their initial positions and paths on the bridge are also required. The input allows for the specification of simultaneous traffic events with vehicles running in the same or opposite directions.

The program generates an entry into the assembled stiffness matrix of the vehicle-bridge system. This entry allows the interaction forces " F_j " at the contact point of each wheel " j " on the bridge to be defined. A compatibility condition between the vertical displacement " $w_j(t)$ " of the wheel " j " and the bridge at the contact point is also established at any time " t " as formulated in equation 1.

$$w_j(t) = \sum_i A_i(t) y_i(t) + \sum_i B_i(t) \theta_i(t) \quad i=1,2,...,N \quad (1)$$

where " $y_i(t)$ " and " $\theta_i(t)$ " are the displacement and rotation at each node " i ", " N " is the total number of bridge nodes, and " $A_i(t)$ " and " $B_i(t)$ " are auxiliary functions. The auxiliary functions $A_i(t)$ and $B_i(t)$ can adopt different values at each node " i " for each instant " t ". Their shape is shown in figure 5. They have zero value outside of the interval between adjacent nodes.

Figure 5 – Auxiliary function $A_i(t)$, auxiliary function $B_i(t)$

Finally, equation 2 illustrates the force " $f_j(t)$ " and moment " $m_j(t)$ " acting on a bridge node " i " at time " t " due to the interaction force " F_j " at each wheel " j ".

$$\begin{Bmatrix} f_1 \\ m_1 \\ f_2 \\ m_2 \\ \vdots \\ f_N \\ m_N \end{Bmatrix} = \begin{Bmatrix} A_1 \\ B_1 \\ A_2 \\ B_2 \\ \vdots \\ A_N \\ B_N \end{Bmatrix}_1 * F_1 + \dots + \begin{Bmatrix} A_1 \\ B_1 \\ A_2 \\ B_2 \\ \vdots \\ A_N \\ B_N \end{Bmatrix}_r * F_r \quad (2)$$

where " r " is the total number of wheels on the bridge. The auxiliary functions " $A_i(t)$ " and " $B_i(t)$ " are different for each " F_j " due to the fact that each axle takes a different length of time to reach the same node and each wheel of a particular axle follows a different path on the bridge.

3.4 VERIFICATION OF FINITE ELEMENT MODEL

In order for analysis to be undertaken using NASTRAN it was first necessary to verify the bridge and truck models with the data collected in Slovenia. An accurate value of structural damping was determined using data collected by the axle detectors. Typical experimental data from a truck crossing was examined and the velocity of the truck determined. The corresponding simulation was then carried out in NASTRAN and the resultant responses compared. Figure 6 shows a typical match that was obtained for each of the single truck events.

Fig 6 – Comparison of measured and finite element responses

It can be seen from the figure that there is a reasonable match between the measured and finite element responses. It is hoped that through further work with the models, particularly in the area of suspension and tyre parameters these matches can be improved.

After each truck was verified separately a simulation of the two truck event was compared to a measured response. This is shown in figure 7.

Fig. 7 - Comparison of theoretical and measured responses for both trucks traversing the bridge at the same time

It was found that there was a reasonable comparison between the two responses illustrated in figure 7. It is important to note the right hand side of the graph in figure 7, this difference in vibration is most likely due to the actual suspension characteristics differing from those in the model. The authors plan to adjust parameters to achieve a closer match in the future.

4. THEORETICAL STUDY OF DYNAMIC AMPLIFICATION FACTORS

Simulations were carried out for the three axle truck traversing the bridge at various speeds from 10 – 80 km/hr. In each case the stress at the centre of the span is examined. Figure 8 shows both the static and the combined midspan stress response at 50 and 80 km/hr for the three axle truck. For low velocities the response of the bridges closely follows the path of the static response. However, as the velocity of the vehicle increases the level of dynamic interaction also increases as shown in figure 8.

Fig. 8 - Mid-span static response & combined response of bridge being traversed by three axle truck travelling at speeds of 50 & 80 km/hr

Similar simulations were carried out for the two trucks crossing the bridge simultaneously. The trucks were travelling in opposite directions and at the same speed. The two trucks met at mid-span, i.e. the front axle of both the two axle and the three axle trucks were at mid-span. Figure 9 shows the static bridge response and the combined bridge response at various velocities for the two truck event.

Fig. 9 - Static & combined response of the bridge being traversed by both trucks at speeds of 50 & 80 km/hr

In each of the simulations the dynamic amplification factor was calculated and the factor for each of the single truck events were compared to that of the corresponding two truck event for various speeds, see figure 10. The values of amplification factor for the two truck event were determined using the assumption that the static response of the bridge for the two truck event was the superposition of the static responses of both the single truck events. This assumption is currently being investigated.

Fig. 10 - Dynamic Amplification Factors for single & two truck events for various velocities

As can be seen from the figure, the dynamic amplification factor with reference to velocity differs for each truck. The peaks in the graph for the single truck events occur at different velocities. For the two axle truck the maximum impact factor occurs at 70 km/hr with a value of 1.11. The maximum amplification factor for the three axle truck occurs at a velocity higher than 80 km/hr. A study of higher velocities should be carried out to determine the maximum impact factor for the three axle truck. The dynamic amplification factor for the two truck event for this particular bridge is generally lower than for the single truck events. This may be explained by a form of destructive interference occurring between the trucks.

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TABLES & FIGURES

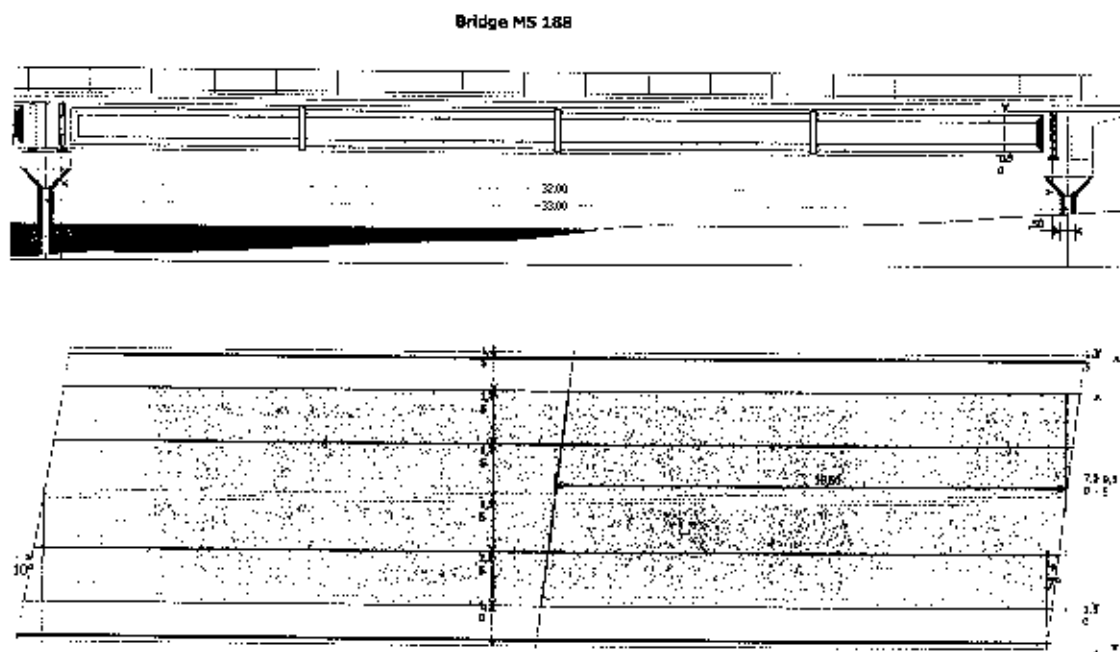


Fig. 1 – Schematic of instrumental bridge



Fig. 2 - Two truck event

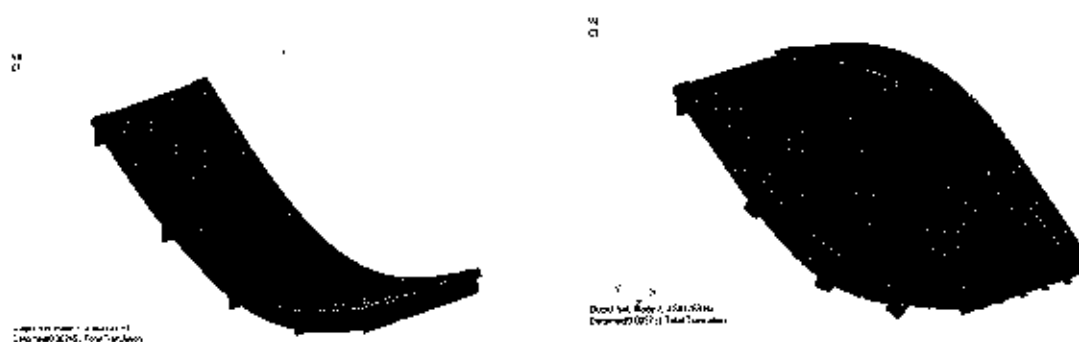


Fig. 3- First mode shape (2.802 Hz), second mode shape (3.681 Hz)

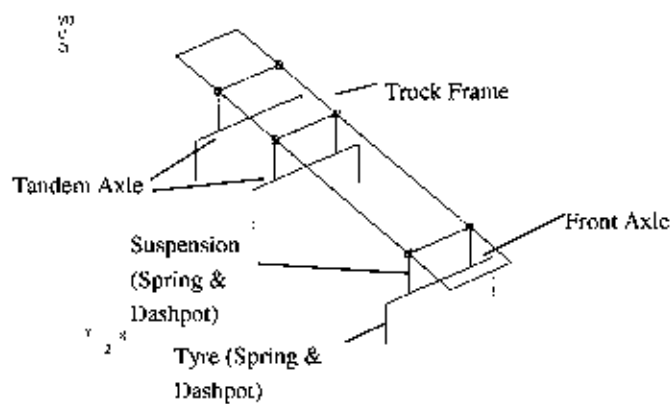


Fig. 4 - Schematic of the three axle truck represented in MSC/NASTRAN

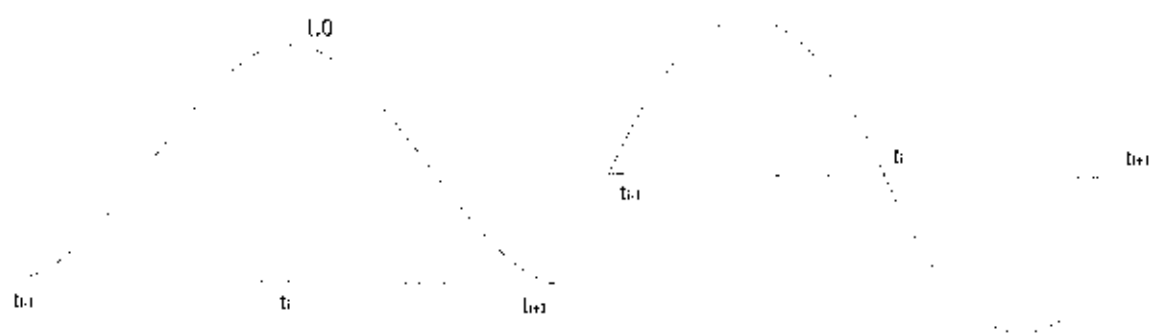
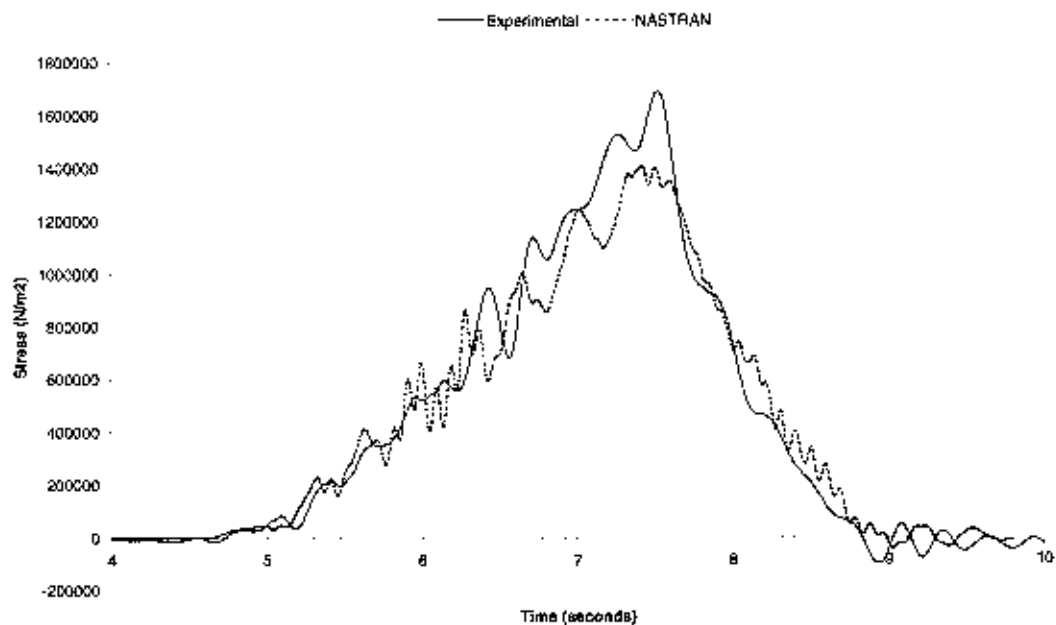
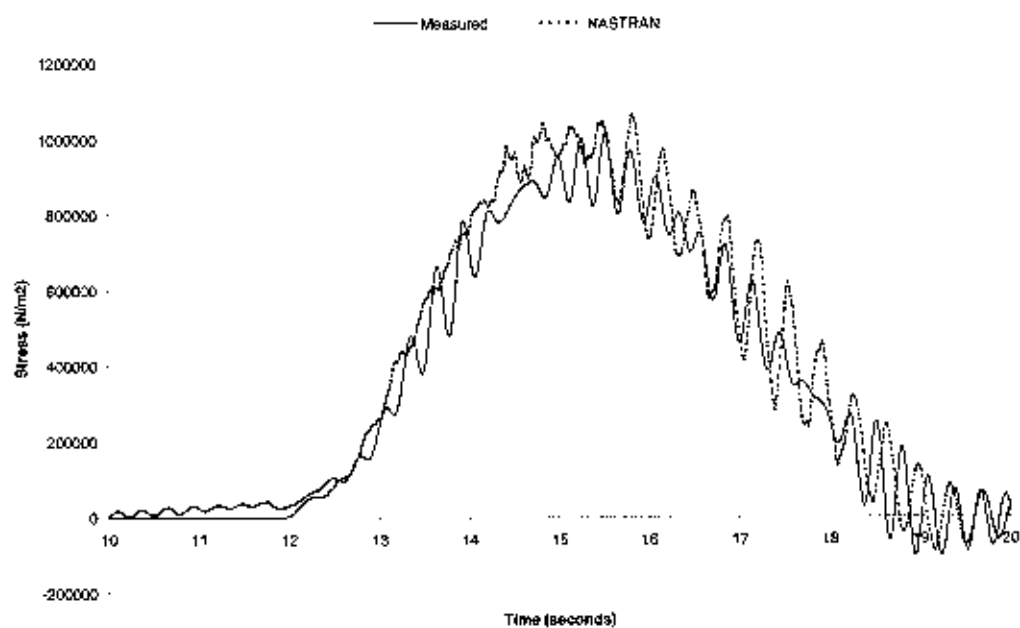


Figure 5 – Auxiliary function $A_i(t)$, auxiliary function $B_i(t)$



2 axle truck



3 axle truck

Fig 6 - Comparison of measured and finite element responses

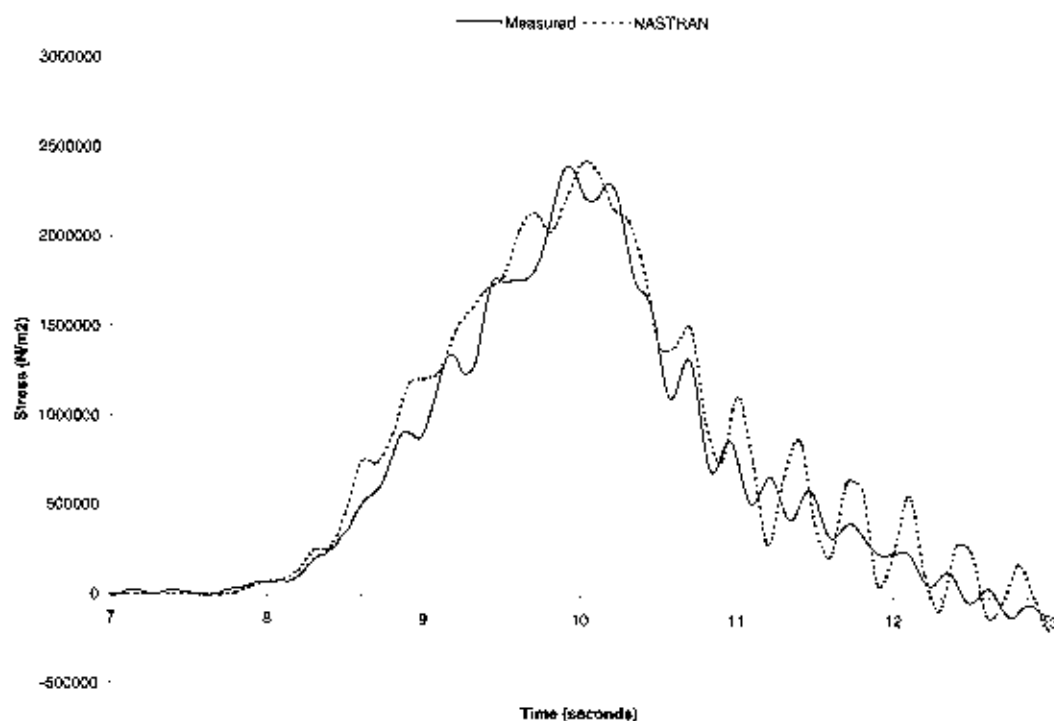


Fig. 7 - Comparison of theoretical and measured responses for both trucks traversing the bridge at the same time

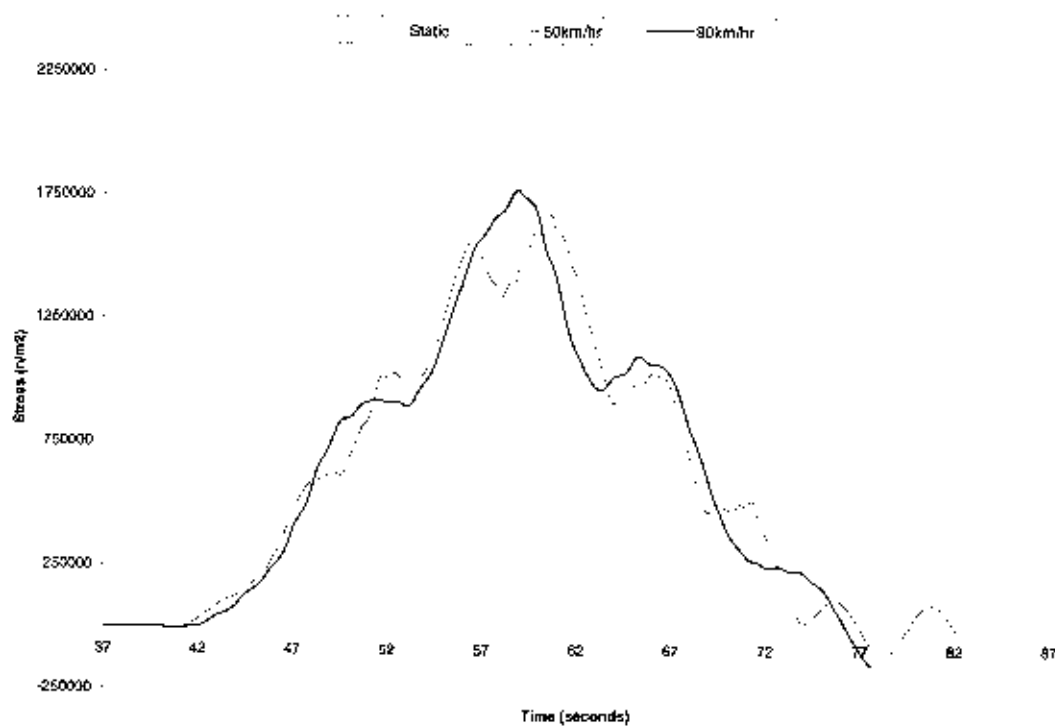


Fig. 8 - Mid-span static response & combined response of bridge being traversed by three axle truck travelling at speeds of 50 & 80 km/hr

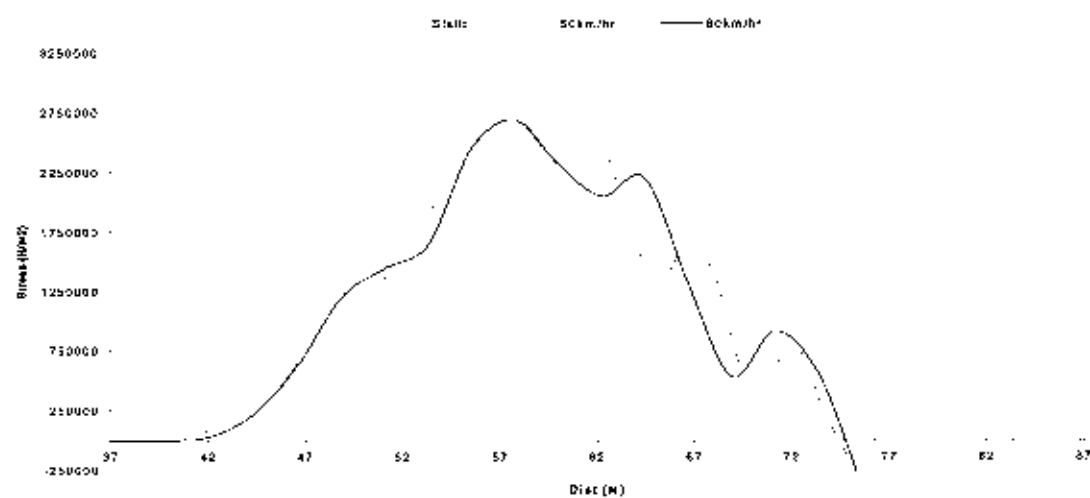


Fig. 9 - Static & combined response of the bridge being traversed by both trucks at speeds of 50 & 80 km/hr

EFFECT OF SURFACE ROUGHNESS ON TRUCK DYNAMIC LOADING AND PAVEMENT DAMAGE

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ABSTRACT

In this paper, some 1,437 pavement sections from ninety-seven projects in Michigan were analyzed to investigate the interaction between pavement surface roughness and distress. The main hypothesis of this research is that an increase in roughness leads to higher dynamic axle loads, which in turn can lead to a tangible acceleration in pavement distress. If this relationship is established, then it will be possible to plan a preventive maintenance (PM) action to smooth the pavement surface. Such a PM action is bound to extend the service life of the pavement by several years. The objectives of this research were to: 1) test the above hypothesis; 2) develop roughness thresholds; and 3) determine the optimal timing of the PM action. The selected projects include all pavement types. The Ride Quality Index (RQI) and Distress Index (DI) were used as measures of surface roughness and distress, respectively. The analysis showed good relationships between DI and RQI for rigid and composite pavements; however for flexible pavements there was significant scatter. A logistic function was used to fit the data. Roughness thresholds were determined as the RQI-values corresponding to peak acceleration in distress. In addition, actual surface profiles of 335 in-service pavement sections from thirty-seven projects were used to generate dynamic axle load using the TruckSim® truck simulation program. Good correlations between dynamic axle load and RQI were obtained. Based on these relationships, roughness threshold values were determined for all pavement types. Rigid pavements had higher RQI threshold values than flexible and composite pavements. The results agreed reasonably with those obtained using MDOT PMS distress data.

INTRODUCTION

All road surfaces have some level of roughness even when they are new, and they become increasingly rougher with age depending on pavement type, traffic volume, environment etc. An increase in pavement roughness leads to higher dynamic axle loads in certain portions of the road. This amplification in the load magnitude can lead to a tangible acceleration in pavement distress; the increased distress, in turn, makes the pavement surface rougher. This process is the result of the interaction between vehicles and pavements. The relationship between pavement damage and roughness (due to truck-pavement interaction) can be used to give an early warning to the pavement management agency under the following hypothesis: that there is a critical value of roughness at which a sharp increase in dynamic load occurs, which would lead to an acceleration in pavement damage.

In this paper, the relationship between the distress index (DI) and the roughness index (RQI) is sought using measured distress and roughness data for 97 projects that have different ages and different levels of distress and roughness. RQI-DI relationships were generated for the three pavement types. The existence of critical roughness values where a sharp increase in distress occurs was confirmed at the network level using these relationships.

SITE SELECTION

Three independent data sets (for a total of 97 projects) were selected from the Michigan pavement network. The first data set has thirty-seven pavement projects: Ten projects with known performance records and having exhibited some distress, and twenty-seven projects where preventive maintenance activities were done during 1997 and 1998. Thirteen of the thirty-seven sites were rigid; fifteen were flexible; and nine were composite pavements. The second and third data sets were selected randomly from the Michigan pavement network. Each data set has

thirty projects: ten rigid, ten composite and ten flexible pavements. These selected projects cover a wide range in pavement age and traffic volume. The length of these pavement projects varies from 2.4 to 36.8 km (1.5 to 23 mi) with an average project length of 11.8 km (7.4 mi). Their ages range from 1 to 39 years. The commercial daily traffic volume ranges from 70 to 12,300. The great majority of rigid pavements were jointed reinforced (JRCF) with slab lengths ranging from 8.2 to 30.2 m (27 to 99 ft). The distribution of these projects in traffic volume and pavement age is shown in Figure 1. This figure shows that, as expected, rigid pavements have higher traffic volumes than flexible and composite pavements. The age for selected rigid pavements is as high as 39 years while composite and flexible pavements have ages less than 25 years.

DATA COLLECTION

For these selected sites, DI and RQI data as well as road surface profiles were obtained from the MDOT PMS database. DI values were available for 1993, 1995 and 1997; whereas RQI values and road profiles were available for the period between 1992 and 1996. The DI and RQI data were available for each 161-m (0.1-mi) long section. Surface profile data were converted to ASCII files containing surface elevations at 76 mm (3 in) intervals. Detailed distress data in the form of distress type, severity and extent were also available at 3 m (10 ft) intervals.

Ride Quality Index (RQI)

As its name suggests, the RQI describes the ride quality of the road. In the early 1970's MDOT conducted a study to determine an objective measure that would correlate ride quality to the subjective opinions of highway users. Using "Psychometric" tests, it was found that some components of a road have a strong effect on user opinion, while others have a significantly lesser effect (1). The Power Spectral Density (PSD) was found to correlate at 90 percent with subjective opinions. Based on this, the profile is split into three wavelength bands: 0.6-1.5 m (2-5 ft), 1.5-7.6 m (5-25 ft), and 7.6-15.2 m (25-50 ft). Wavelengths shorter than 0.61 m (2 ft) mostly create tire noise and those longer than 15.2 m (50 ft) fail to disturb the vehicle suspension. The RQI is calculated from these three PSD wavelength bands according to the equation shown below (2):

$$RQI = 3 \ln(Var_1) + 6 \ln(Var_2) + 9 \ln(Var_3) \quad (1)$$

where Var_1 , Var_2 and Var_3 are variances for 7.6-15.2 m, 1.5-7.6m and 0.6-1.5 m wavelengths, respectively.

An RQI value between zero and 30 indicates excellent ride quality; RQI-values from 31 to 54 indicate good ride quality; values from 55 to 70 indicate fair ride quality, while pavements with RQI-values of more than 70 are considered as having poor ride quality (2). The longitudinal profile for the entire pavement network in Michigan is measured annually using a Rapid Travel Profilometer (RTP). The data is used to calculate both the RQI and the IRI, which is reported to the Federal Highway Administration (FHWA). Figure 2 shows the correlations between RQI and IRI for rigid, flexible and composite pavements.

Pavement Distress Evaluation

The MDOT collects both functional and structural distress data to assess the surface condition of the pavement. Distress data are collected by videotaping 50 percent of the pavement network every year. The videotapes are reviewed in the office and each distress on the pavement surface within each 10-ft (3m) long section is identified, reviewed, checked, scored and stored in the PMS databank. Hence the data includes information on the status of each crack and its location within the 10-ft (3m) long section. The distress data are then grouped into surveying unit sections that are 0.1-mile (161 m) long. Thus the PMS databank contains, for each 0.1-mile (161 m) segment of the road, detailed data for each type of pavement distress and the severity and extent of the 'associated distress'. The term 'associated distress' is used in MDOT rehabilitation practice to denote secondary distresses associated with the principal distress. For example, 'spalling' associated with a transverse crack would be considered as 'associated distress' for the transverse crack.

The MDOT PMS group has developed a rating system whereby each type of principal distress and its associated distress level are ranked and assigned 'Distress Points' (DP) based on their impact on pavement performance and on experience. For any pavement section, the Distress Index (DI) can be calculated as the sum of distress points along the section normalized to the section length. The length of the pavement section (L) is expressed in terms of 161 m (0.1mi) unit-sections. The equation for the DI follows:

$$DI = \sum DP/L \quad (2)$$

where: DI = Distress Index
 Σ DP = Sum of the distress points along the pavement section
 L = Length of the pavement section in 161m (0.1 mile) unit sections

The DI scale starts at zero for a perfect pavement and it increases (without a limit) as the pavement condition worsens. MDOT categorizes DI into three levels: Low (DI < 20), Medium (20 < DI < 40), and High (DI > 40). A pavement with a DI of 50 is considered to have exhausted its service life; hence its remaining service life (RSL) is zero, and it is a candidate project for rehabilitation. This DI threshold-value was established based on historical pavement performance data and on experience.

RELATIONSHIP BETWEEN DI AND RQI

The DI-values for 805 m (0.5 mi) sections were plotted against the corresponding RQI-values from three data sets for rigid, composite and flexible pavements as shown in Figures 3 to 5. The logistic model (3) having the following form was used for the regression analysis.

$$DI = a \times \frac{\exp(b + c \times RQI)}{1 + \exp(b + c \times RQI)} \quad (3)$$

where a , b and c are regression constants.

DI-RQI Relationship from the First Data Set

Regression analysis relating the DI to the RQI for the first data set resulted in R^2 values for rigid and composite pavements of 0.488 and 0.522, respectively. For flexible pavements, there is no good trend and the scatter in the data is very large, with an R^2 -value of 0.311. This probably reflects the higher variability in flexible pavements, indicating that weak spots in the pavement will tend to "attract" damage as opposed to rougher spots inducing higher dynamic axle loads.

Relationships between RQI and DI show that the increased rate in distress is not constant, with the DI sharply increasing at a critical RQI level. This RQI value corresponds to the point at which the acceleration in pavement distress is maximal. Mathematically, it is where the second derivative of DI-RQI function is maximal. Acceleration in the accumulation of DI- vs. RQI-values for each pavement type is shown in Figure 3. The RQI-value where the DI sharply increases was determined to be 57 for rigid pavements. This corresponds to an IRI of 1.70 m/km (106 in/mile). For composite pavements, this RQI-value was found to be 44. This corresponds to an IRI of 1.22 m/km (76 in/mile) for composite pavements. For flexible pavements, the corresponding RQI-value was found to be 45; however this value is not reliable because of the high scatter in the data. Scatter is high in all cases and very high for flexible pavements.

It should be noted that these RQI-values represent the overall behaviour of the pavement network, and therefore cannot be applied to a particular project. In other words, they are useful only for planning at the network level and not at the project level. It is also interesting to note that the critical RQI-value for rigid pavements corresponds to a DI of 8, as opposed to a DI of 18 and 22 for composite and flexible pavements, respectively. This may imply that the optimal time window for preventive maintenance actions corresponds to a lower distress level (higher remaining service life) for rigid pavements than for composite or flexible pavements.

DI-RQI Relationship from the Second and Third Data Sets

For each pavement type, the DI-values for 800 m (0.5 mi) sections were again plotted against the corresponding RQI-values using the data from the second and third independent data sets (see Figures 4 and 5). The same logistic model that was used for the first data set was used in the regression analysis for these data sets. For rigid pavements, plots of DI against RQI from the new data sets have R^2 -values of 0.699 and 0.731. For composite pavements, the R^2 -values from the new data sets are 0.511 and 0.603. For flexible pavements, the R^2 -values from the new data sets are 0.448 and 0.507. Again, the critical RQI-values were determined as the RQI-values where the acceleration in pavement distress (DI) is maximal. The critical RQI-values from the new data sets were determined to be 54 and 57 for rigid pavements. These values agree very well with that from the first data set that (RQI=57). For composite pavements, the critical RQI-values were determined to be 48 and 42. For flexible pavements, they

were 40 and 44. These values agree reasonably well with the values obtained from the first data sets, which are 44 and 45 for composite and flexible pavements, respectively.

The critical RQI-values were also determined using all data sets including the original data set and the two independent data sets (see Figure 6). Using all data sets, the critical RQI-values were determined to be 55, 45 and 41 for rigid, composite and flexible pavements, respectively. Finally, the critical RQI-values determined from the original data set, two independent data sets and all data sets are summarized in Table 1.

Interpretation of the Results

The above results indicate that rigid pavements have higher critical RQI-values than composite and flexible pavements. This seems to be caused by the following three factors:

First, the mechanisms of how the pavement surface becomes rough with time are different in rigid and flexible (or composite) pavements. For rigid pavements, the pavement surface becomes rough because of faulting, curling and warping. These distresses can happen without the existence of cracks. This means that the pavement surface can be rough without the existence of cracks, i.e., a rigid pavement can have high RQI-values without an increase in DI-value under the MDOT distress index pointing system. For flexible pavements, on the other hand, the pavement surface becomes rough mainly because of cracks. This difference makes rigid pavements exhibit high critical RQI-values.

The second factor could be the initial smoothness (or roughness) of a newly constructed or rehabilitated pavement. Generally, the initial roughness for rigid pavements is higher than that for flexible pavements because of the existence of joints. This high initial roughness may cause the critical RQI-values to shift up to a higher value.

Finally, the third factor could be the material behavior. Portland cement concrete is stronger than asphalt concrete; therefore, rigid pavements should be able to sustain higher dynamic axle loads than do flexible pavements. All the above-cited factors may lead to a higher critical RQI-value for rigid pavements.

RELATIONSHIP BETWEEN DYNAMIC TRUCK RESPONSE AND RQI

Correlation between Dynamic Loads from Different Axles and Trucks

The TruckSim™ program was used to generate dynamic loads from three truck types: 2 and 3-axle single unit trucks and a 5-axle tractor semi-trailer (see Table 2). To study spatial repeatability of dynamic loads for all truck axles, the correlation between the different axles were studied. The analysis showed a strong correlation between aggregate axle loads for the different trucks with coefficients of correlation higher than 0.77. Cole has shown that a p -value of 0.707 is indicative of good spatial repeatability (4). Aggregate axle loads for each truck and the second axle of the 5-axle tractor semi-trailer also showed very good correlation with values around 0.7. This indicates that this axle can be used to represent the aggregate load from all trucks. It should be noted that the combination of 5-axle tractor semi-trailers and 2- and 3-axle single unit trucks constitute more than 80% of the truck population in Michigan. The details of this analysis can be found in (5).

Relationship between Ride Quality Index and Dynamic Load

To get dynamic load vs. RQI curves for each pavement type, several 161 m (0.1 mi) sections for each roughness level (RQI=30, 40, 50, 60, 70, and higher than 80) were selected randomly from the 37 projects for a total 333 sections. Table 3 shows the number of samples used for each roughness level and pavement type. Dynamic axle load profiles were generated along each 161 m (0.1-mi) section, using actual pavement surface profiles as input to the truck simulation program, TruckSim™. The second axle of a typical 5-axle tractor-semi-trailer, was considered as representative of the aggregate loads from all trucks, as discussed above. From these dynamic axle-load profiles, DLC (Dynamic Loading Coefficient) and the 95th percentile axle load were calculated and plotted against the corresponding RQI-values. Figure 7 (a) shows the relationship between DLC and RQI for rigid, flexible and composite pavements. The relationship between the 95th percentile axle load and RQI is shown in Figure 7 (b). The data were fit to fourth-order polynomial curves, with the resulting R^2 -values ranging from 0.85 to 0.95. The DLC-RQI curves had slightly better R^2 -values ($R^2 = 0.91$ to 0.95) than the 95th percentile axle load curves ($R^2 = 0.85$ to

0.93), with the highest values being for rigid pavements and the lowest values for flexible pavements. This is expected because of the variability in asphalt-surfaced pavements.

Dynamic-load-induced Pavement Damage and Corresponding Reduction in Pavement Life

The relative dynamic load-induced damage in pavements can be estimated by using a power law (6):

$$\text{Relative Damage} = \left(\frac{L_{\text{dynamic}}}{L_{\text{static}}} \right)^n \quad (4)$$

where n is the damage exponent from the failure criterion (typically, $n = 3-5$).

Using the 4th power law, relative damages from the 95th percentile dynamic load at different RQI levels were calculated and plotted in Figure 8 for all pavement types. The corresponding R^2 -values were between 0.83 and 0.92, with the higher values being for rigid pavements. The general equation for these curves can be written as:

$$y = a \times RQI^4 + b \times RQI^3 + c \times RQI^2 + d \times RQI + e \quad (5)$$

where y is the relative damage and a , b , c , and d are regression constants.

The theoretical percent reduction in pavement life can be calculated as (7):

$$Y = \text{Percent Reduction in Pavement Life} = 100\% [1 - (\text{Relative Damage})^{-1}]$$

Determination of Roughness Threshold Values

A range of RQI-values where pavement damage sharply increases can be determined from the *derivatives* of the above function (Percent Reduction in Pavement Life) as follows:

The *lower bound* for the critical RQI-value can be taken as the *minimum* of the first derivative (minimum slope of the curve), beyond which the rate of damage or reduction in pavement life starts increasing:

$$RQI_{\min} = \min f'(RQI) \text{ where: } f'(RQI) = \frac{dY}{dRQI} \quad (6)$$

The *upper bound* for the critical RQI-value can be taken as the *maximum* of the second derivative (maximum acceleration of the curve), beyond which the acceleration in damage is highest, or the rate of increase in the reduction of pavement life is highest:

$$RQI_{\max} = \max f''(RQI) \text{ where: } f''(RQI) = \frac{d^2Y}{dRQI^2} \quad (7)$$

The functions $f(RQI)$ and $f'(RQI)$ for each pavement type are also shown in Figure 8. The function $f(RQI)$ decreases with increasing RQI down to a minimum point after which it starts to increase. The RQI-value where $f(RQI)$ is minimum can be taken as the *lower bound* value. The function $f'(RQI)$ vs. RQI increases with increasing RQI up to a maximum point beyond which it starts to decrease. The RQI-value where $f'(RQI)$ is maximum can be taken as the *upper bound* value.

The critical RQI-value would be the RQI value that corresponds to this reduction in pavement life. The critical RQI-values from the mechanistic analysis are as follows:

Rigid pavements:	RQI = 61 (lower bound) ; RQI = 77 (upper bound).
Flexible pavements:	RQI = 47 (lower bound) ; RQI = 66 (upper bound).
Composite pavements:	RQI = 50 (lower bound) ; RQI = 70 (upper bound).

At the lower bound value, reduction in pavement life starts to accelerate, while the acceleration is highest at the upper bound value. Beyond the upper bound value, reduction in pavement life decelerates. The optimal timing for preventive maintenance action would be between the lower and upper bound values. However, the range in dynamic load for a given RQI value is wide.

These values are not very sensitive to the exponent used in the power damage law, as shown in Table 4, and the lower-bound values are in reasonable agreement with field-derived values based on surface distress accumulation. The field-derived values are lower than the dynamic-load-based values; this can be explained by the fact that distress accumulation in in-service pavements is due to many factors such as structural and material integrity of the pavement components and environmental effects. These additional factors will cause an earlier increase in distress. Since the field-derived roughness thresholds are based on the rate of increase in distress, they are bound to be lower than those predicted mechanistically solely on the basis of the increase in dynamic loading.

These threshold values represent the overall behaviour of pavements at the network level, and may not be applicable for a particular pavement project. Therefore, such thresholds can be used for network-level pavement management, and not necessarily at the project level.

CONCLUSION

The MDOT PMS database was used to develop relationships between distress (Distress Index, DI) and roughness (Ride Quality Index, RQI) for all pavement types. Three independent data sets for a total of 97 projects, or 1,437 (0.5-mile) sections, that have different ages and levels of distress and roughness were used for this analysis. DI-RQI relationships for rigid pavements had the highest R^2 -values (0.488, 0.699 and 0.731). Flexible pavements had the lowest R^2 -values (0.311, 0.448 and 0.507), and composite pavements had in-between R^2 -values (0.522, 0.511 and 0.603). Critical RQI-values corresponding to maximum distress acceleration were obtained from these relationships. These were 57, 54 and 57 for rigid pavements; and 44, 48 and 42 for composite pavements. For flexible pavements, critical RQI-values were found to be 45, 40 and 44; however, these values are not reliable because of the high scatter in the data. This variability is due to the fact that distress is caused not only by axle loads but also by many other factors.

In the mechanistic approach, the mathematical expression for the reduction in pavement life as a function of roughness allowed for the determination of lower and upper bound roughness (RQI) threshold values. The lower bound values were taken as those corresponding to the minimum slope of the RQI-life reduction curves. These values were found to be equal to 61, 50 and 47 for rigid, composite and flexible pavements, respectively. The upper bound values were taken as those corresponding to the maximum acceleration of the RQI-life reduction curves. These values were found to be equal to 77, 70 and 66 for rigid, composite and flexible pavements, respectively. Mechanistically determined RQI-threshold-values were not sensitive to the exponent used in the power damage law. The lower bound RQI-threshold values compared reasonably well with field-derived values based on surface distress accumulation.

Finally, these threshold values represent the overall behaviour of pavements at the network level, and may not be applicable for a particular pavement project. Therefore, such thresholds can be used for network-level pavement management and not necessarily at the project level.

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TABLES & FIGURES

Table 1 - Summary of Critical RQI Values

Pavement Type	First Data Set	Second Data Set	Third Data Set	All Data Sets
Rigid Pavements	57	54	57	55
Composite Pavements	44	48	42	45
Flexible Pavements	45	40	44	41

Table 2 - Truck Matrix Sizes and Weights




Truck Configuration	Configuration Name	GCVW (kN)	Axle Loads (kN)	Wheel Base (m)
	2 Axle Truck	125	49/76	4.3
	3 Axle Truck	150	56/94	6.1
	5 Axle Semi-Trailer	356	54/151/151	3.6/11.0

Table 3 - Number of Samples (n) for RQI vs. Dynamic Load Analysis

Rigid Pavements	RQI	35	45	55	65	75	> 80	Subtotal
	n	16	15	23	20	13	22	109
Flexible Pavements	RQI	20	30	40	50	60	> 70	
	n	9	22	23	16	18	23	111
Composite Pavements	RQI	30	40	50	60	70	> 80	
	n	12	19	24	21	12	25	113
Total								333

Table 4 - RQI Threshold Values from Different Power Laws

Pavement Type	Power Law	Lower Threshold	Upper Threshold	Field-derived Threshold
Rigid Pavements	4 th Power	61	77	55
	5 th Power	61	76	
	6 th Power	62	75	
Flexible Pavement	4 th Power	47	66	38
	5 th Power	47	64	
	6 th Power	49	63	
Composite Pavements	4 th Power	50	70	45
	5 th Power	52	69	
	6 th Power	54	69	

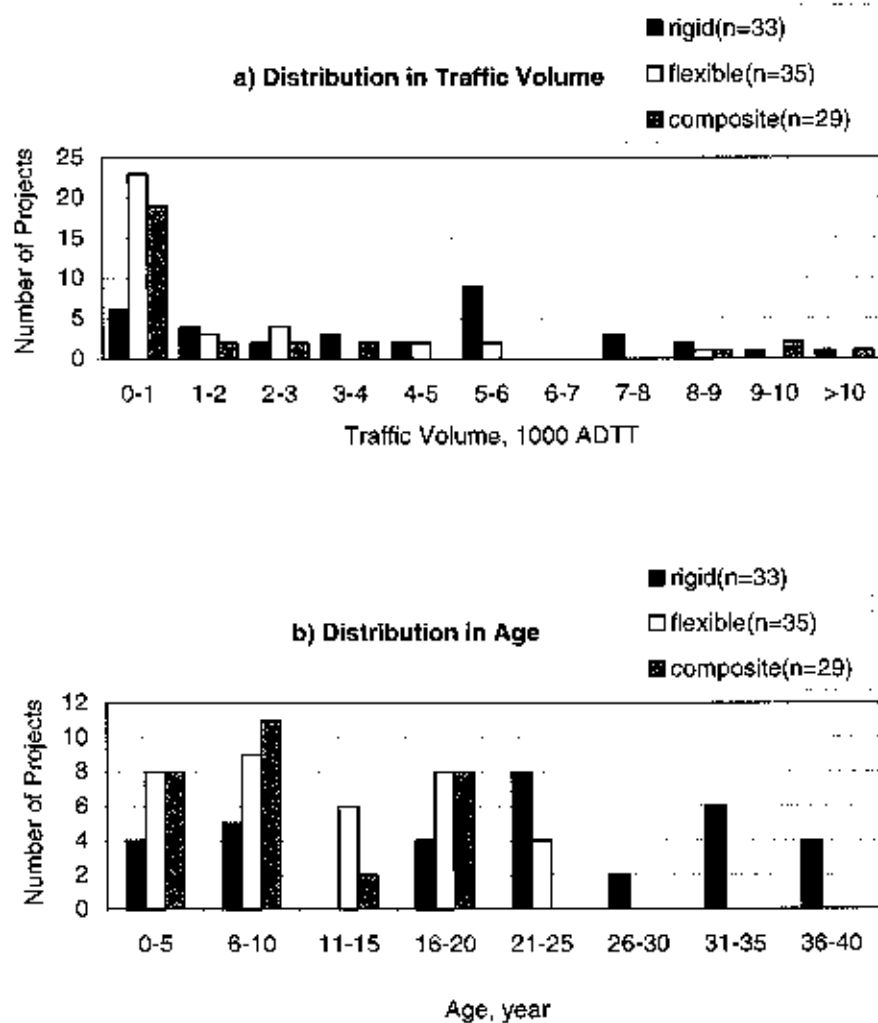


Figure 1 - Distribution of Projects in Traffic Volume and Pavement Age

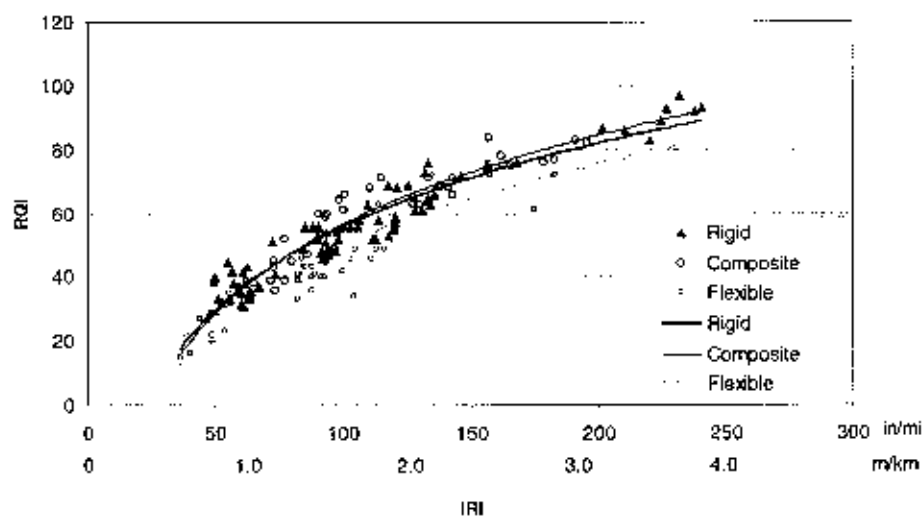


Figure 2 - Relationship between RQI and IRI

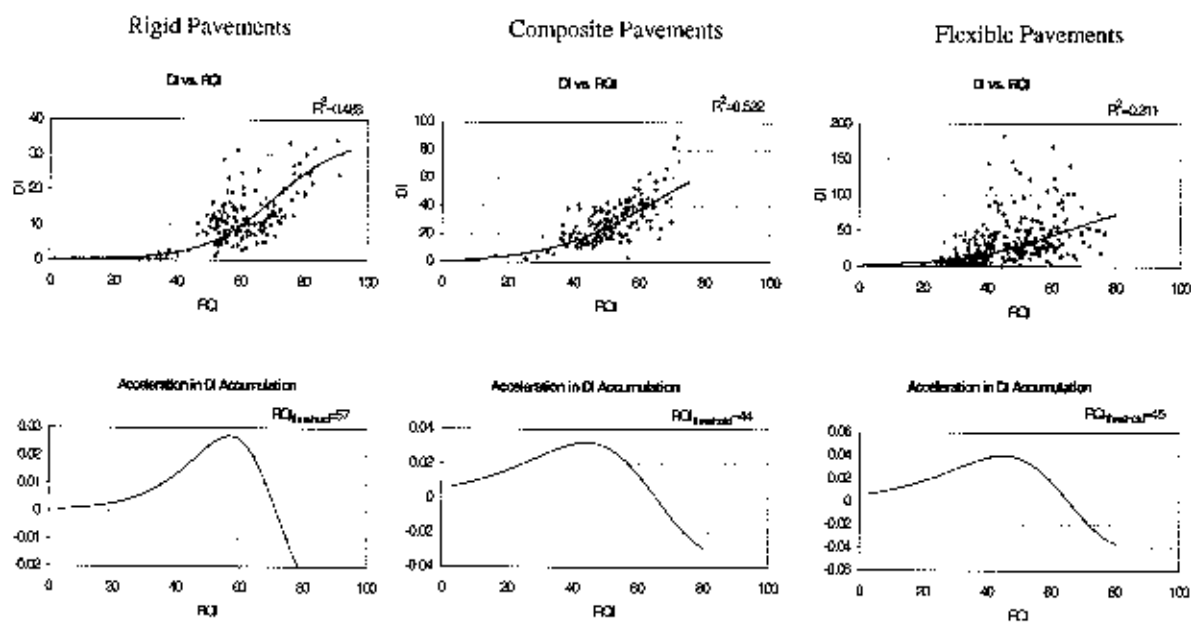


Figure 3 - Relationship Between DI and RQI for All Pavement Types from Data Set #1

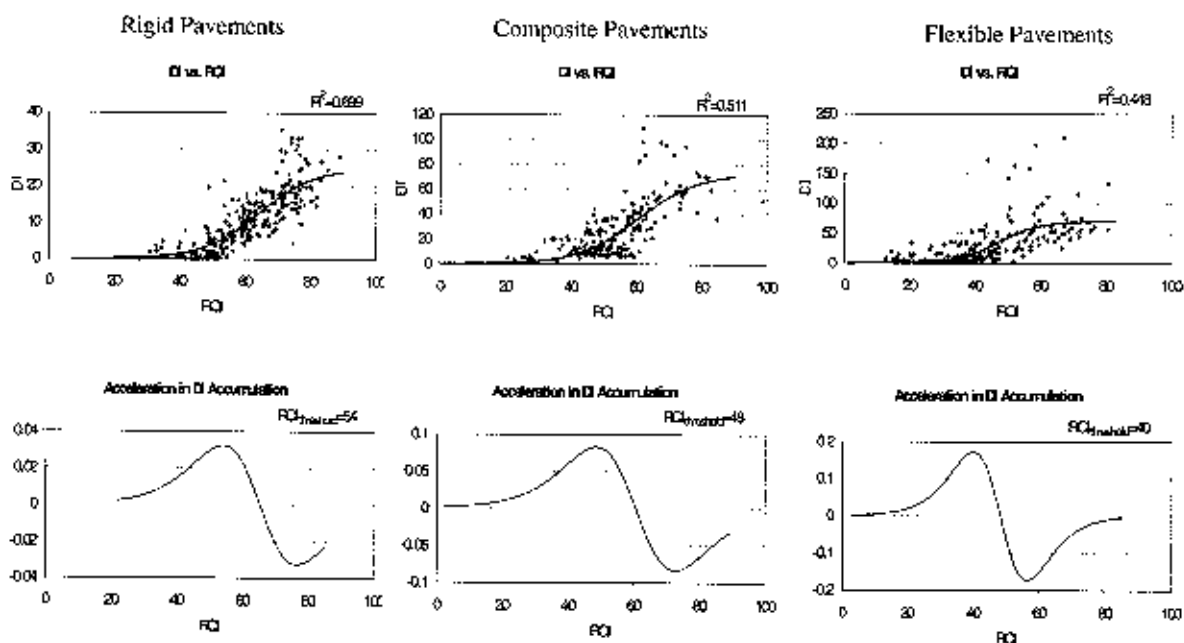


Figure 4 - Relationship Between DI and RQI for All Pavement Types from Data Set #2

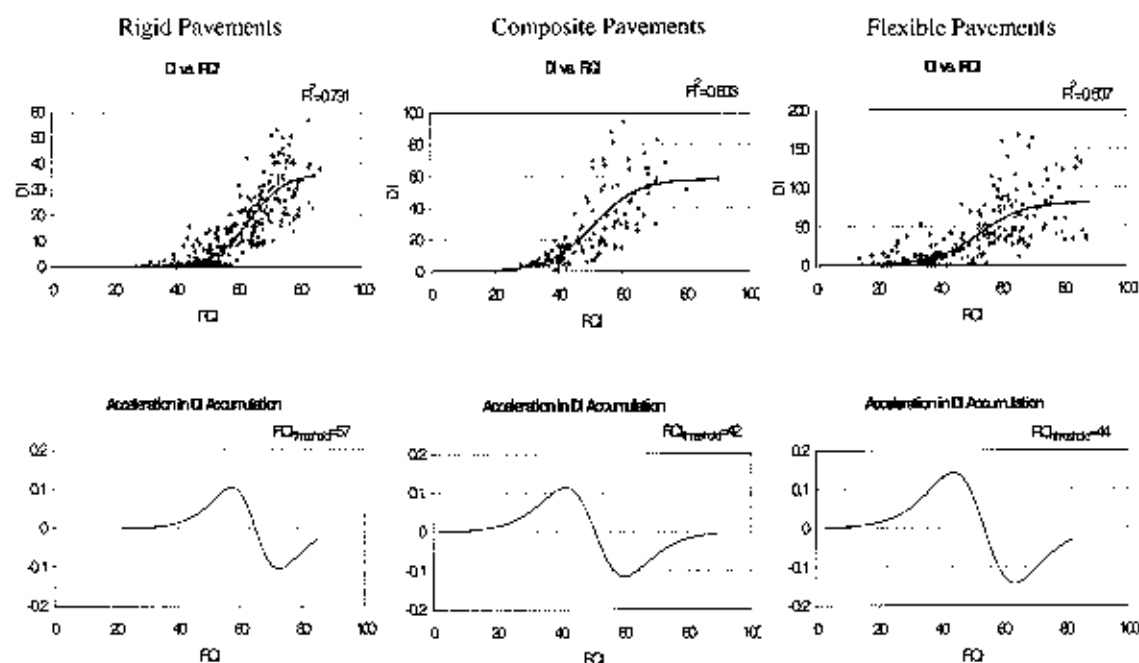


Figure 5 - Relationship Between DI and RQI for All Pavement Types from Data Set #3

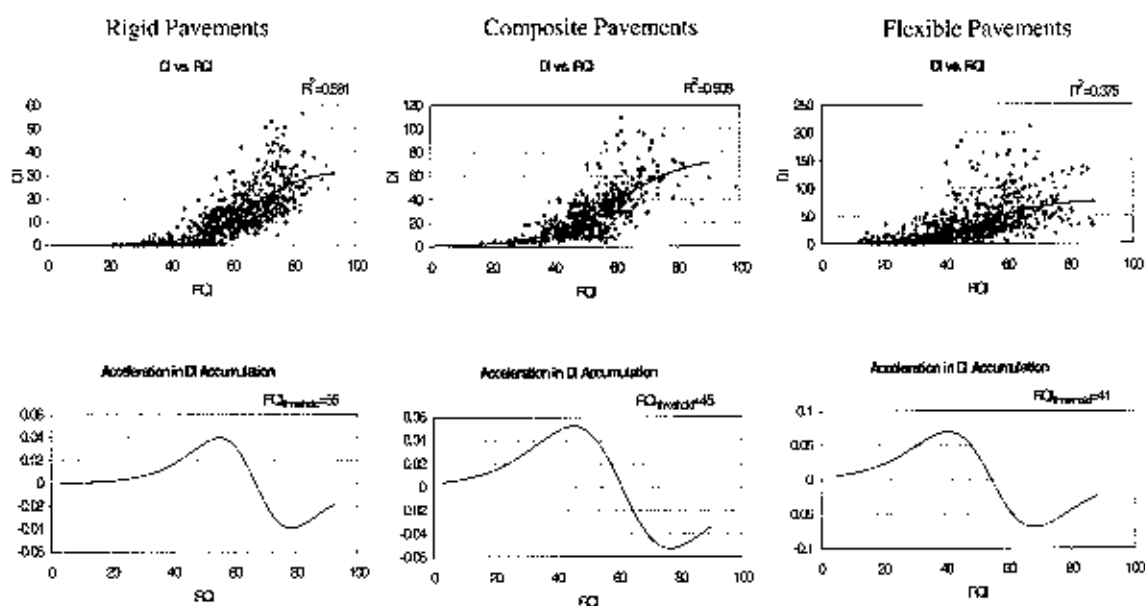


Figure 6 - Relationship Between DI and RQI for All Pavement Types from All Data Sets Combined

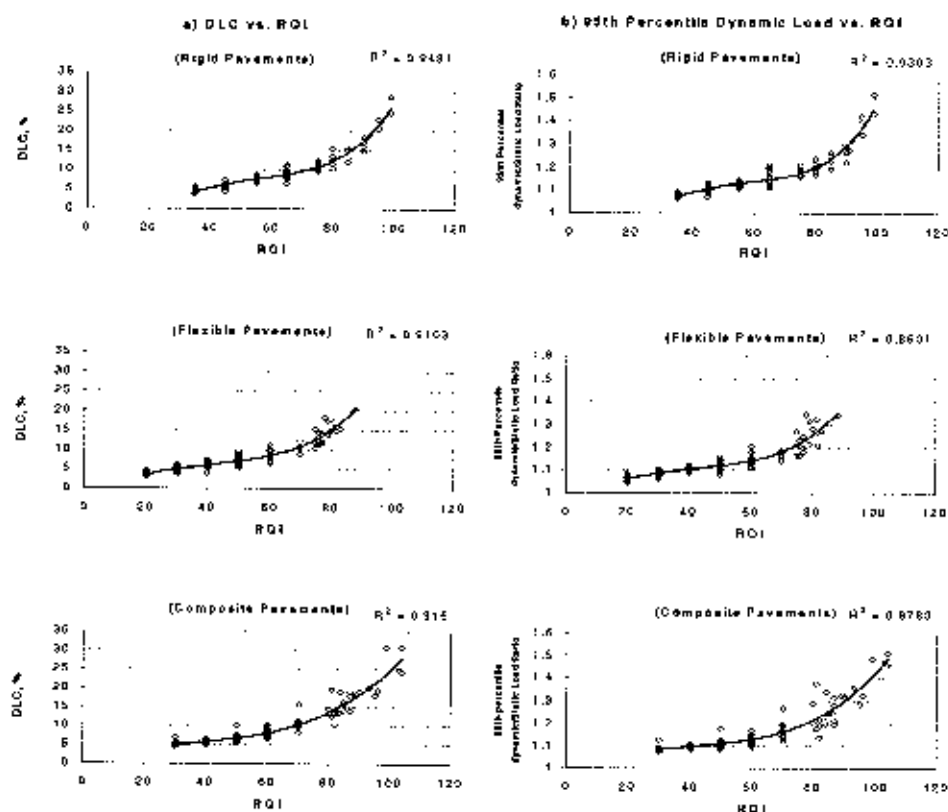


Figure 7 – Relationship between Dynamic Load and RQI

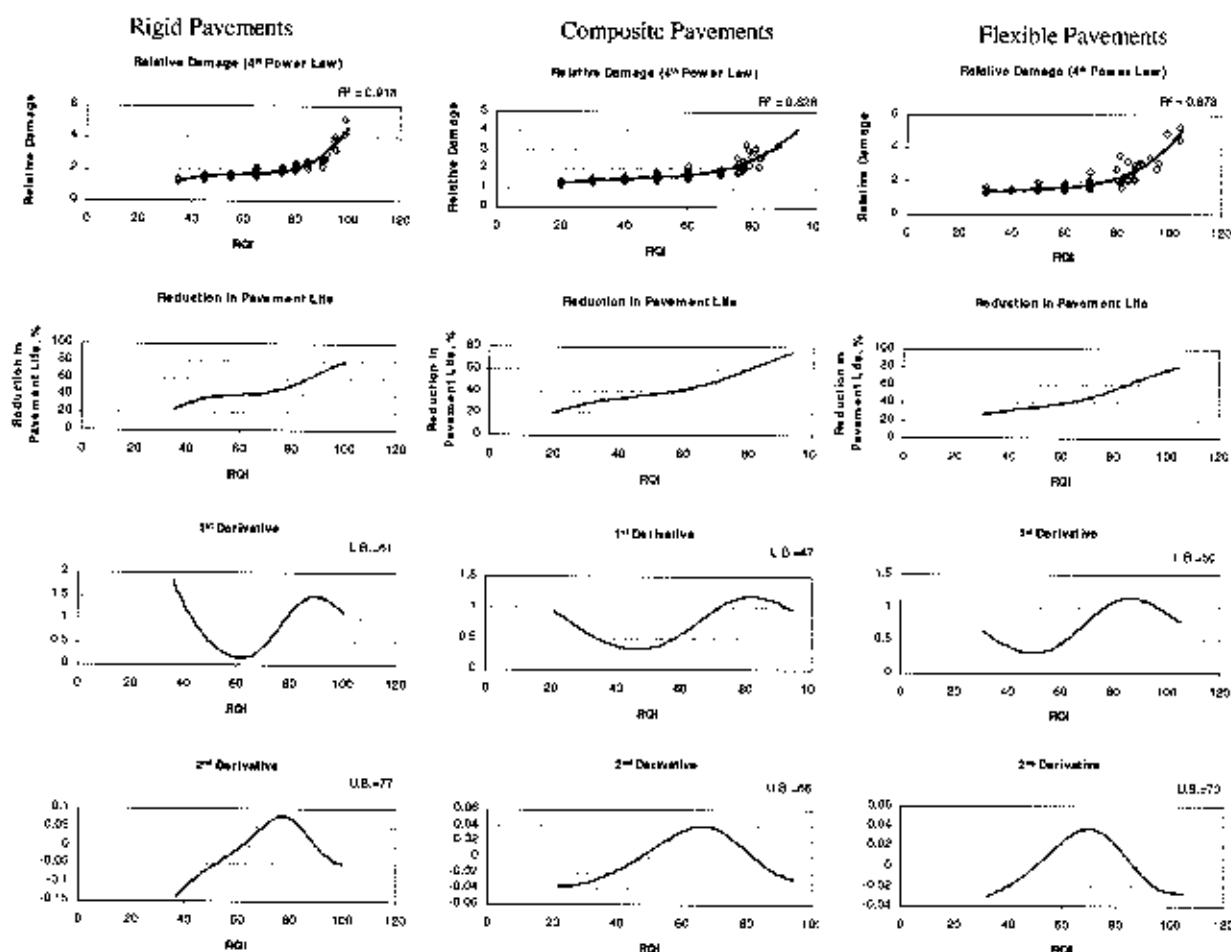


Figure 8 - Relative Damage (4th Power Law) and Reduction in Pavement Life vs. RQI

A PROFILE BASED TRUCK DYNAMIC LOAD INDEX (DLI)

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ABSTRACT

In this paper, a new roughness index called the Dynamic Load Index (DLI) is developed for the purpose of identifying pavement profiles that are likely to generate high dynamic truck-axle loads. The DLI is calculated as a weighted index of variances of the profile elevation in the frequency ranges 1.5-4 Hz and 8-15 Hz. The first frequency range corresponds to truck body bounce, while the second frequency range corresponds to axle bounce. The analysis showed a very good relationship between DLI and dynamic load. The DLI was tested on a range of road profiles from in-service pavements, and it was found that for any particular value of ride quality index (RQI), the DLI can cover a wide range of values, and this variation in DLI was found to correlate very well with dynamic load, as predicted by a truck simulation program. This was not the case for the International Roughness Index (IRI), which gave a low coefficient of correlation with dynamic load for the same range of profiles. Therefore, the new index can differentiate between profiles that generate high dynamic loads and those having the same RQI but generating low dynamic loads. Most importantly, the use of the DLI index negates the need for running a truck simulation program. This makes it possible for a state highway agency to decide whether a particular pavement with a given surface profile needs smoothing (to extend its service life) based on the DLI-value.

INTRODUCTION

A number of profile indices have been developed for characterizing pavement surface roughness in terms of serviceability or ride quality. Such indices include "ride numbers" such as the Ride Quality Index used by the Michigan department of Transportation (MDOT). The most used roughness index is the International Roughness Index (IRI). All of these indices are based on the response of passenger cars to the pavement profile. While they can generally have a good correlation with dynamic truck loading, these indices are not able to determine whether a specific pavement section is "truck friendly" or not. This makes it difficult to decide whether or not a particular pavement section should be smoothed based on such indices because damage in pavements is caused mainly by heavy truck axle loads. An accurate prediction of roughness level that will excite trucks requires the evaluation of dynamic truck axle loading likely to be generated by the profile characteristics of the individual pavement section. One way to predict dynamic axle loads, given a surface profile, is to use a truck simulation computer program. This would require some knowledge of truck dynamics and a minimum fluency in truck parameters for specific components such as the suspension system, the chassis and the tires. Therefore it would be impractical for a state highway agency such as MDOT to adopt such an approach. An alternative method would be to determine the relative increase in dynamic axle loads directly from the profile itself, since dynamic axle loading is a function of the pavement surface profile characteristics. In this paper, a new roughness index termed the Dynamic Load Index (DLI) is developed. This new index negates the need for running a truck simulation program to determine whether a pavement profile is friendly/unfriendly from a dynamic loading aspect. The DLI can be a useful pavement management tool since it makes it possible to decide whether a particular pavement with a given surface profile needs smoothing for the purpose of reducing dynamic truck-axle loads and preventing accelerated pavement damage.

BACKGROUND

The Michigan DOT has funded a research study to develop roughness thresholds aimed at minimizing dynamic truck-axle loads (1). Because MDOT uses the RQI as its roughness index, it was decided to first investigate the feasibility of using RQI for developing these thresholds.

Ride Quality Index (RQI)

In the early 1970's MDOT conducted a study to determine an objective measure that would correlate ride quality to the subjective opinions of highway users. The Power Spectral Density (PSD) of pavement surface profiles was found to correlate at 90 percent with subjective opinions. Based on this, the profile is split into three wavelength bands: 0.6-1.5 m (2-5 ft), 1.5-7.6m (5-25ft), and 7.6-15.2 m (25-50ft). Wavelengths shorter than 0.61 m (2 ft) mostly create tire noise and those longer than 15.2m (50 ft) fail to disturb the vehicle suspension. The RQI is calculated from these three PSD wavelength bands according to the equation shown below (2):

$$RQI = 3 \ln (Var1) + 6 \ln (Var2) + 9 \ln (Var3) \quad (1)$$

where *Var1*, *Var2* and *Var3* are variances for 7.6-15.2 m (25-50ft), 1.5-7.6m (5-25ft) and 0.6-1.5 m (2-5 ft) wavelengths, respectively.

An RQI value between zero and 30 indicates excellent ride quality; RQI-values from 31 to 54 indicate good ride quality; values from 55 to 70 indicate fair ride quality, while pavements with RQI-values of more than 70 are considered as having poor ride quality (3).

Relationship between RQI and Dynamic Load

In this analysis, actual pavement surface profiles of 333 (161-m or 0.1-mile) sections from 37 projects were used as input to the truck simulation program, TruckSim™. The pavements included all types (rigid, flexible and composites) with age varying from zero to 39 years. Rigid pavements were mainly jointed reinforced pavements (JRCF) with slab lengths ranging from 8.2 to 30.2 m (27 to 99 ft). Distress levels included the entire range from no distress to distress levels exceeding the threshold for rehabilitation. The average daily commercial traffic volume varied from 70 to 8,900, and the project lengths varied from 2.4 km (1.5 mi) to 26.7 km (16.6 mi). The spatial repeatability¹ of dynamic truck-axle loading was analysed using three different truck types: 2 and 3-axle single unit trucks and a 5-axle tractor semi-trailer. All trucks were equipped with leaf spring suspensions. Tables 1 and 2 summarize the trucks characteristics. A total of 168 actual pavement profiles from ten in-service pavement projects (5 rigid, 3 flexible and 2 composite pavements) were used for this purpose. Based on the findings of this analysis, the 2nd axle load in the 5-axle tractor semi-trailer was determined to be representative of the three truck types used (5). This "reference" axle was therefore used in developing relationships between RQI and dynamic load.

Using the time histories of the reference axle, the Dynamic Loading Coefficient (DLC) and the 95th percentile axle load were calculated and plotted against the corresponding RQI-values. The DLC is an "average" measure of the magnitude of the dynamic variation of axle load over a given surface profile, and is calculated as the ratio of the standard deviation of the dynamic load fluctuations over the static load. The DLC-value for a perfectly smooth pavement surface would theoretically be zero. DLC-values less than 8% indicate moderately smooth pavements, while DLC-values higher than 10% are considered to be indicative of moderately rough pavements, and DLC-values higher than 15% indicate very rough pavement surfaces. Cases of DLC-values higher than 20% could occur when the truck is equipped with an unfriendly suspension system such as the walking-beam type (an older, rugged suspension system that is used mainly in off-road trucking nowadays), with maximum values possibly reaching 30 to 35% (6). The 95th percentile axle load is an "extreme" measure of dynamic loading that is indicative of "hot" spots within the pavement surface.

Figure 1 (a) shows the relationship between DLC and RQI for rigid, flexible and composite pavements. Figure 1(b) shows the relationship between the 95th percentile axle load and RQI. While these plots of truck dynamic axle loading against RQI have high R²-values, they show a wide range of dynamic load magnitudes for a given RQI value. This is because RQI is calculated from a wide range of wavelengths ranging from 0.3 to 15.1 m (2-50 ft).

DEVELOPMENT OF A DYNAMIC-LOAD-BASED ROUGHNESS INDEX

According to the literature, various experimental and theoretical studies have shown that vehicle bounce occurs in the range of frequencies between 1.5 and 4 Hz, and axle bounce occurs between 8 and 15 Hz (7,8). These

¹ See reference (4) for a literature review on the subject.

frequencies correspond to wavelengths between 6.7 and 17.9 m (22-59 ft) and between 1.8 and 3.3 m (6-11 ft) at a vehicle speed of 96 km/hr (60mph). The remaining wavelength ranges have little to do with dynamic truck-axle loads. Thus, if a profile based index is focused only on the above wavelengths, i.e., 6.7-17.9 m (22-59 ft) and 1.8-3.3 m (6-11 ft), it could have a better correlation with truck dynamic axle loading than car-response based pavement roughness indices such as RQI and IRI.

Formulation of the New Profile Index

According to linear random vibration theory, the PSD of truck response is obtained by multiplying the square of the truck response function by the PSD of the surface profile. Figure 2 shows this relationship schematically. The variance of truck response can be expressed mathematically as (9),

$$V_y = \frac{1}{v} \int \left| G(w) \right|^2 S_x\left(\frac{w}{v}\right) dw = \int \left| G(w) \right|^2 S_x(w) dw \quad (2)$$

where V_y is the variance of truck dynamic load;

$G(w)$ is the truck response function;

$S_x(k = \frac{w}{v})$ is the PSD function of the surface profile;

w is circular frequency;

v is vehicle speed;

k is wavenumber; and

$S_x(w) = \frac{1}{v} S_x(k = \frac{w}{v})$ is the PSD of the temporal input to the truck suspension system.

The area under the PSD curve of the profile at a given frequency range can be approximated as the profile variance for that frequency range. If only the frequency ranges of 1.5-4.0 and 8.0-15.0 Hz, which correspond to truck body and axle bounces, are considered, V_y can be approximated as,

$$V_y = \left| G(w_1) \right|^2 V_1 + \left| G(w_2) \right|^2 V_2 \quad (3)$$

where $G(w_1)$ is the peak value of truck response function at the frequency range of 1.5-4.0 Hz;

$G(w_2)$ is the peak value of truck response function at the frequency range of 8.0-15.0 Hz;

V_1 is the variance of the elevation in the frequency range of 1.5-4.0 Hz; and

V_2 is the variance of the elevation in the frequency range of 8.0-15.0 Hz.

The standard deviation of truck response is therefore,

$$\sigma_y = \sqrt{V_y} = \sqrt{\left| G(w_1) \right|^2 V_1 + \left| G(w_2) \right|^2 V_2} \quad (4)$$

Equation (4) suggests the following form for the new roughness index, called Dynamic Load Index (DLI):

$$DLI = \sqrt{a_1 V_1 + a_2 V_2} \quad (5)$$

where: V_1 is the variance of elevation of Profile 1 (unit: 10^{-2} in, 1in = 25.4 mm); Profile 1 contains only waves in the wavelength range of 6.7 to 17.9 m (22 to 59 ft), corresponding to a frequency range of 1.5-4.0 Hz for a truck travelling at 96 km/hr (60 mph);

V_2 is the variance of elevation of Profile 2 (unit: 10^{-2} in, 1in = 25.4 mm); Profile 2 contains only waves in the wavelength range of 1.8 and 3.3 m (6 to 11 ft), corresponding to a frequency range of 8.0-15.0 Hz for a truck travelling at 96km/hr (60 mph); and

a_1 and a_2 are weighting factors.

Profiles 1 and 2 are obtained by filtering out the content from all wavelengths of the original profile that has been transformed in the wavenumber domain except for the critical wavelength ranges. This process is done according to the following steps:

1. Transform the original profile into the wave number domain using the Fast Fourier Transform (FFT).

2. Split the transformed profile into two profiles that have wavelength ranges of 6.7-17.9 m (22-59 ft) and 1.8-3.3 m (6-11 ft), respectively. This can be done by forcing zero amplitudes for all wavelengths except for the above critical wavelength ranges.
3. Transform the above two profiles back to the space domain using the Inverse FFT (IFFT) algorithm.
4. Calculate variances (V_1 and V_2) from both profiles for each 161-m (0.1-mile) section.
5. Calculate DLI for each 161-m (0.1-mile) section using the above equation.

The weighting factor α_1 for V_1 is set equal to one for convenience. The value for the weighting factor, α_2 , was determined as that which gives the highest correlation between the DLI and dynamic load.

Calibration of the New DLI Index

The overall relationship between DLI and dynamic load was determined for rigid, composite and flexible pavements, respectively. The analysis used all 333 pavement sections representing a large range of RQI values. Figures 3 (a-1) and (a-2) show the variation of DLC and 95th percentile dynamic load, respectively, with DLI for rigid pavements. Figure 3 (a-3) shows R^2 -values for different weighting factors. Figures 3 (b) and (c) show the same things for composite and flexible pavements, respectively. These plots have higher R^2 -values and lower standard error (SE) values than those using RQI. Based on the variation of R^2 with weighting factors for each pavement type a weighting factor of 14, corresponding to the overall highest R^2 -value for all pavement types, was selected for the DLI equation, which can then be written as:

$$DLI = \sqrt{V_1 + 14V_2} \quad (6)$$

with V_1 and V_2 as defined above.

ILLUSTRATIVE EXAMPLE

Figure 4 shows the wavelength ranges used for the RQI and DLI. It can be seen that there are gaps between the ranges of wavelengths used in calculating the RQI and DLI. These gaps help explain the possibility of obtaining an inflated RQI-value because of noise in the profile in the range of 0.6 to 1.8 m (2 to 6 ft) and 3.3 to 6.7 m (11 to 22 ft). On the other hand, if the profile contains high elevations at wavelengths greater than 15.1 m (50 ft), the RQI value will be deflated.

Figures 5 (a-1) and (b-1) show surface profiles of two 161-m (0.1-mile) rigid pavement sections that have the same RQI (equal to 65) but different DLC-values. Section #1 (WB US-10, CS18024, M.P. 7.0-7.1) has an "unfriendly" surface profile with a DLC-value of 11.3%. On the other hand, Section #2 (EB US-10, CS18024, M.P. 2.3-2.4) has a DLC-value of 6.6%, and therefore has a "friendly" surface profile. The corresponding axle load profiles (2nd axle load in 5-axle semi-trailer) are shown in Figures 5 (a-2) and (b-2). The power spectral density (PSD) curves of the dynamic axle load are shown in Figures 5 (a-3) and (b-3). The figures show that in Section #1, large axle loads occurred at frequencies between 1.5 and 4 Hz while there was no such amplification in Section #2. At frequencies between 8 and 15 Hz, both sections show small dynamic loading; while at all other frequencies, the dynamic axle load is negligible. The above clearly illustrates that dynamic truck axle loading is related to profile elevations having a wavelength between 6.7 and 17.9 m (22-59 ft) and between 1.8 and 3.3 m (6-11 ft). As stated above, these frequencies / wavelengths excite the truck body bounce and axle bounce, respectively.

In Figure 6, PSD curves of the two profiles are plotted together. The two wavelength ranges that excite the truck bounce are marked on Figure 6(a), while those used for the calculation of RQI are shown in Figure 6(b). Figure 6(a) shows that at the critical wavelength ranges, the PSD curve of Section #1 has much higher amplitude relative to Section #2. The areas under the profile PSD curve between wavelengths of 6.7 and 17.9 m (22 and 59 ft) are 3,006 mm² (4.66 in²) for Section #1 and 884 mm² (1.37 in²) for Section #2. Areas between wavelengths of 1.8 and 3.3 m (6 and 11 ft) are 136 mm² (0.211 in²) for Section #1 and 78 mm² (0.121 in²) for Section #2. These areas represent the amplitudes of the surface elevations for the critical wavelength ranges. The results indicate that Section #1 has a much larger area for wavelengths between 6.7 and 17.9 m (22 and 59 ft) than Section #2. The high amplitude of these waves for Section #1 excited the body bounce of the truck and led to high dynamic axle loads.

Figure 7 shows DLC values and areas under PSD curves for each wavelength range used in calculating the RQI for both sections. The figure shows that the profile of Section #1 contains high roughness at the high range of wavelengths (7.6-15.1 m, or 25-50 ft). On the other hand, the profile of Section #2 contains high roughness at the low range of wavelengths (0.6-1.5 m, or 2-5 ft). This range of wavelengths does not excite the truck; hence the RQI-value for Section #2 is inflated from the point of view of dynamic loading because of this noise. Note that Section #2 does not contain much roughness in the range of wavelengths between 7.6 and 15.1 m (25 and 50 ft), which does excite the truck. The roughness contents in the wavelength range of 0.6 to 1.5 m (2-5 ft) for Section #1 and in the wavelength range of 7.6-15.1 m (25-50 ft) for Section #2 explain why both sections have the same RQI-value.

VERIFICATION OF THE NEW DLI INDEX

To verify the relationship between the variances of the two filtered profiles (containing the critical wavelengths only) and dynamic load, twenty rigid pavement sections having the same RQI ($RQI = 65$) but different DLC-values were analysed. The profiles have DLC-values ranging from 6.56% (low to no dynamic loading) to 11.32 % (relatively high dynamic loading). This means that the RQI was not able to differentiate between cases of high versus low dynamic loading. On the other hand, based on the above discussion, the variance of the two filtered profiles should differentiate between high and low dynamic loading cases.

Using the filtered profiles, the DLI was calculated for each of the twenty sections using Equation (6). Figure 8 (a) shows the relationship between DLI and DLC, with $R^2 = 0.742$. The curve for DLI versus 95th percentile dynamic load is shown in Figure 8 (b), with $R^2 = 0.839$. A linear equation was used to fit the data. The high R^2 values show that the DLI is able to differentiate between cases of high versus low dynamic loads for these twenty sections. However, when the same analysis was done using the International Roughness Index (IRI), very low R^2 -values were obtained between DLC or 95th percentile load and IRI, as shown in Figure 8 (c) and (d), respectively.

The same analysis described above was done at different RQI-levels (RQI ranging from 35 to 90) and for all pavement types. The analysis used the same 333 pavement sections. The results are shown in Figures 9 through 11. Figure 9 shows the DLI-DLC plots at constant RQI-values for rigid pavements. The figure shows good correlations for most RQI-levels except for lower RQI-values ($RQI=35$ and $RQI=45$). This RQI-level represents a relatively new pavement, with very low DLC-values (DLC less than 8). The pavement surface is essentially smooth, and both RQI and DLI are able to characterize that. Therefore this difference is of no real consequence. Figure 10 shows the DLI-DLC plots at constant RQI-values for composite pavements. Again, good correlations exist for higher RQI-levels. Figure 12 shows the DLI-DLC plots at constant RQI-values for flexible pavements. The figure shows lower R^2 -values, indicating higher variability in flexible pavement surfaces.

Finally, Figure 12 shows the DLI-RQI plots for rigid, composite and flexible pavements. The figure clearly shows that for a given RQI-value the DLI can cover a wide range of values, confirming the results detailed above. As expected, the variation in DLI-values is minimal at low RQI-values. Also, the DLI-RQI relationship is similar to that between DLC and RQI confirming that DLI is representative of truck dynamic loading.

CONCLUSION

In conclusion, it can be stated that the DLI is a good indicator of dynamic truck-axle loads. The DLI was tested on a range of road profiles from in-service pavements, and it was found that for any particular value of Ride Quality Index (RQI), the DLI can cover a wide range of values, and this variation in DLI was found to correlate very well with dynamic load, as predicted by a truck simulation program. This was not the case for the International Roughness Index (IRI), which gave a low coefficient of correlation with dynamic load for the same range of profiles. The DLI can therefore differentiate between profiles that generate high dynamic loads and those having the same RQI but generating low dynamic loads. Most importantly, the use of DLI negates the need for running a truck simulation program. This makes it possible for a highway agency to decide whether a particular pavement section with a given surface profile needs smoothing or not based on the DLI-value. Thus the DLI can be used as a project-level roughness index for deciding whether or not to smooth a pavement based on dynamic load considerations.

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TABLES & FIGURES

Table 1- Truck Matrix Sizes and Weights




Truck Configuration	Configuration Name	GCVW (kN)	Axle Loads (kN)	Wheel Base (m)
	2 Axle Truck	125	49/76	4.3
	3 Axle Truck	150	56/94	6.1
	5 Axle Semi-Trailer	356	54/151/151	3.6/11.0

Table 2- Vertical Suspension Properties

Suspension Location	Suspension Type	Upper Envelope Stiffness (kN/m)	Lower Envelope Stiffness (kN/m)	β^1	Linear Damping Coefficient (kN-s/m)	Unsprung Weight (kN)
Steer Axle	Flat Leaf	28.5	28.5	0.04	0.50	6.2
Single Drive Axle	Flat Leaf	47.8	41.3	0.08	0.02	11.1
Tandem Drive Axle	Flat Leaf	47.8	41.3	0.08	0.02	22.7
Tandem Semi-trailer Axle	Flat Leaf	47.8	41.3	0.08	0.02	16.9

¹ β = Decay Constant

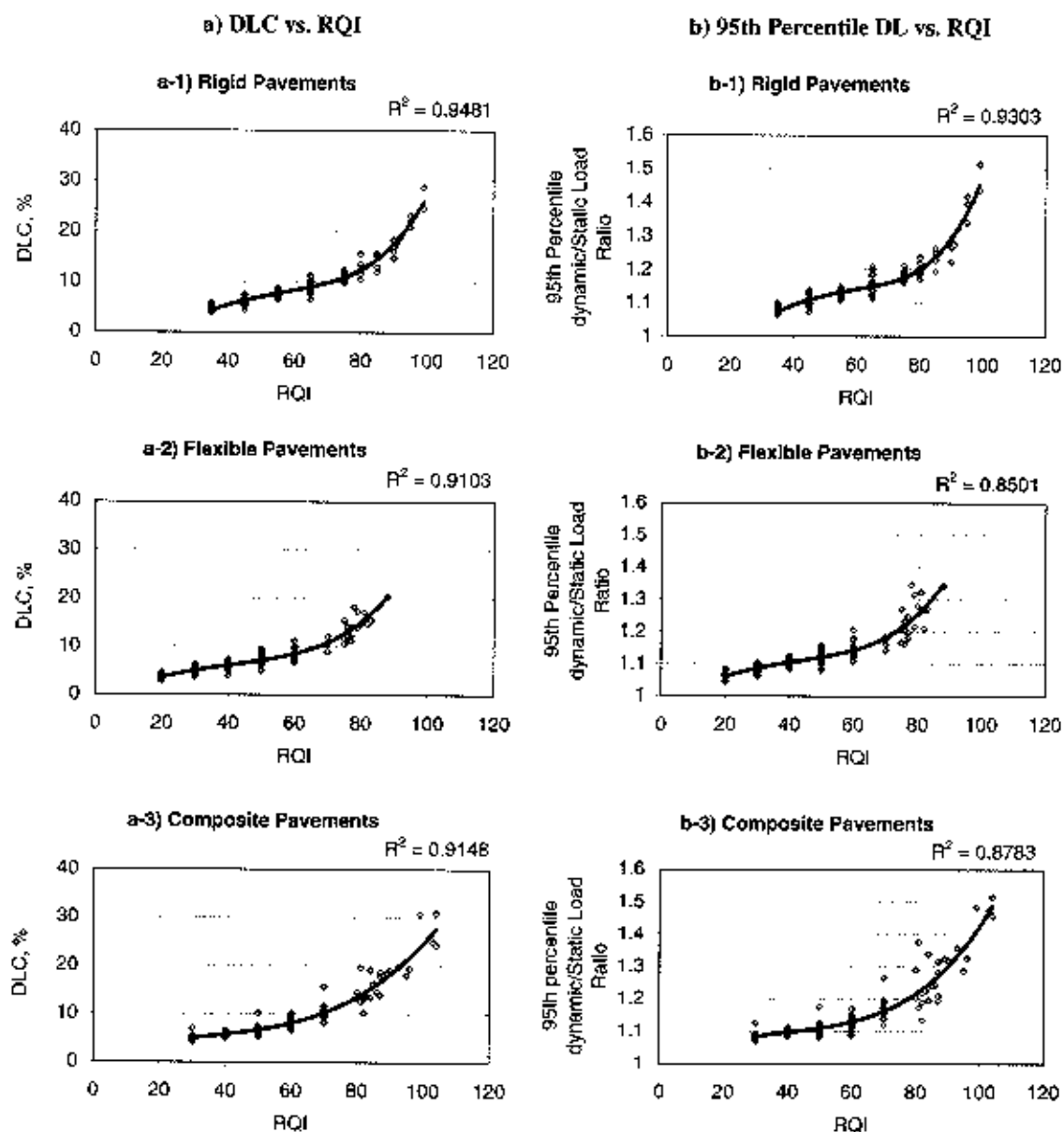


Figure 1 - Dynamic Load versus RQI Curve for Each Type of Pavements

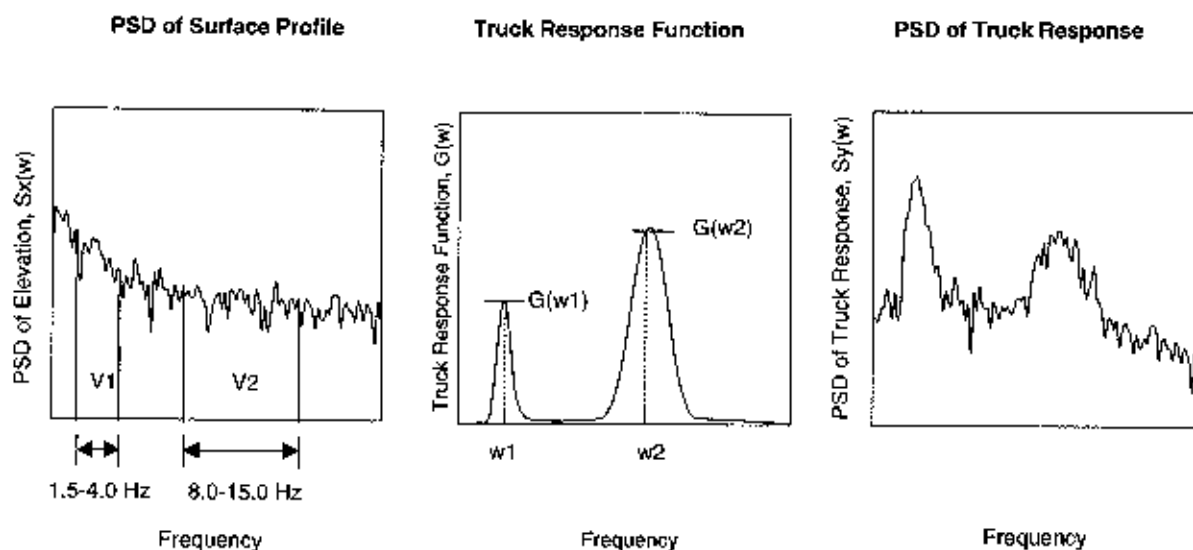


Figure 2 - Schematic Figures of PSD of Surface Profile, Truck Response Function and PSD of Truck Response

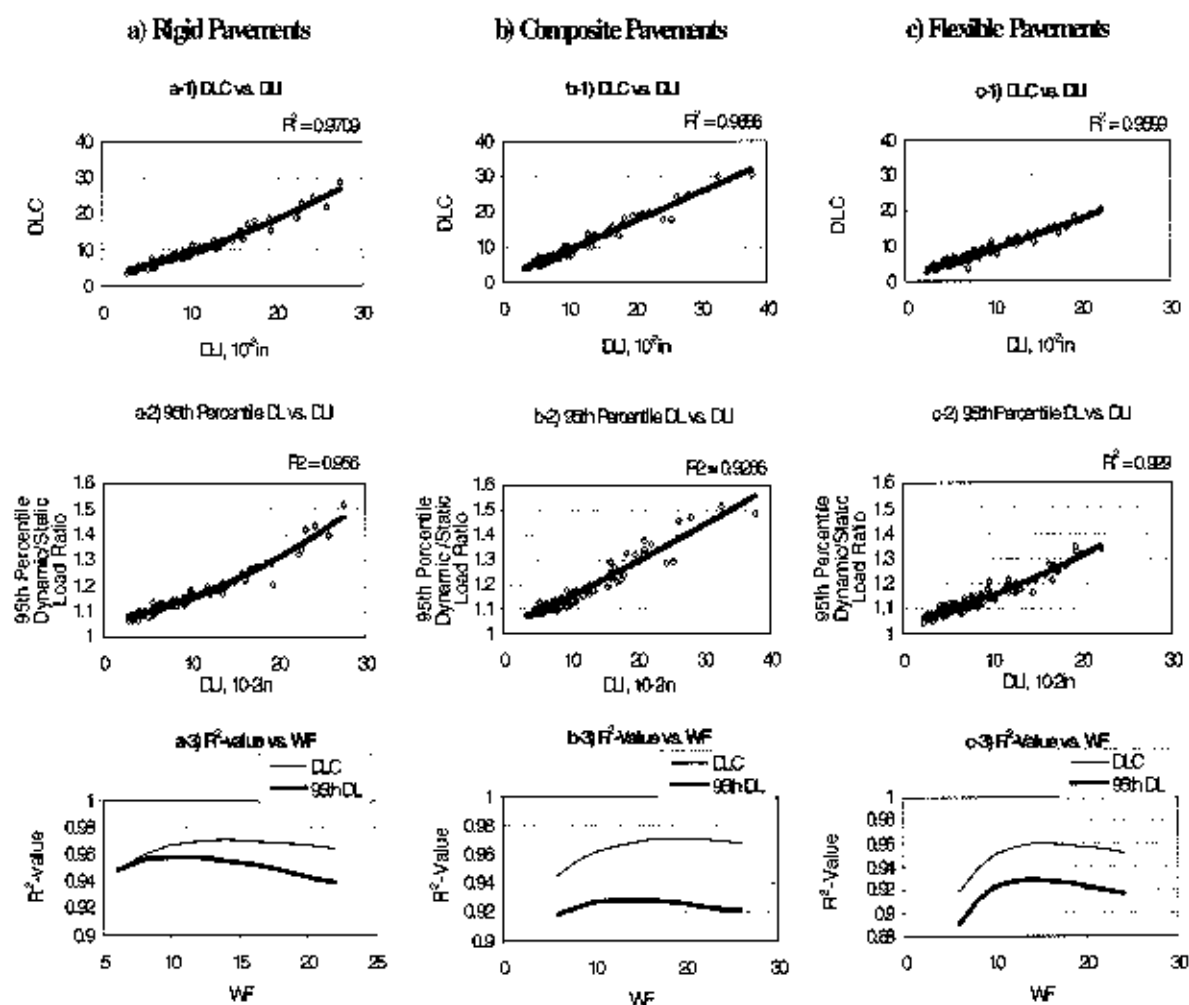


Figure 3 - Dynamic Load vs. DLI for Rigid, Composite and Flexible Pavements

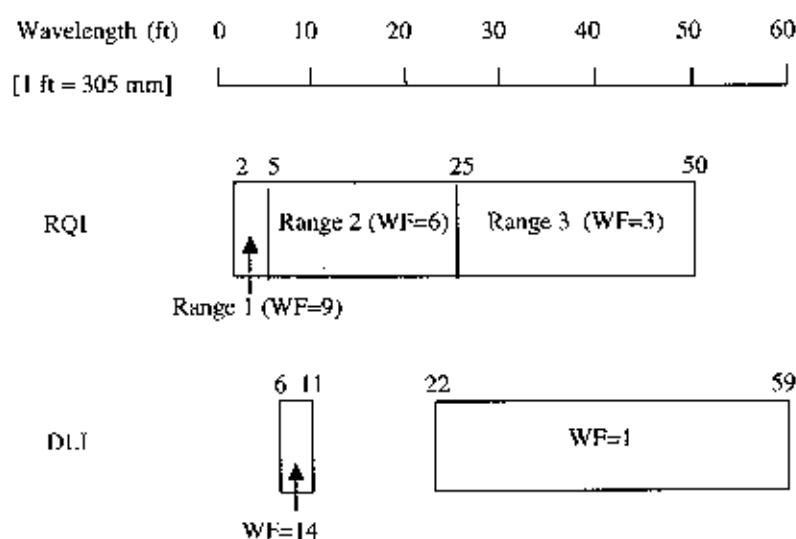


Figure 4 - Wavelength Ranges used for RQI and DLI

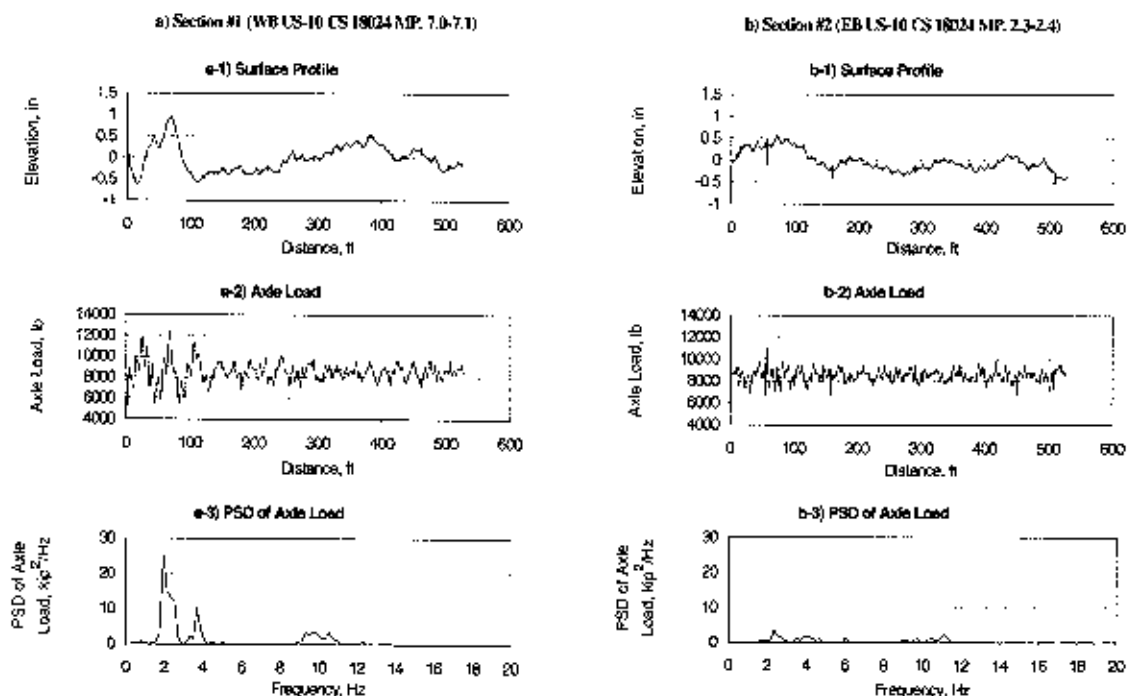


Figure 5 - Surface Profile and Axle Load for Section #1 (WB US-10) and Section #2 (EB US-10)

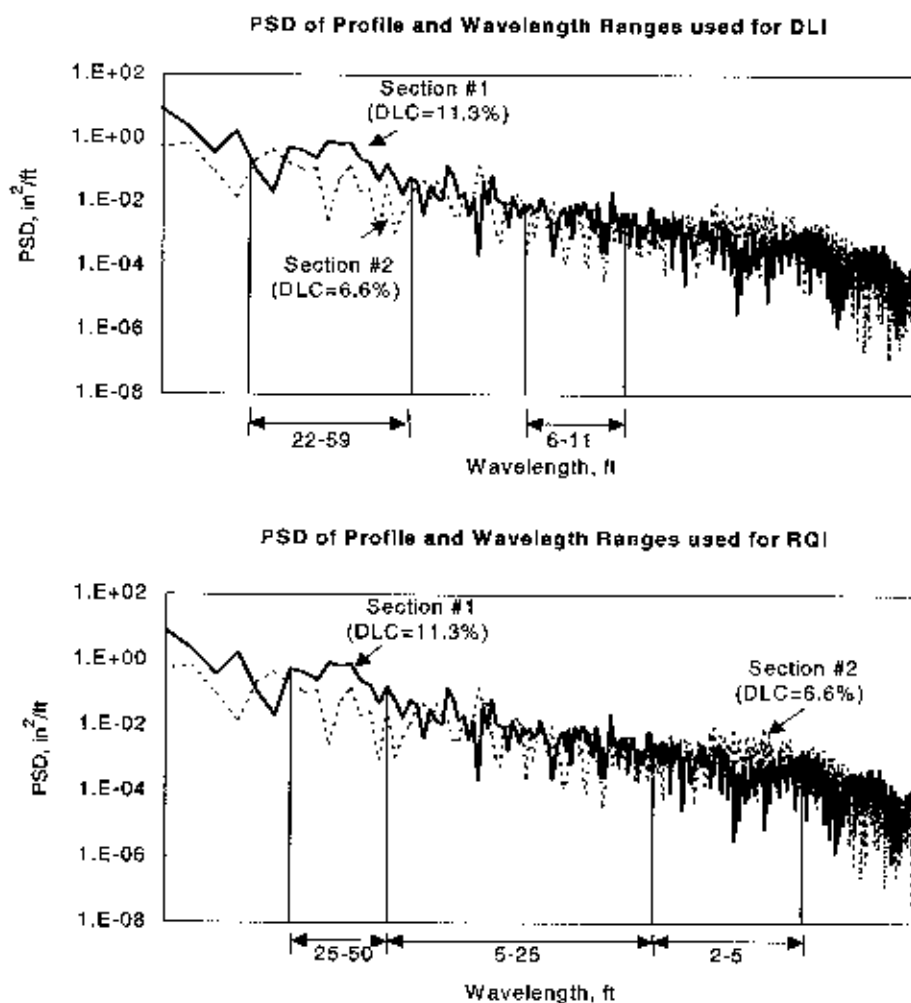


Figure 6 - PSD of Profiles for Sections #1 and #2 and Wavelength Ranges used for DLI and RQI

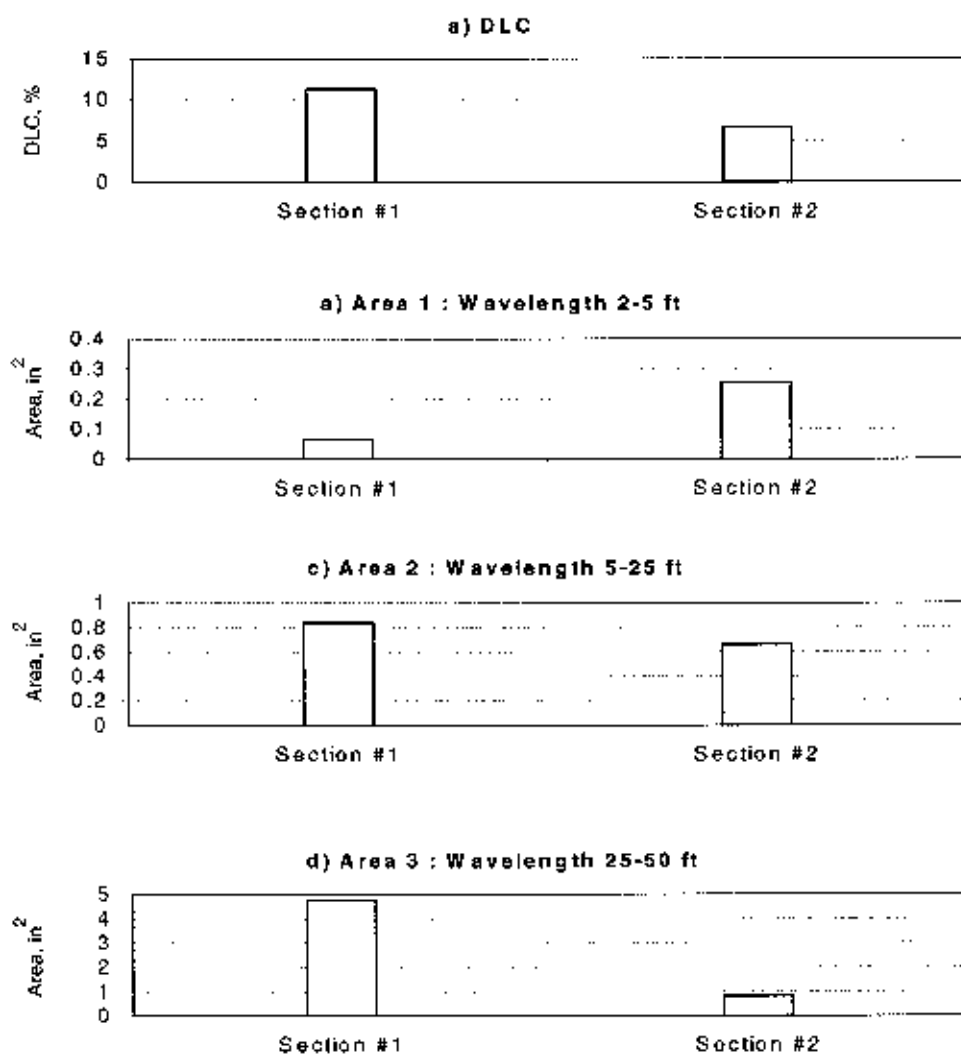


Figure 7 – RQI Wavelength Content and DLC for Example Sections 1 and 2

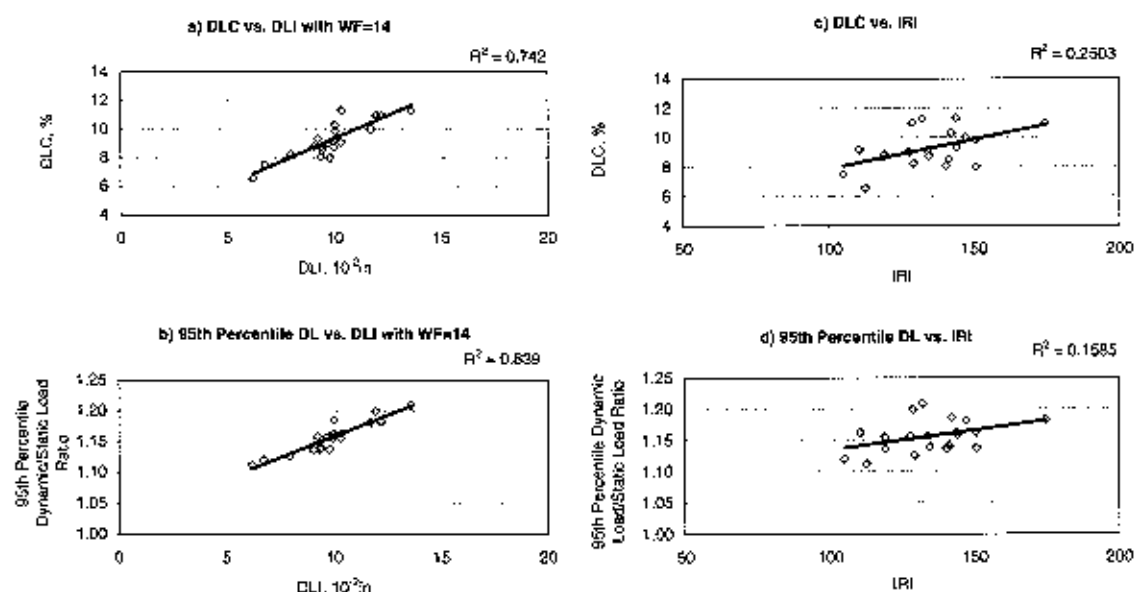


Figure 8 - Relationship between Roughness Indices (DLI and IRI) and Dynamic Load for 20 Sections

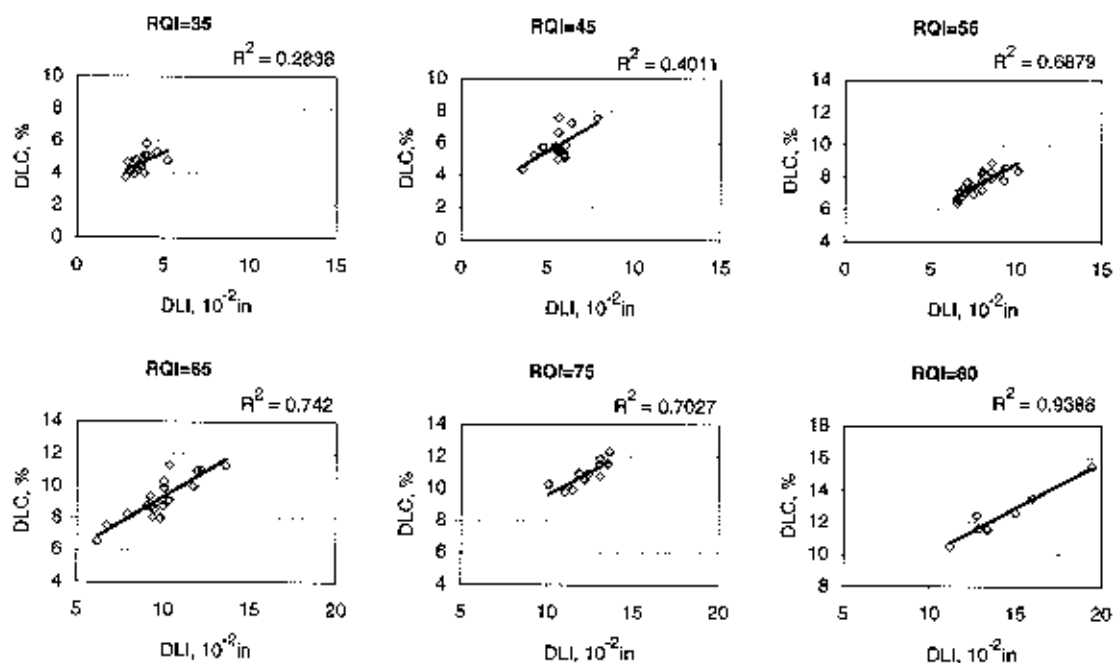


Figure 9 - DLI-DLC Plots at Constant RQI-Values for Rigid Pavements

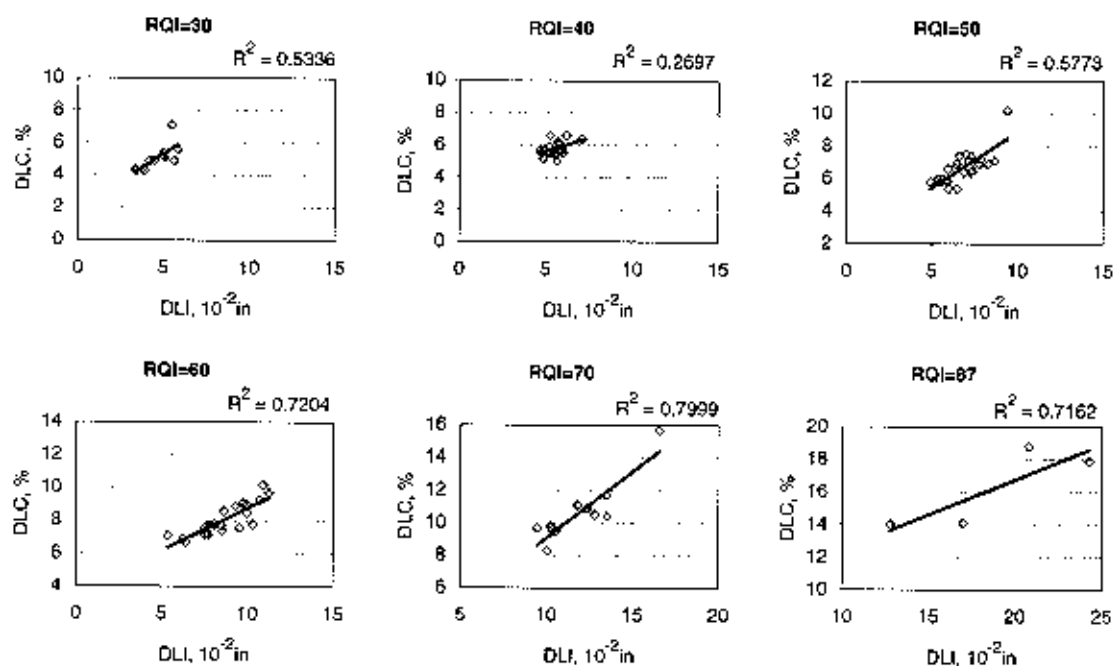


Figure 10 - DLI-DLC Plots at Constant RQI-Values for Composite Pavements

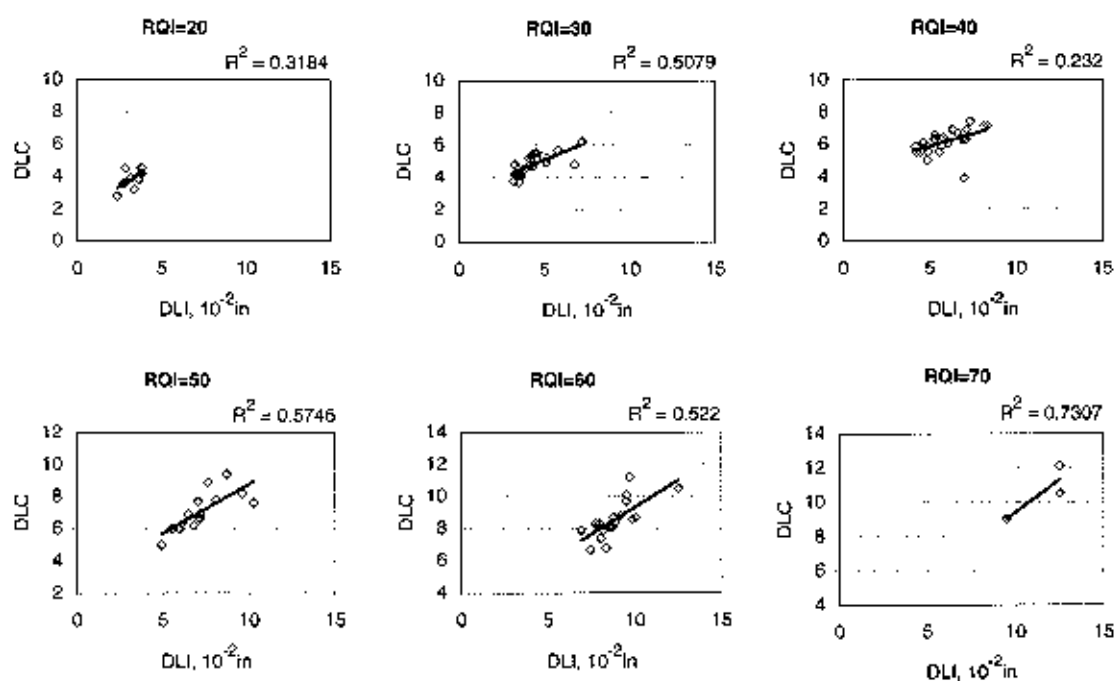


Figure 11 - DLI-DLC Plots at Constant RQI-Values for Flexible Pavements

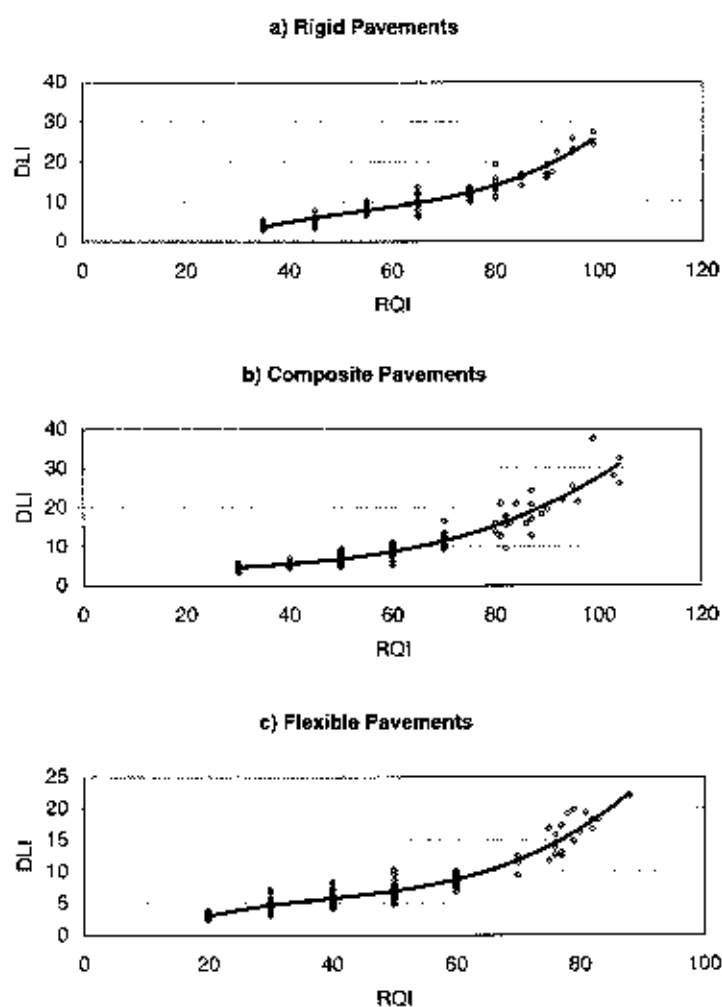


Figure 12 – Relationship between DLI with RQI for Rigid, Composite and Flexible Pavements

TRAFFIC CHARACTERISATION IN FLEXIBLE PAVEMENT DESIGN

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ABSTRACT

This paper investigates the way in which traffic data is used in flexible pavement design and analysis procedures. A deterministic Long Term Pavement Performance Model (LTPPM) is used to calculate flexible pavement damage caused by trafficking from three realistic fleets of commercial vehicles. The fleets have been modelled using seven axle group models representing steer axles, drive axles (single or tandem) and trailer axles (tandem or tridem) with either steel suspensions or air suspensions. Results are compared to the calculated flexible pavement damage caused by a fleet of 80kN standard axles and predictions from the currently used method of traffic characterisation for UK pavement design procedures. Results show that pavements that fail by rutting last longer when trafficked by the realistic vehicle fleets compared to the fleet of 80kN standard axles whereas the pavements that are predicted to fail by fatigue last longer when trafficked by the fleet of 80kN standard axles. Results also show that current UK pavement design procedures are more sensitive to differences in the vehicle fleet compared to LTPPM predictions.

INTRODUCTION

In current UK detailed pavement design procedures, the damage associated with different classes of commercial vehicles is characterised using Vehicle Wear Factors (VWFs). For a particular vehicle, j , the VWF, expressed in terms of an equivalent number of 80kN standard axles, is calculated using [1]:

$$VWF_j = \sum_{i=1}^{N_o} \left(\frac{P_i}{80} \right)^4 \quad (1)$$

where P_i is the static axle load (kN) on axle i , and N_o is the number of axles.

Average VWFs for the different classes of commercial vehicles (based on axle load measurements determined from static weighbridges or, more recently, Weigh-in-Motion (WIM) systems [2]) are then used to calculate the design traffic using the following equation [1]:

$$T = 365 \times 10^{-6} Y \sum_{j=1}^N F_j \times G_j \times VWF_j \times Q_j \quad (2)$$

where T is the design traffic (million standard axles) Y is the design period (years), F_j is the present traffic flow (commercial vehicles per day) for vehicle class j , G_j is the annual growth factor for vehicle class j , Q_j is the proportion of vehicle class j using the left hand lane, and N is the number of vehicle classes.

Typically, the design traffic T is used in conjunction with an empirical design chart to determine the pavement construction that is required to carry the design traffic. It can be seen from Equation (1) that the damage associated with a particular vehicle is calculated using the static axle load raised to a fourth power. The power of four originates from full scale tests carried out in the late 1950s by the American Association of State Highway Officials (AASHO) [3]. Using a regression analysis it was found that the decrease in 'pavement serviceability' caused by a heavy vehicle axle could be related to its static load raised to a fourth power. Serviceability was rated by a panel of experts and expressed in terms of a 'present serviceability index' (PSI) ranging from 0 to 5 (perfect road). It was found that PSI could be correlated with cracking and patching, rutting, and surface roughness.

However, this approach has been found to be subject to a number of limitations [4], namely:

- The sensitivity of pavement damage to vehicle speed and the frequency of applied loads are not included;
- It does not account for the effects of dynamic tyre forces or tyre configuration;
- It does not account correctly for the mode of road failure (ie rutting, fatigue etc).

The validity of the 4th power law is also questionable because current axle group configurations, tyre sizes and pressures, pavement constructions, and traffic volumes are significantly different from the conditions of the AASHO road test.

In more analytical design procedures, the main difference is that a computer model of the pavement structure is used to calculate the response of the pavement to a standard 40kN load applied through dual tyres (or a single tyre), rather than using a design chart. Both of these procedures rely on VWFs for determination of the design traffic. Consequently, it can be seen that in both of these approaches, the effect of truck factors (eg axle spacing, load sharing, suspension type), tyre factors (eg contact area, contact pressure, tyre type etc) and pavement factors (eg surface roughness, variations in asphalt layer thickness) on pavement design cannot easily be incorporated. In the case of empirical design, these factors are implicitly included in the design charts for the vehicle fleets in operation when the empirical data was collected. However, subsequent changes in the vehicle fleet (eg super-single tyres, road-friendly suspensions) and associated changes in pavement performance cannot be predicted using empirical design charts.

The method to be investigated in this paper differs fundamentally from the standard VWF approach which is based on aggregating traffic. Realistic pavement damage models are used, and damage caused by different axle groups in the fleet are calculated and accumulated as the pavement is progressively trafficked. This can be thought of as an aggregate damage approach rather than an aggregate traffic approach. The model used is discussed in the following sections. The research described in this paper forms part of the Highways Agency's (HA's) research programme.

LONG-TERM PAVEMENT PERFORMANCE MODEL (LTPPM)

The model used to calculate long-term pavement performance in the presence of traffic and environmental loads. Areas of the model relating to the calculation of dynamic axle loads are described in the following section. Other details of the model and assumptions can be found in [5,6].

Referring to Figure 1, the initial inputs to the LTPPM are:

- The specification of the pavement being simulated (i.e. layer thicknesses, mixture specifications etc.);
- the time increment to be used in the simulation;
- The rate of traffic loading;
- The climatic conditions.

From this initial specification, the simulation process can be divided into the following steps.

1. A length of pavement surface profile is generated and divided into many equally spaced sub-sections of length 0.5m.
2. A time domain vehicle (or axle group) simulation is used to generate dynamic tyre forces for one vehicle (or more) as a function of distance along the pavement. The vehicle (or axle group) model parameters are chosen to best represent the traffic conditions for the type of pavement being simulated.
3. A set of primary response 'influence functions' is generated, for each pavement sub-section and each mode of damage. The modes of damage that are included in the LTPPM are permanent deformation (rutting) in both the asphaltic and lower pavement layers and fatigue damage to the asphaltic layers.
4. These primary response influence functions are combined with the dynamic tyre forces, to give primary pavement response time histories at a large number of equally spaced discrete points along the pavement.

5. The primary responses are combined with the appropriate pavement damage models and the number of load applications, to predict damage (rutting and fatigue damage) as a function of distance along the pavement for the current time increment.
6. An updated surface profile is then generated by subtracting the calculated rutting in the wheel path from the initial profile used for that time increment. This mechanism accounts for the effects of changing surface roughness on dynamic tyre forces.
7. The calculated fatigue damage is used to reduce the elastic modulus of the asphaltic material for each subsection. This mechanism reflects the effects of cumulative fatigue damage on the primary responses and hence subsequent pavement damage.
8. The above process is then repeated for the next time increment, and so on, until the pavement has reached the end of its serviceable life.

2.1. Axle Group Models

Due to the relatively large amount of computer time required to run the LTPPM [6] it was decided to represent a typical UK commercial vehicle fleet using a number of 2-dimensional axle group models rather than whole vehicle models. This assumes that the effect on dynamic tyre forces caused by interactions between axle groups is small compared to the dynamic tyre forces generated by the axle group itself. In total, seven models were chosen, to represent the axle groups found on a typical fleet of commercial vehicles (see Table 1). The steel spring suspension elements were based on a validated model developed by Fancher et. al. [7]. The parameters used in the models are largely based on results from validated articulated vehicle simulations developed by Cole and Cebon [8].

Single axle steel/air suspension

This axle group model was used to represent a tractor steer axle with steel suspension (STER) and a tractor drive axle with steel or air suspension (SINS, SINA). For all the axle group suspensions (except the steer axle) the first three letters denote the number of axles (ie SIN=single axle, DUA=dual axle, TRI=tri-axle) and the last letter indicates whether the suspension is steel sprung (S) or air sprung (A). The model is shown schematically in Figure 2. It can be seen from this Figure that this model has 2 degrees of freedom (Z_{u1} and Z_s). M_{s1} represents the sprung mass and M_{u1} represents the unsprung mass (axle mass etc). The force versus displacement characteristics of the suspension element joining the sprung and unsprung masses and the tyre element joining the unsprung mass to the pavement surface profile are shown in Figures 3(a) and 3(b) respectively [7].

It can be seen from Figure 3(a) that 6 parameters are required to characterise the force versus deflection characteristics for the suspension. k_u and k_l define the upper and lower stiffness envelopes for the model, β_u and β_l define the form of the exponential function used to join the envelopes, and F_f and C_r define the level of friction and hydraulic damping in the suspension. The values of these parameters for a typical air and steel suspension are given in Table 2. It can be seen from Table 2 that, for the drive axle suspensions, the suspension stiffness is lower for the air suspension; the air suspension has a higher level of hydraulic suspension damping; and there is no friction. The values of tyre stiffness and damping are typical for dual tyres [8]. The steer axle suspension has lower values of sprung and unsprung masses, lower suspension stiffness and a higher level of hydraulic suspension damping. The values of tyre stiffness and damping are typical for single tyres [8].

Tandem axle steel/air suspension

This axle group model was used to represent either a tractor drive axle with steel or air suspension or a trailer axle with steel or air suspension (DUAS, DUAA). The model is shown schematically in Figure 4.

It can be seen from this figure that this model has 6 degrees of freedom (Z_{u1} , Z_{u2} , Z_s , θ_{u1} , θ_{u2} and θ_s). As before M_{s1} represents the sprung mass and M_{u1} and M_{u2} represent the unsprung masses (axle mass etc). The force versus displacement characteristics of the suspension element joining the sprung and unsprung masses and the tyre element joining the unsprung mass to the pavement surface profile are shown in Figures 3(a) and 3(b). It can be seen from Figure 4 that the main difference between this model and the single axle model is the addition of a levelling beam between connecting the axles [9]. The force versus deflection characteristics of the levelling beam are shown in Figure 3(c). Values of the parameters used in the model are given in Table 2. It can be seen from

this table that, as before, the suspension stiffness is lower for the air suspension, the air suspension has a higher level of hydraulic suspension damping and there is no friction. The values of tyre stiffness and damping are typical for dual tyres [8]. It can also be seen from Table 2 that the levelling beam stiffness and damping are large for the air suspension compared to the steel suspension. This is because the axle of an air suspension work essentially independently whereas the axles on a steel suspension are typically coupled and do not behave independently of each other.

Tri-axle steel/air suspension

This axle group model was used to represent a trailer axle with steel or air suspension (TRIS, TRIA). The model is similar to the tandem axle suspension (Figure 4) with the addition of an additional axle station and is not shown for brevity.

This model has 9 degrees of freedom and, as before, M_{s1} represents the sprung mass and M_{u1} , M_{u2} and M_{u3} represent the unsprung masses (axle mass etc). The force versus displacement characteristics of the suspension element joining the sprung and unsprung masses and the tyre element joining the unsprung mass to the pavement surface profile are shown in Figures 3(a) and 3(b). The force versus deflection characteristics of the levelling beam are shown in Figure 3(c). Values of the parameters used in the model are given in Table 2.

Relative Axle Group Proportions

Detailed data from WIM measurements [10] were used to determine the relative proportions of the different commercial vehicles (and hence axle groups) for 3 classes of road (Motorway, Trunk Road, Principal Road). These data, in conjunction with results from Potter et. al. [11] were used to estimate the relative proportions of steel and air suspensions for each axle group model (see Table 3). The resulting axle group proportions for the 3 pavement classes (Motorway, Trunk Road, Principal Road) are denoted Fleet A, Fleet B and Fleet C in the rest of this paper.

Also shown in Table 3 are the *VWFs* for each of the axle group models calculated using Equation (1). In addition, a Fleet Vehicle Wear Factor (*FVWF*) has been calculated using the following equation:

$$FVWF = \sum_{i=1}^{N_g} VWF_i \times p_i \quad (3)$$

where VWF_i is the Vehicle Wear Factor for axle group i , N_g is the number of axle groups, and p_i is the proportion of axle group i .

The *FVWF* can be interpreted as the number of passes of a standard 80kN axle required to cause the same damage as one pass of the vehicle fleet according to the 4th power law. For example, if there were 38 passes of STER, 26 passes of SINS, 4 passes of SINA, 14 passes of DUAS, 4 passes of DUAA, 7 passes of TRIS and, 7 passes of TRIA this would be equivalent to approximately 185 passes of an 80kN standard axle.

To ensure that the dynamic force levels generated by the axle group models were realistic, simulations were performed where each axle was run over a typical Motorway pavement surface at 22m/s (50mph) and the Dynamic Load Coefficients (DLCs) were calculated for each axle. It was found that the DLCs were realistic and, as expected, the values for the steel sprung suspensions were greater than the values for the air sprung suspensions. For example, the DLCs for the tandem axle air suspension were found to be 5.5% and 5.1% compared to 10.3% and 10.2% for the tandem axle steel suspension.

LTPPM SIMULATIONS

LTPPM Input Parameters

The long-term response of the three classes of flexible pavement trafficked by four vehicle fleets (A, B, C and an 80kN standard fleet) have been calculated for typical UK climatic conditions using the LTPPM. It should be noted that Vehicle Fleets A, B and C were determined using traffic data measured from a Motorway, Trunk Road and

Principal Road respectively. In the LTPPM simulations these vehicle fleets have been applied to all three pavement classes (Motorway, Trunk Road and Principal Road) to investigate the sensitivity of long-term performance for each pavement class to changes in the vehicle fleet. Each of the pavements has been trafficked at the same rate (in terms of applied axle loads per month) for ease of comparison of the results and the speed of the axle group models were chosen to represent the average speed of heavy goods vehicles measured using WIM for the particular class of road (24.4m/s for a typical Motorway, 22.2m/s for a typical Trunk Road and 16.1m/s for a typical Principal Road). The only geometrical difference between the different classes is the thickness of the asphalt layer. This was taken to be 350mm for the Motorway, 250mm for the Trunk Road and 150mm for the Principal Road. The length of each simulated pavement section was 100m. A typical 50 pen Hot Rolled Asphalt (HRA) was used and the subgrade was assumed to have an elastic stiffness modulus of 40MPa. For a Motorway, an initial IRI roughness of 2 was used and for a Trunk Road and Principal Road initial IRI roughnesses of 3 were used. The variation in mean monthly air temperature for each type was assumed to be sinusoidal, with a mean temperature of 11°C and a temperature amplitude of 7°C. These are typical of UK climatic conditions [23]. The pavement surface profile was updated every 3 months, and the asphalt layer modulus was degraded in 10% steps to achieve a satisfactory trade-off between convergence of results and computational speed [6]. Two different exponents (1 and 4) were used in the lower layer rutting model to investigate the influence of the sensitivity of lower layer rutting to the level of dynamic load (see [5, 6] for further details).

For each of the pavements the performance for each of the Vehicle Fleets A, B and C has been quantified in relation to the pavement performance for the fleet of 80kN standard axle loads by use of a Pavement Life Reduction Factor (*PLRF*) defined as:

$$PLRF = \left(1 - \frac{N}{N_{std}} \right) \times 100\% \quad (4)$$

where N is the number of load passes required to produce failure conditions (defined as a 20mm rut or a cumulative level of fatigue damage equal to 1), N_{std} is the number of load applications of the fleet of standard 80kN axles required to produce failure conditions.

It can be seen from Equation (4) that the *PLRF* is a measure of the reduction in life of the pavement caused by trafficking by a particular vehicle fleet compared to the same pavement trafficked by a fleet of 80kN standard axles. For example, a *PLRF* of 50% would indicate that the life of the pavement trafficked by the realistic vehicle fleet is half of the life of the same pavement trafficked by a fleet of 80kN standard axles. However, it should be noted that the *PLRF* compares pavement damage in terms of the number of load passes required to achieve a certain level of pavement damage. Consequently, if the vehicle fleet has a different average static load compared to the standard fleet, this will be reflected in the *PLRF* as well as other differences between the vehicle fleet and the standard fleet that influence pavement damage such as dynamic loading and tyre type.

In addition to LTPPM simulations using axle group models operating under fully laden conditions, simulations have also been performed for axle group models operating under one-third and two-thirds of fully laden conditions (for Vehicle Fleet A) to enable *PLRFs* to be predicted for realistic axle load distributions which then can be directly compared with predictions from a standard traffic classification approach using *VWFs*.

Simulation Results (Fully Laden Vehicle Fleets)

A typical example of the output from the LTPPM is shown in Figure 5 where the pavement surface displacement profile at various loading stages is plotted as a function of distance along the pavement for the Motorway trafficked by Vehicle Fleet A using a subgrade rutting exponent of 1. It can be seen from this figure that the degradation in surface profile is relatively uniform, and the frequency content of the initial profile is largely preserved. This is because almost all of the rutting occurs in the upper bituminous material which is most sensitive to the level of static loading on the axle and relatively insensitive to the level of dynamic loading [22].

Figures 6 and 7 show the accumulation of average rut depth and 95th percentile fatigue damage as a function of the number of individual axle load passes for Vehicle Fleet A trafficking the Motorway. The 95th percentile level of fatigue damage has been used to reflect the more localised nature of fatigue damage, where pavement failure can occur when a relatively small proportion of the wheeltrack is extensively cracked [5]. It can be seen from Figures 6 and 7 that, as expected both forms of damage increase as the pavement is progressively trafficked. It can also be seen that the curves are not smooth. This is caused by seasonal variations in air temperature affecting the rate of damage accumulation. For example, rutting in the bituminous material occurs most rapidly in the Summer when the temperature is highest.

For the fleet of 80kN standard axles the average rut depth and the 95th percentile fatigue damage caused by static loads alone has also been plotted in Figures 6 and 7 respectively (dashed lines). The LTPPM simulation for this vehicle fleet did not include asphalt layer modulus degradation caused by cumulative fatigue damage. Therefore, the results for this fleet can be considered to be the "standard" case for comparison of the results with the more realistic vehicle fleets.

It can be seen from Figure 6 that, for the standard fleet, the average rut depth reaches failure conditions (defined as a 20mm rut depth) after approximately 47.5 million load applications. The corresponding life to failure for Vehicle Fleet A is approximately 32.3 million load applications giving a *PLRF* of approximately 32% (see Table 4). It can be seen from Figure 7 that the level of cumulative fatigue damage in this class of pavement is small (0.035 at 32.3 million load applications). This is consistent with recent observations from thick "long-life" flexible pavements where the long-term surface deflection was found to decrease [16] indicating little or no asphalt modulus degradation caused by cumulative fatigue damage. This will significantly increase with thinner pavement constructions where more fatigue damage is likely to occur. For this type of thick pavement where the failure mechanism is permanent deformation in the upper bituminous layers which is relatively insensitive to dynamic load [5], the *PLRF* primarily reflects differences in the average static load applied to the pavement by the 2 vehicle fleets. For example, for Vehicle Fleet A the rutting caused by static loads alone after 35 million load applications was 22.5mm which is almost identical to the average rut depth shown in Figure 6 at the same number of load passes.

Figures 8 and 9 show the corresponding results for the Principal Road trafficked by Vehicle Fleet A and the fleet of standard 80kN axles using a subgrade rutting exponent of 4. It can be seen from these figures that, for this type of pavement, the mode of failure is fatigue damage rather than rutting. This is caused by the pattern of dynamic loading applied by Vehicle Fleet A to the pavement surface causing an accumulation of fatigue damage in particular areas, thus reducing the effective stiffness of the asphaltic material in these areas which, in turn, leads to a more rapid accumulation of damage and, ultimately, failure. The *PLRF* for this case is approximately 89% (see Table 4). This is significantly greater than the corresponding value for the Motorway (32%) because this form of damage is more sensitive to the level of dynamic loading than permanent deformation in the upper bituminous layers which is relatively insensitive to dynamic loading [5]. In addition, this form of damage accelerates significantly as failure is approached owing to the modulus feedback mechanisms in the LTPPM methodology (see Figure 1).

It should also be noted from Figure 8 that the rate of rutting increases rapidly after the 95th percentile cumulative fatigue damage has reached failure. This is because as the level of fatigue damage approaches failure at certain locations on the pavement, the stiffness of the asphaltic material in these locations has decreased to 20% of its initial value and the vertical compressive strain transmitted to the subgrade has increased dramatically, thus increasing the rate of lower layer rutting. This is often observed in practice where extensive subgrade rutting can quickly follow after cracking in the bound bituminous material.

Results for the other combinations of pavement class and vehicle fleet are summarised (in terms of the resulting *PLRFs*) in Table 4. It can be seen from this table that, for all the vehicle fleets and both values of subgrade rutting exponent, the mode of failure for the Motorway is rutting, but fatigue for the Principal Road. The mode of failure for the Trunk Road was rutting for the 80kN fleet of standard axles, but fatigue for Vehicle Fleets A, B and C. It

can also be seen from this Table 4 that, for all 3 vehicle fleets, the *PLRFs* are relatively insensitive to the value of the subgrade rutting exponent and they are similar for all 3 fleets of vehicles.

Simulation Results (1/3 and 2/3 Laden Vehicle Fleets)

In addition to the results presented above, LTPPM simulations have also been performed for each of the three pavement classes trafficked by Vehicle Fleet A where the axle group models are nominally 1/3 laden and 2/3 laden (using a subgrade rutting exponent of 1). This was achieved by adjusting the sprung masses of the axle group models (see Table 2).

LTPPM simulations were performed with Vehicle Fleet A 1/3 and 2/3 laden at the same trafficking rate as in the fully laden case. It can be seen from Table 2 that the axle model representing the steer axle was not adjusted. This is because the steering axle load is not strongly dependent on whether the vehicle is fully laden or empty. Other combinations of vehicle fleet and subgrade rutting exponent were not considered because it was previously shown that the results (in terms of the *PLRF*) are relatively insensitive to these variables (see Table 4). In the cases where the pavement trafficked by the 1/3 or 2/3 laden fleet did not reach failure conditions after 20 years (ie it lasted longer than the same pavement trafficked by the fleet of 80kN standard axles) linear extrapolation was used to calculate the *PLRF*.

The results, in terms of *PLRFs*, are shown in Table 5. It can be seen by comparing the results with those in Table 4 that, for the Motorway, the 1/3 laden condition results in a *PLRF* of -10%. This means that under these loading conditions the life of the pavement is extended compared to the same pavement trafficked by the fleet of 80kN standard axles. The corresponding *PLRF* for the 2/3 laden condition is +16%. The situation for the Trunk Road is similar except that the extremes are greater. It should however be noted that the Trunk Road was predicted to fail by rutting for the 1/3 and 2/3 laden conditions whereas it was predicted to fail by fatigue for the fully laden condition. It can be seen from Table 9 that the situation for the Principal Road is different. In this case all three loading conditions result in positive *PLRFs* ranging from 76% to 89%. The reason for this is that all these pavements fail by fatigue which, as can be seen from Figure 9 occurs extremely rapidly because of modulus degradation. Consequently, although the static axle loads are reduced the peak loads (dynamic plus static) still cause relatively rapid pavement failure.

Realistic Axle Load Variations

The *PLRFs* calculated above are for cases where the fleet is assumed to be either one-third laden, two-thirds laden or fully laden. In reality each axle group in the vehicle fleet will contain a distribution of axle loads between unladen and fully laden. Consequently, the *PLRFs* calculated above have to be adjusted to take into account realistic axle load variations. To achieve this adjustment the measured axle load distribution for a typical UK motorway [3] was divided into three weight bands (1-4 tonnes, 4-7 tonnes and 7-10 tonnes). The proportions of axles in each weight band were estimated to be 0.52, 0.37 and 0.11 respectively. Using the data in Tables 3 and 5 the average axle weight was calculated for Vehicle Fleet A for the 1/3 laden, 2/3 laden and fully laden cases (4.5 tonnes, 6.4 tonnes and 8.4 tonnes respectively). Equivalent *PLRFs* were then calculated for each pavement class (Motorway, Trunk Road and Principal Road) corresponding to the mid-points of the 3 weight bands (2.5 tonnes, 5.5 tonnes and 8.5 tonnes) using cubic spline interpolation/extrapolation. These results were then combined with the relative proportions of axles in each weight band to estimate the overall *PLRFs* for each pavement class which are shown in Table 6.

It can be seen from Table 6 that the *PLRFs* predicted from the LTPPM are negative in all but one case (LTPPM, Principal Road) indicating that, as before, the realistic vehicle fleets are causing less pavement damage than the fleet of 80kN standard axles. It can also be seen from Table 6 that the *PLRFs* calculated from the LTPPM indicate an increase in pavement life of approximately 20% for the Motorway and Trunk Road trafficked by the realistic vehicle fleets compared to the fleet of 80kN standard axles. For the Principal Road the LTPPM predicts a decrease in pavement life of approximately 80% caused by trafficking by the realistic vehicle fleets.

COMPARISON WITH STANDARD VWF APPROACH

It can be seen from the above that the *PLRF* has been chosen as the performance measure for the different combinations of vehicle fleet and pavement structure. However, since the standard approach is in terms of *VWFs* a method is required to transform these into equivalent *PLRFs*.

For a single pass of a particular vehicle, the *VWF* can be considered to be the equivalent number of 80kN standard axles required to cause the same level of pavement damage. Consequently, assuming it takes x vehicle passes (and hence xN_a axle passes, where N_a is the number of axles per vehicle) to reach critical damage conditions it will take $xVWF$ passes of a standard 80kN axle load to reach the same damage conditions. Using Equation (4) an equivalent *PLRF* can be calculated to be:

$$PLRF = \left(1 - \frac{N_a}{VWF} \right) \times 100\% \quad (5)$$

For a vehicle fleet comprising a number of different vehicles the same approach can be used and Equation (5) can be generalised to give:

$$PLRF = \left(1 - \frac{\sum_{i=1}^{N_g} N'_a \times p_i}{\sum_{i=1}^{N_g} VWF_i \times p_i} \right) \times 100\% = \left(1 - \frac{\sum_{i=1}^{N_g} N'_a \times p_i}{FVWF} \right) \times 100\% \quad (6)$$

where N'_a is the number of axles in axle group (or vehicle) i , VWF_i is the Vehicle Wear Factor for axle group (or vehicle) i , N_g is the number of axle groups (or vehicles), and p_i is the proportion of axle group (or vehicle) i .

The *PLRFs*, calculated using the *VWFs* currently used in HD 24/96 [1], the relative axle group proportions (Table 3) and Equation (7), are shown in Table 6 for the 3 vehicle fleets. It can be seen from this table that the *PLRFs* are negative indicating that the realistic vehicle fleets are causing significantly less pavement damage than the fleet of 80kN standard axles. It can also be seen from Table 6 that the *PLRFs* are sensitive to the different vehicle fleets. For example, the *PLRFs* predicted for Vehicle Fleet A show the pavement life to be increased by 74% for the realistic vehicle fleet compared to the fleet of standard 80kN axles whereas, for Vehicle Fleet C, the corresponding increase in pavement life is 170%.

CONCLUSIONS

- A recently developed model of Long-Term Flexible Pavement Performance (LTPPM) has been modified and used to calculate the damaging effect of different realistic vehicle fleets.
- The axle group models give realistic dynamic tyre force results as quantified by the Dynamic Load Coefficient (DLC).
- Three different vehicle fleets (A, B and C), and three different loading conditions for each fleet have been considered; fully laden, 2/3 laden and 1/3 laden.
- The effect of the different vehicle fleets (compared to a vehicle fleet of 80kN standard axles) on long-term flexible pavement performance has been assessed for three classes of pavement (Motorway, Trunk Road and Principal Road).
- Results from the LTPPM show that different failure mechanisms are predicted for different pavement classes. For example, a thick pavement construction (Motorway) is predicted to fail by rutting with negligible cumulative fatigue damage, whereas a much thinner pavement construction (Principal Road) is predicted to fail by fatigue damage and loss of stiffness in the asphaltic material.
- LTPPM predictions show that the results, in terms of a Pavement Life Reduction Factor (*PLRF*) are insensitive to the response of the lower pavement layers (sub-base and subgrade) to dynamic loads, and are relatively insensitive to differences in the vehicle fleet.
- Results from the 1/3 laden and 2/3 laden simulations have been used to adjust the *PLRFs* to account for realistic axle load variations within the vehicle fleet.

- The adjusted *PLRFs* calculated from the LTPPM are negative for those pavements that are predicted to fail by rutting indicating that the increase in pavement life caused by trafficking by the realistic vehicle fleets (compared to pavement life for the same pavements trafficked by a fleet of 80kN standard axles) is typically 20%.
- The corresponding *PLRFs* are positive for those pavements that are predicted to fail by fatigue indicating that the reduction in pavement life caused by trafficking by the realistic fully-laden vehicle fleets (compared to pavement life for the same pavements trafficked by a fleet of 80kN standard axles) is typically 78%.

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TABLES & FIGURES

Table 1: Axle Group Models.

Model Name	No. Axles	Axle Group	Suspension Type	Tyre Type	Axle Weight (Tonnes)
STER	1	Tractor (steer)	Steel (mono-leaf)	Single	7
SINS	1	Tractor (drive)	Steel (multi-leaf)	Dual	10
SINA	1	Tractor (drive)	Air + dampers	Dual	10
DUAS	2	Tractor (drive) Trailer	Steel (multi-leaf)	Dual	19
DUAA	2	Tractor (drive) Trailer	Air + dampers	Dual	19
TRIS	3	Trailer	Steel (multi-leaf)	S-single	22.5
TRIA	3	Trailer	Air + dampers	S-single	22.5

Table 2: Axle Group Model Parameters.

Parameter	STER	SINS	SINA	DUAS	DUAA	TRIS	TRIA
M_{u1} / kg (fully laden)	3100.0	4400.0	4400.0	8500.0	8500.0	10,050.0	10,050.0
(2/3 laden)	3100.0	2933.3	2933.3	5666.7	5666.7	6700.0	6700.0
(1/3 laden)	3100.0	1466.7	1466.7	2833.3	2833.3	3350.0	3350.0
M_{u2} / kg	400.0	600.0	600.0	500.0	500.0	400.0	400.0
M_{u3} / kg	-	-	-	500.0	500.0	400.0	400.0
M_{u4} / kg	-	-	-	-	-	400.0	400.0
I_{u2} / kgm ²	-	-	-	9.65	9.65	9.65	9.65
I_{u3} / kgm ²	-	-	-	9.65	9.65	9.65	9.65
I_{u4} / kgm ²	-	-	-	-	-	9.65	9.65
k_u / MN/m	0.23	0.86	0.5	0.9	0.2	0.9	0.2
k_l / MN/m	0.23	0.86	0.5	0.9	0.2	0.9	0.2
β_u / mm	2.5	2.5	2.5	1.5	1.5	1.5	1.5
β_l / mm	2.5	2.5	2.5	1.5	1.5	1.5	1.5
F_T / kN	5.0	8.0	0.0	4.5	0.0	4.5	0.0
C_s / kNs/m	1.5	6.5	13.0	1.0	8.0	1.0	8.0
k_p / kNm/rad	-	-	-	10.0	150.0	10.0	150.0
C_p / kNms/rad	-	-	-	0.1	2.0	0.1	2.0
k_T / MN/m	1.0	2.0	2.0	2.0	2.0	1.3	1.3
C_T / kNs/m	1.0	2.0	2.0	2.0	2.0	1.3	1.3
b_1 / m	-	-	-	0.495	0.495	0.495	0.495
b_2 / m	-	-	-	0.185	0.185	0.185	0.185

Table 3: Axle Group Model Proportions and Calculated VWFs.

Axle Group Model	VWF	Motorway (Fleet A)	Trunk Road (Fleet B)	Principal Road (Fleet C)
STER	0.54	0.38	0.43	0.45
SINS	2.26	0.26	0.34	0.37
SINA	2.26	0.04	0.06	0.06
DUAS	3.68	0.14	0.09	0.06
DUAA	3.68	0.04	0.02	0.02
TRIS	2.16	0.07	0.03	0.02
TRIA	2.16	0.07	0.03	0.02
FVWF	-	1.85	1.67	1.60

Table 4: *PLRFs* for Vehicle Fleets A, B and C (Fully Laden).

Road Type	Subgrade Rutting Exponent	Failure Mode	<i>PLRF</i> (A)	<i>PLRF</i> (B)	<i>PLRF</i> (C)
Motorway	1	Rutting	32%	32%	34%
	4	Rutting	34%	35%	35%
Trunk Road	1	Fatigue*	51%	56%	56%
	4	Fatigue*	56%	60%	60%
Principal Road	1	Fatigue	89%	90%	90%
	4	Fatigue	89%	90%	90%

Table 5: *PLRFs* for 1/3 and 2/3 Laden Conditions.

Road Type	Subgrade Rutting Exponent	Failure Mode	<i>PLRF</i> (Fleet A)		
			1/3 Laden	2/3 Laden	Fully Laden
Motorway	1	Rutting	-10%	16%	32%
Trunk Road	1	Rutting*	-22%	9%	51%
Principal Road	1	Fatigue	76%	80%	89%

Notes: * The mode of failure for the 1/3 and 2/3 laden conditions was rutting whereas the mode of failure for the fully laden condition was fatigue; -ve *PLRF* = Increase in life relative to fleet of 80kN standard axles; +ve *PLRF* = Decrease in life relative to fleet of 80kN standard axles.

Table 6: Predicted *PLRFs* Including Realistic Axle Load Variations.

Fleet	HD 24/96	LTPPM ($L_0=1$)		
		Motorway	Trunk	Principal
A	74%	-20%	-20%	+78%
B	136%	-	-	-
C	170%	-	-	-

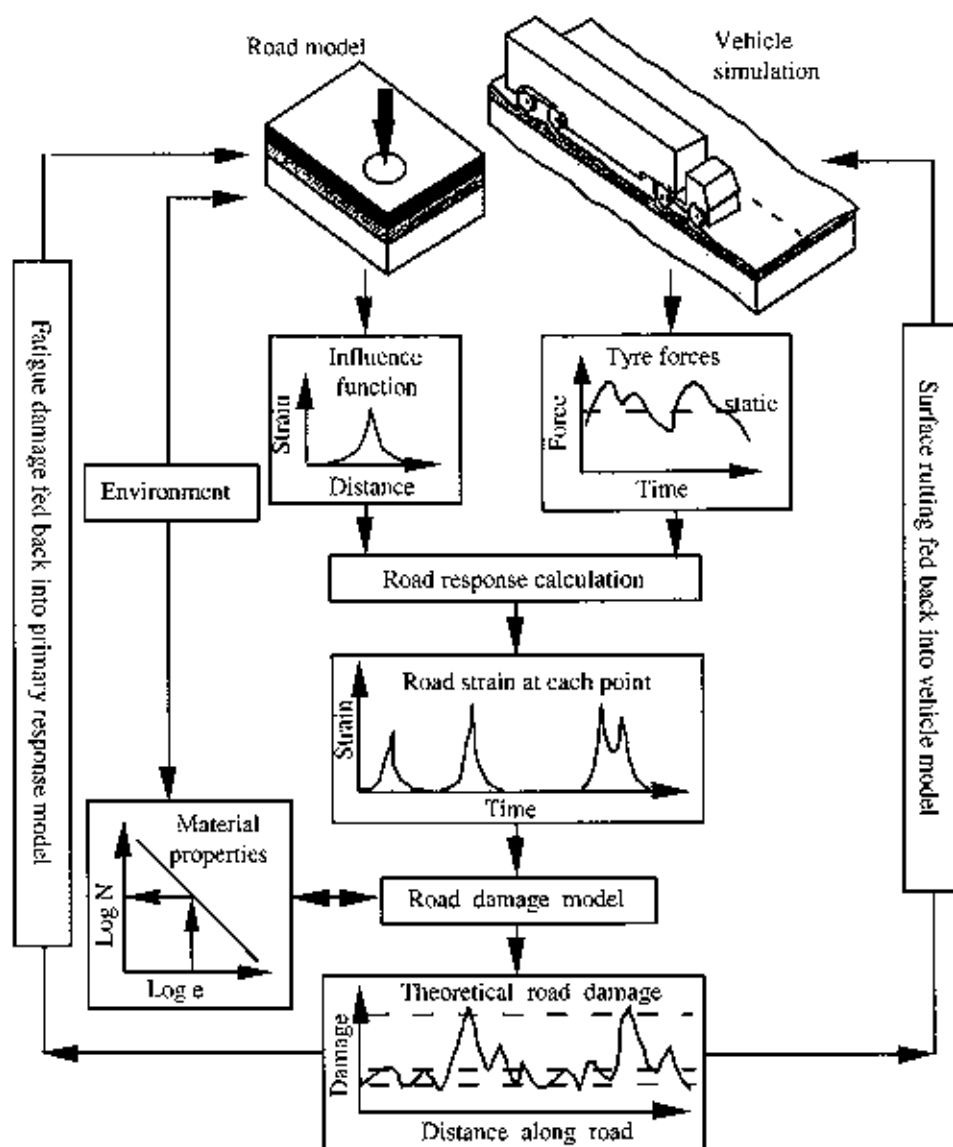


Figure 1: LTPPM Methodology.

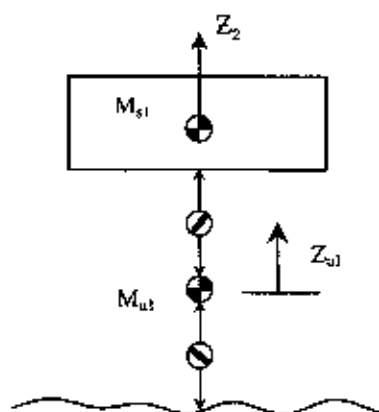


Figure 2: Single Axle Group Model.

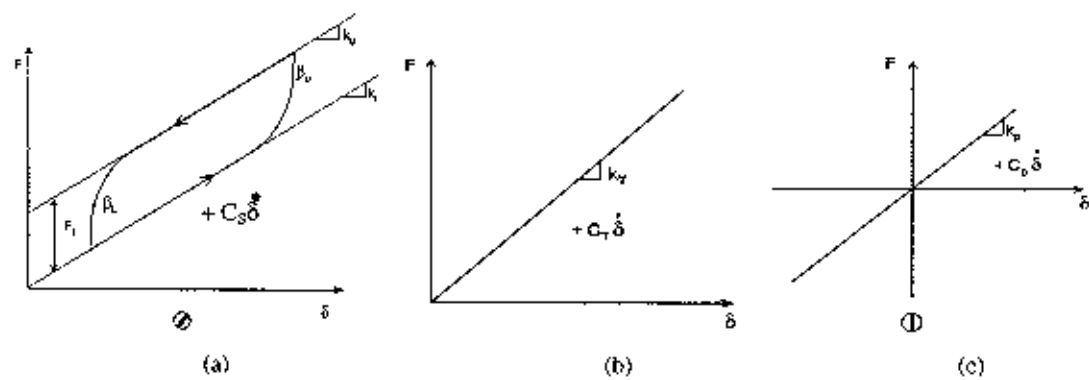


Figure 3: (a) Suspension Element, (b) Tyre Element, (c) Levelling Beam Element.

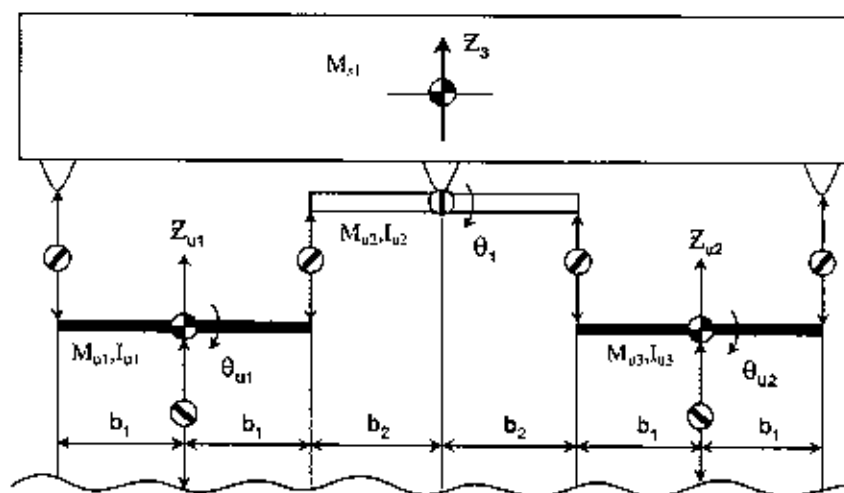


Figure 4: Tandem Axle Group Model.

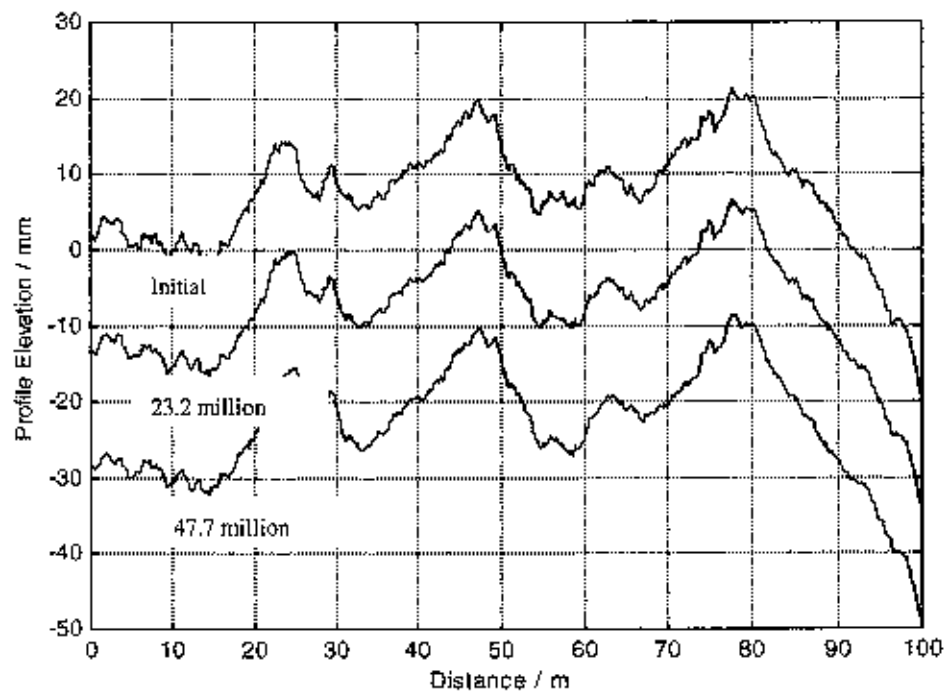


Figure 5: Surface Displacement Profiles for a Motorway Trafficked by Vehicle Fleet A (Subgrade Rutting Exponent=1).

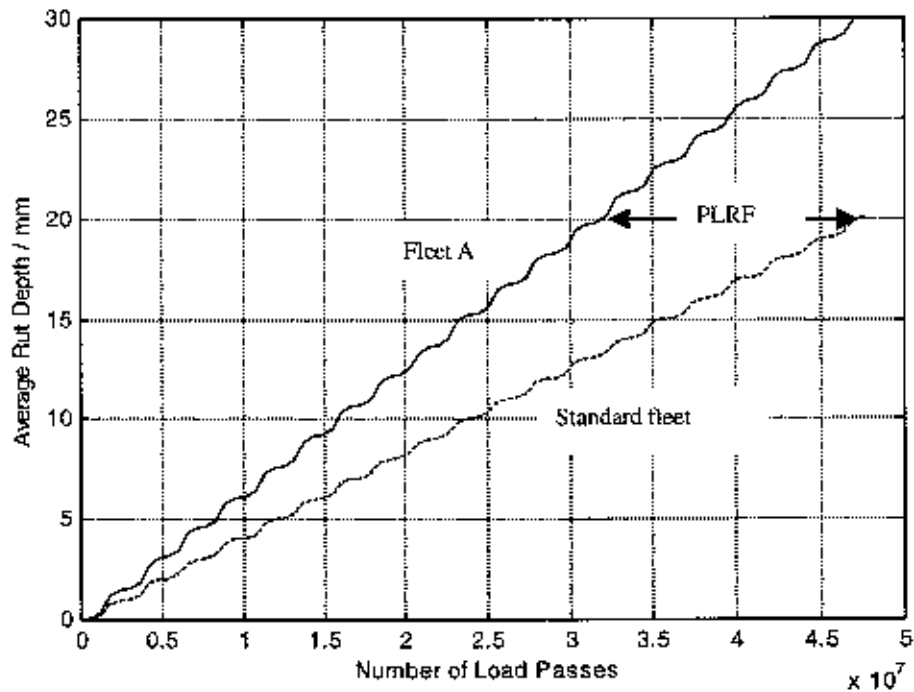


Figure 6: Accumulation of Rut Depth for a Motorway Trafficked by Vehicle Fleet A (Subgrade Rutting Exponent=1).

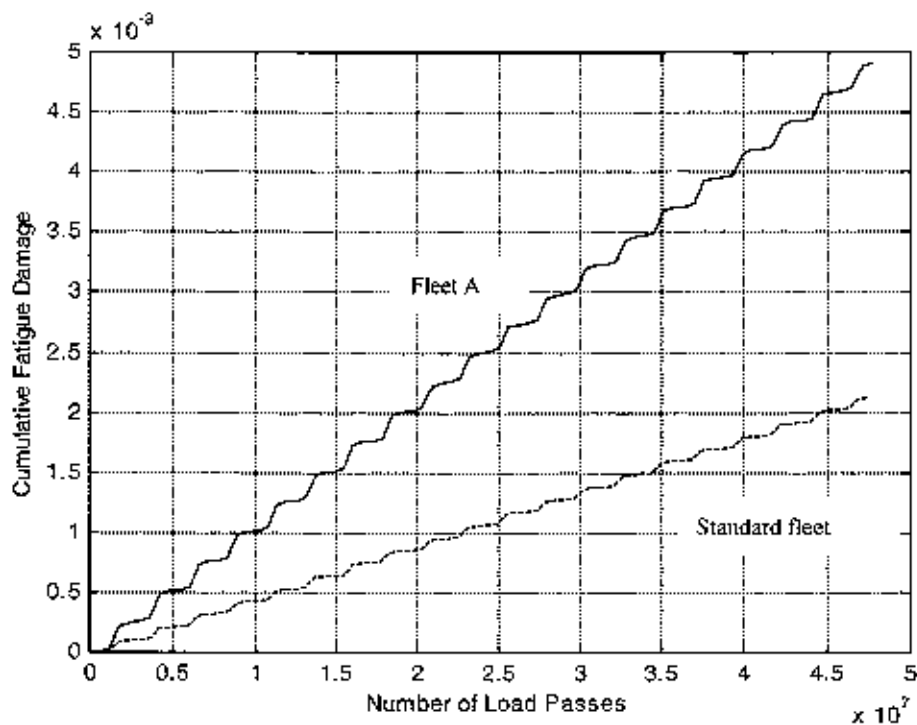


Figure 7: Accumulation of Fatigue Damage for a Motorway Trafficked by Vehicle Fleet A (Subgrade Rutting Exponent=1).

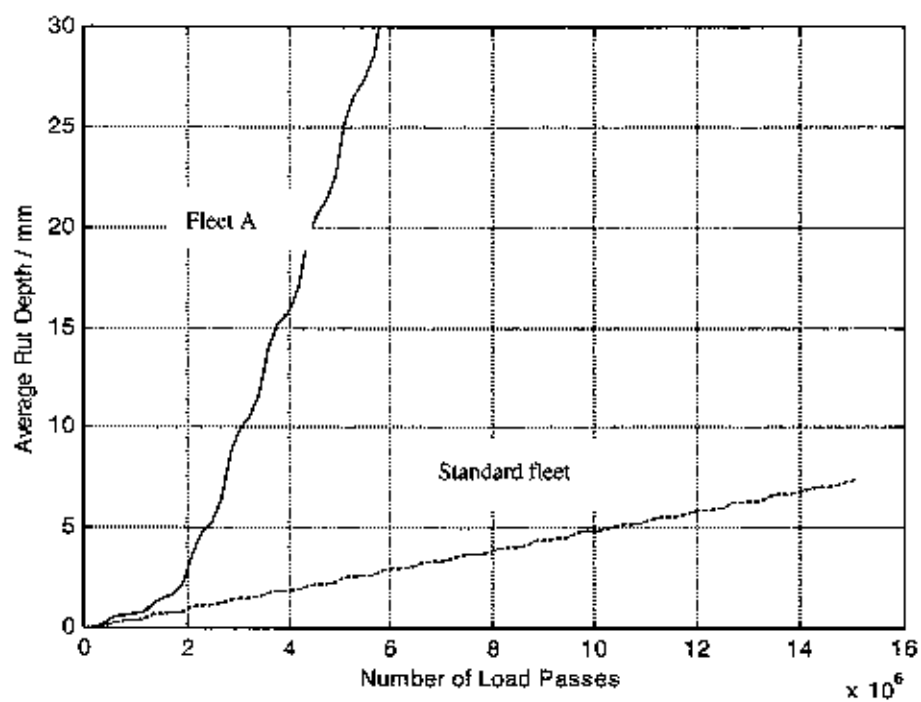


Figure 8: Accumulation of Rut Depth for a Principal Road Trafficked by Vehicle Fleet A (Subgrade Rutting Exponent=4).

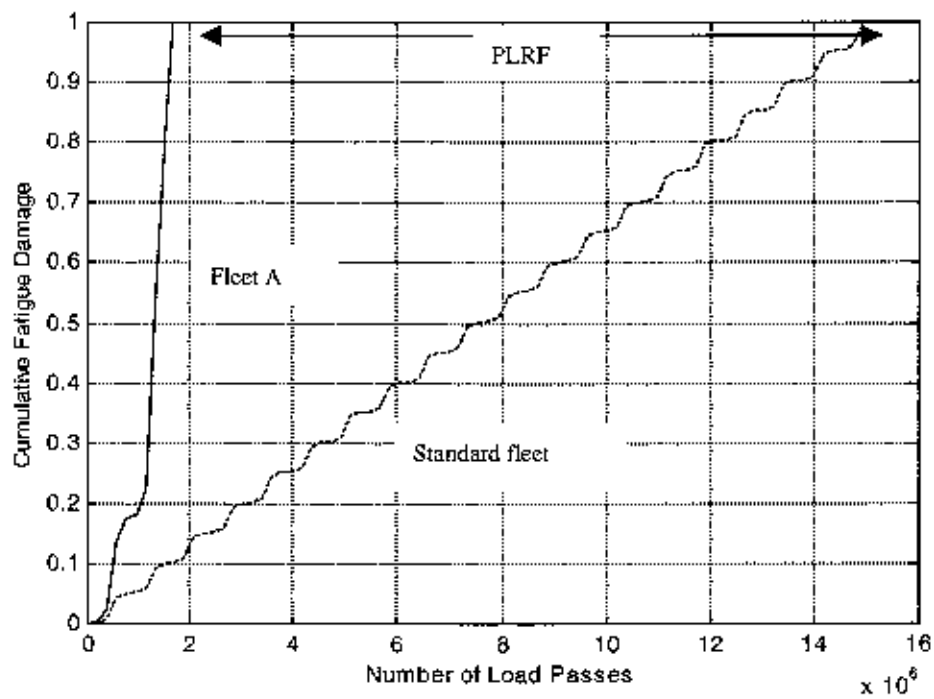


Figure 9: Accumulation of Fatigue Damage for a Principal Road Trafficked by Vehicle Fleet A (Subgrade Rutting Exponent=4).

PARAMETER SENSITIVITY OF THE DYNAMIC ROLLOVER THRESHOLD

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ABSTRACT

Knowledge of commercial vehicle rollover mechanics, required in the development of active dynamic control systems and when designing for increased safety, commonly relies on static analysis providing the steady state rollover threshold, SSRT. In a rolling vehicle kinetic energy is always present and that deteriorates the analysis of roll stability from SSRT. Therefore, knowledge of the dynamic rollover threshold, DRT, is equally relevant. In order to investigate the parameter sensitivity of the dynamic rollover threshold, the Taguchi method is applied: simulations are performed according to a specific plan forming an orthogonal matrix existing of high, medium and low parameter values. The influences from five test parameters on SSRT as well as DRT of a truck and a tractor semitrailer combination are calculated, including the corresponding parameter interaction effects. Investigated parameters are frame roll stiffness plus axle roll stiffnesses and roll center heights of front and rear axles. Results show that the different vehicles are unequally sensitive to parameter changes: the rear axle roll characteristics are the most important semitrailer parameters, while the front axle roll stiffness is most important for the truck. An important result yielding from this is that two vehicles can be equally stable statically but different dynamically.

INTRODUCTION

Commercial vehicle rollover has grave implications: the accident type contributes substantially to injuries but also to environmental damage. Several vehicle occupants are seriously injured or killed every year and vehicles carrying hazardous goods often waste it. When a driver detects the vehicle approaching or exceeding the stability limit, the possibility of returning to a stable driving situation is very small or does not exist. Several of these accidents are hardly avoidable today, e.g. the extremes in the driver-vehicle-environment system: the driver falling asleep at the wheel, the roadway collapsing during cornering or the vehicle having such a low level of roll stability that a simple steering input results in rollover. However, it is possible to identify situations, where the rollover can be avoided by using stability system assistance to the driver and/or the vehicle.

Avoiding the Accident

In 1990, Preston-Thomas and Woodrooffe created a rollover warning device feasibility study [1]¹, showing that 42% of more than 2000 investigated incidents could be avoided with a system installed, warning of the incipient risk. They propose a passive warning device, similar to other passive devices and learning systems presented by e.g. Ervin et al. [2] and Nelligan and Zein [3].

Active stability systems, e.g. Vehicle Dynamics Control, VDC discussed by Hecker et al. [4], avoid rollovers more efficiently than passive systems. The vehicle detects the prevailing conditions and stability margins and supports the driver at an early stage by returning the combination to a stable condition. This is achieved by applying a suitable braking force at each wheel, whereby speed is reduced while a stabilising yaw torque is generated, assisting the driver in managing the situation. Several similar active systems are suggested, e.g. the rearward amplification suppression, RAMS [2] and the pure roll stability system, ARB presented by Wielenga [5]. Active as well as passive systems require *detection of instability* and this implies a need for increased knowledge of rollover mechanics.

THE DYNAMIC ROLLOVER THRESHOLD

The on-road rollover, in contrast to the off-road event, is the most thoroughly documented type, being the easiest to analyse and control: yaw and/or roll stability systems possibly avoid these rollovers. Its most common analysis

¹ Numbers in brackets indicate end paper references

approach is the static, from which the rollover threshold is calculated. During steady state cornering on an even road under no influence of external forces, the level of lateral acceleration determines whether the vehicle rolls over or not. The following definition is applicable:

Definition 1: The Steady State Rollover Threshold, SSRT is the maximum value of lateral acceleration, which the vehicle when driving in steady state may resist in order not to rollover.

Numerous models exist for the calculation of SSRT: generally two-dimensional roll plane models, the static stability factor, SSF being the simplest. It is defined as one half of the average front and rear track-width, divided by the total CG height. Through the simplifying assumption that a vehicle behaves as a rigid body, SSF equals SSRT as a first order approximation. SSF is the least conservative estimation among applied quasi-static models, reported by e.g. Christos and Guenther [6]; when comparing it to any established model considering flexibility, it predicts a higher threshold.

During cornering, the vehicle rolls due to suspension compliance resulting in lateral CG shift toward the outside of the turn. This offset reduces the lever arm on which the gravity force acts to resist rollover. In order to refine the model, this effect is considered by introducing suspension roll center and stiffnesses. Consequently, models with higher precision exist including other flexibilities, such as tire and chassis frame, cf. e.g. [1], Lozia [7] and Ruhl and Ruhl [8].

However, in the real world, rollover does not take off during steady state driving, it occurs during a transient manoeuvre, at acceleration levels lower than SSRT, Rakheja and Piché [9]. Hence there exists a lateral acceleration interval within which rollover can, but does not necessarily occur. Since SSRT is the best case measure of roll stability, the worst case measure is also needed in order to encircle the range where instability is possible. The following definition is applicable when driving on a smooth surface under no influence of external forces:

Definition 2: The Dynamic Rollover Threshold, DRT is the minimum absolute peak value of lateral acceleration of all manoeuvres bringing the vehicle to rollover.

This definition implies that DRT is a worst case measure of roll stability: conditions are necessary but not sufficient for rollover. Other proposals for a dynamic measure of roll instability exist, like the step acceleration threshold by Bernard et al. [10], which is limited to one vehicle model only. Variants of the dynamic acceleration threshold are also presented by e.g. Marine et al. [11] and by Das et al. [12].

A METHOD DETERMINING DRT

The calculation method presented by the author in [13], determines DRT in accordance with the definition above. It is here only briefly represented.

The Roll Energy Diagram

The roll energy diagram includes energy considerations: in rough outline it consists of the potential energy required to bring the vehicle into roll instability and the kinetic energy present in the system, both as a function of lateral acceleration. The diagram, first presented in [14], is successfully developed in [13].

When the vehicle is stable, driving straight ahead, the potential energy is defined as zero. By applying a lateral acceleration and finding the static equilibrium, that acceleration's *equilibrium potential energy* originating in spring deflections and the translation of CG against the gravitational and lateral acceleration potential fields results.

The diagram presents two curves of equilibrium potential energy as functions of steady state lateral acceleration, stable and unstable driving. The lower curve starts at stable driving straight ahead (at the origin) and continues for increasing steady state lateral acceleration up to stable driving at SSRT (rightmost point of the curve). The upper curve on the other hand is more spectacular in its interpretation: it starts at straight ahead driving on the wheels of one side only(!) and continues for that kind of driving at increasing lateral acceleration up to SSRT.

The *rollover energy*, the minimum required bringing the vehicle to rollover, is simply the energy difference between the boundaries, the difference between the two equilibrium states. The rollover energy decreases as acceleration increases, approaching zero at SSRT.

The lower curve is interpreted as the negative *unrestrained kinetic energy*: introducing a step acceleration laterally releases the corresponding energy. Therefore, DRT occurs at the intersection between the upper curve and the

abscissa: the lateral acceleration at which the unrestrained kinetic equals the rollover energy. At this acceleration, the released kinetic energy equals the minimum required causing rollover, fulfilling the criteria of the dynamic rollover threshold definition.

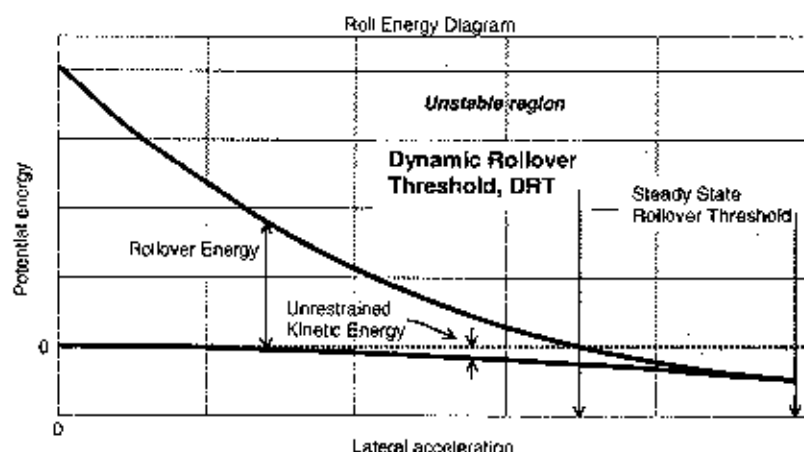


Figure 1 - The Roll Energy Diagram determining DRT

By deriving the roll energy diagram, using a suitable calculation program and a valid model, steady state and dynamic rollover thresholds are identified.

[13] shows that roll damping is considered in a modified DRT, but one lateral acceleration value can not fully describe the dynamic forces acting on the rolling vehicle. The measure is also incapable of describing yaw induced rollover, e.g. instability originating in rearward amplification. It is however a very manageable measure: DRT using the presented method consists of one engineering value only and it is as close to the definition as is possible.

MODEL EXAMPLES

One important quality of DRT is its *model independence*, i.e. it is applicable to any vehicle rollover model. Here, the method is applied to two vehicle models: describing a single truck and a semitrailer combination.

The Single Truck

The single truck model includes five rigid bodies and five roll degrees of freedom (5DOF): front and rear axles, cab and the front and rear parts of the chassis, interconnected by rotational spring-damper elements, giving each body 1DOF. The frame flexibility is also included, but in order to avoid a complex FE-model, two chassis bodies are simply connected with a rotational stiffness. Tires are assumed stiff in the lateral direction to limit the DOF, but they are compliant vertically.

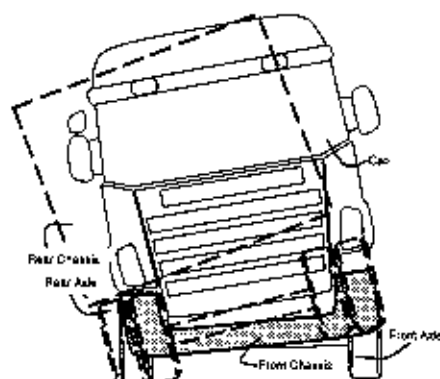


Figure 2 - Front view of truck model

The Semitrailer

The semitrailer pulled by a tractor is the most common combination rolling over, wherefore this investigation also includes such a model, originating in the truck described above. Three bodies are added, representing the

semitrailer (one wheel axle and two chassis bodies), while the cab body is excluded. The fifth wheel is a rotational stiffness in the 7DOF model.

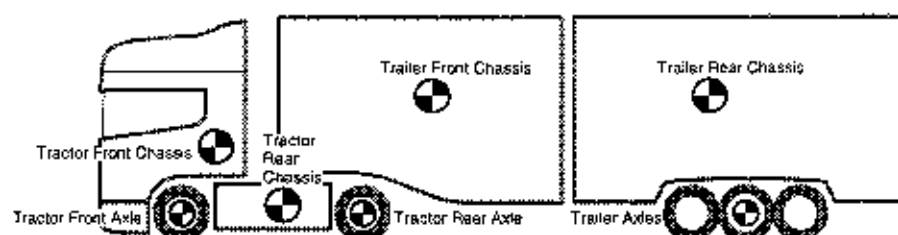


Figure 3 - Side view of semitrailer model

THE TAGUCHI METHOD

The Taguchi method, one of several multivariate analytical methods, is frequently used in quality engineering, but also applicable in product development. A test plan, described by an orthogonal matrix, is used enabling testing of more than one parameter at a time, without repetition, at two or more levels. Using two levels yields a linear and three or more a non-linear estimation of the parameter sensitivity. The method is thoroughly described by e.g. Phadke [15] and Gore and Davis [16].

One of two advantages with the Taguchi method over the traditional one-at-a-time method can be chosen: the number of tests can be reduced considerably (reduced multi-variate testing) or the effects of parameter interaction can be derived. The latter is chosen in this investigation.

PARAMETERS

In this investigation, the Taguchi method is used to analyse the influence of five truck parameters on the static and dynamic rollover thresholds of the vehicles described above. The method determining DRT described above is applied on computer calculations in the multi-body dynamics software ADAMS and simultaneously, SSRT is calculated. Since the parameter sensitivity of SSRT is more thoroughly documented in the literature, this investigation focuses on DRT.

The models are chosen in order to clarify any possible difference in parameter sensitivity to DRT, and its divergence from SSRT, between the rigid truck and the semitrailer combination. The parameter values defining the roll stability properties of the vehicles are therefore chosen as being equal where applicable, so the vehicles have equal total CG height and except for the load on the truck, it is equal to the tractor in the semitrailer combination.

There are a number of parameters that are likely to influence DRT and SSRT, but in order to limit this investigation only five parameters are chosen. Since it is desirable to investigate the improvement in roll stability that is possible through a realistic redesign of the vehicle, CG height and track-width are left out even though they probably influence the rollover thresholds considerably. Instead five parameters that are more easily changed on a real vehicle, all set by the vehicle manufacturer, are chosen: the roll stiffness of the frame and the roll stiffnesses and roll center heights of both the front and the rear axles, cf. table 1. These parameters are known to influence SSRT, cf. e.g. Winkler et al. [17].

Table 1 - Alternated parameters and their levels

Parameters	Low level	Intermediate level	High level
A Frame Roll Stiffness	200 kNm/rad	400 kNm/rad	600 kNm/rad
B Roll Center Height Front	0.20 m	0.40 m	0.60 m
C Roll Stiffness Front	200 kNm/rad	400 kNm/rad	600 kNm/rad
D Roll Stiffness Rear	500 kNm/rad	1000 kNm/rad	1500 kNm/rad
E Roll Center Height Rear	0.30 m	0.60 m	0.90 m

To investigate non-linearities among the influences, each parameter is alternated on three realistically chosen levels, cf. table 1. It is important to keep in mind that the results of this investigation depend on the range within which the parameters vary. All other parameters are listed in Appendix A.

EXPERIMENTS

Several standard orthogonal arrays, designed for multivariate analysis are found in the literature, cf. e.g. [15], and for this investigation an L_{27} is chosen. This is a 3^{13} -array, indicating that it has 13 columns at three levels, enabling testing of up to 13 parameters at three levels each. However in this investigation, since only five parameters are used, several interaction effects are also studied: all interactions between the frame roll stiffness and the other factors are chosen.

In table B.1 (cf. appendix B), the L_{27} array is presented with grey columns showing the varied parameters and their levels. The non-shaded columns do not set any parameter value levels, they are neglected since interactions are studied. The orthogonal array has 27 rows, indicating that 27 experiments are conducted in this test. Since SSRT and DRT are simultaneously yielded once the roll energy diagram is derived, 27 truck simulations are required and an equal number for the semitrailer.

Tables B.1 and 1 are combined into the left part of table C.1 in appendix C defining parameter settings during the 2-27 simulations. In that table, the base for the following experiments, empty columns are left out since they are not used during experiments. Simulations are conducted as specified by the rows of table C.1 and for each computer calculation, the resulting SSRT and DRT are logged.

RESULTS

The grey shaded fields in table C1 include the results from the experiments. Some of the resulting thresholds are very low since the models simulate high CG vehicles and since some parameter combinations become very unfavourable: DRT varies between 1.38 m/s^2 and 2.98 m/s^2 while SSRT varies between 1.87 m/s^2 and 3.51 m/s^2 .

The simulation results are processed and the resulting effect of a parameter level is defined as the deviation caused by the average of the results at that parameter level from the overall mean. Every parameter has thereby three deviation values or main effects: when keeping it low, intermediate and high as shown in the line diagrams, e.g. figure 7. These main effects are linearised by subtracting the effect of keeping a parameter at its low value from the effect of keeping it high, as in the bar diagrams, e.g. figure 4. Hence, the effect of a parameter is the deviation a unit-change in that parameter causes to the result, e.g. the effect of the frame roll stiffness on SSRT equals the change in SSRT when the stiffness is increased from its low 200 to its high 600 kNm/rad.

Furthermore, the resulting SSRT and DRT, when changing more than one parameter, depend not only on the main effects but also on the interaction effects of two or more parameters. These interactions are not as easily calculated as the main effects: the linearised interaction effects of two parameters are calculated, with the intermediate level excluded, by subtracting the average result with the parameters kept at different levels from the average result keeping them at equal. So, the combination effect of two parameters is the *additional* deviation a simultaneous unit-change in both parameters causes to the result compared to sum of the parameter effects only.

Interactions are analysed in diagrams where the deviation from the overall mean when combining one level of a parameter with one level of another is calculated. To derive the combination of two parameters at three levels, nine such calculations are required. Non-linear interactions are recognised by the non-parallelism in these diagrams.

Truck Results

The linearised effects of the truck simulations are given in figures 4 to 6. A Pareto diagram disregards the signs, but presents the sizes of the effects arranged in a descending scale.

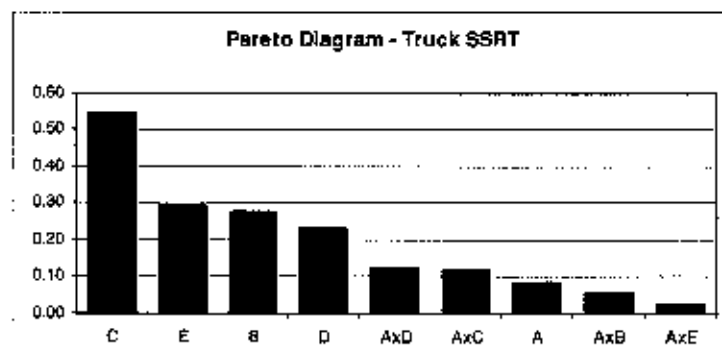


Figure 4 - Linearised main and interaction effects of the truck SSRT

As can be seen in figures 4 and 5, the front axle roll stiffness is by far the most important parameter for both rollover thresholds of this truck. Roll stiffness and roll center height of the rear axle are also important parameters for DRT, while these two plus the roll center height of the front are important for SSRT. DRT appears to be influenced more than SSRT by the frame stiffness, which in interaction with the front roll stiffness also has a slight effect on the dynamic threshold. The other combination effects studied are negligible.

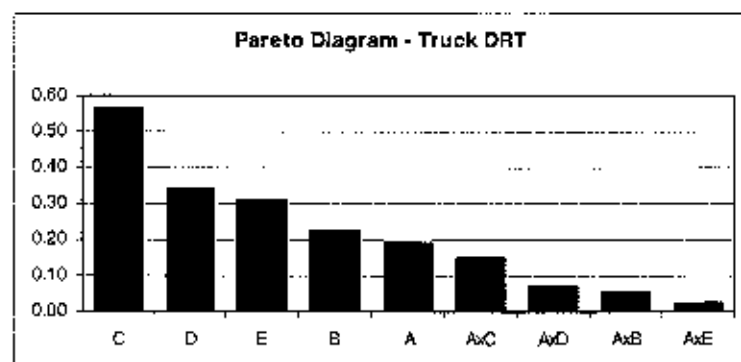


Figure 5 - Linearised main and interaction effects of the truck DRT

Figure 6 shows the linearised parameter effects for DRT divided by SSRT, which indicates how much the parameter change influences relative stability, i.e. the size of the interval within which rollover can occur. By changing a parameter that highly influences that quotient, the stability of the vehicle is more affected than expected if DRT is not considered. The stiffnesses, rear axle the most, clearly affect the relative stability of this truck, while all other effects, except for that from the rear axle roll center height, are negligible.

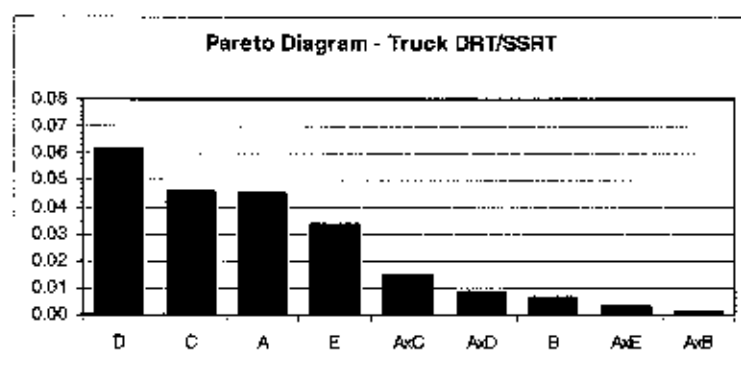


Figure 6 - Linearised main and interaction effects of the truck DRT/SSRT

Figure 7 presents all main effects on the truck DRT, showing that all are positive and slightly non-linear: the effect is higher when changing from the low level to the intermediate than from the intermediate to the high.

Figure 8, showing the non-linear interactions of the truck DRT, reveal that the higher the frame stiffness, the more important is the front axle roll stiffness (AxC) but all other interactions are small.

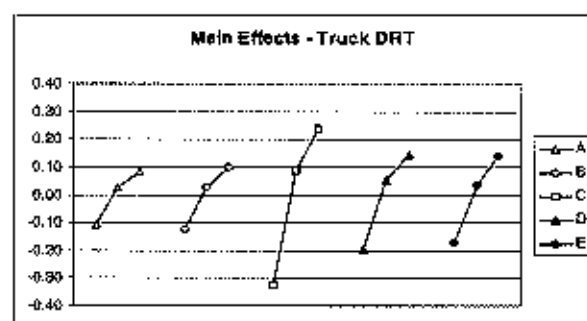


Figure 7 - Main effects of the truck DRT simulations

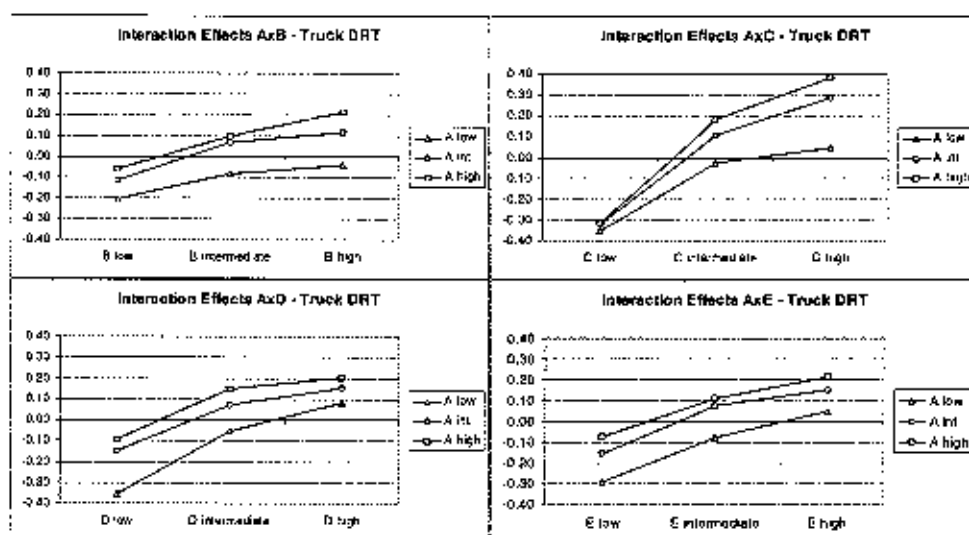


Figure 8 - Interaction effects of the truck DRT simulations

Semitrailer Results

Semitrailer results are shown in figure 9 to 13.

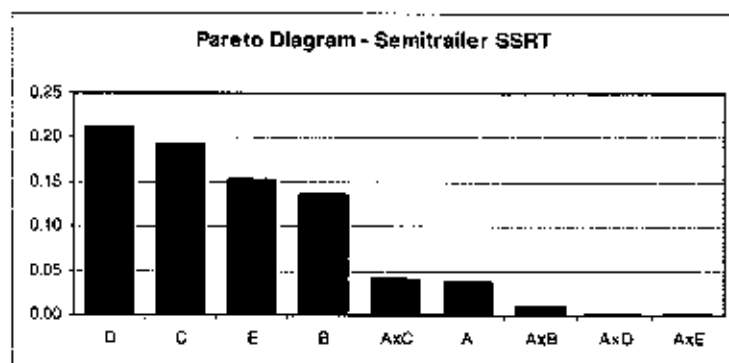


Figure 9 - Linearised main and interaction effects of the semitrailer SSRT

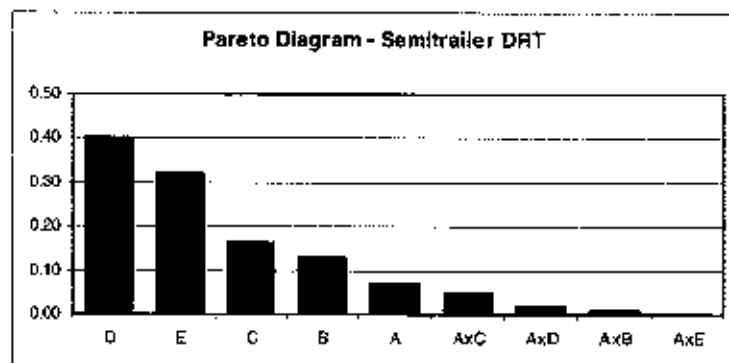


Figure 10 - Linearised main and interaction effects of semitrailer DRT

As shown in figures 9 and 10, DRT effects from the rear axle characteristics are very important for the semitrailer. The rear axle roll stiffness is the most important parameter for both thresholds and for DRT, the rear axle roll center height is also important. The effects of front and rear axle roll stiffnesses and roll center heights are of equal magnitude for SSRT, but for DRT, front axle roll characteristics have only a small influence. Other studied DRT as well as SSRT effects are negligible. It is important to point out that the scales differ between DRT and SSRT results and between truck and semitrailer results.

Figure 11 indicates that only the rear axle roll characteristics have a significant influence on the relative stability of the semitrailer combination. Both relevant effects are positive, i.e. the parameters affect DRT more than SSRT.

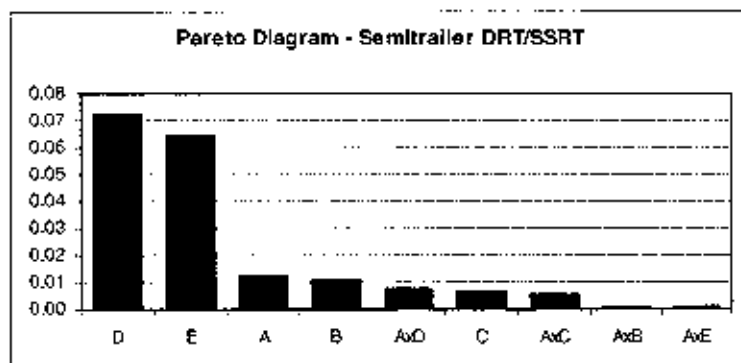


Figure 11 - Linearised main and interaction effects of the semitrailer DRT/SSRT

Figure 12, presenting all main effects on the semitrailer DRT, shows that all are positive and all but the front axle roll stiffness are slightly non-linear with higher gain in the lower range. The front roll stiffness is more non-linear: the highest DRT is reached when that parameter is kept at its intermediate level.

Figure 13, finally shows all non-linear interactions in the semitrailer DRT simulations being small but AxC: the higher the frame stiffness, the less pronounced is the non-linear effect of the front axle roll stiffness. Results in figure 13 are differently scaled than the corresponding truck results in figure 8.

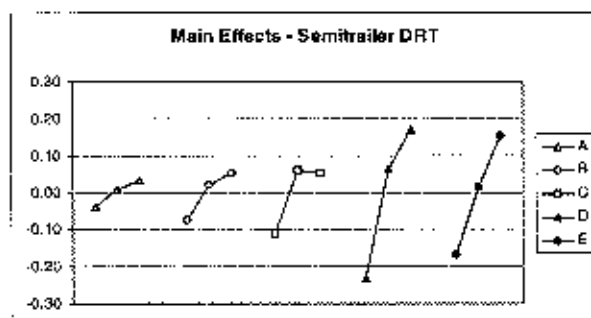


Figure 12 - Main effects of the semitrailer DRT simulations

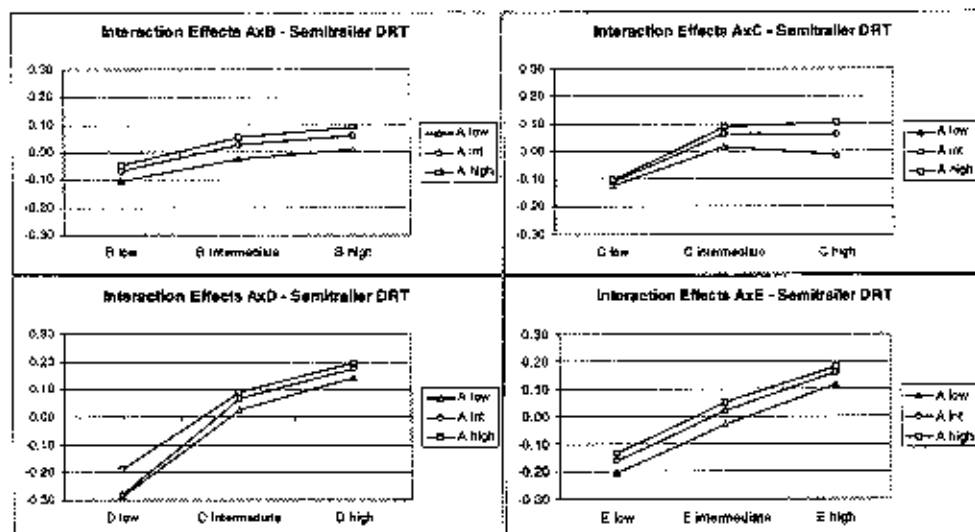


Figure 13 - Interaction effects of the semitrailer DRT simulations

CONCLUSIONS

In this work, a method deriving the dynamic rollover threshold, a complement to the static threshold is analysed. Results from applying the Taguchi method to simulations of the static and dynamic rollover thresholds of a rigid truck and a semitrailer combination, in which five parameters and their interactions are studied, show that the parameters influence the different vehicles unequally:

- Among the studied parameters, the rear axle roll stiffness and roll center height are important to the sensitivity of the vehicles. While they are the most important semitrailer parameters, the front axle roll stiffness is the most important on the truck.
- The frame stiffness is less important: a tendency that is most pronounced in the semitrailer results. However, the interaction effect of the front axle and the frame stiffnesses are noticeable in DRT. In order to gain dynamic stability by increasing the front axle roll stiffness it is important to keep the torsional stiffness of the frame high. The other combination effects are negligible.
- The parameter effect magnitudes on the truck are larger overall than on the semitrailer, since the trailer axle additionally resists the roll overturning moment.

The most important conclusion is that, since DRT and SSRT are not equally affected by parameters, vehicles can be *equally stable statically but different dynamically*. Therefore, if the static but not the dynamic rollover threshold is analysed, a vehicle redesign can reduce the roll stability even though it appears to preserve or even improve it.

ACKNOWLEDGEMENTS

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DEFINITIONS AND ABBREVIATIONS

A	Test Parameter: Frame Torsional Stiffness [Nm/rad]
B	Test Parameter: Front Axle Roll Center Height [m]
C	Test Parameter: Front Axle Roll Stiffness [Nm/rad]
D	Test Parameter: Rear Axle Roll Stiffness [Nm/rad]
E	Test Parameter: Rear Axle Roll Center Height [m]
I ₂₇	Orthogonal Test Array with 27 Rows
ADAMS	Automatic Dynamic Analysis of Mechanical Systems
ARB	Anti Rollover Braking
CG	Center of Gravity
DOF	Degrees of Freedom
DRT	Dynamic Rollover Threshold
RAMS	Rearward Amplification Suppression
SSF	Static Stability Factor
SSRT	Steady State Rollover Threshold
VDC	Vehicle Dynamics Control

APPENDIX A

The values used in the simulations are listed in tables A.1 and A.2. All values are realistic but do not necessarily correspond to any existing vehicle.

Table A.1 - Single truck parameters

CG heights	Front axle	0.50 m
	Rear axle	0.50 m
	Front chassis	1.80 m
	Rear chassis	2.30 m
	Cab	2.00 m
Masses	Front axle	700 kg
	Rear axle	1300 kg
	Front chassis	3000 kg
	Rear chassis	12000 kg
	Cab	1000 kg
(Inter-body) Roll center heights	Chassis front to rear	0.90 m
	Chassis front to cab	1.40 m
Roll stiffnesses	Chassis front to cab	500 kNm/rad
Vertical tire stiffnesses (one side)	Front axle tires	950 kN/m
	Rear axle tires	1900 kN/m
Geometry	Wheel base	3.70 m
	Distance front axle to CG	2.06 m
	Front axle track-width	2.06 m
	Rear axle effective track-width	1.86 m

Table A.2 - Semitrailer parameters

CG heights	Trailer front chassis	2.37 m
	Trailer rear chassis	2.37 m
	Trailer axles (merged)	0.50 m
	Tractor front chassis (merged with cab)	1.20 m
	Tractor rear chassis	1.10 m
Masses	Trailer front chassis	15000 kg
	Trailer rear chassis	15000 kg
	Trailer axles (merged)	2000 kg
	Tractor front chassis (merged with cab)	4000 kg
	Tractor rear chassis	2000 kg

Table A.2 - Semitrailer parameters, continued

(Inter-body) Roll center heights	Fifth wheel	1.15 m
	Trailer chassis front to rear	1.00 m
	Trailer axle to rear trailer chassis	0.40 m
Roll stiffnesses	Fifth wheel	5000 kNm/rad
	Trailer chassis front to rear	2000 kNm/rad
	Trailer axle to rear trailer chassis	4500 kNm/rad
Vertical tire stiffness (one side)	Trailer axle tires	2850 kN/m
Geometry	Distance tractor front axle to CG	1.11 m
	Distance tractor CG to fifth wheel	1.70 m
	Distance fifth wheel to trailer CG	4.07 m
	Distance trailer CG to trailer axle	1.85 m
	Trailer axle track-width	2.00 m

APPENDIX B

In the following table, the L_{27} array defining the parameter levels during the 27 tests is presented. Grey-shaded columns show the parameter levels while non-shaded columns are neglected in the analysis and are shown only since they are necessary to demonstrate that the chosen array is orthogonal, a requirement in the Taguchi method. Due to this orthogonality, not all possible combinations of the five parameters on the three levels are required.

Table B.1 - The standard L_{27} orthogonal array

Exp. no.	1	2	3	4	5	6	7	8	9	10	11	12	13
	A	B			C			D			E		
1	1	1	1	1	1	1	1	1	1	1	1	1	1
2	1	1	1	1	2	2	2	2	2	2	2	2	2
3	1	1	1	1	3	3	3	3	3	3	3	3	3
4	1	2	2	2	1	1	1	2	2	2	3	3	3
5	1	2	2	2	2	2	2	3	3	3	1	1	1
6	1	2	2	2	3	3	3	1	1	1	2	2	2
7	1	3	3	3	1	1	1	3	3	3	2	2	2
8	1	3	3	3	2	2	2	1	1	1	3	3	3
9	1	3	3	3	3	3	3	2	2	2	1	1	1
10	2	1	2	3	1	2	3	1	2	3	1	2	3
11	2	1	2	3	2	3	1	2	3	1	2	3	1
12	2	1	2	3	3	1	2	3	1	2	3	1	2
13	2	2	3	1	1	2	3	2	3	1	3	1	2
14	2	2	3	1	2	3	1	3	1	2	1	2	3
15	2	2	3	1	3	1	2	1	2	3	2	3	1
16	2	3	1	2	1	2	3	3	1	2	2	3	1
17	2	3	1	2	2	3	1	1	2	3	3	1	2
18	2	3	1	2	3	1	2	2	3	1	1	2	3
19	3	1	3	2	1	3	2	1	3	2	1	3	2
20	3	1	3	2	2	1	3	2	1	3	2	1	3
21	3	1	3	2	3	2	1	3	2	1	3	2	1
22	3	2	1	3	1	3	2	2	1	3	3	2	1
23	3	2	1	3	2	1	3	3	2	1	1	3	2
24	3	2	1	3	3	2	1	1	3	2	2	1	3
25	3	3	2	1	1	3	2	3	2	1	2	1	3
26	3	3	2	1	2	1	3	1	3	2	3	2	1
27	3	3	2	1	3	2	1	2	1	3	1	3	2

APPENDIX C

Table C.1 defines the parameter settings during the simulations and presents the results. The non-shaded part of the table is created by combining table B.1 and table 1 presented in the chapter Parameters.

Table C.1 - The 27 experiments, the control factor level combinations and the results

Exp. no.	Parameters					Truck		Semitrailer	
	A kNm/rad	B m	C kNm/rad	D kNm/rad	E m	SSRT m/s ²	DRT m/s ²	SSRT m/s ²	DRT m/s ²
1	200	0.20	200	500	0.30	1.87	1.38	2.92	2.04
2	200	0.20	400	1000	0.60	2.70	2.22	3.44	2.67
3	200	0.20	600	1500	0.90	2.97	2.55	3.43	2.90
4	200	0.40	200	1000	0.90	2.60	2.14	3.37	2.76
5	200	0.40	400	1500	0.30	2.74	2.26	3.39	2.69
6	200	0.40	600	500	0.60	2.66	2.11	3.23	2.41
7	200	0.60	200	1500	0.60	2.67	2.19	3.35	2.76
8	200	0.60	400	500	0.90	2.75	2.21	3.38	2.62
9	200	0.60	600	1000	0.30	2.76	2.24	3.43	2.58
10	400	0.20	200	500	0.30	1.90	1.44	2.92	2.06
11	400	0.20	400	1000	0.60	2.71	2.32	3.45	2.71
12	400	0.20	600	1500	0.90	2.96	2.66	3.47	2.95
13	400	0.40	200	1000	0.90	2.55	2.15	3.36	2.77
14	400	0.40	400	1500	0.30	2.74	2.35	3.42	2.73
15	400	0.40	600	500	0.60	2.93	2.46	3.28	2.51
16	400	0.60	200	1500	0.60	2.63	2.21	3.36	2.77
17	400	0.60	400	500	0.90	2.89	2.41	3.42	2.69
18	400	0.60	600	1000	0.30	2.91	2.50	3.48	2.65
19	600	0.20	200	500	0.30	1.91	1.46	2.91	2.07
20	600	0.20	400	1000	0.60	2.71	2.38	3.46	2.73
21	600	0.20	600	1500	0.90	2.96	2.74	3.50	2.98
22	600	0.40	200	1000	0.90	2.53	2.15	3.36	2.77
23	600	0.40	400	1500	0.30	2.74	2.41	3.44	2.75
24	600	0.40	600	500	0.60	3.07	2.50	3.31	2.57
25	600	0.60	200	1500	0.60	2.61	2.21	3.36	2.78
26	600	0.60	400	500	0.90	2.96	2.52	3.43	2.72
27	600	0.60	600	1000	0.30	2.99	2.67	3.51	2.70

ROAD USER CHARGING FOR HEAVY GOODS VEHICLES

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ABSTRACT

The aim of this project is to develop, field-trial and evaluate a dynamic road user charging system for Heavy Goods Vehicles (HGVs). The need for fairer and more efficient means of pricing road use through the "polluter pays" principle has prompted many governments throughout Europe and other parts of the world to investigate alternative pricing strategies for urban and inter-urban road networks. Of particular concern is the road freight sector, as HGVs cause a disproportionately high level of road pavement wear per vehicle-kilometre compared to other vehicle classes, and fixed annual taxation payment systems are highly inefficient at capturing the external costs of pavement wear which can vary by pavement type, dynamic axle load, HGV class and axle configuration. In order to align better the actual costs of pavement wear and charges for road use more accurately, various distance and weight-based charging approaches have been implemented recently. For example, in Europe and the USA, Weigh-In-Motion (WIM) systems have been installed on certain roads and bridges to assist infrastructure cost recovery (for example the HELP Programme in the USA). In Sweden and Norway, HGVs pay a distance and HGV-class based charge. Australia and New Zealand operate an axle-weight-based distance charge, and in Switzerland, the recently introduced Heavy Vehicle Fee charges HGVs on the basis of actual distance travelled and gross permissible axle weights.

However, the main drawbacks with extending these systems into a pan-European system for charging for actual pavement wear at the point-of-use include the likely roadside infrastructure costs and serious concerns about using measured static axle loads (as opposed to dynamic axle loads) to estimate actual pavement wear. In addition, it is argued here that these systems are not able to satisfactorily align pavement wear costs to road user charges. Therefore this paper concludes with a functional specification of an on-board charging system which seeks to overcome these drawbacks by extending state-of-the-art technologies for variable road-user charging, automatic vehicle locationing and dynamic axle-load measurement to enable HGVs to be charged a price which includes the dynamic effect of loading on structural pavement wear. It is envisaged that the evaluation of the prototype system will lead to recommendations for new approaches to allocating road track costs more efficiently and fairly between individual road-users.

INTRODUCTION

Roads and road transport have undoubtedly played and continue to play a tremendous role in the continued development of human civilisation. The importance of road infrastructure in the economy can be seen from the fact that road freight, which is carried mainly by HGVs (> 3.5 tonnes unladen weight), accounts for about 81% of all freight transport in the UK and other European Union countries (FTA, 1999).

The cost of maintaining and improving this vital infrastructure, which amounts to about 0.2 to 2 percent of a country's Gross Domestic Product, (OECD, 1998), has generally been met through various forms of road and vehicle charges and taxes (e.g. fuel taxes and vehicle excise duty). Throughout the world, various approaches are in use to facilitate the collection and administration of these taxes and charges. However, the underlying philosophy behind them is that 'each road user should pay the highway costs that is creates or occasions' (HCAS, 1997). This approach, known as the Cost Occasioned approach, is used widely all over the world and is in recognition of the fact that various vehicle classes differ in the extent of road wear they cause. However, until the 1960s, there was no way of incorporating this in either pavement design or in cost allocation procedures.

One of the most significant results of the extensive full-scale road tests conducted by the American Association of State Highway and Transport Officials (AASHTO) between 1958 and 1960 was the establishment of a quantitative relationship explaining the extent to which different vehicles cause road wear. It established that 'the decrease in pavement serviceability caused by heavy vehicles could be related to the fourth power of its static load' and is known as the Fourth Power Law (Cebon 1993, p. 9). This 'Holy Grail' has been used by engineers since then to assess the damaging potential of vehicles on pavements and also for allocating road user charges between HGVs.

Over the years, continued research (see for example AASHO Road Test, 1958-1960; OECD, 1992, 1998) has been carried out to investigate and provide scientific evidence of pavement-vehicle interactions and how the characteristics of vehicles and road pavements affect road wear (OECD, 1998). These studies have shown that the major factors which affect pavement wear include the characteristics of the vehicle (in terms of its axle and tyre configuration, suspension system and vehicle speed), the dynamic wheel forces generated by the vehicle and pavement characteristics (type, surface roughness and temperature).

A review of road and highway cost allocation procedures and other charging systems implemented throughout the world reveals that these systems are not able to adequately satisfy the requirements of the cost occasion approach - to charge a user the highway cost he creates - as is being sought. This is evident in the considerable efforts being made by various governments striving to constantly improve existing methodologies (for example the Federal Highway Cost Allocation Studies in the USA) and the introduction of new charging systems by others (for example the Heavy Vehicle Fee in Switzerland). This is because the fundamental factors causing pavement wear are not satisfactorily and adequately employed in the cost allocation and charging methodologies. In addition, temporal and spatial variations in pavement wear associated with vehicle use are not sufficiently catered for. As a result, these systems lack fairness and efficiency in the resulting vehicle class-based charges. These issues were part of the subject matter in the European Commission's Green Paper published in 1995, 'Towards Fairness and Efficient Pricing in Transport', which noted that, because of the apparent lack of fairness and efficiency in transport pricing, it recommended, as a matter of urgency, the introduction of Electronic Kilometre charges based on infrastructure damage and possibly other parameters (European Commission, 1995).

With a view to the introduction of such a scheme, some time in the future, this research project aims to develop, test and evaluate a dynamic road user charging system for HGVs based on pavement wear. To achieve this aim, the following objectives have been set (Thorpe, 1998):

- § to undertake a critical review of current procedures for allocating road track costs between different HGV groups;
- § to design and develop an in-vehicle automatic debiting system for charging HGVs in relation to expected structural pavement wear and other external costs of road-use by HGVs;
- § to test the automatic debiting system through off-road and on-road experiments with suitably-equipped HGVs;
- § to evaluate the system's ability to accommodate varying road-use charges, both spatially and temporally and by HGV type; and
- § to assess the likely impacts of a 'pay-as-you-go' system for charging HGVs on the freight distribution sector.

The purpose of this paper is to describe the design of the functional specification for a dynamic road user charging system for HGVs. Conclusions arrived at from this initial work show that this new dynamic charging system for HGVs will indeed be capable of addressing many of the more critical issues of concern associated with current charging systems by charging HGVs for pavement wear as and when it is caused. The various functional components and technologies necessary for developing the system have also been identified and a functional model of the charging system is presented.

The next section of this paper presents a review of the interaction of HGVs with road pavements and points out the major parameters which determine the extent of pavement wear. This is followed in the subsequent section by describing methods currently employed to recover pavement wear costs from HGVs and identifying the need for new improvements with these methods. The final section then describes preliminary work on the design of a system for charging HGVs based on pavement wear.

VEHICLE - PAVEMENT INTERACTION: A REVIEW

Modern road pavements can be classified as being flexible of bituminous construction, rigid from Portland cement concrete or composite consisting of both flexible and rigid components. In most countries world-wide flexible pavement construction generally dominates. During its service life, road pavements are subjected to a variety of external conditions (e.g. traffic loading and environmental conditions), which leads to a decrease in pavement performance. These external conditions cause the pavement to suffer a variety of distress types, which can be classified as functional or structural and load or non-load associated. Whereas functional pavement wear is primarily non-load associated and is concerned with the 'ride quality' of the pavement from the viewpoint of the road user, structural pavement wear is mainly load associated and describes both visible and / or non-visible defects on the road pavement such as fatigue cracking or pumping (Huang, 1993). A significant development from the Fourth Power Law was the concept of the Equivalent Standard Axle Load (ESAL) which provides a way of comparing the relative damaging potential of different vehicle classes on the road network (Cebon, 1993). Using this concept, it can be estimated that one pass of a single 11.5 tonne axle of a heavy goods vehicle is equivalent to about 276,000 passes of a 0.5 tonne axle of a car. Thus, in comparison to HGVs, cars cause relatively insignificant load associated road wear and as a result, all load associated road wear is usually attributed to HGVs.

Even though the Fourth Power Law has been used extensively in both pavement design and cost allocation methodologies, it has received much criticism (see for example Addis and Whitmarsh, 1981) regarding its validity

in light of differences in tyre sizes and pressures, pavement construction materials and methods, traffic volumes and wheel loads and axle load group configurations which are all significantly different now from the time of the AASHO road tests (Cebon, 1993). Research shows that the exponent in the Fourth Power Law varies widely, with different exponents for the different pavement types. For flexible pavements, values ranging from 1.3 - 6 have been cited while for composite and rigid pavements, it may be as high as 8 - 12 (Cebon, 1993). This variation is important as the value of the exponent in the Fourth Power Law has implications for the allocation of road track costs. A lower exponent (say 1 or 2) would mean that the current practice of attributing all load-associated damage to HGVs is questionable. Alternatively, a higher exponent implies more damage by HGVs. However, the lack of any widely accepted alternative means that the Fourth Power Law is still used by engineers and researchers in studies of pavement design and damage and also to allocate road track costs between different vehicles.

Pavement wear is affected by a variety of static vehicle load characteristics. Research by Gillespie *et al.* (1992) shows that whereas static wheel forces are the main cause of fatigue damage to pavements, gross vehicle weight is the dominant factor in rutting damage. Axle configuration also affects pavement wear with tandem and tri-axle groups carrying more weight than the same number of widely spaced single axles for equal pavement wear to flexible and rigid pavements (Peattie, 1984; Cebon 1993). The effect of tyre pressures and configuration on pavement wear has also been studied. For example, Christison *et al.* (1978) report that, in theory at least, single tyres cause about seven to ten times more damage than a dual tyre pair while a report by the OECD (1992) recommends that wide-base single tyres should be considered to be 2.1 times more damaging and conventional single tyres to be 2.9 times more damaging compared to dual tyres. Cebon (1993) reports similar results that have been reported by Treybig (1983), Huhala (1988) and Gillespie *et al.* (1992).

The magnitude of static wheel forces has been thought to be the main cause of pavement wear and the use of the Fourth Power Law was considered appropriate for cost allocation. However, because of the surface nature of road pavements, construction methods, materials and vehicle dynamics, forces which are applied to the road pavement are not uniform or of a constant value but fluctuate above and below a constant static force. These fluctuations are referred to as *dynamic forces* or *loads*. These forces are generated by two distinct components of vehicles - the sprung mass of the vehicle, which generates low frequency dynamic wheel forces (1.5 - 4 Hz), and the unsprung mass, which generates dynamic wheel forces in a higher frequency range (8 - 15 Hz). The forces are generated in the low frequency range from the body bounce or pitch motions of the vehicle while the high frequency forces are associated with wheel hop vibrations. For most vehicles, dynamic wheel forces are generated from the sprung mass motion in the low frequency range (i.e. 1.5 - 4 Hz). However, vehicles with poorly damped suspensions also generate large amounts of unsprung mass vibration (Cebon, 1999).

Dynamic wheel forces are thought to cause considerably more pavement wear than static wheel forces and, in recent years, research has focused on understanding the full effects of these forces on pavement wear (OECD, 1992). The effects of dynamic wheel forces during vehicle and pavement interactions have been studied by instrumenting either the vehicles themselves or the road pavement (see for example Mitchell and Gyencs, 1987; Cole and Cebon, 1989), or through computer simulations (Captain *et al.*, 1979; Hu, 1988; Cole and Cebon, 1992) to measure or simulate stresses and strains which develop within the pavement structure. These stresses and strains are the primary pavement responses when road pavements are loaded with vehicle wheel forces. The magnitude of dynamic wheel forces is influenced by a number of vehicle and pavement characteristics, the most important of these being the roughness of the road surface, the speed of the vehicle, the axle and tyre configuration, vehicle geometry and mass distribution and the properties of the vehicle suspension. It can be represented by a parameter known as the *Dynamic Load Coefficient* which is defined as the ratio of the root mean square dynamic wheel force to the mean wheel force where the root mean square dynamic wheel force is essentially the standard deviation of the probability distribution. In general the magnitude of dynamic wheel forces increases with vehicle speed and pavement surface roughness with typical root mean square amplitudes of approximately 10 - 30% of static wheel loads.

Research has shown that differences in pavement types and characteristics lead to differences in how pavements respond to the effects of dynamic loading. The OECD DIVINE Project (1998, p.66) concluded that, for thicker bituminous pavement structures (i.e. >150mm), 'there was a direct and proportional relationship between primary pavement responses and the instantaneously applied load' implying that strain levels which lead to pavement wear reflect the degree of dynamic loading. For thinner pavements however, the relationship was less clear. Dynamic wheel loads also cause a relative increase in pavement wear due to fatigue damage while on the other hand, there was a decrease in pavement wear due to rutting damage.

The net effect of dynamic wheel forces on road pavements compared to static wheel loads is a further reduction in the pavement life due to increased levels of stresses and strains. Cebon (1993, p.66) states that 'depending on the method of analysis, assumptions, and mode of failure, dynamic wheel loads increase *average* theoretical road damage by 10 - 40%, and *peak* theoretical damage by a factor of 2 to 4 (over damage due to static loads) for typical vehicles and operating conditions.' However, the presence of spatial repeatability (which is the phenomenon where by under mixed traffic conditions, dynamic loads typically tend to concentrate at points along a road at intervals of 8 - 10 metres) means that peak pavement damage rather than average pavement damage is the critical consideration for pavement wear. In this case (i.e. with spatial repeatability), the relative increase in peak

damage by dynamic wheel forces can range from 2 - 14 times more depending on the suspension type of the vehicle.

Dynamic wheel forces, and in particular the presence of spatial repeatability on road pavements, lead to a relative increase in road pavement damage. Road pavements also wear at different rates depending on factors such as the type and thickness of the pavement, the nature of the surface roughness and the characteristics and speed of passing vehicles.

Clearly therefore, there are a number of truck, pavement and environmental factors which contribute to pavement damage. The nature and extent of these various contributions was investigated by Gillespie *et al.* (1993). Their findings suggest that the different factors affect pavement damage in different ways and to different extents. Interaction between some of the factors also impact on their relative contribution to pavement damage. For example, whereas static axle loads are the primary cause of fatigue damage, gross vehicle weight has a direct influence on rutting with individual axle loads being less significant. The relative importance of these various parameters can be appreciated by looking at the extent of variability in the range of damage ratios of the different parameters in Gillespie's research. Based on this criterion, the most important of these factors are listed below (in no particular order):

- \$ axle loads;
- \$ gross vehicle weight;
- \$ tyre type;
- \$ pavement roughness;
- \$ slab / wear course thickness;
- \$ surface temperature (flexible) / temperature gradient (rigid); and
- \$ wheel path location.

This knowledge of how vehicles and pavements interact can help improve not only the design and maintenance of road pavements and vehicles but also form the basis of providing a more equitable, efficient and transparent way of charging vehicles for their use of road space.

ROAD TRACK COST ALLOCATION AND HGV CHARGING SYSTEMS

Road Track Cost Allocation Models

The recovery of 'sunk' road transport infrastructure cost has traditionally been informed by Highway Cost Allocation (HCA) models. HCA is the assignment of highway-related costs to various classes of highway users (and sometimes non-users), usually to estimate the share of highway costs that various users pay and to evaluate the equity of highway user fees (USDOT, 1997). A number of approaches to highway cost allocation exist and are summarised below (see US Highway Cost Allocation Study 1997 for details). These are:

- \$ the *Benefit Based Approach*, which allocates costs according to the relative benefits realised by different vehicle classes from highway investments. The greater the benefits, the greater the share of user fees a vehicle class should pay, regardless of its contribution to highway costs;
- \$ the *Cost Occasioned Approach* where the physical and operational characteristics of each vehicle class are related to expenditures for pavement, bridge, and other infrastructure improvements; and
- \$ the *Marginal Cost Approach* which charges vehicles according to environmental, congestion, pavement, and other marginal costs associated with their highway use. Unlike other approaches, the objective of the marginal cost approach is not to assign all highway agency expenditures to different vehicle classes, but rather to estimate user fees that cover the marginal costs of highway use by different vehicle classes.

Of these three approaches to highway cost allocation, the Cost Occasioned Approach has been the most widely used (see for example the UK Road Track Cost Allocation Model). The underlying philosophy of this approach is that each user should pay the highway costs that it creates or "occasions". Traditionally, these costs have been limited to the obligations and expenditures of the various highway agencies responsible for the improvement and maintenance of the road infrastructure. The focus has thus been on the highway cost paid from highway user charges and how close these user charges reflect the true cost of each vehicle's use of the highway. Different user charge mechanisms and structures are then evaluated to ensure equity in the user charge structure.

An example of a road track cost allocation model is the UK Road Track Cost Allocation Model (RTCAM) which was, until recently, used to estimate and allocate costs to different classes of road users based on the cost occasioned approach (UK Department of Transport, 1994). Other models used in different countries, such as the Highway Cost Allocation (HCA) Model used in the USA (HCAS, 1997) and the 'PAYGO' system in Australia (National Road Transport Commission, 1998), are also based on the cost occasioned approach with a similar methodology to the UK RTCAM. However, unlike the UK model, the US HCA model allocates costs separately for new and/or added lanes and for maintenance. For new pavement construction, costs for a hypothetical 'base facility' that would serve the purposes of all vehicle classes are shared evenly after which costs for additional requirements such as thicker pavement structures relating to HGVs are allocated to the vehicle class responsible for such costs.

A number of important issues of concern regarding HGVs regarding these systems are the continued use of static loading as against dynamic loading in cost allocation procedures, the inability of these systems to distinguish and align costs to temporal and spatial variations and ensuring equity in user charges to all vehicle classes. These limitations, some of which are inherent in the adopted methodologies, arise because of the lack of resources required to obtain all the necessary data and ensure a minimum level of data quality (Urquart and Rhodes, 1990). The result is that there is generally a mismatch between pavement costs and the prices paid, which varies both within and across different vehicle classes.

Heavy Goods Vehicles Charging Systems

The need for fairer and more efficient ways of charging (especially HGVs) for road use has led to the introduction of charging systems aimed at addressing the issue of aligning pavement costs with prices paid in a more equitable and efficient way. For example, the New Zealand Heavy Vehicle Fee (HVF) introduced in 1977 and still in use today, attempts this by allocating heavy vehicle charges based on the number and spacing of axles, the number of tyres per axle and the actual distance travelled as monitored by a Hubodometer - the rationale being that these additional vehicle parameters are known to affect the extent of pavement wear caused by HGVs and therefore offer more differentiation in vehicle charges (Galenson, 1990).

The Heavy Vehicle Electronic License Plate (HELP) Programme introduced in the early 1990s in the USA is a multi-state heavy vehicle monitoring and charging system using Automatic Vehicle Identification (AVI), Automatic Vehicle Classification (AVC) and Weigh-in-Motion (WIM) technologies. Vehicles equipped with transponders pass key data (such as distance travelled and registration details) to roadside infrastructure, while vehicle loads are measured at normal driving speeds by fixed WIM stations on the highway. Charges are then calculated based on a range of vehicle characteristics, distances travelled and measured vehicle loads. Although a welcome step in the right direction, such a system is unable to evaluate key pavement wear factors such as dynamic wheel forces and differences in pavement types and roughness and could involve considerable capital investments if implemented on (say) a European scale.

The Swiss Heavy Vehicle Fee introduced in 2001 is an electronic charging system for charging HGVs on the country's road network (Liechti *et al.*, 2000). The system consists of On-Board Units (OBUs) and roadside infrastructure for charging and enforcement purposes. The OBU consists of a distance recording device which is complemented with a satellite positioning system (GPS), a tachograph, a movement sensor, a chip card interface and a microwave communications link interface - Dedicated Short Range Communication (DSRC). The roadside infrastructure consists of radio beacons installed along the carriageways at national borders, roadside video/license plate recognition cameras and scanners. Charging is performed by the OBU based on fixed parameters (emission values of vehicles and the maximum laden weight of the tractor and trailer) and on a variable parameter (kilometres driven). Clearly, with this system, an unloaded vehicle is charged the same as a fully laden vehicle per unit distance although in reality, the extent of pavement wear caused is different not only because of differences in weight but also in pavement type and surface characteristics.

A similar electronic charging scheme has been announced for implementation in Germany (Williamson, 2000), while other EU countries also have plans to introduce electronic schemes. Clearly, a considerable amount of effort is being made to improve cost allocation procedures and HGVs charging mechanisms by constantly improving on the data used and introducing more parameters. However, the fundamental issues of concern regarding pavement wear such as dynamic wheel forces and pavement surface roughness remain unresolved.

FUNCTIONAL DESIGN OF A NEW DYNAMIC CHARGING SYSTEM

System Objective

The design of the new road user charging system for HGVs based on actual pavement wear proposed here is based on the principle that such a system should be able to charge individual vehicles for the actual costs of pavement wear caused. In order to do this there is the need to identify, measure and record the necessary factors that contribute to pavement wear. The major factors are:

- § pavement type and thickness;
- § pavement surface roughness;
- § dynamic wheel forces;
- § axle and tyre configuration;
- § vehicle speed; and
- § environmental factors (temperature)

In the UK, flexible pavements make up 85% of all road pavements, with 10% being rigid and 5% of composite construction (DETR, 2000). Depending on ground conditions and other environmental factors, pavement thickness varies between 300mm for 'thin' pavements to 1200mm for 'thick' pavements. Pavement surface roughness also varies depending on the traffic and environmental conditions. These pavement and environmental factors, coupled with vehicle factors such as the axle and tyre configuration, vehicle speed and suspension type, cause differences in the dynamic wheel forces generated as the vehicle travels along the carriageway.

Ideally, during each individual journey, vehicle speeds and dynamic wheel forces are monitored continuously and recorded, and the different pavement types and surface characteristics of these roads travelled on noted. These data, together with details of axle and tyre configuration, are used to determine the appropriate road use charge based on an estimate of actual pavement wear caused.

One argument against such an approach has been that it might discourage road authorities from maintaining the road network to a satisfactory standard. Although possible, this is considered unlikely because of the political and socio-economic importance of roads in modern societies. Highway maintenance is also formulated within the framework of legislation (including the important issues of health and safety) and national specifications, Codes of Practice, European Standards, Road Notes and Quality Assurance Schemes. In the UK for example, legislation is embodied in the Local Authority Association's Highway Maintenance Code of Good Practice (LGACP).

Functional System Components

The following functional components are identified for a possible dynamic road user charging system for HGVs:

1. an Automatic Vehicle Positioning (AVP) system to provide vehicle route information;
2. a digital road map database, which combines with the positioning system to provide vehicle location on the road network as well as characteristics of the pavements travelled on;
3. a Dynamic Wheel Force (DWF) measuring device capable of measuring continuously dynamic wheel forces as the vehicle moves;
4. a distance measuring device;
5. a road use charging model to estimate pavement wear;
6. an on-board computer for processing and storing data;
7. a data retrieval system;
8. a man-machine interface device to provide system information to the driver; and
9. a device for communicating with the roadside for enforcement and transmission purposes.

Modelling Pavement Damage

One of the key components of the charging system is the charging model. This is responsible for converting a range of vehicle characteristics (e.g. axle/tyre configuration and loads, vehicle speed and position) and pavement properties (e.g. pavement type, surface roughness) recorded on the vehicle during its journey for estimating pavement wear. Extensive research into the mechanisms of pavement wear has led to a considerable understanding in this area. Of particular interest are road damage models, which relate road damage at specific points along the road surface by aggregating the dynamic tyre forces applied by the vehicle. Two methods based on this approach to estimating pavement damage are the 'single-vehicle pass' calculations and 'whole-life models'. A summary of these two approaches is provided below. A more detailed review is provided by Cebon (1999).

The Single Vehicle Pass Calculation determines the incremental road damage due to the passage of a vehicle over a particular road. It is very useful for comparing the effects of vehicle features on road damage. The procedure for the single vehicle pass calculation involves measuring the dynamic tyre forces generated by a vehicle travelling over a specific road profile. The forces generated are used as inputs to a pavement model (flexible or rigid) where primary pavement responses (stresses, strains and displacements) are calculated at points along the road. Equations for estimating material damage (fatigue and permanent deformation) are then used with damage accumulation calculations to determine the theoretical road damage at each location.

Whole-Life Models are similar in methodology to the single vehicle pass calculation model but also attempt to predict the deterioration of a pavement's structural integrity and surface profile over time. The calculation attempts to include the effects of environmental/seasonal factors on the strength of the road and variations in the structure and properties of the road. However, these models require an empirical relationship between wheel forces and the degradation of the road surface profile. According to Cebon (1999), this is still an area of much uncertainty and at present no such relationship exist. He points out however that whole-life models could offer the best prospects for understanding the complex interactions that occur in the vehicle-road-environment system.

The single-vehicle pass calculation provides a relatively simple and effective way of comparing the damaging potential of different vehicle types operating under different conditions such as vehicle speed, axle loads and pavement properties. Implicit in this system however is the assumption that pavement characteristics remain constant and hence, under the same operating conditions, pavement wear caused by vehicles in the same category remains unchanged. In reality, this is not so as the degradation of the pavement surface would generally lead to increased pavement wear by subsequent vehicles. In this respect, the whole-life model provides a more realistic approach to pavement damage modelling because of the presence of a feedback loop which updates the condition of the pavement after each vehicle pass. However, as has been mentioned, the whole-life model has yet to be fully validated. Consideration is therefore being given to the choice of model to be adopted since this has implications for the development of the charging system being proposed.

System Architecture

Two candidate system architectures have been identified for the proposed dynamic heavy goods vehicle charging system. The first separates the charging system into two main subsystems; an on-board processing sub-system and an post-processing subsystem (Figure 1). Within this system architecture, the on-board system acts as a data collector - recording dynamic wheel forces and position data and vehicle speeds as the vehicle undertakes its journey. The data collected is then communicated to the post-processing subsystem where it is processed to determine information such as the roads travelled on and the properties of the road pavement (e.g. pavement type and surface roughness). These data are then used in the charging model to enable the appropriate road use charge to be calculated. Bills could then be issued to vehicle operators on a monthly or annual basis.

An alternative system architecture involves combining the two subsystems described above into one on-board, real-time processing system (Figure 2). With this system architecture, each equipped vehicle has all the necessary functional components to estimate autonomously road use charges in real-time. The choice of which system architecture to adopt depends on an evaluation of both systems based on the overall system objectives. The interactions and data flow between the various components of the charging system architecture is shown in Figure 3. Charges may then be collected in real-time, possibly via a smartcard payment system.

Technologies for Functional Components of Charging System

Various technologies exist for each of the functional system components identified in the real-time system architecture depending on factors such as accuracy, cost and the requirements of the dynamic charging system (Drane *et al.*, 1998; Zhao, 1997). For example, the need for the vehicle to be located on the road network in real time requires a sufficiently reliable and accurate positioning system (say less than 10m positional accuracy). It also requires an on-board database for mapping purposes. The dynamic wheel forces of all the axles should also be measured and recorded continuously throughout the vehicle journey. In addition, the amount of data to be stored requires a robust and sufficient capacity data storage device.

For the purposes of this research project the following technologies are proposed to satisfy the functional requirements of the charging system:

- § a GPS unit to provide positioning information as well as vehicle speed, distance travelled and direction data;
- § a dead-reckoning sensor to provide backup to the GPS unit;
- § a strain gauged wheel force sensor for dynamic wheel force measurements;
- § an on-board computer to provide data processing and storage requirements;
- § a GSM interface for external communications; and
- § a man-machine interface for visual and audible signs.

The main focus of the work completed to date has been a review of the various aspects of HGV charging systems and an outline proposal for a new approach to align better road track costs and the charges incurred by vehicle-operators. Currently, alternative methodological approaches for estimating pavement wear by HGVs are under investigation. It is hoped that a decision as to the most appropriate approach can be made soon. This will enable the development of software and hardware systems to commence with a view to pilot installation and field-testing of the prototype system in early 2003.

CONCLUSIONS

In this paper, a brief review of vehicle - pavement has been carried out, summarising some of the important factors and characteristics affecting road pavement wear. An investigation of current road track allocation systems and models and HGVs charging systems reveal that in as much as the ultimate aims of these systems are to provide a fair, equitable and efficient means of charging for road use, it is argued that this has not been achieved to date. This is because the fundamental issues and factors of concern, which relates road use and pavement wear, are not addressed satisfactorily.

This paper has thus described the functional specification for a dynamic road user charging system for HGVs, which will be able to charge road users for the pavement wear. The next steps in the project involve the formulation of the charging model for estimating pavement wear caused by individual vehicles after which the software and hardware development of the charging system will commence. It is hoped that the development of this prototype system will make a considerable progress towards to a fairer, efficient and transparent means of cost allocation and road user charging.

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TABLES & FIGURES

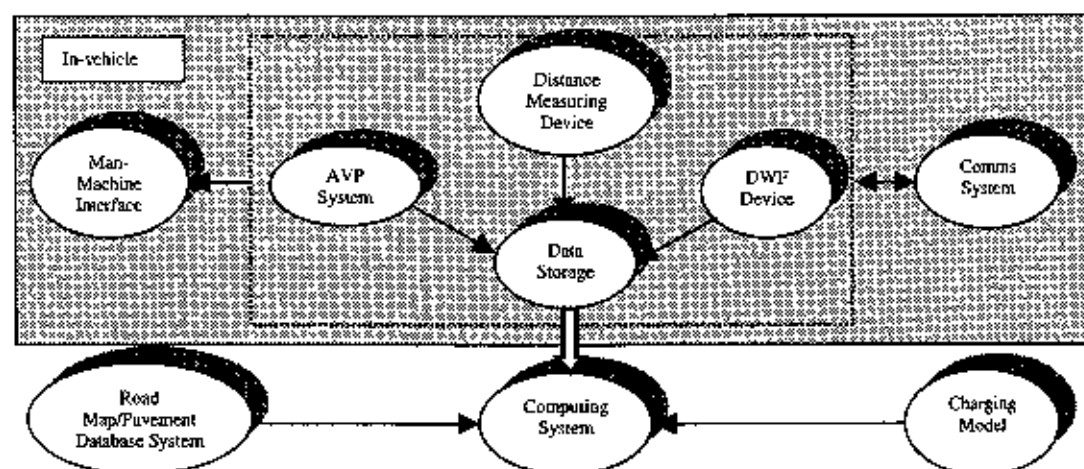


Figure 1 – Distributed Post-Processing System Architecture

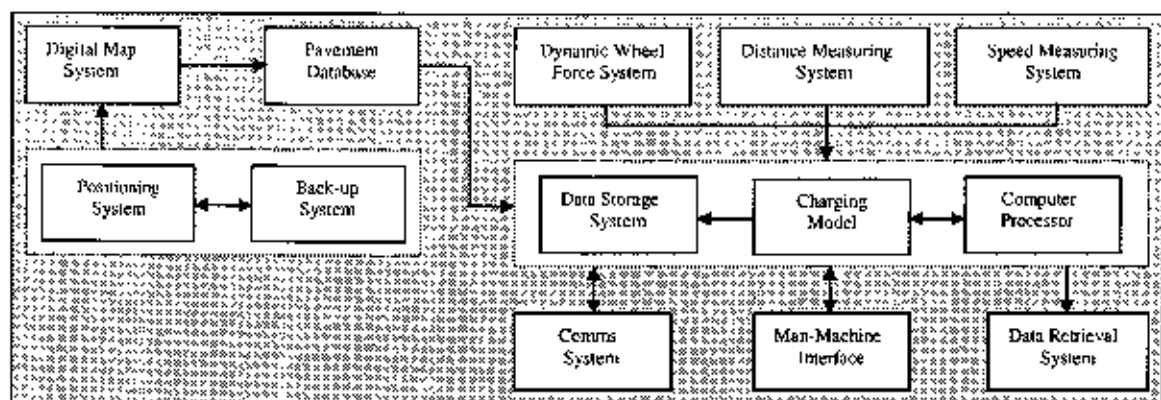


Figure 2 - Real-time System Architecture

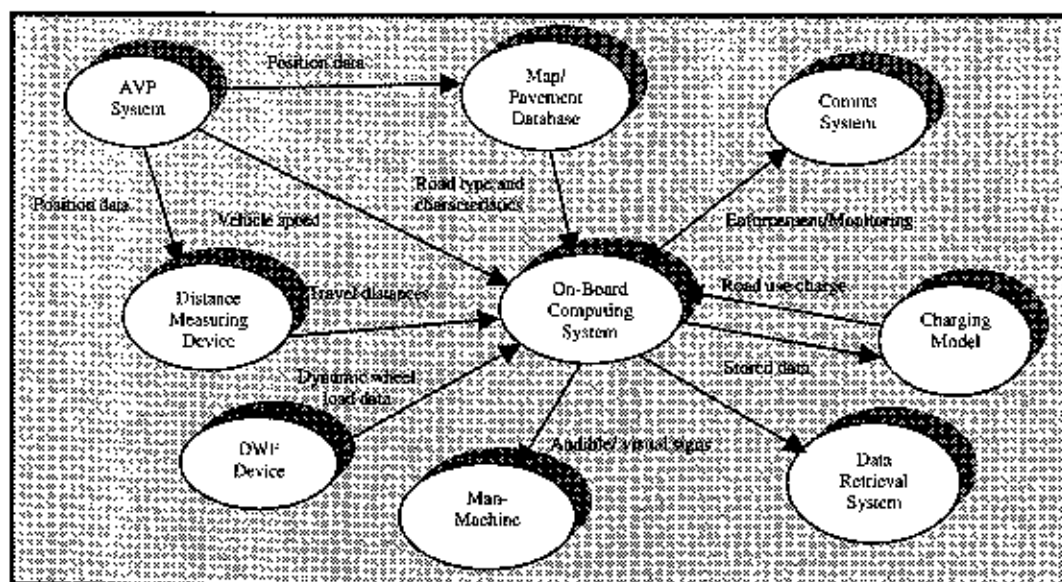


Figure 3 - Data Flow Diagram for Heavy Goods Vehicle Charging System Architecture

APPLYING PERFORMANCE STANDARDS TO THE AUSTRALIAN HEAVY VEHICLE FLEET

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ABSTRACT

In 1999 the National Road Transport Commission (NRTC) and Austroads initiated a major joint project to develop Performance Based Standards (PBS) for heavy vehicle regulation in Australia and New Zealand.

This paper discusses the principles that form the foundations of PBS and the process that will be followed in determining how vehicles can operate under the system. The benefits of moving to Performance Standards and the key issues in implementing a PBS approach are discussed.

A set of 25 proposed performance standards were developed against which the Australian heavy vehicle fleet was tested. Fifteen of these measures have been selected for further development and implementation.

This paper reviews a large body of work being undertaken over a three-year period. A number of reports published by the NRTC provide in-depth technical background on specific issues discussed here: policy principles, selecting and setting performance standards, and assessing computer simulation models. These papers may be accessed through the NRTC Website (www.nrtc.gov.au), where more information on the wider project may also be obtained.

1. BACKGROUND

Heavy vehicles in Australia and New Zealand are regulated predominantly by prescriptive standards that evolved over a long period and often differed between States and Territories. Through the reforms progressed by the National Road Transport Commission (NRTC) many inconsistencies have been removed. Nevertheless, some remain particularly in relation to innovative approaches to solving transport needs. Modernising regulations by moving to a nationally consistent performance based approach to regulation of heavy vehicle operations is now being considered as a voluntary optional alternative to the existing prescriptive regulations.

Under a performance-based approach to regulation, standards would specify the performance required from vehicle operations rather than mandating how this level of performance is to be achieved. In Australia and New Zealand this approach to regulation has been adopted in other sectors, such as occupational health and safety and food standards, and is now well established as the approach preferred by regulatory review agencies.

The performance based standards (PBS) project seeks to align regulatory requirements more closely with the realities of how vehicles perform, how they are driven and operated, and the characteristics of the road network. It aims to improve productivity, increase safety, and to better protect the infrastructure.

Traditionally, heavy vehicles have been regulated by tightly defined prescriptive limits, such as mass and size limits, which provide little scope for innovation. This method of control is very crude, with no guarantees that vehicles meeting the current requirements do not have relatively poor performance. Many of the intrinsic safety issues such as stability, handling and controllability, high-speed tracking, and gradeability are not evaluated and are only indirectly controlled, if at all.

Under PBS, the interactions of vehicles with the roads they will be used on are taken into account more explicitly. In determining whether a specific vehicle can operate on a particular road, the vehicle's capabilities and the relevant road standards and traffic conditions can be examined jointly to decide whether the operation meets the performance standards.

A wide range of performance measures has been developed over many years of research for the evaluation of heavy vehicle performance. A key selection of these performance measures has been considered and found to be both practical and relevant for the evaluation of the Australian heavy vehicle fleet. Accident studies have found relationships between these measures and crash risk, providing a sound basis on which to set minimum performance levels for the key performance measures.

The following are the key objectives and benefits that can be attributed to a PBS approach to the regulation of heavy vehicles:

- increased productivity and innovation in vehicle design and operation
- improvements in road safety, traffic operations and asset management (infrastructure)
- a national basis for the regulation of heavy vehicles
- consistency in the application of assessment techniques that are performance based
- better matching of the capabilities of vehicles and the road system; and
- consistency in permitting local and specific-use vehicles.

In defining the project, the NRTC and Austroads established the following six inter-related phases:

- | | |
|----------|---|
| Phase A: | <i>Performance Measures and Standards</i> – identifying the appropriate performance measures and standards and surveying the performance of the current heavy vehicle fleet. |
| Phase B: | <i>Regulatory and Compliance Processes</i> – establishing a regulatory system in which PBS can operate as a seamless national alternative to existing prescriptive regulations including national compliance and enforcement arrangements. |
| Phase C: | <i>Guidelines</i> – preparing guidelines detailing the procedures and processes for the consistent application of PBS. |
| Phase D: | <i>Legislation</i> – developing the legislative arrangements for PBS to operate as an alternative to prescriptive regulations. |
| Phase E: | <i>Case Studies</i> – assembling work previously conducted and demonstrating the practical application of PBS to nationally agreed priorities. |
| Phase F: | <i>Implementation</i> – putting in place the necessary legislative and administrative systems to allow PBS to operate nationally and providing the training and information to support these changes. |

A number of reports have been prepared as part of Phase A of the project. These are listed in the bibliography. Together with the report *Performance Characteristics of the Australian Heavy Vehicle Fleet (The Fleet Report)*, and the *Regulatory Impact Statement*, they will form the completion of Phase A of the PBS project. Phase A is the main focus of this paper.

The NRTC budget for completion of the PBS Project to the stage of final recommendations to ministers is expected to be approximately A\$2 million. Additionally parts of the project are being undertaken and funded by state road agencies.

2. THE NEED FOR A PERFORMANCE-BASED APPROACH

The main reasons for investigating performance-based approaches to heavy vehicle regulation are that:

- road transport is a vital component of the Australian economy and consequently any improvements to regulation that PBS can provide are significant;
- there is continuing pressure to improve the safety and amenity of heavy vehicles;
- there is little room for further wholesale relaxation of prescriptive standards, as has occurred in the past.

The search for regulatory solutions that will support Australia's high and growing dependency on road freight is critical to improving Australians' standard of living and the nation's economic wellbeing. Large increases in the size of the road freight task are forecast (NRTC, 2001d), highlighting the importance of continued efforts to improve the overall safety, efficiency and fairness of the road transport system. It is unlikely that these trends can be maintained without the adoption of mechanisms that promote innovation and provide the flexibility for transport operators to improve productivity, where this has no detrimental impact on safety or road infrastructure. In the road transport sector this includes a more sophisticated approach to heavy vehicle regulation.

At the same time as providing for innovations in the road transport sector, governments must also meet the community's expectations for improved health, safety and quality of life (NRTC, 2001d).

The introduction of PBS is expected to:

- encourage innovation;
- provide a better match between vehicles and roads;
- increase regulatory transparency by providing a more consistent and more rational regulatory approach;
- improve performance (by providing better controls on safety and infrastructure wear); and
- improve compliance.

3. TERMS AND CONCEPTS

Performance standards specify the outcomes required of vehicle operations, but leave open the ways in which these outcomes are achieved. For example, *performance standards* might specify that a vehicle must be able to travel along the road and negotiate turns without tipping over or intruding on the road space of other road users. They might specify how well the vehicle should be able to stop and how much road wear it can cause. In comparison, *prescriptive standards* would specify the dimensions and mass of a vehicle to achieve these outcomes.

Each *performance standard* assigns a numerical limit to a *performance measure*, defining a boundary between what is acceptable and unacceptable. A *performance measure* quantifies how a vehicle performs for a specific circumstance or manoeuvre. The manoeuvre and the method of measuring the vehicle's performance must be specified in detail, in order for the *performance measure* to be objective.

For example, low-speed off-tracking is a *performance measure* for the tendency of the rear trailer or rear axle to track inside the path taken by the steering axle in low-speed turns. It is measured at a specific speed, angle of a turn and so on. Without this, comparisons between vehicles and tests would be meaningless.

4. PBS PRINCIPLES

Five principles form the basis of the policy framework for the PBS proposal. These principles, developed through a stakeholder consultation process, have been agreed by the Australian Transport Council (ATC). The ATC consists of the Australian Commonwealth, State and Territory Transport Ministers.

They are:

Principle One:

Performance-based standards will be a national system of regulating heavy vehicle safety and infrastructure impacts that operates as an alternative to existing prescriptive regulations.

- Some existing heavy vehicle regulations will continue to apply, but PBS will provide alternative controls in the main areas of mass, dimension and configuration controls.
- If an operator chooses to operate outside PBS, the relevant prescriptive standards must be satisfied.

Principle Two:

Performance standards should be matched to road and traffic conditions.

- Consequently, standards established for some performance measures will differ between a limited group of road classes, based on variations in road and traffic conditions that characterise each class of roads.
- Road authorities will determine which classification applies to the roads they manage.

Principle Three:

Compliance with performance based standards will be ensured by nationally consistent and practical methods that are based on certifying that vehicle-related features meet the performance standards and identifying simple operating conditions.

- Where warranted, vehicle operators will be required to be accredited to demonstrate that they comply with these conditions.

Principle Four:

All parties in the transport chain will be held responsible for factors in their control that ensure Performance Standards are achieved and maintained.

Principle Five:

An approval process will apply to each proposed PBS operations. Features of the approval process will be:

- Anyone should be able to apply for a vehicle, component or operation to be approved under the performance based standards.
- Lower cost ways of accessing the benefits of PBS must be available to smaller operators and those with fewer technical resources.
- The approval process and compliance arrangements should provide vehicle operators with the flexibility to choose at different times whether to operate under the existing prescriptive regulations or the performance based standards.
- Procedures should be incorporated to provide for mutual recognition of PBS approvals nationally.
- All performance assessors will need to be accredited to ensure their assessments are consistent and of sufficient quality.

The PBS system will apply to both general access and access to limited routes/regions. The approval process will involve:

- identifying which set of performance standards applies for the circumstances of the proposal, eg the roads to be accessed;
- assessing whether these standards are met by the proposal and identifying simple conditions to ensure they will be met on-road;
- certifying that the vehicle(s) to be used are consistent with the proposal assessed; and
- recording the approval and any operating conditions.

5. DEVELOPMENT OF PROPOSED PERFORMANCE MEASURES

As a first step in establishing an appropriate set of performance standards under a PBS approach to the regulation of heavy vehicles in Australia, the entire field of potential performance measures relevant for heavy vehicles were determined and documented (NRTC, 1999a; NRTC, 1999b). The next step in the process was to select an initial set of potential regulatory performance measures using the methodology described in (NRTC, 2000a). Several steps were used in the selection process, detailed in (NRTC, 2001a), to reduce the more than 100 potential performance measures to a set of 25 that cover safety and infrastructure related issues.

(NRTC, 2001c) provides definitions for the set of 25 potential regulatory performance measures that emerged from (NRTC, 2001a) reviewed by stakeholders (Appendix A). The report (NRTC, 2001c) also specifies an initial set of performance levels based on the review of the literature (available records) by the project team. The potential regulatory performance measures when combined with the associated performance levels lead to an

initial set of potential regulatory performance standards (Appendix B). Appendix B also indicates for each measure whether physical testing or computer simulation methods are available for determining vehicle performance.

The potential regulatory performance standards that are considered to have been developed to a useable level were tested against the Australian heavy vehicle fleet using computer modelling. A set of 139 generic vehicle combinations which characterise the Australian Heavy vehicle fleet were developed for this purpose. This process and the results reported in (NRTC, 2002), the Fleet Report, are discussed below.

6. THE FLEET REPORT

6.1 Overview of the Fleet Report

The Fleet Report aimed to do two things:

- assess the performance of the existing fleet of heavy vehicles in comparison to the proposed measures; and
- review the performance standards in light of additional work, the results of the fleet assessment and stakeholder responses to previous work.

The proposed performance measures were reviewed in light of the fleet performance results from the simulations, the further findings generated by the study, and the feedback received from stakeholders. Further analysis through correlation was also undertaken The Fleet Report to reduce the total number of measures that would cover all the areas of interest to ensure safety performance was not compromised and infrastructure impacts would be acceptable.

After rigorous review, a total of twenty performance measures survived the process, which constitute the proposed final set, summarised in Table 1.

Of the twenty performance measures presented, fifteen have been developed to a stage where they are considered to be both useable and suitable for performing heavy vehicle assessments for regulatory purposes. In Table 1, the fifteen measures that are proposed to remain under consideration are presented in normal text. Those considered to be relevant but not yet developed to a stage where they can be fully implemented – requiring further research and development – are underlined and shown in italics.

Further details of the proposed set of performance standards, including details of the way in which they would be assessed and the standards to be required, are set out in Appendix C, reproduced from the Fleet Report.

In the review and evaluation process two new performance measures were proposed, Acceleration Capability and Maximum Effect Relative to Reference Vehicle (or MERRV). These are designed to replace, respectively, Intersection Clearance Time and Maximum Bridge Stress.

A number of relatively minor revisions were made to several of the performance standards, as detailed in the Fleet Report. However, a major revision was made to the performance level specification for Rearward Amplification. In its revised form it links the performance requirement to the rollover stability of the critical rear-most, roll-coupled unit — providing a very transparent and tangible safety outcome. This leads to a significantly higher proportion of the fleet meeting the rearward amplification standard, particularly for truck/trailers and road trains.

From the original set of performance measures the following two, Load Transfer Ratio and High-Speed Steady-State Offtracking, were determined to be redundant, and they were removed from further consideration. Load Transfer Ratio being highly correlated in the fleet with Static Rollover Stability, Rearward Amplification, and the interaction between the two, whereas High-Speed Steady State Offtracking and Tracking Ability were found to be highly correlated with each other, essentially providing the same information.

A comprehensive series of parametric studies was conducted on a set of mid-range vehicles, which were selected from the main set. Briefly, parameters found to be highly significant were: engine power/torque, driveline gear ratio, centre-of-gravity (CG) height, axle loads, wheelbase dimensions (trailers in particular), tyre cornering stiffness and speed. The less significant parameters included coupling rear overhang and suspension roll stiffness.

This analysis provides an indication of how vehicles can achieve additional productivity while meeting the necessary safety and infrastructure protection criteria. A table summarising these results is included as Appendix D, an extract from the Fleet Report.

For example, if an articulated vehicle needs to improve its steer tyre friction demand in order to take advantage of the productivity increases potentially available under PBS, it may be able to do so by increasing the prime mover wheelbase and decreasing drive axle group spacings. Alternatively, if an operator wishes to utilise increased axle loads, the vehicle may need to have spare 'capacity' in its current level of performance against some of the performance standards.

6.2 Further Issues to be Considered

Several performance measures were identified as requiring further development. The further issues are summarised below.

- i. Overtaking Time: Overtaking effects needed to be considered because this is clearly a safety issue on two-lane two-way rural roads. However, additional work will be required to develop this important safety-related performance measure to a useable form. It is considered this issue could be addressed in the context of the road environment rather than as a vehicle performance issue *per se*, through the judicious use of overtaking lanes, for example. Some background material presented in The Fleet Report, builds on this notion.
- ii. Ride Quality (Driver Comfort): There is currently insufficient information to determine an acceptable performance level for this performance measure. Discussions with industry identified that this performance measure was important and should be retained, even if there is insufficient data at this stage to justify the setting of performance levels. Further research is required before an acceptable performance level can be determined for this measure.
- iii. Rearward Amplification: There appear to be two interpretations by industry of the requirement of the SAE J2179 lane change test; one that correctly requires the lateral acceleration test value of $\pm 0.15g$ to be achieved at the centre of the steer axle, and the other - the more common but incorrect - requiring lateral accelerations of $\pm 0.15g$ to be achieved at the hauling unit CG. The second is based on the original definition of rearward amplification but it is not consistent with SAE J2179 and should not be used. The consequences of these different interpretations, while not immediately obvious, have very subtle (and potentially serious) implications, as discussed in the report. The correct version of SAE J2179 should be promoted and used.
- iv. Handling Quality (Oversteer/Understeer): Few fleet vehicles were found to comply with the proposed performance requirements for heavy vehicle handling. However, an estimated 20% of the fleet vehicles considered were found to possess handling qualities that are similar to those of the vehicle identified as having the worst handling in the recent study into heavy vehicle handling funded by the Federal Office of Road Safety (FORS). As the implications of this finding are potentially serious, it warrants urgent further investigation.
- v. Braking Stability in a Turn: The standard in its present form appears to be technically feasible, and the fleet results suggest performance compliance would be high. However, before this standard can be recommended, field-testing of a range of vehicles is suggested, both to confirm the findings of the fleet analysis and to determine if there are any practical issues that may prevent the use of ABS on a much wider range of Australian fleet vehicles and operating conditions.
- vi. Gross Mass per Standard Axle Repetition: The standard in its present form, with a performance level of 8.4t/SAR applicable to vehicles operating on granular pavements with thin surfacing, appears to be acceptable and would cover the majority of combination vehicles operating on the major part of the network. However, further research will be required for other pavement types and operating environments, including development of a suitable measure for rigid pavements.

- vii. Horizontal Tyre Forces: During stakeholder consultation there was general agreement on the need for this, as controls on horizontal forces were seen as particularly important to develop. Further research was recommended aimed at measuring horizontal forces and stresses in pavements to confirm the predictions from the simulations.
- viii. Tyre Contact Pressure Distribution: This standard will not proceed at this stage because further research is required before an adequate performance level can be established. In the consultation process there was general agreement of the need for this standard and support for further research aimed at developing a performance standard is therefore recommended.

6.3 Fleet Performance

Each fleet vehicle was evaluated against each of the proposed standards using the following three broad levels of route access/road environment.

- unrestricted access to the entire network (urban and arterial);
- access to major freight routes; and
- access to remote area routes.

If the vehicle met all the performance requirements for the full set of the proposed standards then its overall performance, in the context of The Fleet Report, was considered to be acceptable.

It is important to note that the analysis presented refers to a sample of representative vehicles that were taken from the Australian heavy vehicle fleet. The overall proportion of vehicles using the roads that meet the standards will depend on the numbers of each representative vehicle actually operating on the road. This aspect of the project will be analysed and considered in much greater detail in the Regulatory Impact Statement (RIS), which will also quantify in general terms the potential benefits and costs of the proposed set of performance standards.

6.4 Unrestricted Access

Around 20% of the fleet considered met the performance standards applicable to unrestricted access. Within this group, 18% of rigid trucks (18%), 47% of prime mover and semi-trailers, 22% of truck and pig/tag-trailers and 21% of truck and dog-trailers met all the performance standards. Details of the proportion of each vehicle category that met each of the performance standards for unrestricted access are shown in Appendix E, an extract from the Fleet Report.

The performance requirements that were set for gradeability (maximum grade), low-speed offtracking and acceleration capability associated with unrestricted access to all roads in Australia, could not be met by almost all the B-doubles, the B-triple, the A-double, and the A-triple and AAB-quad road trains. This was not an unexpected result, given that all these vehicles operate under access restrictions at present.

Most of the rigid trucks, buses/coaches, and various configurations of truck/trailer failed to comply with the GM/SAR performance requirement. This standard is based on a performance level of 8.4t/SAR, applicable to vehicles operating on granular pavements with thin surfacing, and appears to be acceptable, covering the majority of combination vehicles operating on the major part of the network. Consequently, these vehicles may not be able to vary from existing prescriptive mass limits, although they may be able to vary dimensions if they meet the safety-related performance standards.

Analysis of weigh-in-motion data indicates that the majority of smaller vehicles—two and three axle rigid trucks and buses—often operates empty or well below their rated legal loads. The proposed standard in its current form would therefore have little practical impact on these fleet vehicles. The exception to this would be smaller vehicles operating in high-density bulk haul applications, where vehicles may be loaded to their rated gross capacity. The low proportion of truck and pig/tag trailers meeting this standard also appears to be due to the low GM/SAR values. However, for the truck and dog-trailer combinations the low overall compliance appears to be the outcome

of an unfortunate mix of only moderate performances for several measures; gradeability, static and dynamic stability as well as the infrastructure measures.

6.5 Major Freight Routes

Overall 27% of the fleet met performance standards for access to major freight routes. Details of the proportion of each vehicle category that met each of the performance standards for major freight routes are shown in Appendix F. Slightly more prime mover and semi-trailers met these requirements compared to those applying to unrestricted access, up from 47% to 53%.

For the B-doubles, 30% were found to meet the standards due to the slightly relaxed requirements on gradeability, acceleration capability and low-speed offtracking. However, gradeability (maximum grade) was the main reason for the somewhat low overall proportion of B-doubles meeting the standards. Reducing the gradeability requirement from 20% to 16% would significantly improve the overall proportion of B-doubles meeting the standards, raising it from 30% to 70%.

It would also increase the proportion of A-doubles meeting these standards to 83%, and elevate the overall proportion of the fleet meeting the standards from 27% to 41%. However, The Fleet Report states that the 20% gradeability performance requirement for major freight route access is based on current practice and that it should be retained.

6.6 Road Train Routes

On road train routes the performance measures controlling vehicle access were found to be the static and dynamic stability measures and the infrastructure/pavements measures. Overall 42% of current vehicles were able to meet these standards. This increase is due mainly to more of the fleet vehicles being able to meet the low-speed longitudinal and directional performance standards. Details of the proportion of each vehicle category that met each of the performance standards for road train routes are shown in Appendix G.

For the A-triple and AAB-quad road trains, the three measures having the greatest influence on the proportion of the vehicles meeting the standards are rearward amplification, high-speed transient offtracking and horizontal tyre forces.

6.7 Route-Specific Access

The Fleet Report suggests there is little scope at present for relaxing the performance levels for these three key measures—namely, rearward amplification, high-speed transient offtracking and horizontal tyre forces—and improvements in performance to achieve the desired safety outcomes would need to come from design-specific changes. Alternatively, route-specific performance levels for the key measures could be determined based on the prevailing local conditions and applied by the Road Agency.

It is important to also note that route specific requirements may be different to the generic set of conditions assumed in the analysis of the generic fleet vehicles. For specific, or tightly managed, applications the precise conditions on the route/environment will be well defined in most cases.

Also, for operations on some routes it may not be necessary to apply all the standards, and a selection of applicable measures may be able to be made by the Road Agency, together with a set of route and operation specific performance levels. These can be adjusted to suit the specific application.

6.8 Conclusions from the Fleet Report

The attached tables from the report provide the basis for the conclusions discussed above. From the information, it can be concluded:

- Large numbers of existing vehicles already meet the performance standards proposed for unrestricted access to the entire road network.

- A greater number of existing vehicles meet the requirements proposed for operation on either major freight routes or remote area routes.
- The study of parametric effects indicates a range of design features that, with adjustment, will enable new or modified vehicles to meet the standards.

Analysis of the costs and benefits of retaining each proposed performance measure has not been completed at the time of writing. It is expected the implementation costs of some measures will exceed the benefits in terms of productivity and safety; these performance measures will not be used.

The NRTC expects a number of existing vehicles to be able to operate in the PBS regime without modification, and many more should meet the performance standards with some design changes. The operator of these vehicles could have access to productivity gains at minimum cost.

7. COMPARISON OF MODELLING SYSTEMS

The initial potential performance standards were tested against the Australian heavy vehicle fleet as mentioned in chapter 6. The performance of the heavy vehicle fleet will be primarily determined using computer-based modelling techniques.

Some stakeholders have expressed the concern that performance predictions from computer-based modelling packages may not be reliable and may substantially differ with software packages and with the computer simulation practitioners that use them. Given that PBS is intended to encourage and foster innovation in road transport, and that computer-based modelling is expected to play a central role in both the development and initial demonstration of innovative vehicles, the concerns that were expressed by stakeholders needed to be addressed promptly and as a priority.

A report "Comparison of Modelling Systems for Performance-Based Assessments of Heavy Vehicles" addresses these concerns and, additionally, resolves calibration issues associated with computer-based modelling in a way that is transparent and open to scrutiny. This is an essential step in building stakeholder confidence in the use of computer-based modelling and in its application to the regulation of heavy vehicles in Australia using performance based standards. The report is on the NRTC Website.

Computer-based models of two vehicles were created in the course of this project by two Consultants using three separate computer-based modelling packages; ADAMS, UMTRI's constant velocity Yaw/Roll program and AUTOSIM. Comprehensive input datasets were developed for a non-descript B-double and a non-descript truck/trailer. The same datasets were supplied to each Consultant and identical simulations were performed using the same test manoeuvres comprising a pulse steer, step steer, standards SAE lane change and a low-speed 90 degree turn.

Time histories of a wide range of variables from the simulations were compared as well as numeric values from a selection of performance measures. For the more stable of the two vehicles models, the B-double, the time histories from the pulse steer and step steer simulations were almost indistinguishable showing excellent agreement between all three modelling packages. Agreement in the outputs from the simulations in all manoeuvres was generally better than 10% for the performance measures considered. These were marginally influenced by the characteristics of the steer controller in the lane change manoeuvre though agreement was still generally better than 10%.

The truck/trailer model, representing a less stable and dynamically more active vehicle compared to the B-double, produced larger but acceptable amounts of variation between simulations in the pulse and step steer simulations and low-speed 90° turn. However, in the SAE lane change the differences between the models were much too large as a result of the greater deviations in the path followed. To achieve acceptable agreement in the lane change manoeuvre between models a deviation from the desired path not greater than $\pm 30\text{mm}$ is required and is recommended. This is significantly less than the current recommended tolerance of $\pm 150\text{mm}$ specified by the Society of Automotive Engineers (SAE).

Simulations that provide a direct measure of only the vehicle responses to precisely defined steer inputs generally lead to more consistent results than simulations that require steer controllers and closed loop path following. When there is a choice, open loop manoeuvres should be selected in preference to closed loop manoeuvres that require the use of steer controllers.

8. REGULATORY AND COMPLIANCE PROCESSES

Phase B forms the second major part of the PBS Project, addressing regulatory and compliance issues. The objective of Phase B – Regulatory and Compliance Processes is to establish a regulatory system in which performance based standards can operate as a seamless national alternative to existing prescriptive regulations, utilising common national compliance and enforcement arrangements. While PBS is to be an optional alternative to the current regulations, it will rely on many of the same mechanisms and processes used to administer the existing prescriptive rules. The regulatory and compliance systems needed to implement the nationally agreed performance standards are being designed with this in mind.

To date five Phase B reports have been published dealing with a selection of specific aspects of the issues in establishing PBS as an optional, nationally consistent alternative to the current prescriptive regulations on mass, dimension and configuration.

9. COMPARISON OF BENEFITS AND COSTS

At the time of writing a Regulatory Impact Statement (RIS) is being prepared to assess the costs and benefits of the proposed set of performance standards. The purpose of the RIS is to analyse the policy impacts of the range of technical proposals discussed in this paper. This comprehensive analysis addresses fleet, operational and infrastructure issues such as:

- Composition of the fleet
- Potential PBS vehicles
- Take up rate
- Vehicle replacement savings
- Productivity improvements
- Costs of the PBS standards
- Assessment costs
- Additional vehicle construction costs
- Savings in pavement costs
- Additional vehicle operating costs

Preliminary indications from work on the RIS suggest net present values between about A\$100 million and A\$300 million depending on cost scenarios.

This RIS will be completed by April 2002. A further RIS will then be undertaken on options for assessment, compliance and enforcement processes.

10. SUMMARY AND CONCLUSIONS

This paper has outlined the structure and technical development of a major project to implement performance standards in Australia on a national basis. At the time of writing the development of the technical proposals has been largely completed. Assessment of the Australian heavy vehicle fleet against each of the proposed performance standards indicates that safety and productivity gains can be achieved with resulting net economic benefits. Therefore, the NTRC is progressing to put in place a comprehensive performance-based alternative to the heavy vehicle mass and dimension regulatory system currently used in Australia. The project is now moving from technical development to implementation.

Implementation of agreed performance standards will primarily use computer-based modelling techniques. Work undertaken to compare computer models and the means of using them indicates that the specifications for the use of models require special attention in order to ensure consistent results. International standardisation on some aspect of heavy vehicle computer simulation modelling would be an advantage to all agencies and practitioners who use these methods

Some areas of heavy vehicle performance, for which would be desirable to have performance standards as part of a comprehensive performance-based regulatory system, have not yet been sufficiently developed for use in a regulatory environment. Some other potential performance measures would require broader industry support than is currently the case. These areas suggest opportunities for research to develop new or enhance performance measure and to achieve a more international agreement.

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TABLES & FIGURES

Table 1 – Summary of Proposed Final Set of Performance Measures

#	Performance Measures
SAFETY RELATED	
Longitudinal Performance (Low Speed)	
1	Startability
2	Gradeability
3	Acceleration Capability
Longitudinal Performance (High Speed)	
4	<u>Overtaking Time</u>
5	Tracking Ability on a Straight Path
6	<u>Ride Quality (Driver Comfort)</u>
Directional Performance (Low Speed)	
7	Low-Speed Offtracking
8	Frontal Swing
9	Tail Swing
10	Steer Tyre Friction Demand
Directional Performance (High Speed)	
11	Static Rollover Threshold
12	Rearward Amplification
13	High-Speed Transient Offtracking
14	Yaw Damping Coefficient
15	<u>Handling Quality (Understeer/Oversteer)</u>
16	<u>Braking Stability in a Turn</u>
INFRASTRUCTURE RELATED	
Pavements	
17	Gross Mass per Standard Axle Repetition
18	Horizontal Tyre Forces
19	<u>Tyre Contact Pressure Distribution</u>
Bridges	
20	Maximum Effect Relative to Reference Vehicles

APPENDICES

APPENDIX A – INITIAL SET OF PERFORMANCE MEASURES

#	POTENTIAL PERFORMANCE MEASURE	DEFINITION
1	Static roll stability	The steady-state level of lateral acceleration that a vehicle can sustain during turning without rolling over.
2	Rearward amplification	Degree to which the trailing unit(s) amplify or exaggerate lateral motions of the hauling unit.
3	Load transfer ratio	The proportion of vertical load imposed on the tyres on one side of a vehicle unit that is transferred to the other side of the vehicle unit during a standard lane change manoeuvre.
4	High-speed transient offtracking	The lateral distance that the last-axle on the rear trailer tracks outside the path of the steer axle in a sudden evasive manoeuvre.
5	High-speed offtracking steady-state	The lateral distance that the last-axle on the rear trailer tracks outside the path of the steer axle in a high-speed steady turn.
6	Yaw damping	The rate at which "sway" or yaw oscillations of the rearmost trailer decay after a short duration steer input at the hauling unit.
7	Tracking ability on a straight path	Amount of variation in the lateral position of the trailing unit (last trailer) measured relative to the path or track followed by the hauling unit (rigid truck or prime mover).
8	Braking stability (in a straight line)	The vehicle's ability to stay within a traffic lane under heavy braking on a straight path.
9	Braking stability (in a turn)	Amount of loss of control when braking in a turn.
10	Handling (understeer/oversteer) quality	No change
11	Low-speed offtracking	Maximum distance that the rear axle of a vehicle or combination tracks inside the path taken by the steering axle in a low speed turn.
12	Frontal swing	The maximum lateral displacement between the path of the front outside corner of the vehicle (or vehicle unit) and the outer edge of the front-outside steered wheel of the hauling unit during a small-radius turn manoeuvre at low speed.
13	Tail swing	The maximum lateral distance that the outer rearmost point on a vehicle moves outwards, perpendicular to its initial orientation, when the vehicle commences a small-radius turn at low speed.
14	Friction demand (steer tyres in corner)	The maximum friction level demanded of the steer tyres of the hauling unit in a tight-radius turn at low speed.
15	Ride quality	The level of vibration that a vehicle's driver is exposed to during a working shift that leads to reduced comfort and decreased proficiency, and contributes to fatigue.
16	Startability	The maximum uphill gradient, expressed as a percentage, on which the vehicle is capable of starting forward movement from rest.
17	Gradeability	The maximum uphill gradient, expressed as a percentage, on which the vehicle can climb at a specified constant speed.
18	Intersection clearance time	The time taken for the rear of the vehicle to clear a given intersection (either straight through or turning) with the vehicle starting from rest with its front immediately behind the intersection stop line.
19	Overtaking time	The time taken for another vehicle to safely overtake the vehicle.
20	Payload mass per ESA	<i>This measure replaced by:</i> <u>Gross Mass per Standard Axle Repetition</u> The Gross Mass (GM) of a heavy vehicle divided by the Standard Axle Repetitions (SARs) applied to the pavement by a single pass of the vehicle.

#	POTENTIAL PERFORMANCE MEASURE	DEFINITION
21	Horizontal tyre forces	Degree to which horizontal forces are applied to the pavement, primarily in a low-speed turn and at constant speed on uphill grades, by the tyres of multi-axle groups (drive-axle group tyres in particular) and the effect on remaining pavement life.
22	Tyre contact pressure distribution	The maximum local vertical stress under a tyre's contact patch for a given vertical tyre load and tyre inflation pressure.
23	Upper bound on axle/axle-group load	<i>These two performance measures have been replaced by the performance measure Gross Mass per Standard Axle Repetition (refer above under #20).</i>
24	Upper bound on GVM/GCM	
25a	Bridge Loads (Axle spacing mass schedule)	<i>These two measures have been combined into the following single performance measure:</i>
25b	Critical design vehicle (bridges)	<u>Maximum Bridge Stress</u> The maximum stress that a bridge can sustain under repeated loading without incurring damage.

APPENDIX B – INITIAL SET OF PERFORMANCE STANDARDS

PERFORMANCE MEASURE	TEST SPECIFICATION	PERFORMANCE LEVEL	TEST METHOD	
			Physical Testing	Calculation or Computer-Based Modelling
SAFETY RELATED				
Longitudinal Performance (Low Speed)				
Stability	Ability to commence forward motion on specified grade.	<ul style="list-style-type: none">• Not less than 15% for unrestricted access to the entire network;• Not less than 10% for arterials and major freight routes; and• No less than 5% for remote areas.	✓	✓
	Ability to maintain forward motion on specified grade.	1) Low-Speed Environment (maximum grade that the vehicle can climb at any speed)	✓	✓
Gradeability		Unrestricted access to the entire network: 25%		
		Urban roads of higher standard: 20%		
		Urban roads in remote areas: 8%		
		2) High-Speed Environment (minimum speed on a 1% gradient)		
		Unrestricted access to the entire network: 80km/h		
		Remote areas: 50km/h		
Intersection Clearance Time	Time required travelling a distance of 50m starting from rest to clear an intersection on a road with no grade. If location specific then suitable test conditions required.	No more than 12s for unrestricted access to the road network; No more than 15s for arterials and major freight routes; No more than 25s for routes designated for long combination vehicles. (May be location specific and require a separate performance level).	✓	✓
Longitudinal Performance (High Speed)				
Overtaking Time	Test specifications specific to road and traffic conditions.	Specific to delay caused to other road users, which in turn is dependent on route characteristics and traffic volumes. Table 2 in the body of the report provides a guide for various road classes.	✓	✓
Tracking Ability on a Straight Path	Traverse 1000m road segment at two test speeds (60 and 90km/h), road roughness in each wheelpath 4.0m/km IRI (±0.4m/km) and average cross-slope 4% (±0.4%). Vehicle laden.	Specified in terms of required lane width. If route specific requirement do not exist then the following is proposed: In the range 3.1 to 3.5m for urban arterials; no greater than 3.5m on rural and regional roads; in the range 3.5 to 3.7m on national highways and freeways; no greater than 3.7m in remote areas.	✓	✓

TEST METHOD

PERFORMANCE MEASURE	TEST SPECIFICATION	PERFORMANCE LEVEL	Physical Testing	Calculation or Computer-Based Modelling
Ride Quality	Traverse 1000m road segment at two test speeds (100 and 60km/h), road roughness in each wheelpath 4.0m/km IRI (± 0.4 m/km). Vehicle laden and unladen.	Performance level required. However, vehicles can be compared on a relative basis using the procedures outlined in British Standard BS 6841, or International Standard ISO 2631, to estimate the frequency weighted RMS vibration.	✓	✓
Braking Stability on a Straight Path	As required by and specified in ADR35/01.	The ability to stay within a 3.5m wide lane.	✓	✓
Directional Performance (Low Speed)				
Low-Speed Offtracking	Centre of steer axle to follow a path comprising a straight entry segment that is tangent to a 11.25m radius 90° circular arc followed by a straight exit segment. Vehicle speed is 10km/h.	Maximum width of the swept path: 5m for local roads, 7.4m for arterial roads, 10.1m for major freight routes, and 13.7m for road train areas.	✓	✓
Frontal Swing	Same as for low-speed offtracking	Not greater than 1.5m for unrestricted access to the entire road network.	✓	✓
Tail Swing	Same as for low-speed offtracking	Not greater than 0.5m.	✓	✓
Steer Tyre Friction Demand in a Low-Speed Turn	Same as for low-speed offtracking	No greater than 80% of the maximum available tyre/road friction. Table 4 in the body of the report provides friction values for a range of surfaces.	not demonstrated	✓
Directional Performance (High Speed)				
Static Rollover Threshold	Procedures defined in SAE J2180 (see Society of Automotive Engineers, 1993a). If by computer-based modelling then 100m radius circular path, centre of steer-axle follows path, test speed slowly increased from 60km/h until rollover occurs.	For road tankers and buses at least 0.40g, for all other heavy vehicles at least 0.35g.	✓	✓
Rearward Amplification	Procedures defined in SAE J2179. Lane change manoeuvre - test speed 88km/h, 1.46m lateral displacement, 61m manoeuvring length, 0.15g peak lateral acceleration (see Society of Automotive Engineers, 1993).	Not greater than 2.0	✓	✓
Load Transfer Ratio	Same as for rearward amplification.	Not greater than 0.6. Where maximum speed is less than 75km/h a load transfer ratio not greater than 0.75 may be considered acceptable on a provisional basis.	not demonstrated	✓

PERFORMANCE MEASURE	TEST SPECIFICATION	PERFORMANCE LEVEL	TEST METHOD	
			Physical Testing	Calculation Computer-Based Modelling or
Yaw Damping	Application of a 3.2° (half sine) steer angle pulse at the road wheel over a 0.1s period, test speed 100km/h.	Not less than 0.15	✓	✓
High-Speed Transient Offtracking	Same as for rearward amplification	Not greater than 0.8m	✓	✓
High-Speed Steady-State Offtracking	39.3m radius circular path, test speed 100km/h, centre of steer axle follows path.	No greater than 0.3m for unrestricted access to the entire network; no greater than 0.5m for arterials and major freight routes; and no greater than 0.7m for low-volume roads in remote areas.	✓	✓
Handling (Understeer/Oversteer)	Quality As specified in El-Gindy, Woodroffe and White (1991), or equivalent. Vehicle speed of 100km/h, the understeer coefficient, K_u , is evaluated over the range 0.15g to 0.3g.	Three-point measure. First point (evaluated at $a_y = 0.15$) $0.5 < K_u < 2.0$ deg/g; second point (transition from understeer to oversteer) $a_y > 1.2g$; third point (evaluated at $a_y = 0.3$) $K_u >$ critical understeer coefficient.	✓	✓
Braking Stability in a Turn	As specified in US FMVSS 121. The vehicle is stopped from an initial speed of 48.3km/h or 75 percent of the maximum drive through speed, whichever is less, on a 152.4m radius curve with a wet surface having a peak friction coefficient of 0.5. Both laden and unladen conditions considered.	The vehicle, when stopped four consecutive times, must stop at least three times within a 3.66m wide lane.	✓	✓
INFRASTRUCTURE RELATED				
Pavements				
Gross Mass per Standard Axle Repetition	Laden vehicle, pavement type and configuration specific.	For granular pavements with thin surfacings 8.3t/SAR for all heavy vehicles.	✓	✓
Horizontal Tyre Forces	Same as for low-speed offtracking, and separately on uphill grades of 2% and 5%	Pavement wear for PBS vehicle for a particular freight task no greater than for the same task being performed by current common vehicles.	not demonstrated	✓
Tyre Contact Pressure Distribution	Laden vehicle, travel speed up to 100km/h.	Further research required to establish a performance level.	✓	not demonstrated yet
Bridges				
Maximum Bridge Stress	Representative loads imposed on the bridge by the proposed vehicle.	A load factor of at least 1.8 for general heavy vehicles. For vehicles carrying indivisible loads a suitable load factor remains to be determined.	✓	✓

APPENDIX C - EXTRACT FROM FLEET REPORT - SUMMARY OF PROPOSED PERFORMANCE STANDARDS

#	PERFORMANCE MEASURE	TEST SPECIFICATION	PERFORMANCE LEVEL
1	Startability	Ability to commence forward motion on specified grade.	<ul style="list-style-type: none"> • Not less than 15% for unrestricted access to the entire network. • Not less than 10% for arterials and major freight routes; and • No less than 5% for remote areas.
2	Gradeability	Ability to maintain forward motion on specified grade.	<p>1) Low-Speed Environment (max. grade that the vehicle can climb at any speed)</p> <p>Unrestricted access to the entire network: 25%</p> <p>Arterials: 20%</p> <p>Remote areas: 8%</p> <p>2) High-Speed Environment (min. speed on a 1% gradient)</p> <p>Unrestricted access to the entire network: 70km/h</p> <p>Remote areas: 60km/h</p>
3 ^b	Acceleration Capability	Ability to accelerate either from rest or to increase speed (no grade).	Performance requirement for unrestricted access, access to arterials and major freight routes, and access to road train routes, as specified in the distance/time charts shown in Fig 4(a) of Section 5.3.1.3 of main body of the report.
4 ^c	Overtaking Time	To be addressed in the context of the road environment rather than as a vehicle performance issue - as detailed in Section 5.3.2.1 of main body of report.	Specific to delay caused to other road users, which in turn is dependent on route characteristics and traffic volumes. Further work required as detailed in Section 5.3.2.1 of main body of report.
5	Tracking Ability on Straight Path	Traverse a 1000m road-segment at a test speed of 100km/h (or the highest speed attainable); road roughness in each wheelpath of at least 4.0m/km IRI and an average cross-slope of at least 3.0%. Vehicle laden.	Specified in terms of required lane width. If route specific requirements do not exist then the following is proposed: in the range 3.1 to 3.5m for urban arterials; no greater than 3.5m on rural and regional roads; in the range 3.5 to 3.7m on national highways and freeways; no greater than 3.7m in remote areas.
6 ^c	Ride Quality (Driver Comfort)	Traverse 1000m road segment at two test speeds (100 and 60km/h); road roughness in each wheelpath at least 4.0m/km IRI and an average cross-slope of at least 3.0%. Vehicle laden and unladen.	Performance level required. However, vehicles can be compared on a relative basis using the procedures outlined in British Standard BS 6841, or International Standard ISO 2631, to estimate the frequency weighted RMS vibration.
7	Low-Speed Offtracking	Centre of steer axle to follow path on straight approaches to a 11.25m radius 90° circular arc. Vehicle speed is 10km/h.	Maximum width of the swept path: <ul style="list-style-type: none"> 5m for local roads, 7.4m for arterial roads, 10.1m for major freight routes, and 13.7m for road train areas.
8	Frontal Swing	Same as for low-speed offtracking	Not greater than 1.5m for unrestricted access to the entire road network.

#	PERFORMANCE MEASURE	TEST SPECIFICATION	PERFORMANCE LEVEL
9	Tail Swing	Same as for low-speed offtracking	For unrestricted access to the entire road network not greater than 0.35m on both approaches to the turn.
10	Steer Tyre Friction Demand	Same as for low-speed offtracking	No greater than 80% of the maximum available tyre/road friction. Only applicable to hauling units that feature tri-axes on the drive group.
11	Static Rollover Threshold	Procedures defined in SAE J2180 (see Society of Automotive Engineers, 1993a). If by computer-based modelling then 100m radius circular path, centre of steer-axle follows path, test speed slowly increased from 60km/h until rollover occurs.	Dangerous goods vehicles and buses: at least 0.40g. All other heavy vehicles at least 0.35g.
12	Rearward Amplification	Prescribed-path lane-change manoeuvre as defined in SAE J2179 (Society of Automotive Engineers, 1993b), or in accord with ISO 14791.	Rearward amplification no greater than 5.7 times the static rollover threshold of the rear-most roll-coupled unit.
13	High-Speed Offtracking	Transient Same as for rearward amplification	Not greater than 0.8m
14	Yaw Damping	Application of a 3.2° (half sine) steer angle pulse at the road wheel over a 0.1s period. Test speed 100km/h or maximum attainable. Alternatively, in accord with ISO 14791.	Not less than 0.15 for unrestricted access. For road trains at lower test speeds not less than defined by Eqn (3) in Section 5.3.4.4 of this report.
15	Handling Quality (Understeer/Oversteer)	As specified in <i>El-Gindy, Woodroffe and White (1991)</i> , or equivalent. Vehicle speed of 100km/h, other speeds may need to be considered. The understeer coefficient, K_u , is evaluated up to a lateral acceleration of 0.3g.	Further follow-up work is highly recommended as detailed in the main body of the report. Fleet vehicles identified as potentially having poor handling quality also should be assessed.
16	Braking Stability in a Turn	As specified in FMVSS 121.	In accord with FMVSS 121. Appears to be technically feasible but further work required as detailed in Section 5.3.4.8 of the main body of the report.
17	Gross Mass per Standard Axle Repetition	Laden vehicle, pavement type and configuration specific.	For granular pavements with thin surfacings no less than 8.4t/SALE for all heavy vehicles. Further work is required to establish suitable performance levels for other pavement types and operating environments.
18	Horizontal Tyre Forces	Same as for low-speed offtracking, and separately (if applicable) on uphill grades of 2% and 5%	Pavement wear for PBS vehicle for a particular freight task no greater than 1.8 times damage caused by conventional vehicles performing the same task.
19	Tyre Contact Pressure Distribution	Laden vehicle, travel speed up to 100km/h.	Further work is required to establish a suitable performance level.

# PERFORMANCE MEASURE	TEST SPECIFICATION	PERFORMANCE LEVEL
20 ^b Maximum Effect Relative to Reference Vehicle	Range of representative bridges considered (generic or route specific).	<p>Bending moments and shear forces to be no greater than the moments and forces induced in the bridge by Austroads BAG Reference Vehicles.</p> <p>On routes that are not satisfactory for BAG vehicles, the worse case legal vehicle operating on that route shall be used as the reference vehicle.</p> <p>For vehicles transporting indivisible freight, reference loads to be determined by the relevant Road Agency.</p>

Notes:

- a) These are considered essential but require further research and development.
- b) "Acceleration Capability" and "Maximum Effect Relative to Reference Vehicles" are designed to replace, respectively, "Intersection Clearance Time" and "Maximum Bridge Stress".

APPENDIX D – EXTRACT FROM FLEET REPORT – BROAD SUMMARY OF PARAMETRIC EFFECTS

Performance Measure	Parameter												
	Increase Engine Power/Torque	Increase Driveline Gear Ratio	Increase CG Height	Increase Axle Loads	Longer Prime Mover Wheelbase	Longer Trailer Wheelbase	Longer Dolly Wheelbase	Increase Number of Articulation Points	Increase Axle Group Spread	Increase Coupling Rear Overhang	Increase Suspension Roll Stiffness	Increase Tyre Cornering Stiffness	Increase Front Overhang
Startability	+	+	+	+	+	+	+	+	+	+	+	+	+
Gradeability a) Maximum Grade	+	+	+	+	+	+	+	+	+	+	+	+	+
b) Speed on 1% Grade	+	+	+	+	+	+	+	+	+	+	+	+	+
Acceleration Capability	+	+	+	+	+	+	+	+	+	+	+	+	+
Tracking Ability on a Straight Path	+	+	+	+	+	+	+	+	+	+	+	+	+
Low-Speed Offtracking	+	+	+	+	+	+	+	+	+	+	+	+	+
Frontal Swing	+	+	+	+	+	+	+	+	+	+	+	+	+
Tail Swing	+	+	+	+	+	+	+	+	+	+	+	+	+
Steer Tyre Friction Demand	+	+	+	+	+	+	+	+	+	+	+	+	+
Static Rollover Threshold	+	+	+	+	+	+	+	+	+	+	+	+	+
Rearward Amplification	+	+	+	+	+	+	+	+	+	+	+	+	+
High-Speed Transient Offtracking	+	+	+	+	+	+	+	+	+	+	+	+	+
Yaw Damping Coefficient	+	+	+	+	+	+	+	+	+	+	+	+	+
GM per SAR	+	+	+	+	+	+	+	+	+	+	+	+	+
Horizontal Tyre Forces	+	+	+	+	+	+	+	+	+	+	+	+	+
Max. Effect Relative to Ref. Vehicles	+	+	+	+	+	+	+	+	+	+	+	+	+

Key to Descriptors:

- ++ Denotes a significant positive effect on performance
- + Denotes a moderate positive effect
- blank Little or no influence
- Moderate negative effect
- Significant negative effect

APPENDIX E - EXTRACT FROM FLEET REPORT - FLEET VEHICLE COMPLIANCE FOR UNRESTRICTED ACCESS TO ENTIRE NETWORK (URBAN AND ARTERIAL) (%)

#	Vehicle Class	Performance Measures															
		Performance Levels															
		15	25	70	12	3.1	7.4	1.5	0.35	80	0.35	5.7	0.80	0.15	8.4	1.8	95
17	rigid trucks	100	100	100	100	100	100	100	100	100	71	100	100	100	18	29	100
6	buses/coaches	100	100	100	100	100	100	100	100	100	100	100	100	100	-	83	100
43	prime-mover and semi-trailer	100	67	100	98	100	95	100	100	100	79	100	100	100	72	93	100
23	B-double	100	-	100	35	100	4	100	100	100	78	100	100	100	78	100	100
1	B-triple	100	-	100	-	100	-	100	100	100	100	100	100	100	100	100	100
9	truck and pig/tag-trailer	100	78	100	100	100	100	100	100	100	56	56	56	100	22	89	100
14	truck and dog trailer	100	79	100	93	100	100	100	100	100	86	50	71	93	71	79	93
12	A-loobie	100	-	100	-	100	8	100	100	100	92	92	83	100	100	100	92
12	A-triple road train	-	-	25	-	100	-	100	100	100	75	17	8	83	100	92	100
2	AAB-quad road train	-	-	-	-	100	-	100	100	100	100	50	-	100	100	50	-
139	entire fleet	90	50	92	68	100	64	100	100	100	79	83	83	98	65	84	97
																	20

APPENDIX F - EXTRACT FROM FLEET REPORT - FLEET VEHICLE COMPLIANCE FOR ACCESS TO MAJOR FREIGHT ROUTES (%)

#		Vehicle Class	Performance Measures																OVERALL (%)
			Startability (%)	Gradeability (Max. Grade) (%)	Gradeability (Max. Speed on 1%Grade) (km/h)	Acceleration Capability (s)	Tracking Ability (m)	Low-Speed Offtracking (m)	Frontal Swing (m)	Tail Swing (m)	Steer Tyre Friction Demand (%)	Static Rollover Threshold (g)	Rearward Amplification (-)	High-Speed Transient Offtracking (m)	Yaw Damping Coefficient (-)	Gross Mass per SAR (vSAR)	Horizontal Tyres Forces (-)	Max. Effect Relative to Ref. Vehicle (%)	
		10	20	70	15	3.5	10.1	1.5	0.35	80	0.35	5.7	0.80	0.15	8.4	1.8	80	-	
17	rigid trucks	100	100	100	100	100	100	100	100	100	71	100	100	100	18	29	100	18	
6	buses/coaches	100	100	100	100	100	100	100	100	100	100	100	100	100	-	83	100	-	
43	prime-mover and semi-trailer	100	86	100	100	100	100	100	100	100	79	100	100	100	72	93	100	53	
23	B-double	100	39	100	100	100	100	100	100	100	78	100	100	100	78	100	100	30	
1	B-triple	100	-	100	100	100	100	100	100	100	100	100	100	100	100	100	100	-	
9	truck and pig/tag-trailer	100	100	100	100	100	100	100	100	100	56	56	56	100	22	89	100	22	
14	truck and dog trailer	100	93	100	100	100	100	100	100	100	86	50	71	93	71	79	93	21	
12	A-double	100	-	100	100	100	100	100	100	100	92	92	83	100	100	100	92	-	
12	A-triple road train	100	-	25	-	100	8	100	100	100	75	17	8	83	100	92	100	-	
2	AAB-quad road train	50	-	-	-	100	-	100	100	100	100	50	-	100	100	50	100	-	
139	entire fleet	99	65	92	90	100	91	100	100	100	79	83	83	98	65	84	99	27	

APPENDIX G – EXTRACT FROM FLEET REPORT – FLEET VEHICLE COMPLIANCE FOR ACCESS TO ROAD TRAIN ROUTES (%)

Performance Measures		Performance Levels																
#	Vehicle Class	Stability (%)	Gradeability (Max. Grade) (%)	Gradeability (Max. Speed on 1%Grade) (km/h)	Acceleration Capability (s)	Trucking Ability (m)	Low-Speed Offtracking (m)	Frontal Swing (m)	Tail Swing (m)	Steer Tyre Friction Demand (%)	Static Rollover Threshold (g)	Rearward Amplification (-)	High-Speed Transient Offtracking (m)	Yaw Damping Coefficient (-)	Gross Mass per SAR (t/SAR)	Horizontal Tyres Forces (-)	Max. Effect Relative to Ref. Vehicle (%)	OVERALL (%)
		5	8	60	25	3.7	13.7	1.5	0.35	80	0.35	5.7	0.80	0.15	8.4	1.8	75	-
17	rigid trucks	100	100	100	100	100	100	100	100	100	71	100	100	100	18	29	100	18
6	buses/coaches	100	100	100	100	100	100	100	100	100	100	100	100	100	-	83	100	-
43	prime-mover and semi-trailer	100	100	100	100	100	100	100	100	100	79	100	100	100	72	93	100	53
23	B-double	100	100	100	100	100	100	100	100	100	78	100	100	100	78	100	100	70
1	B-triple	100	100	100	100	100	100	100	100	100	100	100	100	100	100	100	100	100
9	truck and pig/tag-trailer	100	100	100	100	100	100	100	100	100	56	56	56	100	22	89	100	22
14	truck and dog trailer	100	100	100	100	100	100	100	100	100	86	50	71	93	71	79	93	21
12	A-double	100	100	100	100	100	100	100	100	100	92	92	83	100	100	100	92	83
12	A-triple road train	100	100	100	100	100	100	100	100	100	75	17	8	83	100	92	100	-
2	AAB-quad road train	100	100	100	100	100	100	100	100	100	100	50	-	100	100	50	100	-
139	entire fleet	100	100	100	100	100	100	100	100	100	79	83	83	98	65	84	99	42

COMPATIBILITY IN TRUCK TO CAR FRONTAL IMPACTS

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ABSTRACT

One of Volvo's core values is safety. In our policy it is stated that we should not only comply with legal demands regarding safety, but also strive to develop vehicles that are safe in a real traffic environment. The demands from the public to make trucks more impact friendly are constantly increasing. Today, only a few percent of the collisions where trucks are involved causes serious injuries or fatalities to the truck occupants, the majority (about 54 %) of injured persons are car occupants. This means that the front underrun protection system, FUPS, is one of the most effective tools to reduce the number of fatalities in accidents involving trucks. The use of an effective FUPS on the truck will dramatically reduce the number of fatalities in truck to car collisions. In Europe, an estimate is 800 saved lives per year if all trucks were equipped.

The FUPS for the new Volvo FH and FM trucks has been designed not only to comply with the static legal demand, Directive 2000/40/EC, effective August 2003, but also to absorb energy for smaller cars while still being rigid enough to withstand the force from a large car at 65 km/h closing speed and 75 % overlap. This has been achieved by optimising a crumpling tube behind the FUPS beam that connects to rigid parts on the chassis.

The system was developed using FE simulations and verified in full scale testing. Sensitivity studies were performed using an FE car model with different impact heights and offset ratios. The influence of impact angle between the car and the truck was also analysed. For future systems, increased ability for the front of the truck to absorb energy is needed to increase the critical closing speed, and thereby further reduce the number of fatalities in truck to car accidents.

BACKGROUND

When developing the front end of the new Volvo FH and FM range, quite a few demands were set on the structure. One objective was to not increase the weight of the lower front in order not to increase the front axle load. Another was to include a front underrun protection system, FUPS. It should not only comply with the legal demand, Directive 2000/40/EC, which will be effective in August 2003. It should also manage to absorb some of the impact energy in a real frontal impact with a smaller car, while withstanding the forces from a large car. The solution was a beam that carries all the components of the module, such as headlights, washer reservoir, lower insteps etc. That beam is connected to rigid parts of the chassis through a crumpling tube. Finally, the level of energy absorption and the maximum force involved in a crash was investigated for optimisation of the system with the limited space and weight available. It was decided that the Euro-NCAP offset test should be the template. A large car (in this case a Volvo V70) impacts the FUPS at a relative velocity of 65km/h with 75 % car overlap.

COMPATIBILITY

When discussing compatibility, it is obvious that the mass difference between the truck and the car is the outstanding mismatch that cannot be changed. That leaves us the opportunity to optimise the force versus displacement of the FUPS for cars with different weight, stiffness and strength. The height of the front end of the truck varies quite substantially depending on vehicle specifications. This, together with the overlap ratio in an accident, needs to be considered when dimensioning the system. The following paragraphs will describe how the FUPS for the new Volvo FH and FM was developed with compatibility in mind.

ENERGY ABSORPTION

In order to choose optimal parameters for the energy-absorbing element, the following limitations were considered:

- Space available for intrusion into the truck structure.
- The maximum triggering force allowed without causing intrusion into cockpit on a small car.
- Energy absorption level of the crash beams in a car.

Space

Very limited space is available in the front of a Cab-Over-Engine, COE, truck design. The steering and suspension systems together with the engine and its cooling systems occupy the front end. For most trucks, the suspension brackets are the limiting components. Legal demands on vehicle lengths effectively reduce the amount of space available. Any protrusion in the front end of the truck would cause a decreased length of the payload. An exception from the rule for safety installations would make it possible to dramatically increase the performance of the FUPS.

Triggering force

What is a small car? It is a very difficult question to answer. Most modern cars have high safety cage strength, regardless of size. As an example, the Renault Clio develops over 400 kN when impacting a rigid FUPS according to the Group 14 report [1]. The force versus deformation curve was examined for a few cars, and the amount of energy absorbed by the crash structure before the bumper hits the engine was extracted. This energy is not absorbed if the bumper beam of the car misses the FUPS beam. A minimum requirement on the energy absorbing capability could be that amount. To define the triggering force, the results from the Group 14 report [1] was used together with our own experience from tests.

MAXIMUM FORCE

The maximum force is dependent more on the structural characteristics of the car than on the velocity involved. For a modern large car, the maximum force involved in a full barrier crash may exceed 800 kN. This force is reduced since the FUPS is not completely rigid and is not covering the complete front of the car. Due to design, it is not common that the FUPS beam takes the full load. Cooling systems and chassis components are often engaged in the crash. To define the maximum force that the FUPS should withstand, FE crash analyses were performed with data from a large car, in this case a Volvo S80.

VEHICLE HEIGHTS

Commercial vehicles are used in a wide range of applications, all with different demands on the truck configuration. For the FUPS, the most significant parameter is the frame height, and thereby the height of the lower front of the vehicle. The predicted number of vehicles in each segment was used when defining the height to obtain the most effective FUPS. The effect of vehicle height is described in PARAMETER STUDY below.

OVERLAP

Accidents are ranging from a small fraction of the car engaged to a full lateral overlap. Generally, large overlap makes the accident better defined in terms of deformations. The Volvo FUPS has two crumpling tubes, one at each frame member. This means that for a small overlap, only one of the tubes is activated. For larger overlap, both tubes are activated while the triggering force is doubled. If the overlap is very small, the car engine misses the FUPS and it is not possible to stop the car for the velocity used in the test. The effect of overlap is described in PARAMETER STUDY below.

ANGLED IMPACT

At higher speeds, the impact is usually head-on between the truck and the car. At lower velocities, the impact angle may often be larger. This makes it important to verify a good behaviour of the system for angled impacts as well.

PARAMETER STUDY

To analyse the effect on the FUPS considering compatibility, all parameters were analysed in a full-scale finite element analysis model. For the study, the new Volvo V70 was used. The following matrix of parameters was used. Maximum deformation pictures are included in the table.

An angle of 20 degrees with 70% of the head-on impact velocity, i.e. 46 km/h was used to verify the behaviour for the angled impact.

FUTURE

For future FUPS systems, the energy absorbing capacity will be the key issue. One way of increasing the behaviour is of course to allow the front end of the truck to protrude somewhat forward. Another way is to remove

rigid objects from the front end of the truck. As cars are constantly improving in frontal crash behaviour with stiffer and stronger bodies, the triggering force can probably be increased as well to allow for more energy to be absorbed.

REFERENCES

- [1] BEVC Working Group 14 Report: Development of Test Procedure for Energy-absorbing Front Underrun Protection Systems for Trucks. G. Blaauw et al. 1996

FIGURES & TABLES



Figure 1 – Euro-NCAP type of impact set up. The figure shows maximum deformation of the car in the crash. Crash FE simulation from above.

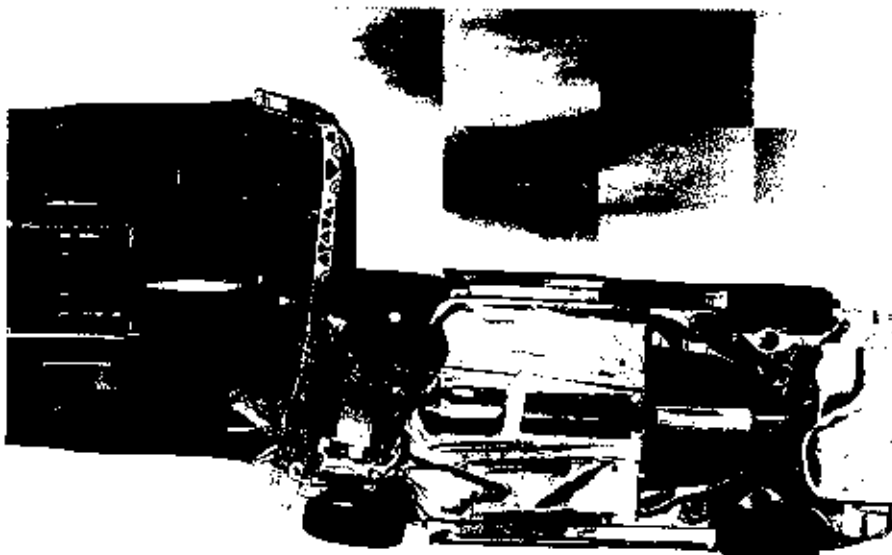

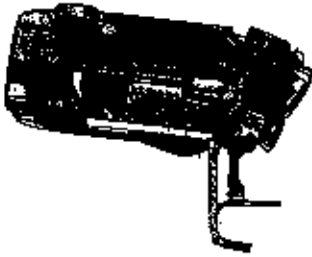
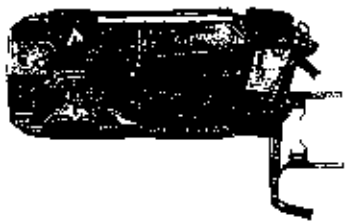

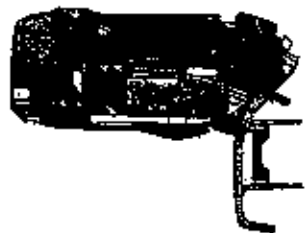


Figure 2 – Euro-NCAP type of impact set up. The figure shows maximum deformation of the car in the crash. Crash FE simulation from below.

Table 1. Matrix of analyses performed to verify the FUPS behaviour. Intrusions at the instrument panel for the car and deformation pictures are included.

Height	75 % Overlap	50 % Overlap
150 mm higher truck	 <p>Intrusion at instrument panel < 100 mm</p>	 <p>Intrusion at instrument panel < 200 mm</p>
Nominal height of truck	 <p>Intrusion at instrument panel < 50 mm</p>	
150 mm lower truck	 <p>Intrusion at instrument panel < 100 mm</p>	 <p>Intrusion at instrument panel < 100 mm</p>

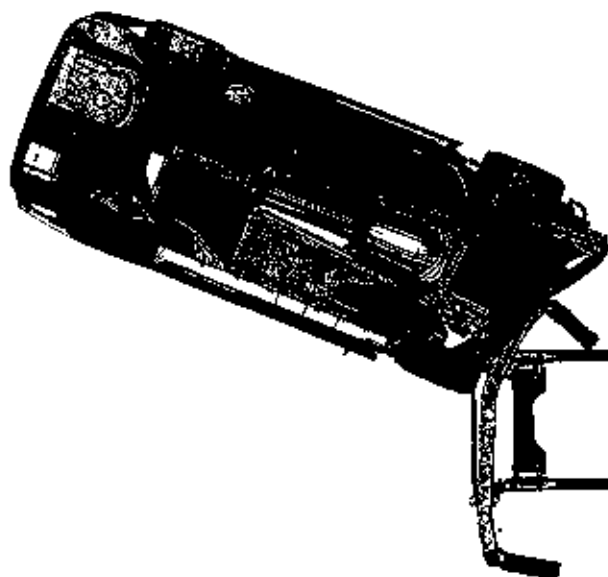


Figure 3 - Angled impact on FUPS, 20 degrees angle, 50 % overlap and 46 km/h.

THE FIRST WIM SYSTEM DESIGNED IN POLAND

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ABSTRACT

In last years the traffic on Polish roads rapidly increases, more and more of shipment is transported by lorries. The infrastructure and condition of Polish roads isn't good, than appears necessity to protect them. One of possibilities is to control the vehicle overloading. Up to now six weighing sites are installed in Poland. The sites consists of static scale and preselection systems for each direction. Unfortunately the preselection systems equipped with capacitive sensors are damage after a few month. Than the first Weighing in Motion – WIM preselection system, fully designed, programmed and assembled in Poland, has been put into service in April 2001. The system operates in standard configuration, i.e. it is equipped with two strip piezoelectric sensors and one inductive loop located between them. Each axle is weighed twice due to the employment of two WIM sensors. The arithmetic mean value of both results is assumed as the final axle load. The measuring range of the system lies between few hundreds of kilograms to 20 tons for the vehicle velocity included in the range 25 – 100 km/h. The described system allows vehicle detection in the measurement zone as well as estimation of a single axle and the whole vehicle loads, measurement of the vehicle velocity, number of axles, the distance between successive axles. Trailer detection is also possible, based on the vehicle magnetic profile, registered with the inductive loop. The statistical characteristics describing the traffic observed on the chosen road section are presented in the paper, specially the distribution of the vehicles number in the course of a day and night as well as in the course of a week. The effectiveness of detection of permissible parameters, such as velocity and axles load, is also estimated.

WIM SYSTEM DESCRIPTION

Presented in this paper WIM system consists of two piezoelectric strip sensors, each 4m long, placed perpendicularly to traffic direction, with mutual distance 1.7m. Between them inductive loop of dimensions 2m x 1.5m is placed. The sensors used in WIM system, are I Class weighing sensors of Measurement Specialties Inc. [4] [5]. The highly compressed piezoelectric polymer there are used in the sensor, and positive signal is generated as tires pass over. Thanks to its flat shape and proper installation directly in the pavement (see Fig. 1), the effect of "axle ghost" can be significantly reduced.

Result of weighing is calculated by integration of sensor output with respect to time and using calibration coefficient. Two strip weighing sensors in the WIM system make possible repeated weighing of each vehicle axle. The result of axle weighing is calculated as arithmetic mean. The total load of the vehicle is the sum of all its axles load.

Additionally such configuration of the measurement system enables counting of the vehicle axles, measurement of the distance between them, velocity and electrical length measurements, as well as the trailer detection on the basis of magnetic profile. The measurement results enable to include the weighed vehicle to one of thirteen classes according to FHWA classification. The system measures also the road surface temperature to correct weighing

results depending of thermic and mechanical surface properties. The results are transmitted to superior system with the RS232 interface.

The system assignement is preliminary weighing of travelling cars in purpose to select these, whose total load or axle load exceed pennissible values. Such vehicles are directed to nearby static scales for final weighing. Fig. 2. shows the electronic part of the preselection system [3].

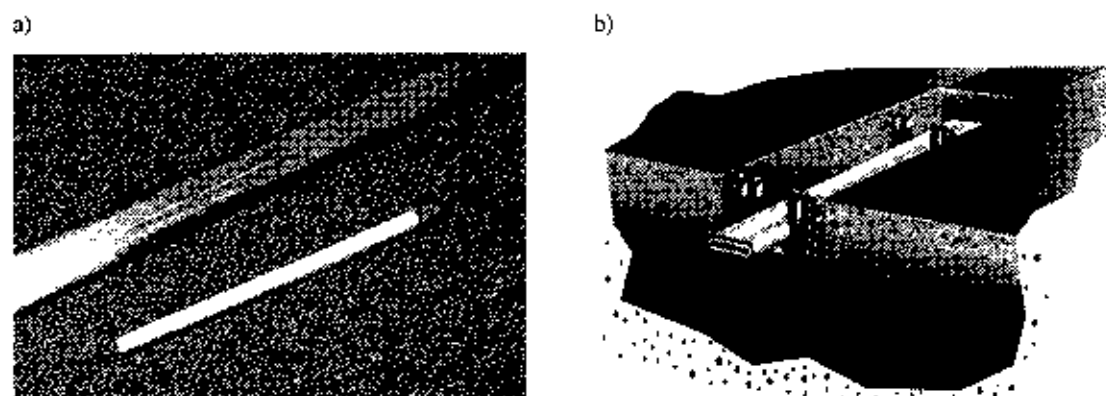


Figure 1. The piezoelectric WIM sensor (a), and method of its installation (b).

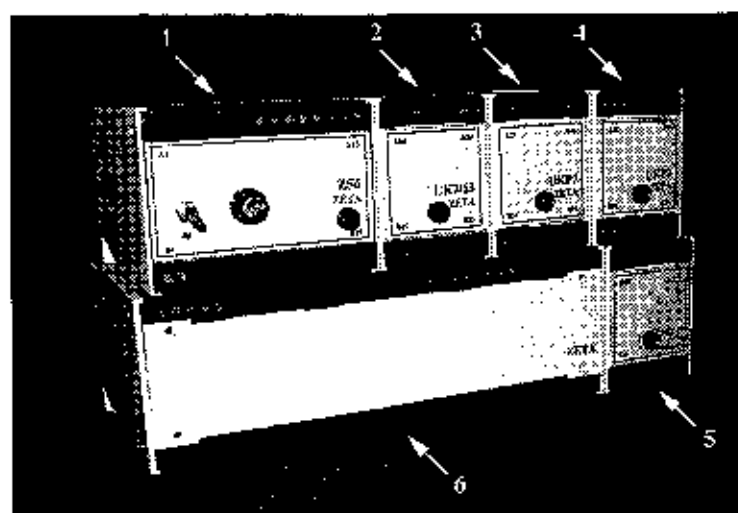


Figure 2 – The electronic part of the WIM system. 1 – power supply, 2 – inductive loop conditioning system, 3,4 – conditioning system of the piezoelectric WIM sensors, 5 – temperature measuring unit, 6 – microprocessor system.

SYSTEM CALIBRATION

The system calibration has been performed using the method of pre-weighed vehicles [1]; two and four axle, with total loads of 17 100 kg and 33 940 kg respectively. The calibration has been done for three different speeds (30, 50 and 70 km/h), 20 runs by each lorry for each speed.

Calibration coefficients calculated on this base for each of WIM sensors allows permanent weighing of moving vehicles for statistical and preselection purposes. The standard deviation of the moving vehicles weighing results, obtained in the described system does not exceed 7% and the bias error is less than 1.5%.

In the Fig. 3. the relative differences between weighing results obtained from two WIM sensors for 162 trucks crossing the tested WIM stand are presented. Their relative average value amounts 3.7%, and its relative standard deviation equals 10.5%.

Since the error of weighing of vehicles moving with road velocities includes significant random component, distribution of weighed vehicles number as a function of weighing error as well as distribution function of

weighing error (absolute value) are presented in Fig. 4. Presented characteristics make possible estimation of vehicles weighing uncertainty, coming out in the constructed WIM system. For example, the probability of committing of weighing errors exceeding 20% (point A on the curve 2) equals 0.22. A half of vehicles however has been weighed with an error not exceeding 10%.

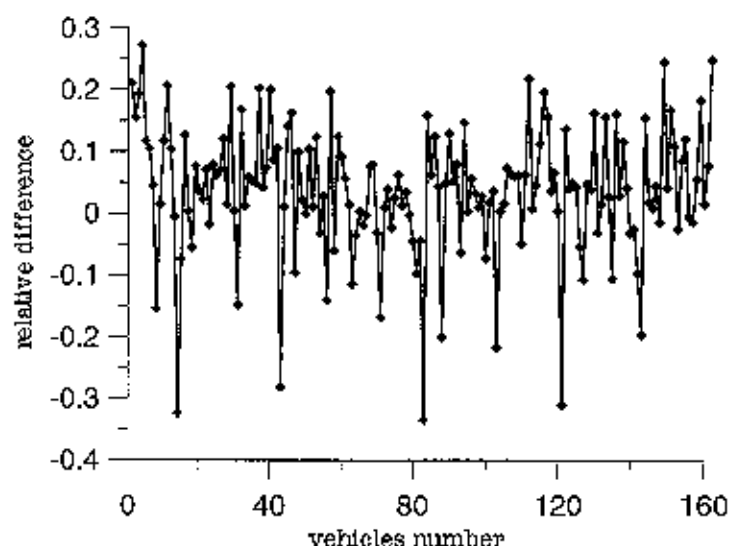


Figure 3 – The relative differences between weighing results issued from two piezoelectric sensors entering into the composition of WIM stand.

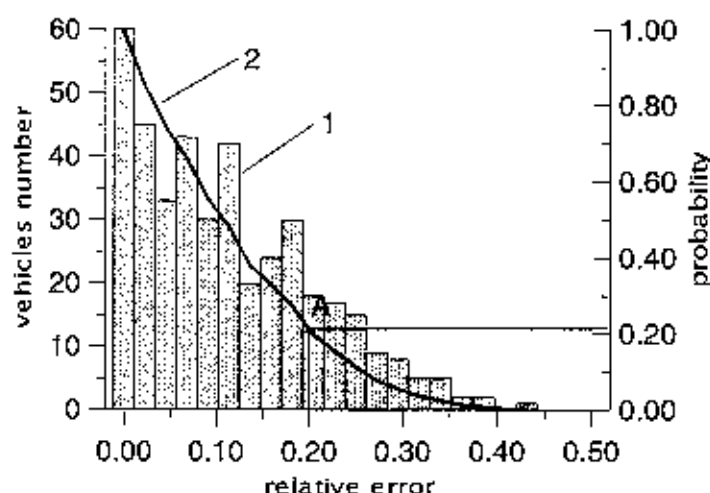


Figure 4 - Distribution of weighed vehicles number as a function of weighing relative error (absolute value).
1 – histogram, 2 – distribution function of absolute weighing error.

WEIGHING RESULTS

Results of traffic parameters measurements, stored by the constructed measurement system, make possible to determine basic characteristics, describing the traffic at the system location. Selected characteristics are, determined on the basis of measurements results, obtained during one week (2001.12.24 – 2001.12.30). The days 25 and 26 December are holidays in Poland; the day 2001.12.30 was Sunday.

The highway, at which the measurement system has been located, shows great differences of traffic between working days and holidays. During working days total load of all vehicles passing one lane within 24 hours exceeds 30 000 tons. During holidays the total load decreases even to 4 000 tons. Much lower differences characterize the total number of passing vehicles. Greater differences are noticed within particular classes, e.g. in the groups of four and five axes vehicles. Detailed data are presented in the Table 1.

Table 1 - Total vehicles number in respective days taking into account partition for lanes and vehicles classes as well as the total load of all vehicles on each lane.

Date	Lane number	Number of vehicles					Total load [tons]
		Vehicle class – number of axles				Total number of vehicles	
		2	3	4	5		
24.12.01	1	3 497	39	51	68	3 655	7 729
	2	4 981	58	73	139	5 251	8 578
25.12.01	1	2 925	14	7	6	2 952	4 056
	2	3 178	19	10	8	3 215	3 640
26.12.01	1	5 183	23	11	23	5 240	8 738
	2	3 309	23	11	23	3 366	4 176
27.12.01	1	7 374	112	169	406	8 061	28 056
	2	5 429	124	132	429	6 114	20 342
28.12.01	1	6 501	127	224	445	7 297	30 228
	2	5 846	140	154	489	6 629	25 683
29.12.01	1	5 399	83	112	234	5 828	17 512
	2	5 195	110	124	319	5 748	16 506
30.12.01	1	4 029	36	50	52	4 167	8 532
	2	3 494	52	39	56	3 641	6 615
31.12.01	1	3 813	35	50	71	3 969	9 689
	2	663	17	17	33	730	2 268

Described measurement system makes also possible traffic monitoring with simultaneous vehicles classification. Fig. 5. presents distribution of vehicles number in time for all types and for 5 - axes vehicles. Both characteristics are in different degree sensitive to holidays. The traffic of 5 - axes vehicles during holidays practically decays. Characteristics of vehicle number distribution in time enable to notice distinct 24 hours fluctuations of vehicles number passing chosen traffic lane. These characteristics show for instance that during holidays maximal traffic intensity (vehicles number per hour) is two times less than during working days. Distortion of distribution during the day 24 December is plainly visible. Traffic intensity maximum is on similar level as during working days, but the distribution characteristic is much more concentrated around the mean value. Also the minimum between holidays has decidedly lower value. The peak of traffic during working days occurs at afternoon hours, while during holidays is delayed about 4 hours.

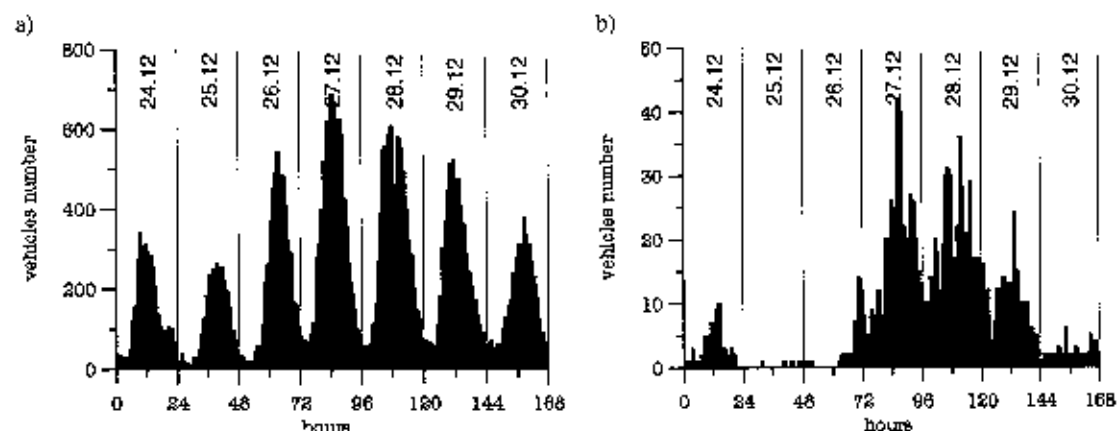


Figure 5 - Time distributions of: a) vehicles total number, b) 5 - axes vehicles number.

Characteristics describing the vehicles load distribution in time allow to observe the changes of transported load depending on the time of day, and the day of a week. Presented characteristics (Fig. 6.) show distinct transport decrease during holidays. The load amount increases at midnight between 26 and 27 December. Minimal amount of transported load occurs about 2 a.m. (500 tons per hour), maximal amount – between 2 p.m. and 4 p.m. (2 200 tons per hour).

From the Fig. 6b. it is evident, that during holidays goods traffic is quite marginal. In Sunday (30 December) observed goods traffic was 4 times less than during working days. In working days the total mass of the load transported by single lane of investigated road reaches even 1000 tons per hour.

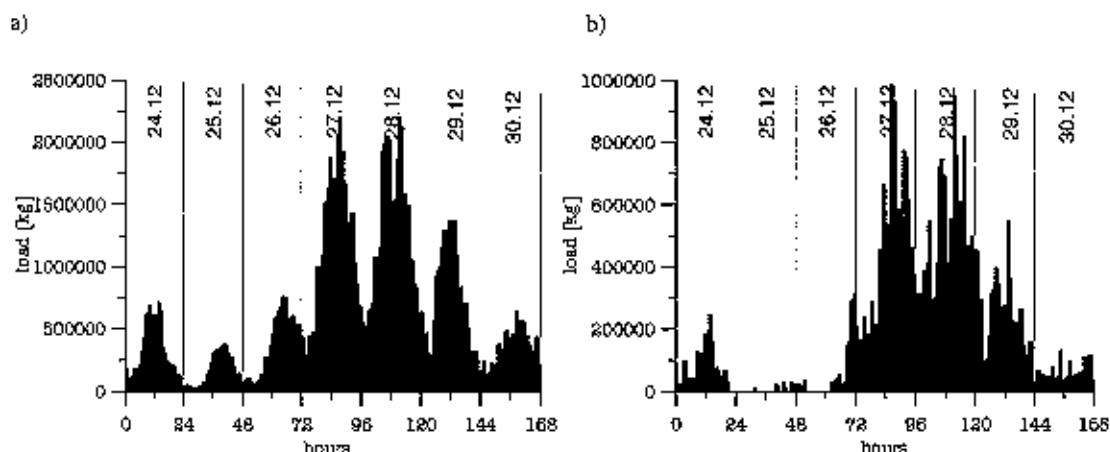


Figure 6 - Time distributions of total load: a) for all vehicles, b) for 5 – axle vehicles.

Characteristics in the Fig. 7. present time variability of average load of 2- and 5- axle vehicles. Since the class of 2- axle vehicles includes passenger cars as well as lorries and trucks, observed changes of average load are caused by the change of proportions of these types within the whole population. The increase of lorries number is particularly well noticeable at 4 a.m., when the average load of 2 – axle vehicle increases even up to 5 tons (Fig. 7a.).

The time distribution of the average load of 5 – axle vehicles allows to choose these times of day, in which the probability of overloaded vehicles detection increases. The characteristic presented in the Fig. 7b. shows, that such periods are 2 hours before and after midnight. Just in this time control weighing of the vehicles should be particularly profoundly performed.

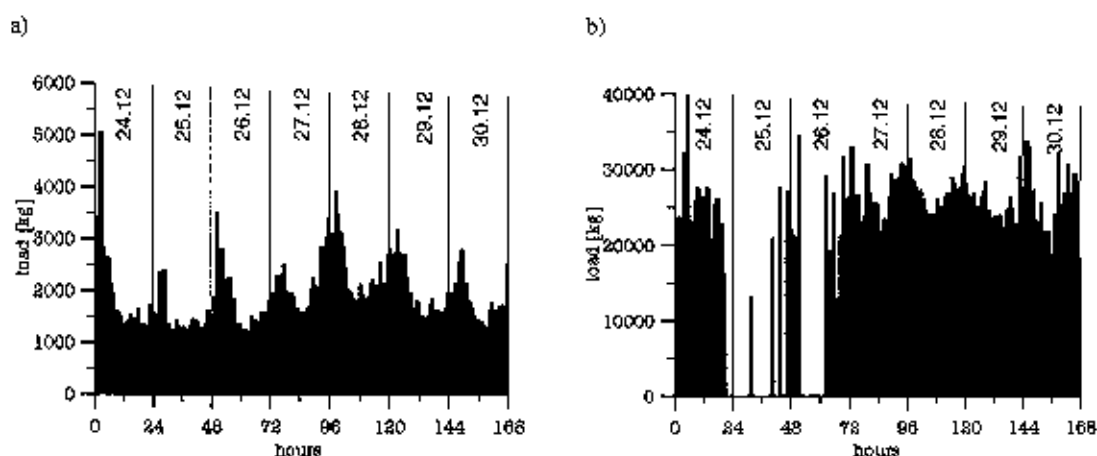


Figure 7 - Time distribution of the average load of: a) 2 – axle vehicles, b) 5 – axle vehicles.

The data concerning exceeding of permissible axle load are compiled in the Table 2. Most of permissible load exceedings are observed in working days within the class of 5 – axle vehicles (65 in one lane). 10 tons was assumed as permissible load of single axle. However, on the road chosen for investigations vehicles may be found exceeding this limitation even over 50% (Fig.8.).

Table 2. Number of permissible axle load exceedings in subsequent days for particular classes.

Date	Lane number	Number of vehicles exceedings permissible axle load			
		Vehicle class – number of axles			
		2	3	4	5
24.12.01	1	1	1	0	5
	2	0	0	0	2
25.12.01	1	0	0	0	0
	2	0	0	0	0
26.12.01	1	3	0	0	6
	2	0	0	0	2
27.12.01	1	11	0	1	65
	2	2	0	0	7
28.12.01	1	8	4	5	61
	2	4	1	0	16
29.12.01	1	9	2	8	24
	2	0	0	1	11
30.12.01	1	3	1	1	7
	2	0	0	0	4
31.12.01	1	1	2	0	13
	2	0	0	0	2

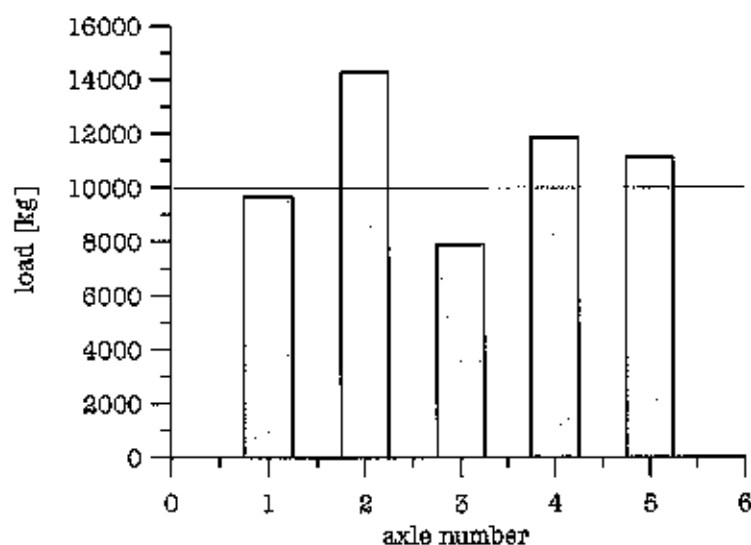


Figure 8 - Axle load distribution for exemplary overloaded 5 – axle vehicle.

Presented system also enables registration of permissible velocity limit breaking. On the road section, where measuring system has been installed, the velocity limit is equal to 70 km/h. Majority of drivers are breaking this limit (see Fig. 9.).

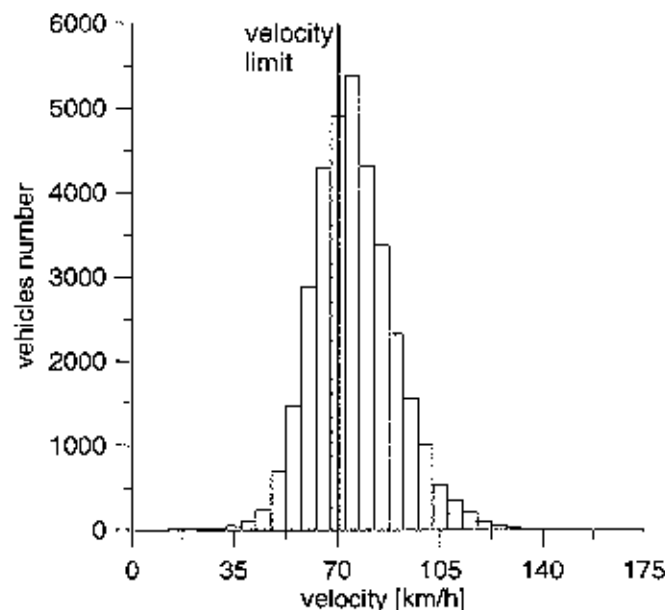


Figure 9 - Velocity distribution characteristic.

CONCLUSIONS

Presented in the paper WIM system is the first one completely designed and constructed in Poland. Over 1 year exploitation period in changeable climatic conditions confirmed its reliability and efficiency. In 100 % of events vehicles indicated as overloaded, actually exceeded permissible axle load. It was confirmed by means of static weighing. Measurements results yielded by described system are stored in superior system and are used for elaborating of summary reports characterizing traffic conditions on selected highway. For authorized persons these informations are accessible through Internet.

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THEORETICAL TESTING OF A MULTIPLE-SENSOR BRIDGE WEIGH-IN-MOTION ALGORITHM

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ABSTRACT

A Bridge Weigh-In-Motion (B-WIM) system is based on the measurement of the flexure in a bridge and the use of measurements to estimate the attributes of passing traffic loads. The information provided by strain sensors and axle detectors is converted into axle weights through the application of an algorithm. Because the dynamic interaction between bridge and vehicle has many parameters for which estimation is very difficult from raw bridge strains, the traditional B-WIM algorithm consists of static equations of equilibrium. Hence, dynamics can be a significant source of inaccuracy in B-WIM systems depending on the bridge and vehicle characteristics. In this paper, the influence of dynamics on B-WIM accuracy is assessed numerically by simulating the passage of a number of vehicles over a bridge. Then, the theoretical bridge response is used to test a new B-WIM algorithm based on multiple longitudinal sensor locations and the traditional algorithm based on one single location. In smooth road conditions, the multiple sensor B-WIM achieved better results, while the traditional B-WIM failed to predict individual axle weights accurately. In rough conditions, results were much poorer due to high dynamic excitation and only gross vehicle weight was predicted accurately.

Keywords: accuracy, bridge, vehicle, dynamics, WIM.

INTRODUCTION

In the 1970's in the USA, the Federal Highway Administration (FHWA) started studying Bridge WIM systems to acquire WIM data. Moses (1979) introduced an algorithm based on the assumption that a moving load will cause a bridge to bend in proportion to the product of the load magnitude and a reference curve representative of the bridge behaviour, the influence line. In the 1980's, Peters (1984) developed AXWAY in Australia. This B-WIM system is based on the same concept of influence line. A few years later, he derived a more effective system for weighing trucks using culverts, known as CULWAY (Peters 1986). Both the American and Australian systems have been used for commercial applications on bridges and culverts. Bridge Weighing Systems Inc. developed one of the first commercial B-WIM systems in 1989 on the basis of Moses' algorithm (Snyder 1992). In the 1990's, three new independent B-WIM systems were developed in Ireland, Slovenia and Japan (O'Connor 1994, Žnidarič & Baumgärtner 1998, Ojio et al. 2000). All of these B-WIM systems use algorithms based on static equations of equilibrium combined with measurements at one single longitudinal section.

Because the data being recorded is the sum of static and dynamic components, the dynamic component is generally ignored. However, bridge and truck dynamics have been proven to be one of the sources of inaccuracy of static algorithms. Traditional averaging or filtering techniques are not suitable for the removal of dynamics in every case. The data input is particularly difficult to analyse from the dynamic response of bridges due to the wide range of trucks on bridges with different classifications and with different dynamic properties. In this paper, theoretical simulations are used to evaluate the influence of suspension type and road profile on the accuracy of a B-WIM algorithm. Additionally, the static equations of the traditional algorithm are extended to multiple longitudinal sensor locations along the bridge. Both one-sensor and multiple-sensor based algorithms are tested in a highly dynamic scenario with the first natural frequency of the bridge close to truck body frequencies.

MULTIPLE SENSOR BRIDGE WEIGH IN MOTION (MS-BWIM) ALGORITHM

Kealy & O'Brien (1998) first attempted to develop a weigh in motion system capable of giving a history of axle weights at each point in time as a truck crosses a bridge at normal highway speed by using multiple strain gauge sensor locations. However, simultaneous equations resulted dependent in some cases and a solution was only feasible for reduced number of axles on the bridge (equal to the number of independent equations at that instant). It has been found that combining more sensors with a least squares fitting technique allows determination of a higher number of axles and makes the algorithm applicable to bridges with longer spans. This instantaneous calculation along the length of the bridge requires many sensors; well in excess of the number of axles. Thus, if there are a number of sensors, m , equal to or greater than the number of axles n , it is possible to minimise the error function defined in Equation 1 at each instant t :

$$f = \sum_{k=1}^{k=n} (\epsilon_k - \tilde{\epsilon}_k)^2 \quad (1)$$

where ϵ_k is the theoretical total strain due to the applied load at location k and $\tilde{\epsilon}_k$ is the measured strain due to the applied load at sensor location k .

By assuming linearity and applying the principle of superposition, it is possible to express the strain at each sensor as:

$$\epsilon_k = \epsilon_{k1}W_1 + \epsilon_{k2}W_2 + \dots + \epsilon_{kn}W_n \quad (2)$$

where:

- ϵ_{kj} : value of influence line of strain at sensor location i due to moving load located at j
- W_j : applied load at location j .
- n : total number of axles on the bridge.

By combining Equations 1 and 2 at each instant t :

$$f = \sum_{k=1}^{k=m} (\epsilon_{k1}W_1 + \epsilon_{k2}W_2 + \dots + \epsilon_{kn}W_n - \tilde{\epsilon}_k)^2 \quad (3)$$

Differentiating Equation 3 with respect to the axle weight, W_i and setting it equal to zero gives:

$$\frac{df}{dW_i} = 2 \sum_{k=1}^{k=m} (\epsilon_{k1}W_1 + \epsilon_{k2}W_2 + \dots + \epsilon_{kn}W_n - \tilde{\epsilon}_k) \epsilon_{ki} = 0 \quad (4)$$

The full set of equations can be expressed in matrix form as:

$$\begin{bmatrix} \sum_{k=1}^{k=m} \epsilon_{k1}\epsilon_{k1} & \sum_{k=1}^{k=m} \epsilon_{k1}\epsilon_{k2} & \dots & \sum_{k=1}^{k=m} \epsilon_{k1}\epsilon_{kn} \\ \sum_{k=1}^{k=m} \epsilon_{k2}\epsilon_{k1} & \sum_{k=1}^{k=m} \epsilon_{k2}\epsilon_{k2} & \dots & \sum_{k=1}^{k=m} \epsilon_{k2}\epsilon_{kn} \\ \vdots & \vdots & \ddots & \vdots \\ \sum_{k=1}^{k=m} \epsilon_{kn}\epsilon_{k1} & \sum_{k=1}^{k=m} \epsilon_{kn}\epsilon_{k2} & \dots & \sum_{k=1}^{k=m} \epsilon_{kn}\epsilon_{kn} \end{bmatrix} \begin{bmatrix} W_1 \\ W_2 \\ \vdots \\ W_n \end{bmatrix} = \begin{bmatrix} \sum_{k=1}^{k=m} \tilde{\epsilon}_k \epsilon_{k1} \\ \sum_{k=1}^{k=m} \tilde{\epsilon}_k \epsilon_{k2} \\ \vdots \\ \sum_{k=1}^{k=m} \tilde{\epsilon}_k \epsilon_{kn} \end{bmatrix} \quad (5)$$

Finally, applied loads (weights) can be calculated from Equation 6:

$$\{W\}_{n \times 1} = [\epsilon]_{m \times m}^{-1} \{\tilde{\epsilon}\}_{m \times 1} \quad (6)$$

If the number of strain sensor locations is high enough, Equation 6 provides a solution along most of the bridge. The static value can be obtained from the root mean square of the calculated instantaneous axle forces.

THEORETICAL SIMULATIONS

The bridge and truck are modelled separately and combined in an iterative procedure. The prediction of bridge response involves convolution of the vehicle loads with modal responses of the bridge and the convolution integral is solved by transformation to the frequency domain using the fast Fourier transform. The method is then extended by an iterative procedure to include dynamic interaction between the bridge and an arbitrary mathematical model of a vehicle. Green & Cebon (1994) illustrate the effectiveness of this calculation method, the convergence of the iterative procedure, and the good agreement with experimental data.

The Bridge Weigh In Motion (B-WIM) system is calibrated with a two-axle fully laden linear sprung vehicle (four degrees of freedom). Then, the system is evaluated with a four-axle vehicle and two different suspension systems (air and steel leaf). The bridge is modelled as a beam 30 m long. Strain output is calculated every 0.01 s (100 Hz) at 5, 10, 15, 20 and 25 m from the bridge support. Figure 1(a) shows the records in free vibration after the vehicle has left the bridge and Figure 1(b) the corresponding spectra for the three runs of the two-axle calibration vehicle. From these Figures, 0.97% damping and a 3.33 Hz first natural frequency are obtained.

The vehicle models developed by Green et al. (1995) are a steel sprung four-axle articulated vehicle validated experimentally, and a similar vehicle fitted with air suspensions and hydraulic dampers. The dynamic response of the 30 m beam model is obtained for crossings of each of these two vehicles. The four-axle articulated vehicle has 11 degrees of freedom as shown in Figure 2. In this Figure, elements A, B and C represent a spring with adiabatic behaviour, Coulomb friction, and a linear spring/damper, respectively. For the vehicle with air suspension, models of air springs with parallel viscous dampers replace the steel-spring elements on the drive axle and the two trailer axles. The suspension on the steer axle is the same for both vehicle models. Two surface profiles, three different speeds (55, 70 and 85 km/h), and three different loading conditions are chosen for the simulations.

Figures 3 and 4 represent the applied dynamic forces over same length of smooth and rough road profile respectively. It can be seen how these forces increase with speed and road roughness. Steel suspensions are clearly more sensitive to these changes, except for a singularity localised at about 90 m. This singularity is due to a bump in the road that excites the wheel-hop mode of the vehicle. A steel suspension is better damped at high excitation amplitudes than an air suspension. However, for low excitation the friction in the steel-springs is not overcome and damping is only provided by the tires. Under such circumstances any viscous shock absorber can provide higher damping than the friction in the steel springs. The load sharing between axles within the tandem of the air-sprung vehicle is less than in the steel-sprung vehicle. The amplitude of the applied dynamic wheels is higher for worse road conditions as shown in Figure 4. Thus, bridge response will increase with road roughness. Also, body bounce modes of vehicle vibration are excited by longer wavelength variations in road profile whereas axle hop are excited by short wavelength defects such as pot-holes, road debris, a rough repair or a mis-aligned joint in a bridge, more likely to occur in a road in poor conditions.

The strain response at the bridge midspan for the fully laden four-axle vehicle on a smooth profile is represented in Figure 5. As expected, the steel suspended vehicle causes higher dynamic oscillations on strain than the air-suspended vehicle. Figure 6 shows the bridge response at midspan for the fully laden vehicle on a rough profile. The maximum dynamic response takes place at 70 km/h for the steel-sprung vehicle and 85 km/h for the air-sprung vehicle. It can be seen that the air-suspended vehicle causes significantly lower dynamic bridge response than the steel-spring suspended vehicle and it is less sensitive to a change in speed. In comparison to the smooth profile, the dynamic response due to the steel-sprung vehicle increases by over 100%. These high dynamics suggest the occurrence of frequency matching between the steel-sprung vehicle and the bridge. Consequently, Bridge WIM will tend to be less accurate in the cases of steel-spring suspensions, rough road profile, and, for this bridge, for vehicle speeds near 70 km/h. Xie (1999) investigates the maximum interaction of bridges with natural frequencies close to air-sprung vehicles and vehicle axle-hop vibrations.

CALIBRATION

The traditional static and multiple-sensor B-WIM algorithms are calibrated with longitudinal bending at midspan and bending at five points equally spaced along the bridge respectively. Accuracy classes are determined according

to the COST323 European Specification on WIM (1999). Thus, the calibration takes place in full repeatability conditions (r1): A two-axle vehicle passing at different speeds, one load and one lateral position on the road. Then, the following section evaluates the system in limited reproducibility conditions (R1): two 4-axle vehicles passing over the bridge at different speeds with different loads. Results are given for two different road profiles.

Concerning calibration, the shape of the theoretical influence line is known from beam theory and the static algorithm only requires a calibration factor to adjust the magnitude of the strains to the theoretical model. If the exact influence line for bending moment is used, the calibration factor is the product of the modulus of elasticity and the section modulus. Several approaches are available for obtaining the real shape of the influence line from an experimental record (González 2001). For this analysis, the calibration factor is obtained by dividing the real static gross vehicle weight by the predicted weight of the calibration truck. A linear sprung two axle vehicle with 4 m axle spacing, 32.42 kN static weight in the front axle and 59.94 kN in the rear axle is used for calibration.

Smooth Road Profile

At one particular location i , bending moment, M_i , is proportional to strain, ϵ_i , through the elasticity, E , and section modulus, S_i , as shown in Equation 7.

$$M_i = ES_i \epsilon_i \quad (7)$$

Bending moment depends on the shape of the influence line, bridge length, static axle weights, axle spacings and position of the calibration vehicle. The results of the static calibration are shown in Figure 7. The calibration factor changes for each speed very slightly (2.11×10^{10} at 55 km/h, 2.10×10^{10} at 70 km/h and 2.08×10^{10} at 85 km/h). An average value of 2.10×10^{10} is adopted.

Other sensor locations used for MS-BWIM are calibrated in the same way. The axle forces predicted by MS-BWIM are compared to the simulated applied forces in Figures 8(a) and (b). A clear correspondence between predicted and simulated instantaneous wheel forces is not evident, but the average value is very similar in both cases. Dynamic wheel forces are strongly excited by a bump located at about 5 m from the bridge end. Values tend towards infinity at both ends of the instantaneous calculation (very small value of the determinant in Equation 6) and they are not taken into account in the determination of the static weight.

Figure 9 illustrates the results in static weights for the calibration vehicle. The front axle is the lightest and the percentage error tends to be higher than in the rear axle as shown in Figures 9(b) and (c). MS-BWIM is slightly worse than B-WIM for predicting gross vehicle weight, but the improvement in individual axle weights is very significant.

Rough Road Profile

The calibration of the static algorithm is shown in Figure 10. The scale factor between real and predicted gross weight is 2.12×10^{10} at 55 km/h, 2.12×10^{10} at 70 km/h and 2.08×10^{10} at 85 km/h. An average value of 2.09×10^{10} is adopted as the calibration factor.

Figure 11 shows the instantaneous axle forces from the simulation run at 55 km/h and the corresponding prediction by MS-BWIM. As in the smooth road profile, the forces increase enormously at about 5 m from the bridge end and the instantaneous solution from here to the end of the record is ignored. There is a large increase in oscillations over the smooth case. Figure 12 shows the variation of axle forces at 70 km/h. The prediction of the static answer proves more difficult at higher speeds, especially in the case of a rough profile. Figure 13 shows the calculation at 85 km/h. The front axle is overweighed. MS-BWIM is not able to distinguish which is the force applied by each axle due to the strong dynamics and the limitation in the number of sensors. Thus, Figures 13(a) and (b) show how the prediction of the front axle follows a pattern similar to the simulated rear axle.

The errors in individual axle weights for the test vehicle are given in Figure 14. Calibration results are much poorer than in the case of a smooth profile (Figure 9). Results in individual axle weights at 85 km/h are very inaccurate, but static B-WIM can predict gross weight with less than 2% error.

CHECK OF ACCURACY

A four-axle vehicle (axle spacings 3.49, 6.76 and 2 m) is chosen for assessment of the calibration carried out in the previous section (influence line and calibration factors obtained with the two-axle calibration vehicle do not suffer any further manipulation). The static weights of this vehicle were unknown to the first author prior to the calculations. Two types of suspensions are investigated: air and steel leaf sprung. These two vehicles are driven at three different speeds (55, 70 and 85 km/h) and three loading conditions (unloaded, half and fully laden).

Smooth Road Profile

Figure 15 shows the approximation by the static B-WIM algorithm to the response caused by a fully laden 4-axle truck at 70 km/h. The results illustrated in Figure 15 are based on the concept of an influence line. The contribution of each axle separately and all axles to the expected static strain are represented. The strain record caused by a steel leaf suspension exhibits a higher deviation from the fitted response than air suspension. Hence, predictions of individual axle weights are expected to be more accurate for air suspensions.

Figure 16 shows the simulated load history and the prediction by MS-BWIM for the case of a fully laden 4-axle truck with air suspension travelling at 70 km/h. Figures 16(c) and (d) show that the third and fourth axles allow for an instantaneous solution only when the first axle is located between 17 and 25 m from the bridge start (as result of the number of truck axles, their spacings and the number of sensors, their locations and influence lines). Figure 17 shows the results of MS-BWIM when using the same truck and speed as Figure 16, but with a steel suspension. From Figures 16(c), 9.16(d), 9.17(c) and 9.17(d), the prediction of the third and fourth axles can be seen to be more difficult for the leaf sprung than for the air sprung vehicle.

Tables 1 and 2 give accuracy classes according to the COST323 WIM European Specification (1999) for each criterion and algorithm. MS-BWIM is very accurate for all single criteria, even axles of a group in class B(10). The static B-WIM algorithm can predict gross weight accurately (A(5)), but fails to predict single axles (B(45)). All results are represented in Figure 18.

Rough Road Profile

Results for a rough road profile are very poor and only gross weight give sensible levels of accuracy. The poor accuracy of a B-WIM algorithm on a rough profile can be explained by Figure 19. Figure 19 represents the approximation by the static algorithm to the total strain generated by a fully laden truck at 70 km/h. The total response is far from this static response due to the high dynamic oscillations. The static algorithm is very inaccurate and their approximations (by minimising difference between total response and expected static response) can lead to negative values for individual axle weights. For instance, the second and fourth axles in Figures 19(a) and (b) (thin line representing bending moment diagram due to a single axle is over the x-axis). As in the case of a smooth profile, steel leaf suspensions give worse results than air suspensions.

If the road is in a poor condition, algorithms based on linearity and superposition do not appear to offer a valid solution. MS-BWIM is also extremely inaccurate because it minimises the instantaneous solution by using influence lines at many locations. Figure 20 shows the results for the same run as Figure 19(a). The static value can be estimated from instantaneous values when the first axle is located between 18 and 25 m from the bridge start. However, the third axle and fourth axles are strongly overweighed and underweighed respectively (Figures 20(c) and (d)).

According to Tables 3 and 4, gross weight is the only criterion which gives reasonable levels of accuracy. The static B-WIM gives the best result – C(15) for gross weight (Table 3). The poor results in individual axles are due to the failure of the static algorithm to estimate the static component caused by each axle from the total bending response (illustrated in Figure 19). Figure 21 illustrates the estimation of every identity in the sample. However, if the test took place in extended repeatability conditions (r2) taking into account only the runs of the air suspension truck, the accuracy class for gross weight would be raised to A(5) and C(15) for the static B-WIM and MS-WIM algorithms respectively.

CONCLUSIONS

A B-WIM system has been theoretically tested in very adverse dynamic circumstances. A two-axle linear sprung vehicle model has been used for calibration and accuracy has been assessed with a four-axle non-linear sprung vehicle (11 degrees of freedom). Air and steel suspensions on both smooth and rough pavements were considered. In smooth road conditions, MS-BWIM achieved the most accurate overall class B(10) (corresponding to the criterion of axle weights within an axle group, but B+(7) for the criterion of individual axle weights). The traditional static B-WIM had the same accuracy class, A(5), for gross vehicle weight as MS-BWIM, but it failed to predict individual axle weights accurately (E(45)). The 30 m span length made it difficult to identify individual axles from strain at only one location, and MS-BWIM derived a more accurate value from the load history.

B-WIM accuracy decreased for steel suspensions because bridge dynamic response was more important than for air suspensions. The poorest results were obtained with steel suspensions on rough profiles. Bridge strains oscillated with higher amplitudes when crossed by the steel suspension truck due to the proximity of the frequencies of the truck and the bridge, which resulted in huge errors in any B-WIM algorithm. In these conditions, the estimation of individual axle weights is very inaccurate (class E), but the traditional static B-WIM algorithm can still provide reasonable values for gross vehicle weight (C(15)). In order to deal with high dynamic bridge excitation, the use of dynamic models and the solution of the dynamic inverse problem in B-WIM practice need to be analysed in the near future. In the interim, bridges selected as B-WIM sites should have natural frequencies that do not coincide with the dominant natural frequencies of heavy vehicles.

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TABLES & FIGURES

Table 1- Accuracy classification for static B-WIM algorithm (smooth profile) (R1) (n: Total number of vehicles; m: mean; s: Standard deviation; π_0 : level of confidence; δ : tolerance of the retained accuracy class; δ_{min} : minimum width of the confidence interval for π_0 ; π : Level of confidence of the interval $[-\delta, \delta]$)

Criterion	Relative error statistics				Accuracy calculation				Class Retained
	n	m (%)	s (%)	π_0 (%)	Class	δ (%)	δ_{min} (%)	π (%)	
Single axle	36	-0.31	21.36	93.1	E(45)	54	47.3	96.5	E(45)
Group of axles	18	-2.34	7.71	90.3	C(15)	18	17.9	90.5	
Gross Weight	18	-0.79	0.93	90.3	A(5)	5	2.5	100.	

Table 2- Accuracy classification for static MS-BWIM algorithm (smooth profile) (R1) (n: Total number of vehicles; m: mean; s: Standard deviation; π_0 : level of confidence; δ : tolerance of the retained accuracy class; δ_{min} : minimum width of the confidence interval for π_0 ; π : Level of confidence of the interval $[-\delta, \delta]$)

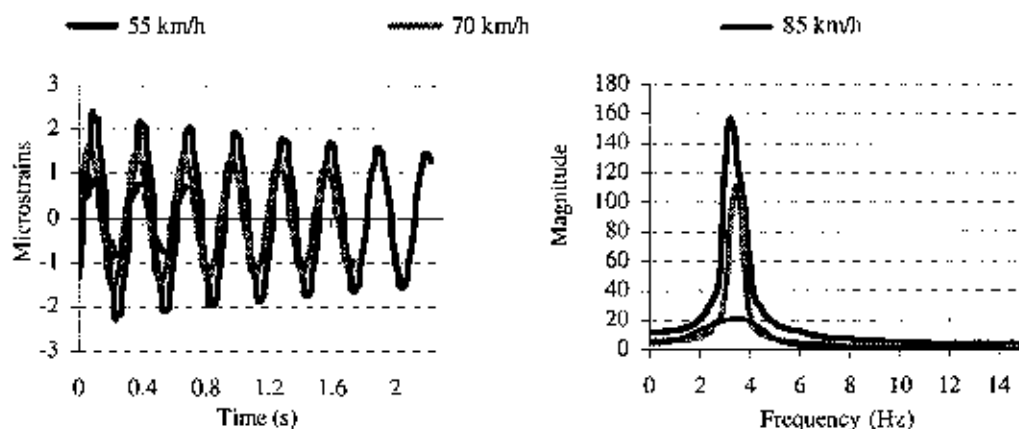
Criterion	Relative error statistics				Accuracy calculation				Class Retained
	n	m (%)	s (%)	π_0 (%)	Class	δ (%)	δ_{min} (%)	π (%)	
Single axle	36	-0.89	3.74	93.1	B+(7)	11	8.5	98.3	B(10)
Axle of group	36	-0.32	8.31	93.1	B(10)	20	18.4	95.4	
Group of axles	18	-0.07	2.70	90.3	A(5)	7.14	6.1	95.3	
Gross Weight	18	-0.56	1.46	90.3	A(5)	5	3.4	98.8	

Table 3- Accuracy classification for static B-WIM algorithm (rough profile) (R1) (n: Total number of vehicles; m: mean; s: Standard deviation; π_0 : level of confidence; δ : tolerance of the retained accuracy class; δ_{min} : minimum width of the confidence interval for π_0 ; π : Level of confidence of the interval $[-\delta, \delta]$)

Criterion	Relative error statistics				Accuracy calculation				Class Retained
	n	m (%)	s (%)	π_0 (%)	Class	δ (%)	δ_{min} (%)	π (%)	
Single axle	36	-9.46	135.4	93.1	E(300)	360	300.4	97.3	E(300)
Group of axles	18	15.60	42.52	90.3	E(100)	110	100.1	93.6	
Gross Weight	18	0.95	5.71	90.3	C(15)	15	13.0	94.9	

Table 4- Accuracy classification for static MS-BWIM algorithm (rough profile) (R1) (n: Total number of vehicles; m: mean; s: Standard deviation; π_0 : level of confidence; δ : tolerance of the retained accuracy class; δ_{min} : minimum width of the confidence interval for π_0 ; π : Level of confidence of the interval $[-\delta, \delta]$)

Criterion	Relative error statistics				Accuracy calculation				Class Retained
	n	m (%)	s (%)	π_0 (%)	Class	δ (%)	δ_{min} (%)	π (%)	
Single axle	36	-4.03	14.90	93.1	E(30)	36	33.9	94.8	E(50)
Group of axles	18	-4.89	21.42	90.3	E(50)	55	49.1	94.1	
Gross Weight	18	-4.07	12.13	90.3	E(30)	30	28.3	92.4	



(a) Time Domain

(b) Frequency Domain

Figure 1 – Bridge response in free vibration due to 2-axis calibration truck (smooth profile)

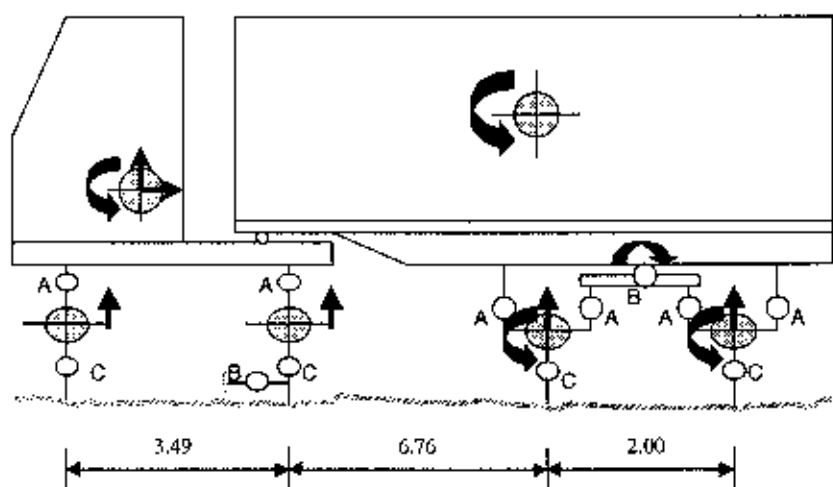
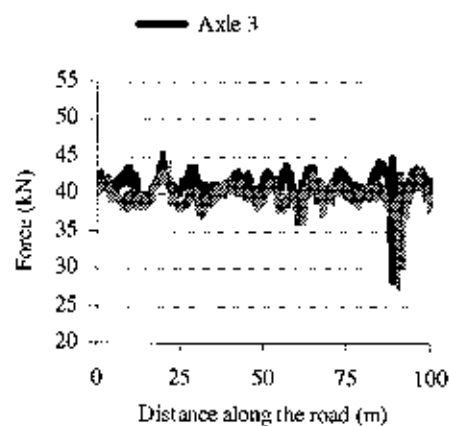
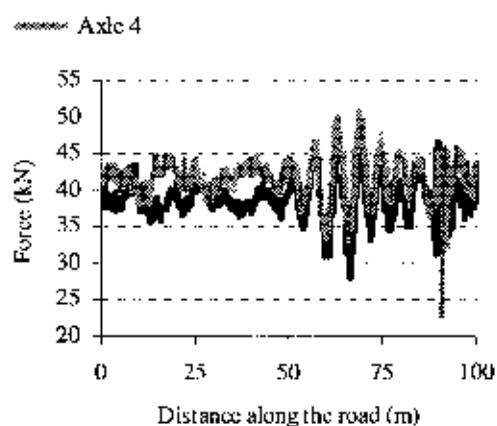


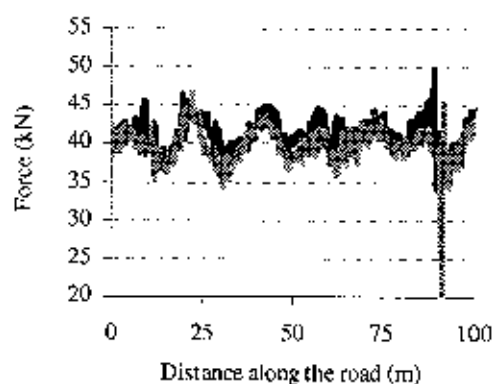
Figure 2 - Two dimensional tractor and trailer vehicle model with steel-spring suspensions (11 degrees of freedom)



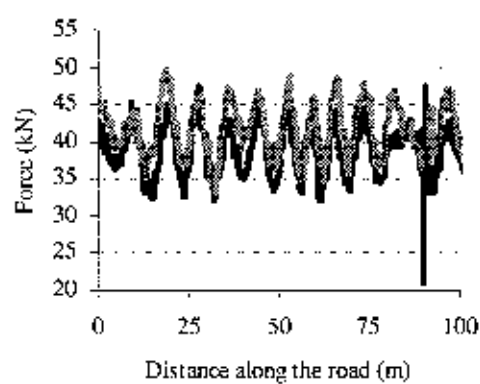
(a) Air suspension at 55 km/h



(b) Steel suspension at 55 km/h

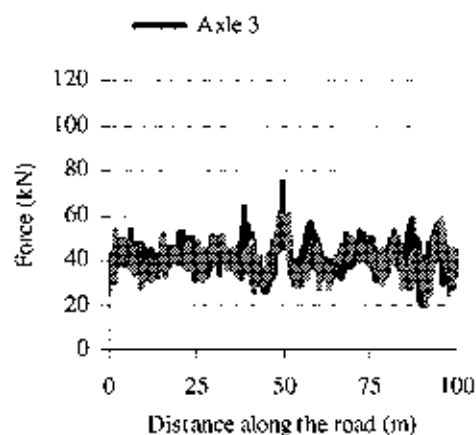


(c) Air suspension at 85 km/h

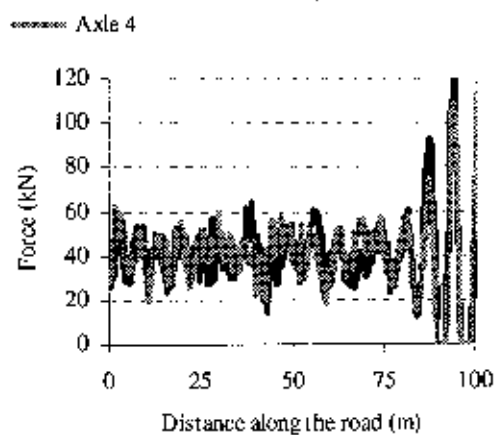


(d) Steel suspension at 85 km/h

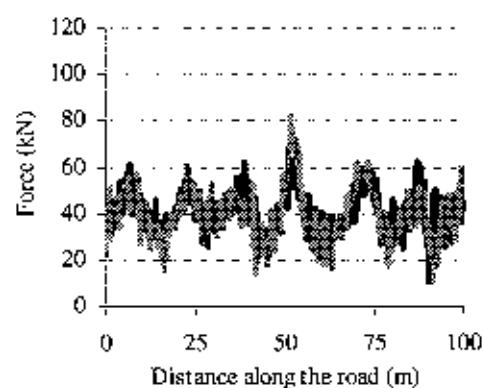
Figure 3 - Applied forces by air- and steel-sprung fully laden tandem (smooth profile)



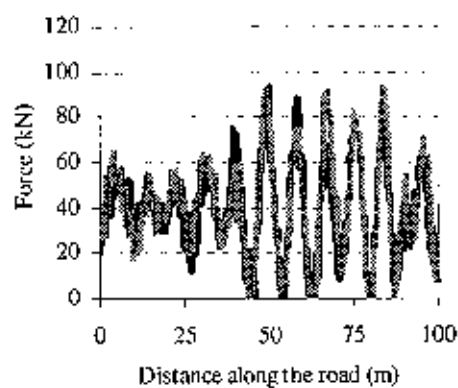
(a) Air suspension at 55 km/h



(b) Steel suspension at 55 km/h

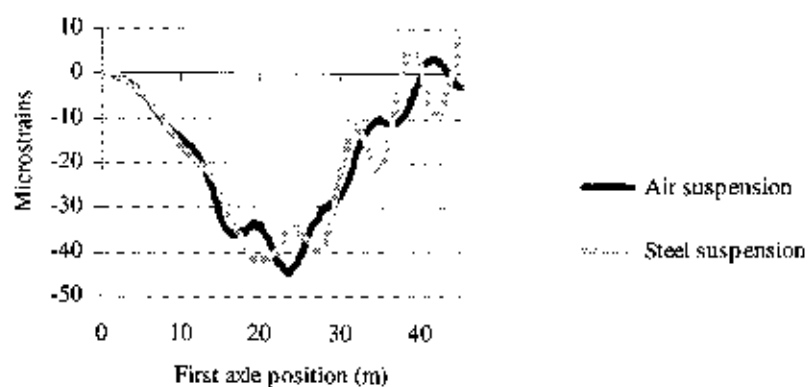


(h) Air suspension at 85 km/h

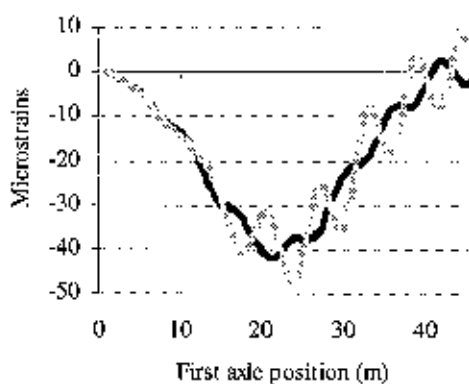


(c) Steel suspension at 85 km/h

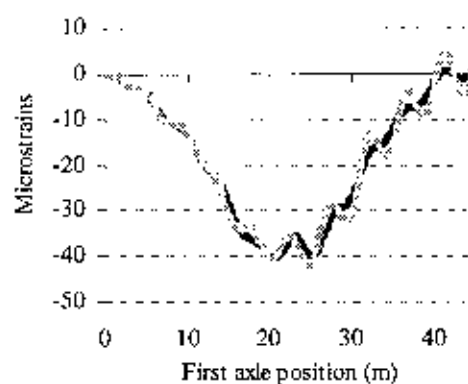
Figure 4 - Applied forces by air- and steel-sprung fully laden tandem (rough profile)



(a) 55 km/h

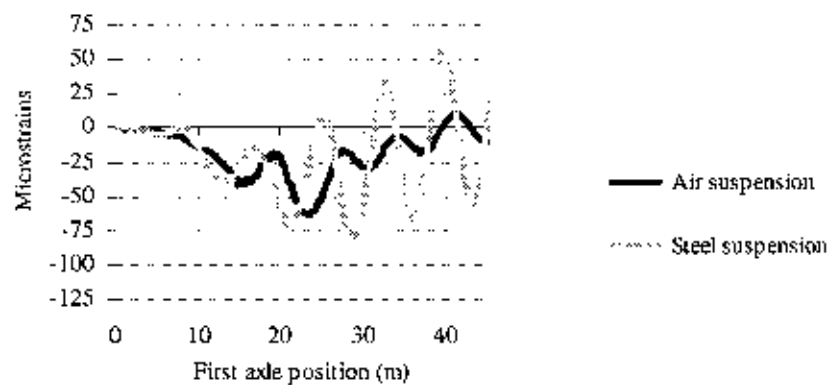


(b) 70 km/h

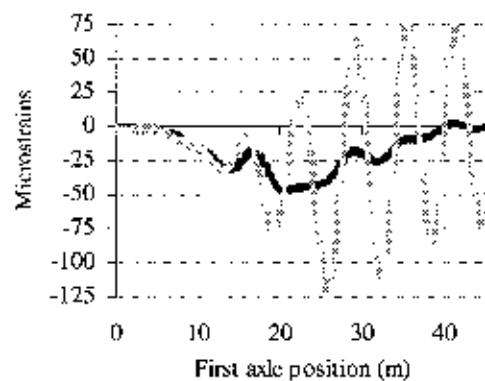


(c) 85 km/h

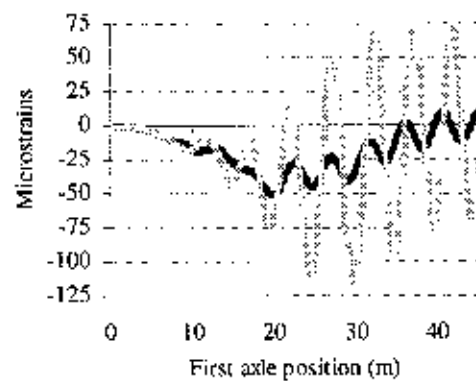
Figure 5 - Effect of vehicle speed and suspension on bridge response (smooth profile)



(a) 55 km/h

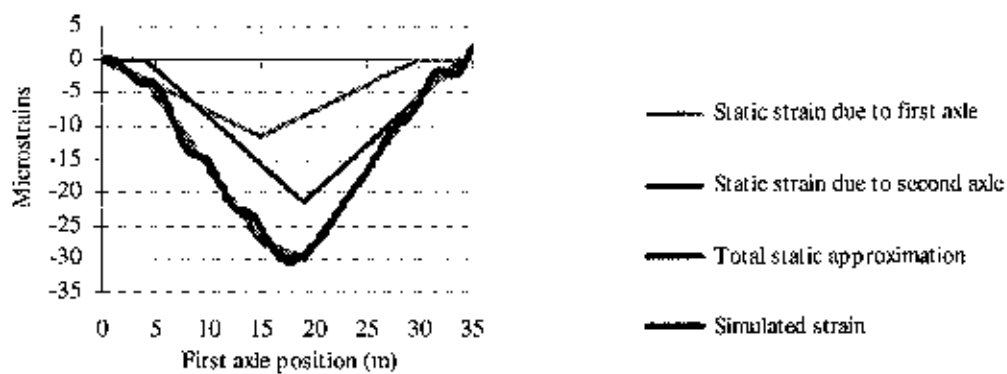


(b) 70 km/h

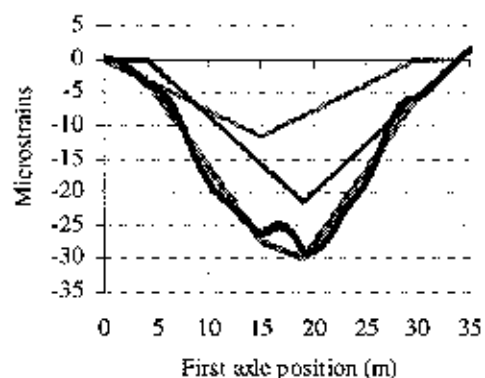


(c) 85 km/h

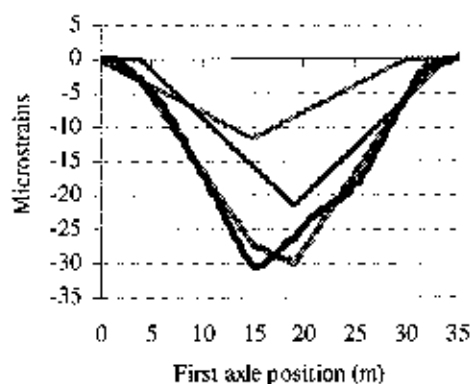
Figure 6 - Effect of vehicle speed and suspension on bridge response (rough profile)



(a) 55 km/h

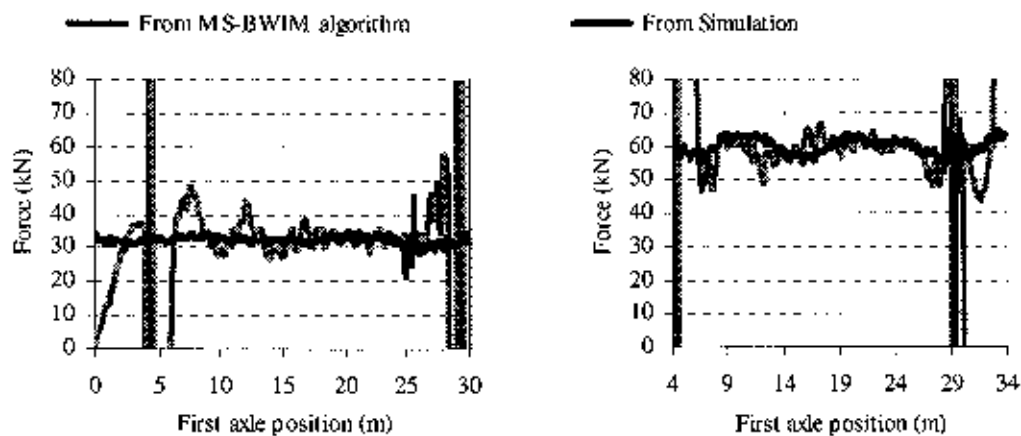


(b) 70 km/h



(c) 85 km/h

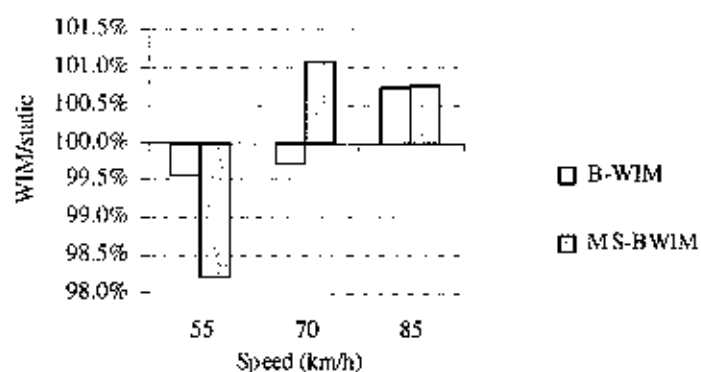
Figure 7 - Calibration of a static B-WIM algorithm (smooth profile)



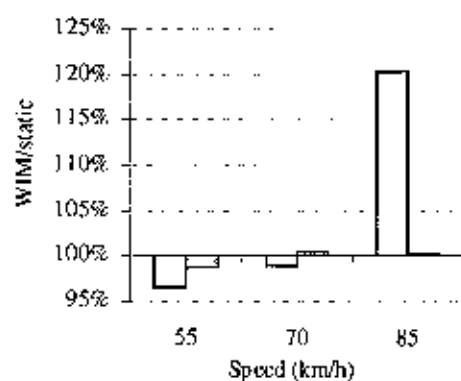
(a) Front axle

(b) Rear axle

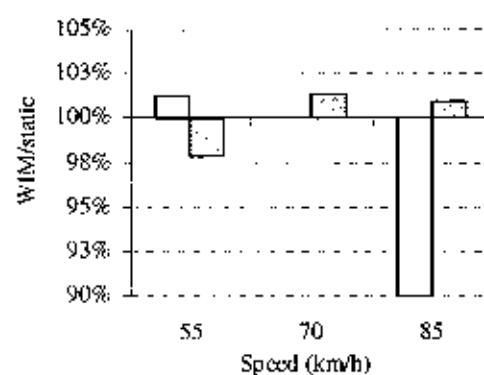
Figure 8 - History of axle forces for 2-axis calibration truck at 55 km/h (smooth profile)



(a) Gross Weight

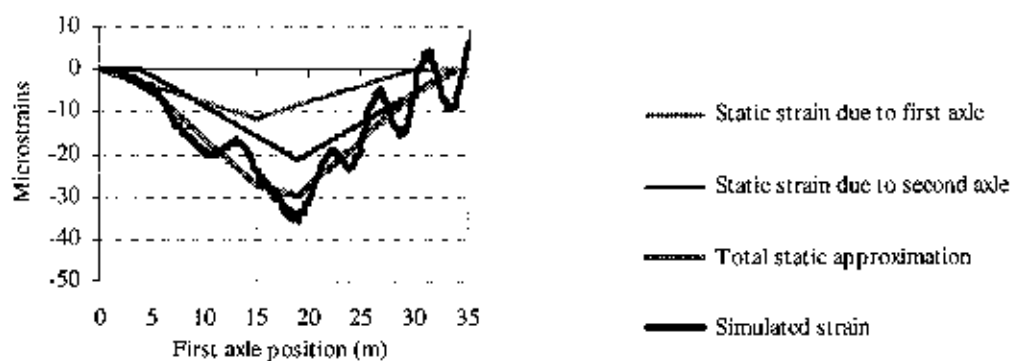


(b) Front axle

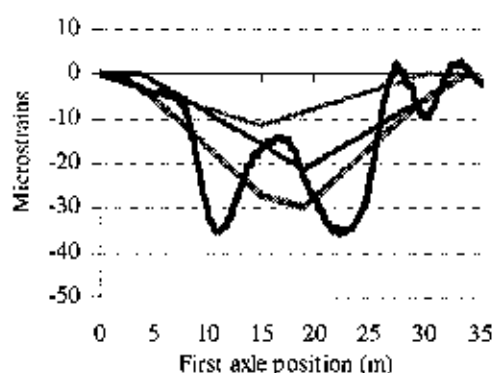


(c) Rear axle

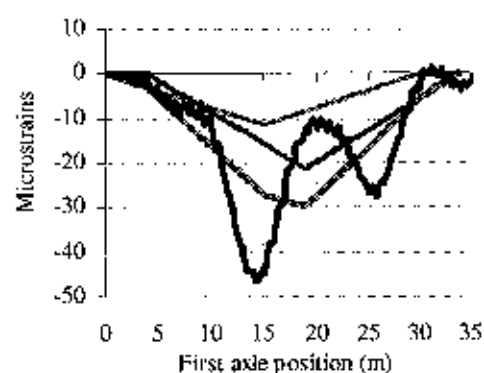
Figure 9 - WIM/static weight versus speed for 2-axle calibration truck (smooth profile)



(a) 55 km/h

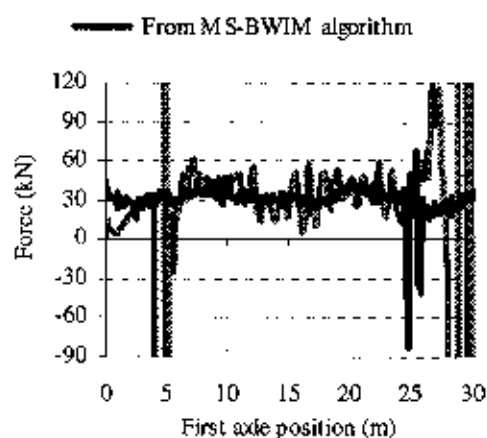


(b) 70 km/h

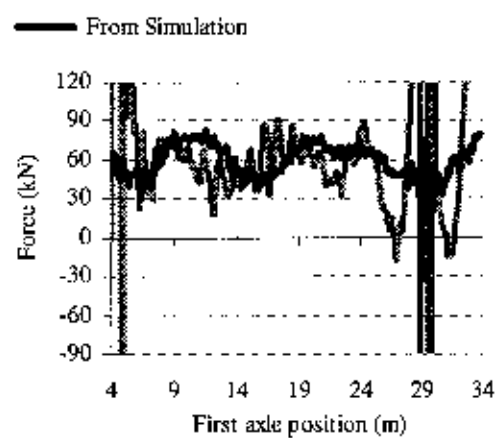


(c) 85 km/h

Figure 10 - Calibration of a static B-WIM algorithm (rough profile)

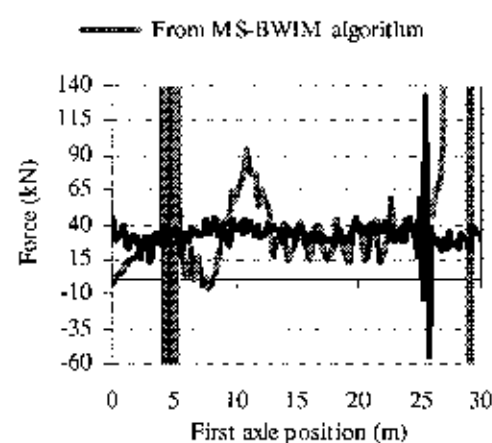


(a) Front axle

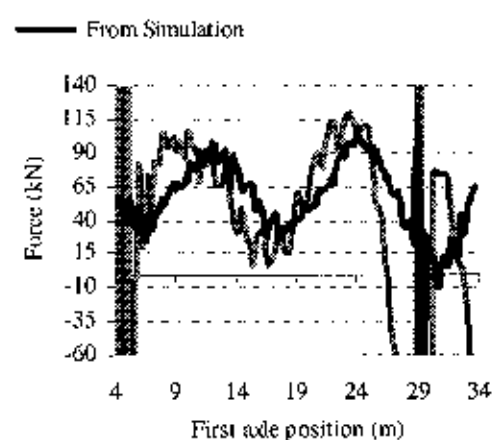


(b) Rear axle

Figure 11 - History of axle forces for vehicle at 55 km/h (rough profile)

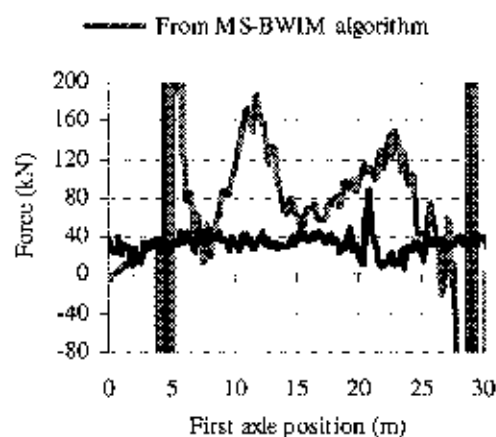


(a) Front axle

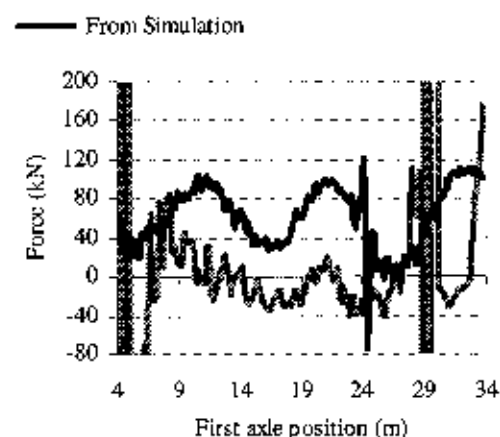


(b) Rear axle

Figure 12 - History of axle forces for vehicle at 70 km/h (rough profile)

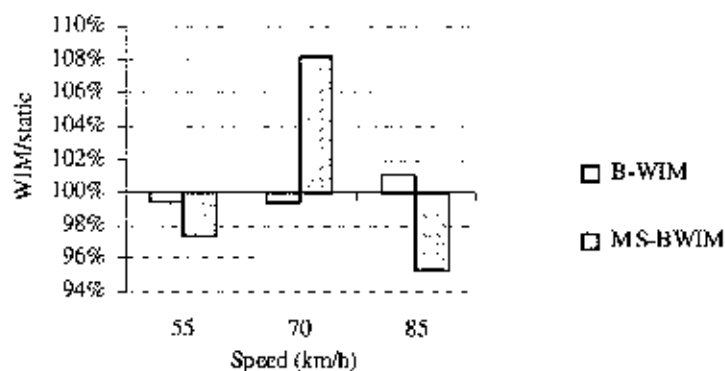


(a) Front axle

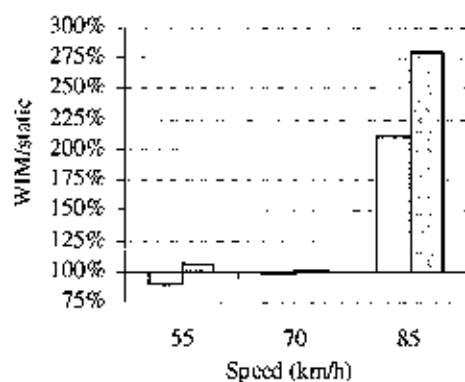


(b) Rear axle

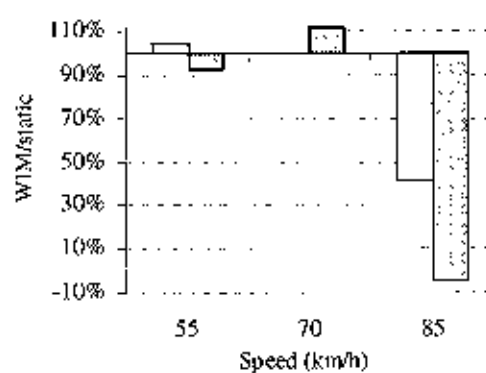
Figure 13 - History of axle forces for vehicle at 85 km/h (rough profile)



(a) Gross Weight

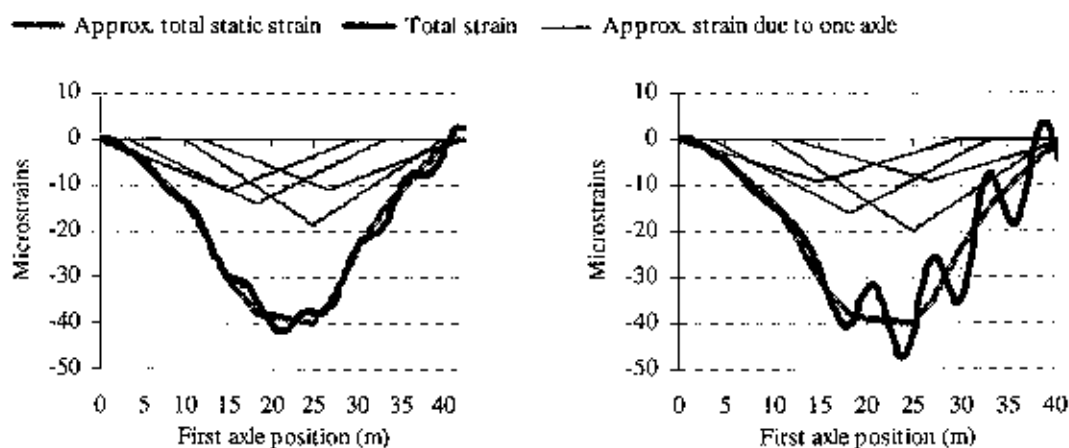


(a) Front axle



(b) Rear axle

Figure 14 - WIM/static weight versus speed for 2-axle calibration truck (rough profile)



(a) Approximation for air suspension

(b) Approximation for steel suspension

Figure 15 - Influence of suspension on static B-WIM (smooth profile)

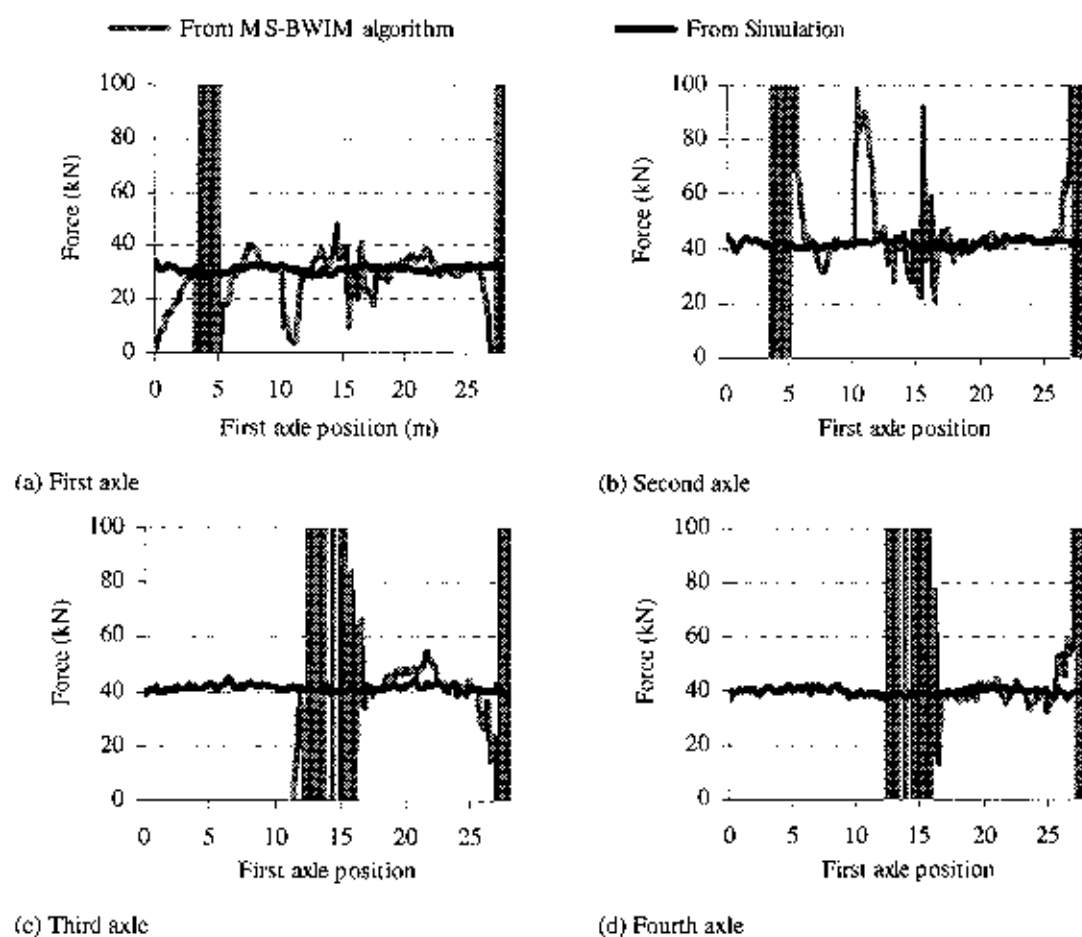


Figure 16 - Instantaneous calculation for a leaf-sprung 4-axle truck (smooth profile)

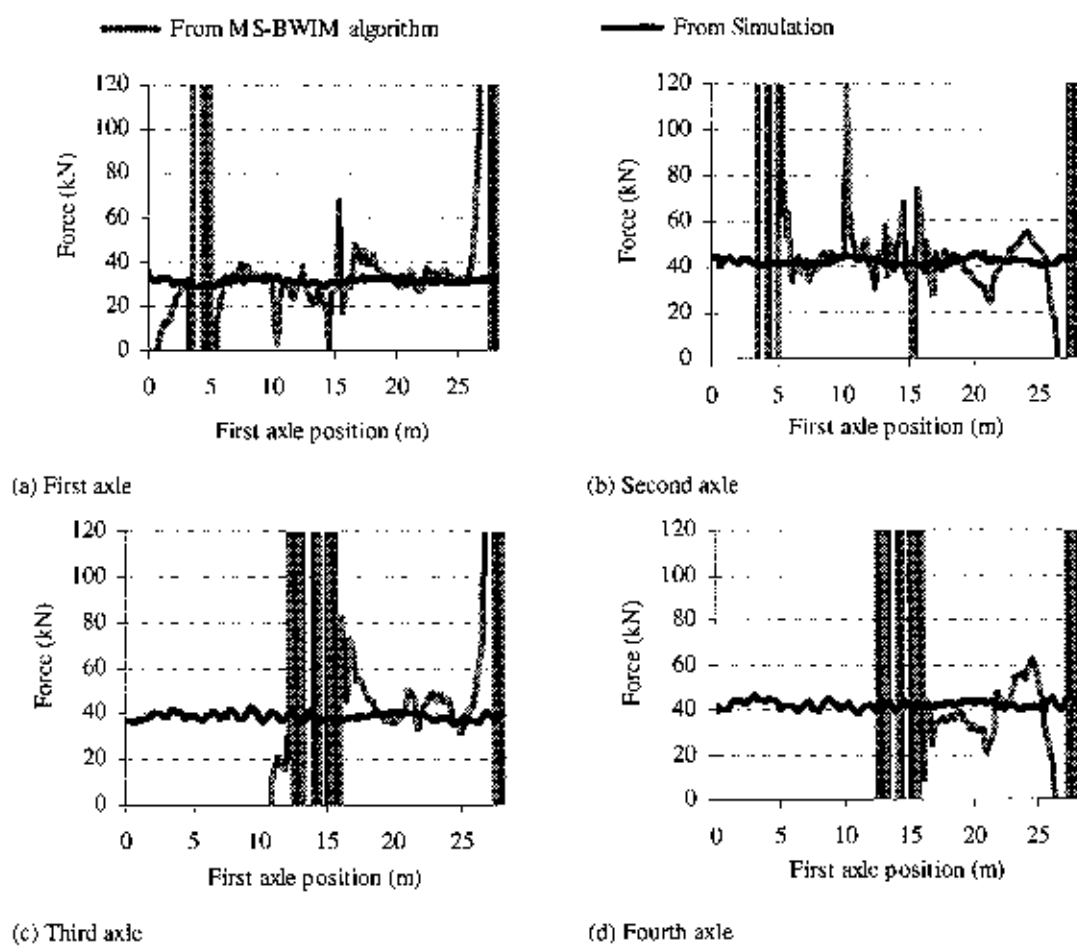
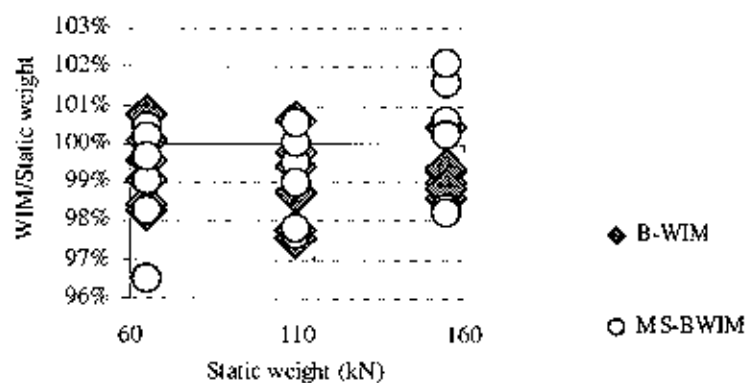
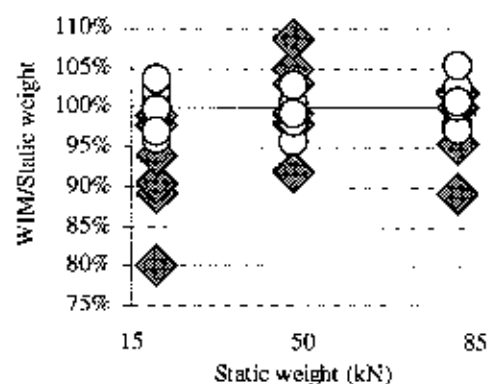


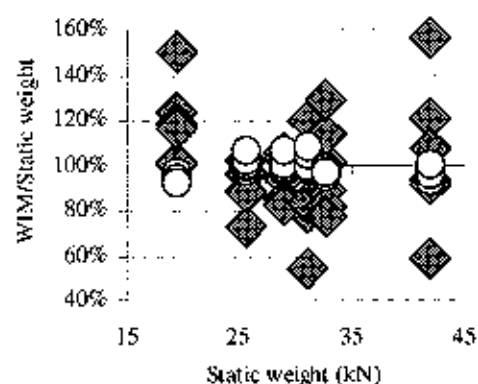
Figure 17 - Instantaneous calculation for a steel sprung 4-axle truck (smooth profile)



(a) Gross vehicle weight

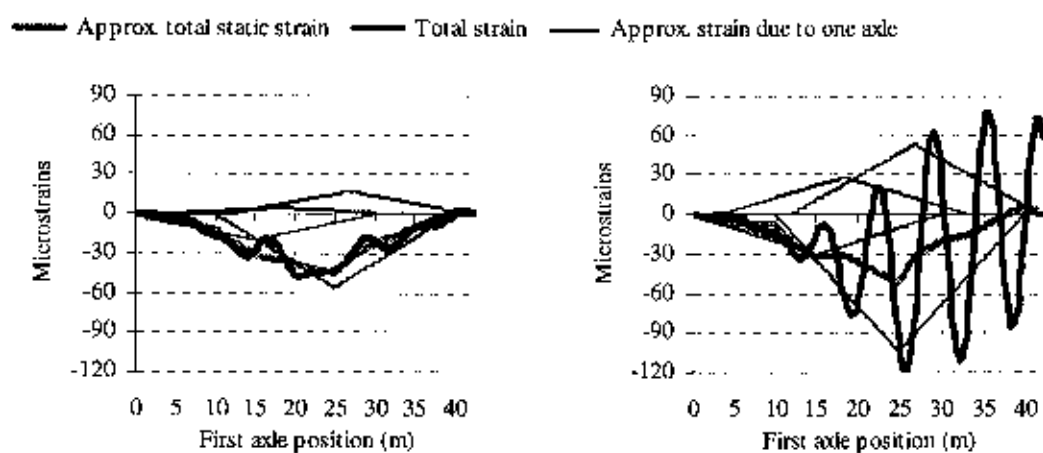


(b) Axle group



(c) Individual axle weights

Figure 18 - WIM/Static versus real static weights for 4-axle truck (smooth profile)



(a) Approximation for air suspension

(b) Approximation for steel suspension

Figure 19 - Influence of suspension type on static B-WIM (rough profile)

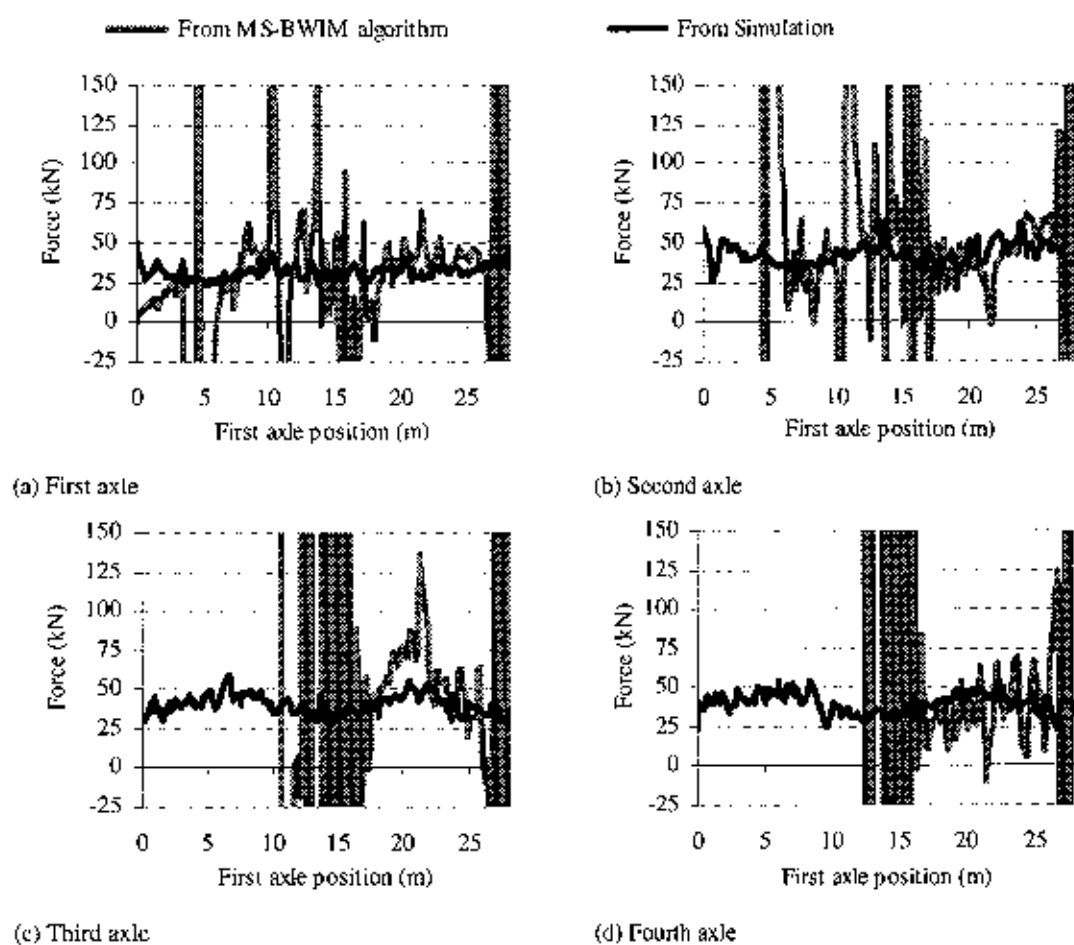
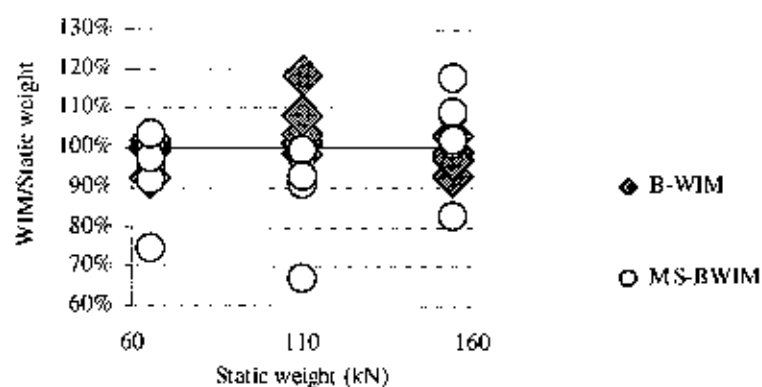
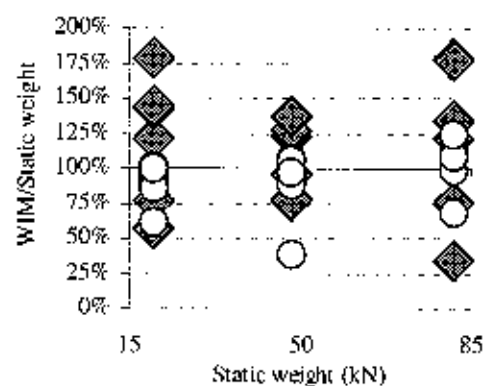


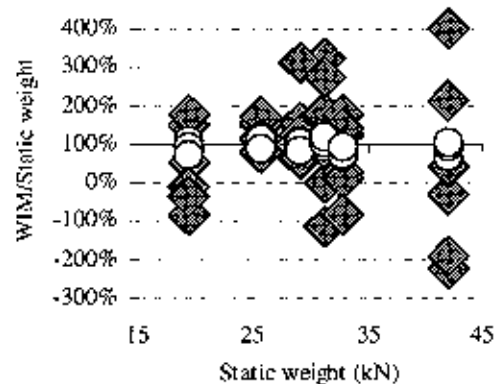
Figure 20 - Instantaneous calculation for a 4-axle truck (rough profile)



(a) Gross vehicle weight



(b) Axle group



(c) Individual axle weights

Figure 21 - WIM/Static versus real static weights (rough profile)

WHEEL LOAD MEASUREMENT, WIM, ACCURACY, TOP TRIAL

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Youssef, Nawaf, Bauhaus-Universität, Marienstraße 13, 99423 Weimar, Germany.
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ABSTRACT

The article focuses on methods for the design of WIM - sensor fields and algorithms for evaluation of dynamic axle load measurement data developed in the TOP TRIAL project.

Design methods and evaluation algorithms aim to increase measurement accuracy of dynamic weighing systems and to reliably determine static axle loads and gross vehicle weights with a high degree of precision.

Introduced are simulative, graphic and numerical methods for calculating and designing WIM sensor fields to minimize dynamic axle load variations measurement errors.

The various methods of calculation and design are based on deterministic as well as stochastic 1- and 2-sinusoidal wave approaches for modelling actual dynamic gross vehicle weight and axle load vibrations. The error minimization methods developed are based on the use of mean value and tolerance interval methods as well as the curve fitting method.

All methods introduced have been programmed and tested under real-life conditions.

1. TOP TRIAL PROJECT

The TOP TRIAL project, executed in the European Commission 5th Framework programme deals mainly with automated truck (overload) control and generation of enforcement rules for harmonization.

TOP TRIAL project partners are:

- Bauhaus-Universität Weimar
- Bundesanstalt für Straßenwesen, Bergisch Gladbach
- Bundesministerium für Verkehr, Bau- und Wohnungswesen, Bonn
- ETH Zürich
- Instituto das Estradas de Portugal, Lisboa
- Ministry of Transport Public Works and Water Management, Delft
- mit Autobahndirektion Südbayern, München
- Oberste Baubehörde im Bayerischen Staatsministerium des Innern
- PAT GmbH, Ettlingen
- Polizeipräsidium Oberbayern, München
- Produtos para Análise e Controlo de tráfego, Oporto
- PTV Planung Transport Verkehr AG, Stuttgart
- Rigobert Opitz Consulting & Engineering, Malsch

TOP TRIAL info: 1)Web: www.TOP-TRIAL.de; 2)E-Mail: kontakte@TOP-TRIAL.de

The project is funded by the European Community in the framework of the "Information Society Technology (IST)" Programme.

2. TOP TRIAL TECHNICAL PROJECT DESCRIPTION

2.1. Overview

Overloaded heavy vehicle traffic implies a considerable safety risk for other road users. The objective of the TOP TRIAL project is to create the preconditions for an increased traffic safety and an improved protection of the road

infrastructure by new technologies and methods of automated truck overload control. The solution worked out within the framework of the TOP TRIAL project, combines the currently two suitable weighing technologies for truck axle loads (bending plates and crystal sensors) in a simulatively tested multi-sensor array. Efficient heavy vehicle control on automatic weighing stations to enforce overloads on motorways means:

- protection of the road infrastructure from damage
- improved traffic safety and less traffic incidents
- fair competition for fleet operators
- cost savings and efficiency by means of direct control on site without staff

2.2. Top Trial Objectives

The first step of the TOP TRIAL project was to agree on the following objectives:

- development and construction of a optimised test site for "High-accuracy Staggered Multi-sensor Arrays" for dynamic weighing of heavy vehicles and overload enforcement
- to implement WIM experiences in an usable approach: simulated and optimised sensor configuration – use of twin WIM technology- sparsely coded and
- operation of the test system on a highly frequented motorway under real traffic conditions using more than thousand random overloaded truck probes
- development of measurement improvement algorithms for automatic WIM
- preparation of technical requirements, procedures of automated controlling truck overloads and initiation of a European Forum „Weigh-in-Motion for Enforcement“
- organization of an international workshop Munich 2002 on overload control

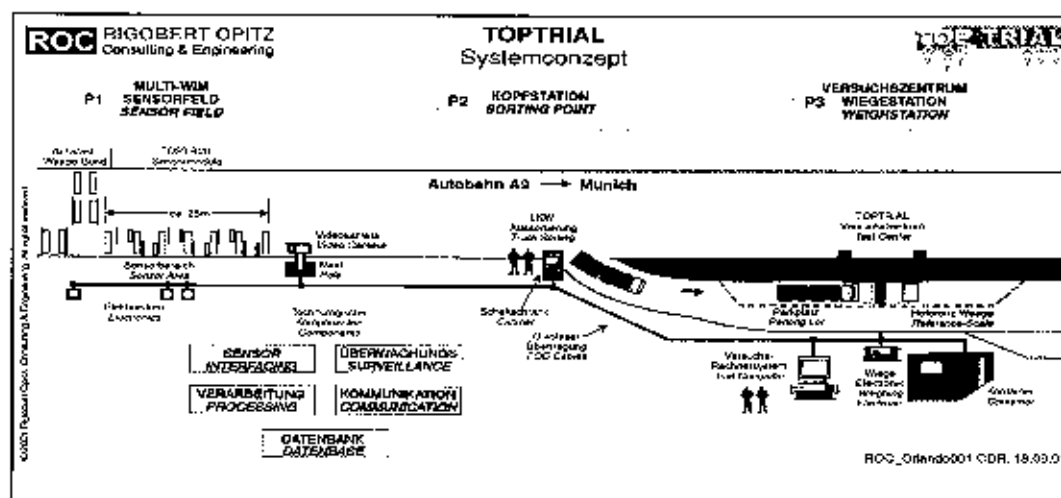


Figure 1 -TOP TRIAL System Concept

2.3. Top Trial System Concept

The overall TOP TRIAL test system includes 3 major components located in an area of 3 km along the Autobahn A9 in Germany, interconnected via a high speed communication system.

The conceptual studies and the simulation analyses finally led to TOP TRIAL sensor configuration consisting of 16 sensors. The weighing sensors of the multi-sensor, sparsely coded twin technology array have been installed in an area of approx. 25 m on the right lane of the Autobahn A9 near Ingolstadt. All passing heavy vehicles are weighed dynamically at ordinary speed. A video camera takes front pictures of overloaded trucks and sends them to the sorting point and the trial container. The weighing electronics record the signals of the sensors and preprocess them. All vehicle details are transmitted to a database and later combined with the data of a static reference scale system.

A special concrete area and a reference scale located on a parking lot 2km behind of the test site was designed and installed. The calibrated axle load weighbridge is used in slow motion in order to provide accurate axle loads as a basis for comparison with the calculations by the algorithms. The best algorithms are analyzed.

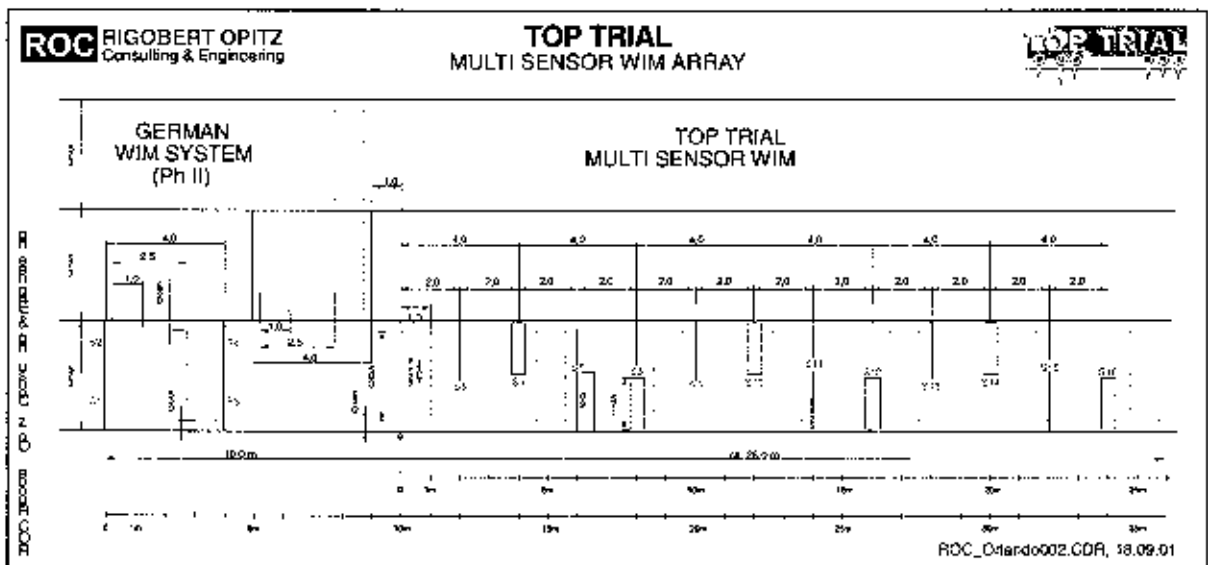


Figure 2 - Detailed Sensor Layout Drawing Optimized by Simulations

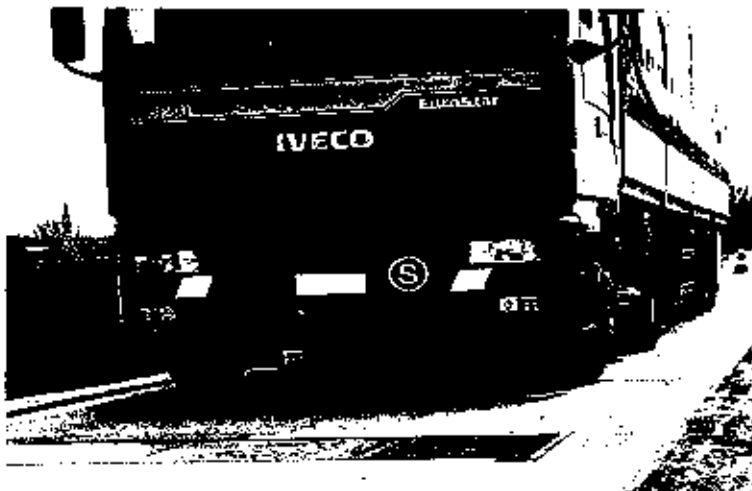


Figure 3 – Reference and Enforcement Scale.- Designed as Dynamic Axle Weighbridge

3. SCIENTIFIC ANALYSIS OF SENSOR ARRAYS AND DESIGN OF ALGORITHMS

3.1. Simulation and Construction

Part of the project was the analysis and simulation of the best sensor field combination on a scientific base. Different researches and the design of special simulation tools (by the Bauhaus-University Weimar) supported the optimisation of the layout of the sensor array. Key issues of these research work were:

- Design of algorithms for the minimisation of measurement errors of dynamic weight measurements by the use of multi-sensor WIM systems
- Simulative design of the sensor array with "minimum" measurement errors

Basic literature of this work is:

- 'Design of multiple-sensor weigh-in-motion systems' [1],
- 'WAVE-project' [2],
- 'Measurement and analysis of dynamic tyre forces generated by lorries' [3] and others.

This literature describes the sinusoidal wave approach which we considered to be a possible theory for creation and validation of WIM-sites.

The sinusoidal wave approach is not a very sophisticated one for studying vehicle/pavement interaction, however it still has some scientific potential.

The main objective of the 8th man month TOP TRIAL project was a usable runtime version and not a high sophisticate solution.

It was intended to use the potential of the sinusoidal approach for construction of WIM sites as well as for validation of measurement data.

Therefore the existing sinusoidal wave approach was extended by a 2-sinusoidal wave model and an optimisation tool for minimising error deviation which is based on a sinusoidal wave model too in order to get a consistent solution for construction and validation of data.

Furthermore an increasing of measurement accuracy was expected by the use of suitable sinusoidal based algorithms such as curve fitting algorithms.

Second objective of TOP TRIAL was a minimum cost at a minimum number of the sensors used in the WIM array.

Most of the man month were planned to develop and algorithms for validation test data. (All experiences made during the TOP TRIAL project have shown that this was right as the parallel developing of algorithms, gathering of measurement data and calibration of sensors caused a lot of efforts. Due to the permanently changing conditions of the weighing technology algorithms had to be adapted often.

For more efficiently use of the construction/algorithms complex load frequency and speed of the traffic flow are a necessary inputs.

Additional to the methods described in literature the construction method which was developed allows to take the distributions of these parameters in to account and calculates optimum sensor distances.

Finally the developed algorithms for validation of measurement data test the sensor outputs whether they are sinusoidal waves or not. If data is tested positive, static loads will be calculated, otherwise it is rejected.

During the TOP TRIAL project 2 digital simulation tools were created allowing generation of WIM sensor fields in substantial diversity and determination as well as comparison of important qualitative and quantitative parameters. These tools also enable simulations of dynamic axle load oscillations of actual vehicles featuring highly detailed characteristics so the influences of real parameters on to the sensor layout could be estimated.

Especially the interrelations between speed, load, type, frequency, dynamic load amplitude and other factors are more or less unknown and could be analysed using the simulator in combination with specific mathematical tools.

With the first simulator a great diversity of sensor array variants was designed and evaluated by simulation using the so-called "Sinus Wave Approach" to design MS-WIM sites by calculating and evaluating maximum and mean values of measurement errors, deviation, distribution and envelope errors of equidistant and inhomogeneous layouts of WIM - sensor arrays depending on number of sensors used. To find out one or more good test sites, Numerous possible equidistant sensor arrays with a maximum length of less than 24 meters and all kinds of vehicles types, which are represented by wave lengths of axles and body load oscillations, moving over these sensor arrays within the framework of the variability of the most important parameters were simulated. All variants of sensor arrays with different sensor distances from 0,25 to 7 meters were evaluated by calculating and comparing their mean values of maximum measurement errors depending on wave lengths from 2 to 20 meters using an increment of 0,25 meters. However, a mathematical model was needed for simulation of dynamic behaviour of moving vehicles. The results of the 1- sinusoidal wave approaches and the 2-sinusoidal wave model - were compared and tested in order to estimate the difference of measurement error.

The 2-sinusoidal wave model works on base of a superimposed load wave of vehicle weight and axle load, (see figure 4 and 5). The results of the simulation are:

- all maximum measurement errors - worst case;
- the mean value of maximum measurement error for each different sensor array;

- the mean value of maximum measurement error dependent on wave lengths for selected individual sensor arrays;
- the distributions of measurement error of individual sensor arrays and for each wave length;
- the distributions of the measurement error of a sensor array – over all wave lengths;
- the n %-quantiles of measurement errors and other interesting values.

The second simulator was developed to create individual vehicles crossing sensor arrays and to calculate dynamic axle loads and measurement errors of axle loads as well as vehicle weights and errors. These individual vehicles feature numerous specific properties such as speed, frequencies, weight, number of axles, distribution of weight and load on the axles. The interrelations between speed, wave length and dynamic load amplitude are variable too. (constant, digressive, progressive, proportional, or exponential functional interrelations are selectable).

The results of simulation were compared to the real-life data (sensor outputs) in order to validate and upgrade the basic approach by the use of an additional random procedure and others for improving the construction tool.

A specific mathematical optimisation model was developed to calculate optimum sensor distances for a given number of sensors, so that the deviation of mean values of the measurement errors over all wave lengths considered is at a minimum. The effect of changing the number of sensors used on the shape of an optimum WIM site was estimated and taken into account for selecting the best solution.

A graphic method for design of 'good' inhomogeneous sensor layouts on base of the maximum measurement errors depending on wave length was developed tested too.

An example of graphic construction of sensor arrays based on the use of the results of error simulation is shown in figure 6.

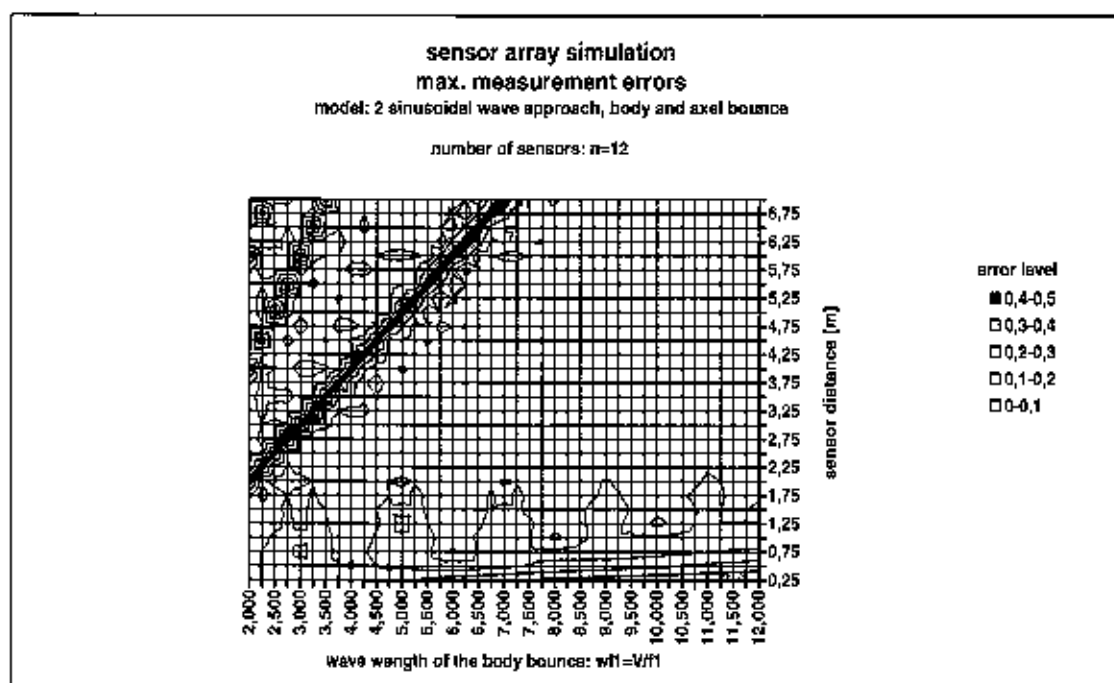


Figure 4 – Sensor array simulation, 2-sinusoidal wave approach

Figure 5 shows the optimum envelope measurement error depend on wave lengths calculated by the use of the numerical optimisation tool.

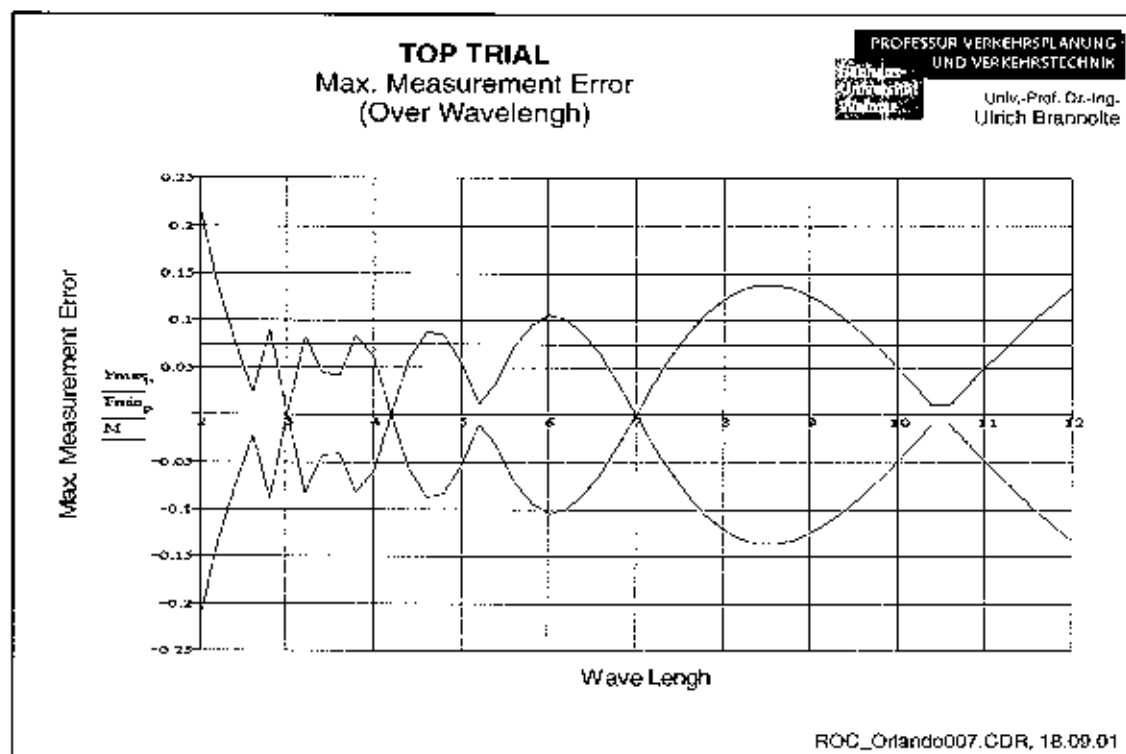


Figure 5 – Envelope measurement error diagram, optimum solution – 2 sinusoidal wave approach

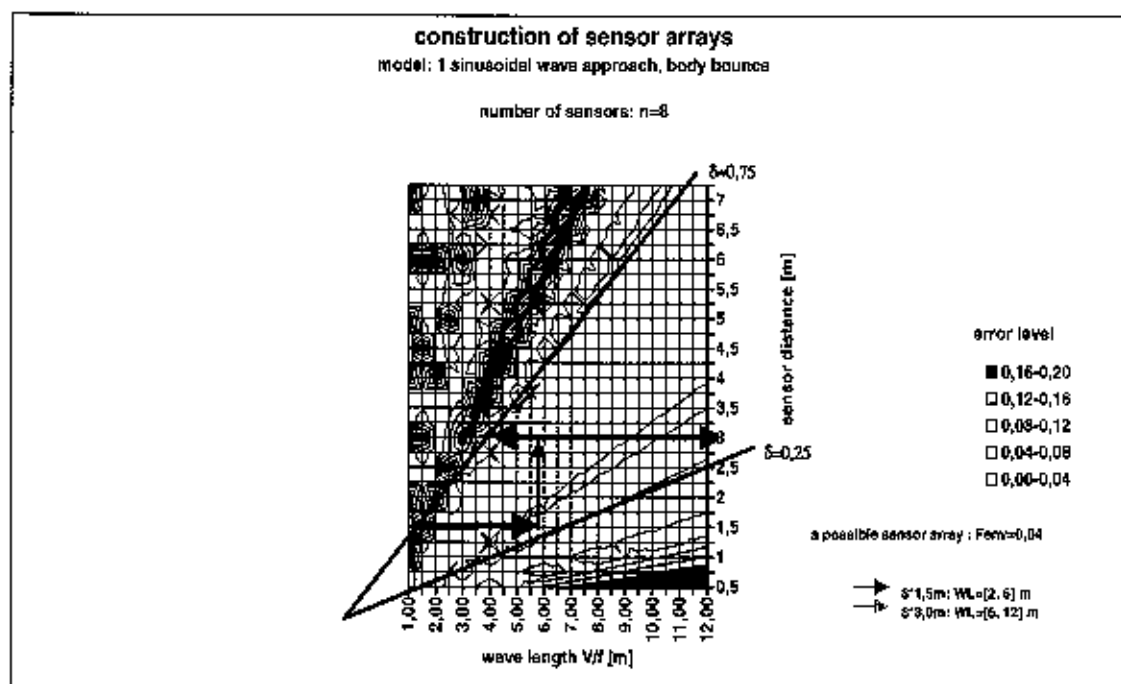


Figure 6 – graphic construction of a combined sensor array, - 1 sinusoidal wave approach

The maxima of all mean values of the measurement errors are calculated with regard to the variety of all possible phase cuts of the load oscillations and in evaluation of approx. 1.7 billion variants, a select few preferred variants were chosen, compared to each other and to the optimum sensor layout and proposed for the configuration of the equidistant MS WIM test site. The proposed sensor array with 12 sensors is an equidistant one with a sensor spaces of 2 meter.

The simulations of this sensor array have shown a good performance in range from 3 to 12 meters wave length. The measurement errors are relative high for wave lengths less then 3 meters.

3.2. Measurements

The dynamic measurement data were directly reported into the controlling container near the sensor array in order to be pre-checked and saved.

In the first two month 228 trucks with about 900 axles were analysed. The data quality of this two month was as like as expected and relative good static weights could be calculated. By the use of the averaging method an measurement error less than 5% of the static axle load with a probability of 83% was the result. Next two month the data deteriorated more and more. The probability of the 5% measurement error reduced to 80% and less.

Unfortunately the data quality reduced rapidly in the following month and the sensor array didn't work well since this time. So data of only 560 trucks could be used to develop effective algorithms for reducing measurement errors. The first measurement data will be saved under <http://www.toptrial.de/> soon.

3.3. Algorithms

A lot of algorithms for minimisation of measurement errors of dynamic weight measurements data of the multi-sensor WIM array were designed and tested by the use of. Different strategies were used to find out a precise static value for each axle load and vehicle weight. These strategies base on the averaging method, clustering method, tolerance interval method, Fourier-Transformation, curve fitting method, filtering methods and others. The results of the first two strategies are described in this article. A special extreme-value-removing (replacing)-method and a value-correcting-method have been used for calculation of errors of the articulated truck type '98' (two-axle tractor / three-axle trailer) and effected a significant improvement of the accuracy. The most effective methods have been combined and employed to improve measurement results, (see figure 7, 8 und 9).

These methods have been executed more times during the one-year TOP TRIL period but the impact of seasons and climatic conditions couldn't be calculated as the weighing technology used didn't work stable over month.

The measurement accuracy is a product of construction and the use of the algorithms as well. The sinusoidal approach of construction of the sensor layout and the application of the developed algorithms effected an error of 5% with a probability of 87% only for axle 2 of all truck types.

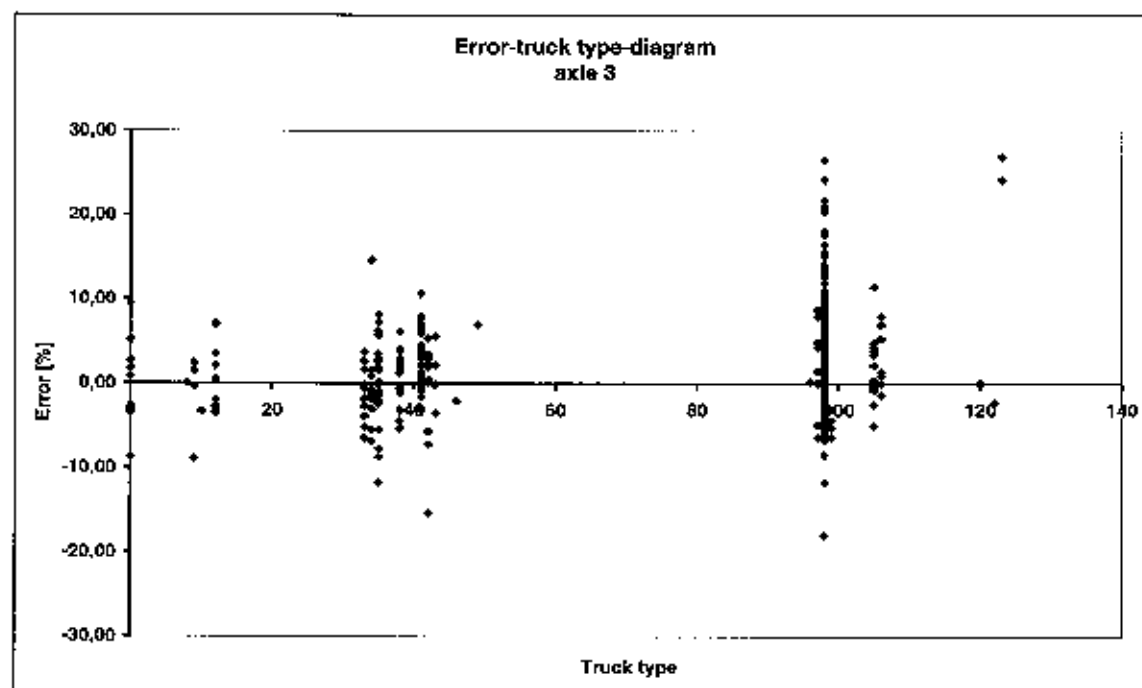


Figure 7 – Error-truck type-diagram, axle 3

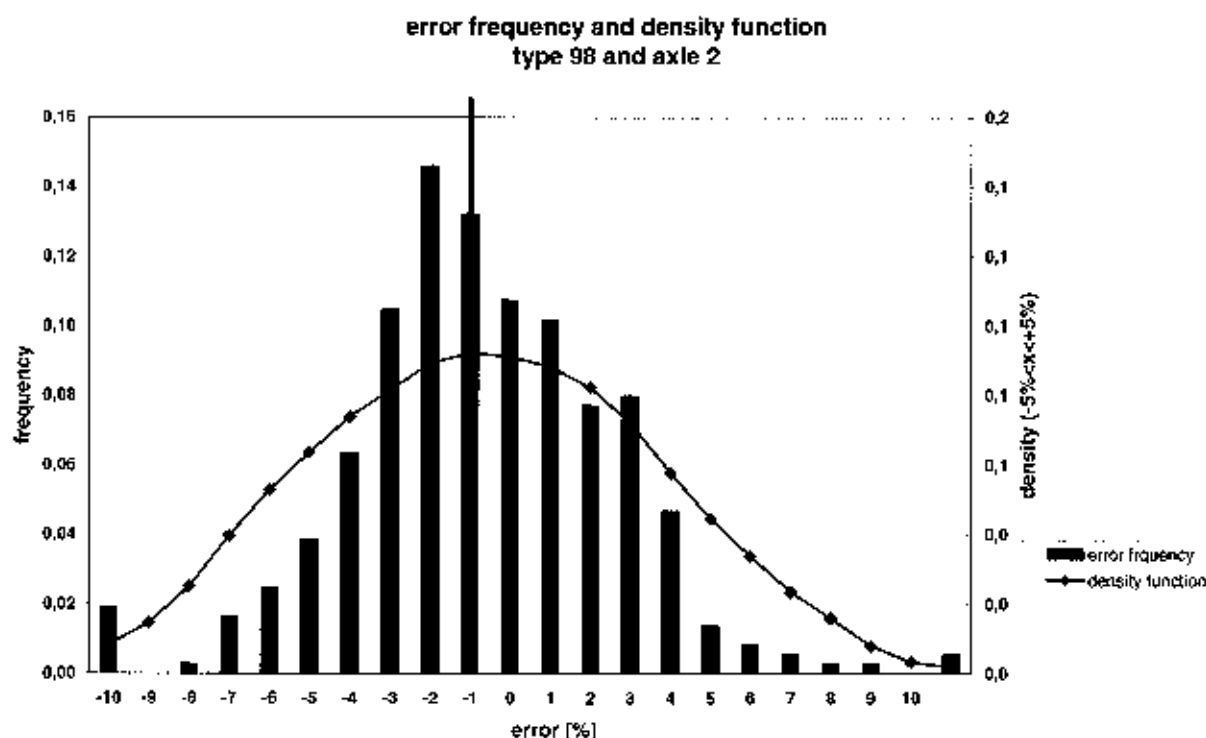


Figure 8 – Error frequency and density function, truck type 98 (two-axle tractor / three-axle trailer), axle 2

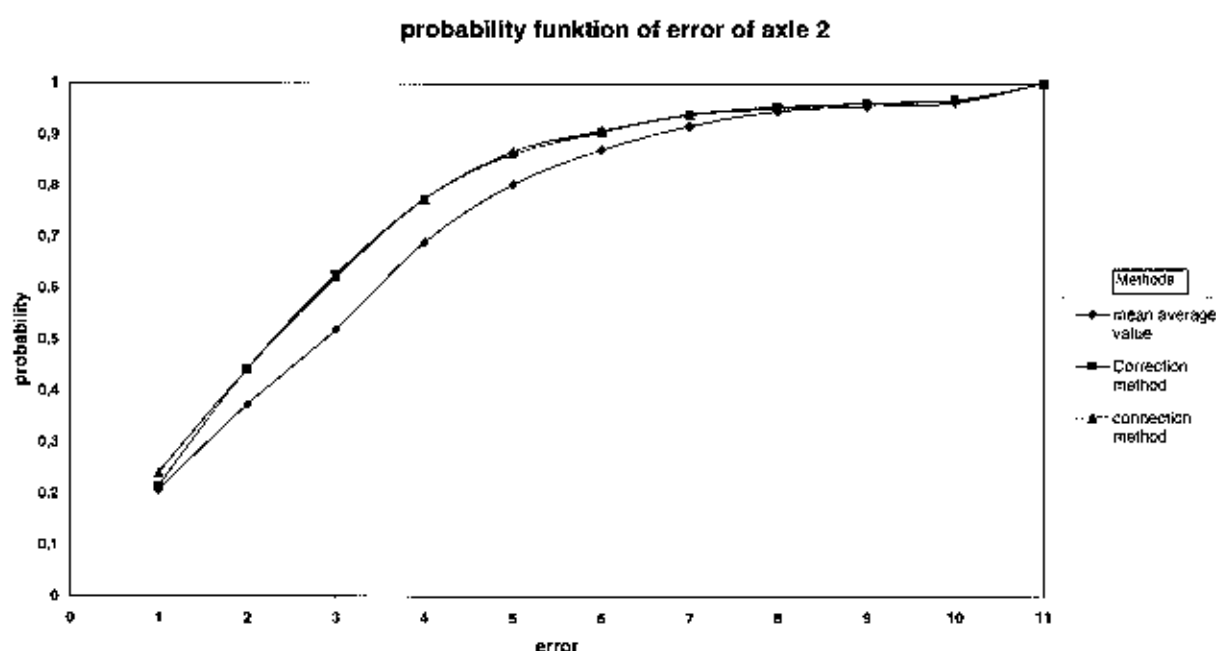


Figure 9 – probability function of axle 2 , all truck types

3. CONCLUIONS

In this project the WIM-sensor field was designed under the assumption of an uniform distribution of speeds and frequency of the traffic flow. Due to the lack of precise data of truck speed and load oscillation-frequency distribution the reality wasn't taken into account sufficiently. The feed back of the main characteristics, which should be worked out of the traffic flow, will lead to more suitable sensor layouts with lower measurement errors.

Other methods and algorithms such as curve fitting or Kalman filtering method should be examined too. These methods were tested already but for single vehicles only, as a statistical validation needs much more programming work caused by connection of different software tools.

Further, researches are necessary to improve the measurement accuracy and probability in order to apply the automatic enforcement system.

The results of future measurements and analysis of the WIM test site, such as the empirical distributions of vehicle speed or the frequency of vehicle and axle load oscillations and others will substitute the model assumptions of the past phase and will have a decisive impact on the design of commercial WIM sites too.

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TANKER TRUCKS IN THE CURRENT ACCIDENT SCENE AND POTENTIALS FOR ENHANCED SAFETY

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ABSTRACT

The transport of hazardous goods, which today is governed by extremely restrictive laws, and which constitutes approximately 10% of road transport, involves great risks to people, the environment and material objects. This is especially true if flammable liquid hazardous goods are released. Large-scale damage or even disasters may be the result. Terrifying incidents in the past, such as Herborn, Germany (1987) or San Carlos de la Rapita Alfauques, Spain (1978), clearly illustrate the scale of the damage that could be involved.

Under this circumstances the development of road accidents involving vehicles carrying hazardous goods will be shown by using statistical data of Germany. In summary, due to a variety of measurements a decrease in road accidents involving dangerous substances can be recognized. Nevertheless, a risk analysis of hazardous goods transport shows that the road transport accident rate is at least seven times higher than that of rail and inland waterway transport. Furthermore, major accidents (involving the escape of more than 10.000 liters) occur most frequently on the road and least frequently by rail.

Therefore the primary sources of risk leading to hazardous substances escaping from a tank will be focused. These are mainly single accidents with rollover of the tank vehicle (roughly 60%) and collisions with other heavy vehicles, in which the rear or side of the tank are involved. Finally active and passive safety measures will be proposed which are in line with the state-of-the-art technology and which are effective to reduce the probability of accidents resulting in the release of liquid hazardous goods from protective tanks.

INTRODUCTION

In 1998, the total volume of goods transported in the European Union was 2,870 billion tonne kilometres (Eurostat, 2000). The largest proportion (1.255 billion tonne kilometres or 43.7%) was transported by truck on European roads. Road transport also had the largest growth rate in the period between 1990 and 1998 at 35%, while rail transport decreased by 6% (Figure 1). In 1988, cross-border transport accounted for 20% of the volume of goods carried by road in the EU, tri-country transport for 2%, intra-country transport for 77% and cabotage for roughly 0.2% (Hedbrand, 2001). These general statistics prove that the truck is the number one means of transportation within the EU. Moreover, forecasts predict that there will be a further increase in the volume of goods transported by road (BMVBW, 2000).

Furthermore, our modern, highly industrialized economic system is inconceivable without the use and transport of hazardous goods. Under § 2 of the German law on hazardous substances, these are "a danger to public safety or order, particularly to the general public, important public property, the life and health of people and to animals and objects".

Nevertheless, the transport of hazardous goods, which today is governed by extremely restrictive laws, and which constitutes approximately 10% of road transport (Staebler, 2001), involves great risks to people, the environment and material objects. This is especially true if flammable liquid hazardous goods are released. Large-scale damage or even disasters may be the result. The damage potential from fire (Figure 2) and explosions and the diffusion of harmful substances carried through the air and atmosphere connected with these is particularly high (Gwehenberger, 1998). Terrifying incidents in the past, such as Herborn, Germany (1987) or San Carlos de la Rapita Alfauques, Spain (1978), clearly illustrate the scale of the damage that could be involved.

ACCIDENT STATISTICS

Unfortunately, there are no detailed statistics on the number of tanker truck accidents within the EU available. However, since the German federal statistics (German Federal Statistics Office, StBA, 2000) started separately recording road accidents involving vehicles carrying hazardous substances, both with and without the release of the substances, a downward trend in the number of accidents involving injury to people and serious material damage has been seen (Figure 3). The official statistics also show that with 52 accidents in 2000, in which hazardous substances were released and which also caused injury to people or serious material damage, accidents involving hazardous substances are infrequent.

If we examine the accidents from 1999 according to the hazard class of the substances transported (Figure 4), the majority of accidents involving released substances (34 of 52 cases or approximately 65%) involved hazard class 3 ("flammable fluids"). This correlates well with the volume of goods carried which, at 66.4%, is also at its highest in this hazard class (KBA/BAG, 2000).

Most flammable fluids (usually gasoline, diesel and fuel oil, as well as other related liquid hydrocarbon compounds) are transported in tank trucks or tank trains. However, up to now, there have been no official detailed accident data statistics in this area. There are no detailed statistics on tank truck accidents available in other EU countries either, which means that other sources of information must be used.

Although it is not directly comparable with the German Federal Statistics Office data, the "hazardous substances accident database" (GUNDI, 2001) compiled by the editors of "Gefährliche Ladung" ("Hazardous Load") provides a good overview of the accident situation for tank trucks. These statistics are primarily based on an analysis of 600 German daily newspapers and additional research carried out by the editorial department with the police, fire department and local authorities. In this context, the graph at the bottom of Figure 5 shows the number of accidents involving tank trucks, and is subdivided into accidents where the tanks remained intact and accidents where hazardous substances were released. According to these statistics, in 194 of the 328 accidents and incidents recorded between 1995 and 2000 (which roughly corresponds to 60%), hazardous substances were released in varying quantities. A total of 752.6 t of hazard class 3 substances were released, the majority of these being gasoline, diesel and fuel oil (683.2 t).

The GUNDI database also shows that in the six year period examined there were 19 cases of fire and explosion and 153 cases of damage to the environment (graph at the top of Figure 5). An extract of the accidents involving fire, including details of how the accidents happened, can be found in the appendix (GUNDI, 2001). Even if, fortunately, no large-scale damage has occurred in Germany in the recent past, accidents involving fire and explosion can cause terrible damage, with a probable maximum loss (PML) of Euro 50 million to Euro 100 million (Gwehenberger, 2000). Economic damage and human suffering, which cannot be expressed in monetary terms, are not included in this estimate.

In summary, it can be said that the German federal statistics and the GUNDI hazardous substances database show that, on the whole, there has been a decrease in road accidents involving hazardous substances. According to the federal statistics, the main responsibility for accidents lies less frequently with drivers of vehicles transporting hazardous substances (excluding cars) – 513 with main responsibility per 1000 parties involved – than with drivers of goods trucks (571). One essential reason for this is the continuing improvement in the safety "chain" for transporting hazardous substances by road (Staebler, 1995, 2001). This includes the requirements that have to be fulfilled by vehicles carrying hazardous substances regarding electrical installations, the braking system, fire protection and speed limiters. In the GGVS/ADR regulations regarding the transport of dangerous goods by road, the "human" factor is also taken into account with driver training, which considerably increases the driver's awareness of the various risks and dangers involved in transporting hazardous substances and in handling the vehicles carrying them. And finally, the ADR/RID structural reform has been contributing to safety since July 01, 2001 with its call for a hazardous substance officer at the location where the goods originate, the intermediate storage facility and with the carrier.

Nevertheless, very serious accidents involving fire, explosions and damage to the environment continue to occur (see Figure 2). According to a risk analysis of hazardous goods transport (Brenck and Mondry, 1998), the road transport accident rate is at least seven times higher than that of rail and inland waterway transport. Furthermore, major accidents (involving the escape of more than 10,000 litres) occur most frequently on the road and least frequently by rail.

Official accident statistics continue to be an inadequate indicator of risk. Accidents monitored in the individual EU countries currently neither suffice to cover the entire spectrum of all possible accident scenarios, nor can the significance of changes to individual, "risk-influencing" factors (such as active safety systems or new types of transport containers) be reasonably represented with regard to the overall risk (see also Brenck and Mondry, 1998). Varying accident definitions and recording methods in the individual countries complicate the problem even more.

This is why the creation of a uniform European database is essential. It is especially important because in the area of hazardous substances, which already has restrictive regulations imposed upon it, decisions concerning safety are today only accepted if they are based on transparent risk analyses.

The particularly large damage potential means that the hypothetical risk of transporting hazardous substances must also be considered. In addition to accidents which have actually happened and which emerge from the statistics retrospectively, in a pragmatic approach, hypothetical accidents must also be considered when attempting to eliminate deficiencies. Work on eliminating the sources of risk and on improving the safety of transporting hazardous substances must be carried out step by step. Two major sources of risk are described below, together with measures for overcoming or reducing them.

PRIMARY SOURCES OF RISK AND COUNTERMEASURES

Although there are no statistics to support this, it is sufficiently well-known from accident research that the main source of risk leading to hazardous substances escaping from a protective tank is a single accident with tilting or rollover of the tank vehicle (roughly 60%). Collisions with other heavy vehicles, in which the rear or side of the tank are hit, are also critical (Podzuweit, 1990; THESEUS, 1995; Rompe and Heuser, 1996). The following types of tank stress occur:

- point load
- collision with an obstacle of the same type
- collision with a bulk object.

In order to reduce the probability of tank truck accidents involving the release of hazardous substances, it is therefore necessary first of all to reduce the risk of rollover for tank vehicles, and to increase the stability of both the protective tanks containing the hazardous substances, and their components.

Sources of Risk: Rollovers

In contrast to trucks without tanks, the centre of gravity of partly filled tank trucks during steady-state turning and with increasing speed shifts radial in the direction of the effective centrifugal force and, at the same time, upwards. The rollover axis is adversely shifted so that the rollover threshold is considerably lower. In this context, Figure 6 shows the adverse shift in the centre of gravity during a steady-state turn of a tank truck at three different speeds.

In addition to this, during an abrupt evasive manoeuvre or turning of the vehicle, the liquid surges so that the vehicle's centre of gravity is shifted even more unfavourably. The natural frequency for this lateral oscillation in a tank 8 feet in diameter and half-full is roughly 0.5 Hz (McLean and Hoffmann, 1973). This means that a stimulus caused by a manoeuvre such as changing lanes causes a dynamic shift of the liquid and also reduces the rollover threshold. Figure 7 shows the rollover threshold for a cylindrical tank depending on the load (in percent) for a steady-state turn and a transient turn during a manoeuvre with a frequency of 0.5 Hz (Winkler et al., 2000). The rollover threshold for the transient turn is reduced from 0.8 g (tank empty) to a level of only 0.3 g (tank half-full).

In summary, technical measures are already being carried out today to raise the rollover threshold and reduce the risk of rollover connected with this. Appropriate measures are the use of anti-surge plates, the subdivision of tanks into a number of chambers and lowering of the centre of gravity by using special tanks. Changing from the cylindrical tank to the rectangular tank is not a suitable measure, however. A lower centre of gravity can, of course, be achieved, but, as we describe later, the stability of a rectangular tank is far lower in the face of global and local loads on the tank.

What augurs well for the prevention of tilting/rollover accidents is ESP (Electronic Stability Program). However, in addition to the features ESP offers for cars, this system must be capable of preventing or reducing dangerous jack-knives or rollovers by means of rollover stabilisation. Fortunately, systems of this kind are about to be or have already been launched on to the market (Hecker et al., 2000; Neuhaus et al., 2000).

Sources of Risk: Protective Tanks

The tanks carried by tank trucks are subject particularly to local and global load initiation during accidents (Figure 8). In the case of local loads, where only specific points on the tank are affected, the strength properties of the material determine the failure threshold. In the case of global loads, however, failure tends to occur where abrupt transitions in rigidity, for example in the bases, bracing rings or welded bracing bands, impede distortion (THESEUS, 1995).

If we take a look at the ADR/GCVS special provision regarding the construction of tanks for transporting gasoline, diesel and fuel oil by road, the requirements are far lower than those for tanks on rail tank cars. They may

- have a low test pressure of roughly 0.4 bar, instead of 4 bar in the case of rail tank cars,
- have forms other than cylindrical forms,
- have abrupt wall transitions,
- be designed as "open tanks" (permanent source of ignition), and
- an increase in wall thickness over and above the minimum wall thickness is not required (Droste et al., 1990)(mailto:droste@all).

From the point of view of accident research, this is extremely surprising, especially from the background of the far lower accident rate of rail transport and its more consistent transport conditions. In contrast to road transport, constantly changing risk situations such as volumes of traffic, routes, quality of road surface or varying bend radii are not normally expected (see also Droste et al., 1990).

Unfortunately, this special provision for tanks transporting gasoline, diesel and fuel oil has resulted in the fact that, today, rectangular tanks made of aluminum alloys (e.g. AlMg 4.5 Mn W 28) with a wall thickness of approximately 5 mm are predominantly in use, even though aluminum has a number of disadvantages as compared to austenitic steel (stainless steel) with regard to global and local stress.

The schematic stress-strain diagram in Figure 9 shows that the specific resilience of austenitic steel (which corresponds to the surface integral of the curves) is several times greater than the specific resilience of the aluminum alloy frequently used (Ludwig and Schulz-Forberg, 1998).

The German Federal Institute for Materials Research and Testing (BAM) was able to ascertain by experiment that the resilience of the aluminium alloy with a wall thickness of 5 mm is 6,900 Nm, while an austenitic steel (X6 Cr Ni Mo Ti 17 122) with a wall thickness of 3 mm has a resilience of 30,000 Nm (4.4 times greater). Stainless steel, which is highly ductile, can therefore be considered to be the better material against local stress even if the wall is thinner.

According to the THESEUS results (1995), if we consider local and global stress, a stainless steel tank with a wall thickness of 3 mm is almost twice as safe for transporting class 3 hazardous substances by road than a tank made of aluminium alloy of the conventional type. Moreover, due to the low melting point of aluminium alloys (ap-

proximately 660 °C), there is the danger, in contrast to steel, of tanks becoming damaged as a result of thermal stress from fire occurring for any reason.

In summary, it can be said that the way rectangular tanks, which are widely used for transporting petroleum, behave in an accident is not acceptable. Aluminum alloys with a minimum wall thickness of around 5 mm are normally used. For this reason, a complete switch should be made from unpressurized tanks to medium-pressure tanks with a test pressure of around 4 bar. At the same time, the dome cover and tank components must be designed with the same level of safety. Both the results of research and practical experience confirm the advantages with regard to resilience against leakage (Fath, 1996).

Rustproof, highly ductile austenitic steel should be the first choice of material, even if additional weight has to be accepted because of the higher density of stainless steel (7.85 kg/dm³ rather than $\rho_{Al} = 2.7$ kg/dm³). The additional weight is counterbalanced by a far safer tank with a high damage reduction potential as regards injury to people and material damage as well as the reduction of human suffering. A statutory load incentive for increased safety could speed up implementation.

DEMANDS AND RECOMMENDATIONS IN SUMMARY

Society's threshold of acceptance for the risks involved in transporting hazardous substances is low. This is why it is necessary to prevent accidents, or at least reduce them to a minimum. The primary and attainable objective should be to aim for safety comparable to that of rail transport.

This can be achieved by systematically analyzing accidents involving tank trucks on a European level and by overcoming the main sources of risk described above, and especially by

- introducing ESP with rollover stabilization for all new tank trucks,
- using medium-pressure tanks (test pressure of 4 bar) instead of aluminum tanks with reduced wall thickness operated at atmospheric pressure,
- installing effective collision-protection systems at the rear and sides of vehicles,
- and granting a weight incentive for vehicles with medium-pressure tanks.

In addition to this, the Institute for Vehicle Safety in Munich, Germany (IFM) calls for the following for all tank trucks:

- compulsory introduction of contour marking as per ECE R 104
- an electronic braking system in the truck tractor and semitrailer/trailer
- a tire pressure monitor
- Accident Data Recorder (ADR), the emphasis being on learning from accidents
- GPS systems with automatic accident warning and synchronized transmission of detailed information on the hazardous substance transported.

In a first "best case" assessment it was ascertained (Langwieder et al., 2000) that active safety systems/driver assistance systems can be of considerable advantage, provided that they include all necessary features ("fail safe"), and support the driver when at risk. The current recommendation is to equip trucks and tank trucks with adaptive cruise control systems (ACC) and lane departure warning systems.

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TABLES & FIGURES

Mode of Transport	1990	1995	1996	1997	1998	1990-1998
Road	932	1,146	1,152	1,205	1,255	+35%
Rail	255	221	220	238	241	-6%
Inland waterways	108	114	112	118	121	+12%
Pipelines	75	83	85	85	87	+17%
Sea (intra-EU)	922	1,071	1,076	1,124	1,167	+27%
Total	2,293	2,635	2,645	2,770	2,870	+25%

Figure 1: Evolution of the total volume of goods carried in the EU by mode of transport from 1990 to 1998 in 1,000 million tkm (EUROSTAT, 2000)



Figure 2: Tank truck accident on August 29, 1998 in Zurich, Switzerland; 23,000 l of gasoline caught fire: the driver was injured and 20 residents had to be evacuated (photo: Keystone)

Accidents involving drivers of vehicles transporting hazardous substances	1996	1997	1998	1999	2000
Accidents causing injury to people	358	338	319	291	279
with release of hazardous substances	31	44	36	38	35
Serious accidents causing material damage	157	148	118	143	98
with release of hazardous substances	17	18	17	14	17

Figure 3: Evolution of accidents involving drivers of vehicles transporting hazardous substances, broken down into personal and material damage

Hazard class	Accidents causing injury to people	With release of hazardous substances		Accidents causing material damage (serious)	With release of hazardous substances	
		Number	%		Number	%
1 Explosive materials	21	-	-	8	1	16.7 %
2 Compressed or fluidized gases or gases released under pressure	32	3	9.4 %	14	-	-
3 Flammable liquids	147	25	1.0 %	74	9	12.2 %
4.1 Flammable solid materials	3	2	6.7 %	9	-	-
4.2 Spont. combustible materials	4	-	-	4	1	25.0 %
4.3 Materials that generate flammable gases in contact with water	-	-	-	1	-	-
5.1 Mats. that promote combustion	4	1	25.0 %	2	-	-
5.2 Organic peroxide	-	-	-	-	-	-
6.1 Toxic materials	8	-	-	3	-	-
6.2 Materials causing infection	-	-	-	-	-	-
7 Radioactive materials	1	-	-	-	-	-
8 Caustic materials	16	3	18.8 %	12	2	16.7 %
9 Various dangerous materials and objects	14	1	7.1 %	5	-	-
Misc. hazardous materials and collective goods	41	3	7.3 %	19	1	5.3 %
Total	291	38	13.1 %	143	14	9.8 %

Figure 4: Accidents involving drivers of vehicles transporting hazardous substances for the year 1999, broken down into the type of substances transported

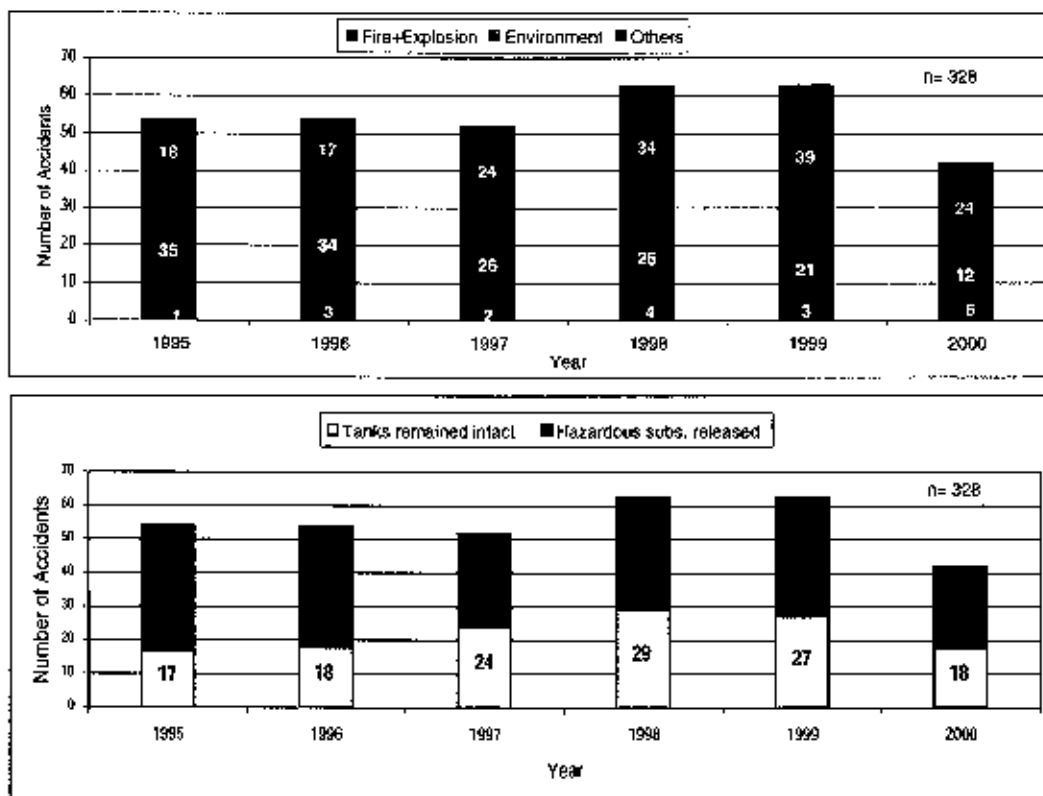


Figure 5: Accidents involving hazardous substances broken down into categories (top) and release of substances (bottom)

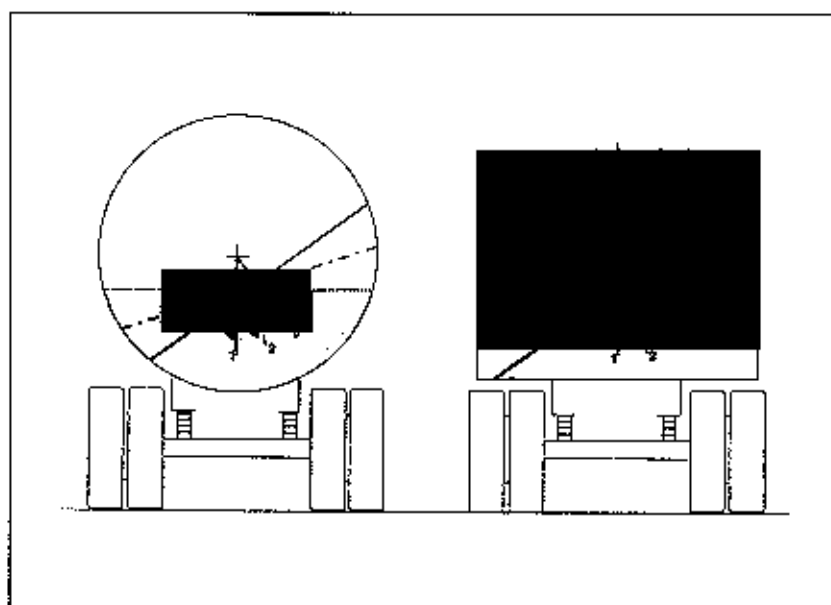


Figure 6: Liquid position and shift in the center of gravity during steady-state turning, for circular and rectangular tanks (as per Winkler et al., 2000)

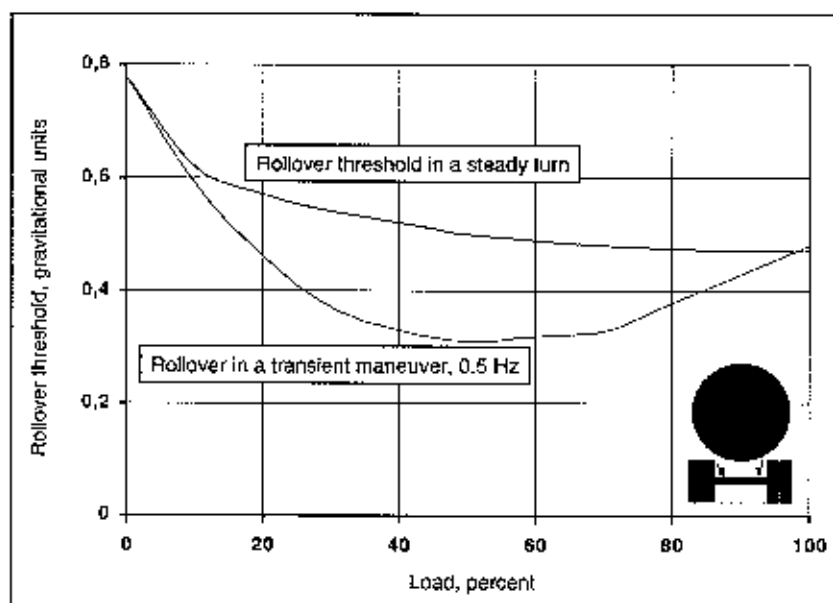


Figure 7: Rollover threshold in a steady and transient turn as a function of the percentage of unrestrained load (as per Winkler et al., 2000)

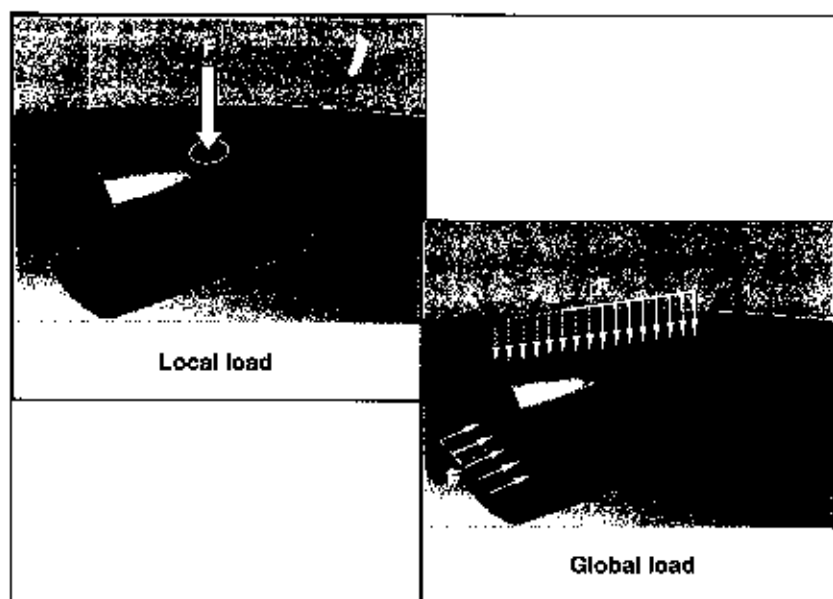


Figure 8: Schematic representation of local and global load initiation on tanks

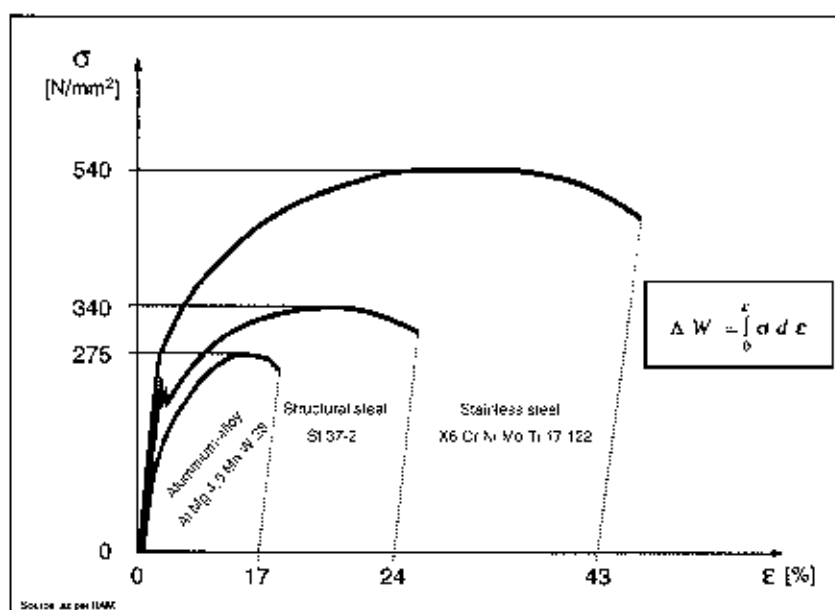


Figure 9: Schematic stress-strain diagram of aluminum alloy, structural steel and stainless steel (as per Ludwig and Schulz-Forberg, 1998)

APPENDIX

Examples of actual tank truck accidents involving fire from 1996 through 2000 (GUNDI, 2001)

Location of accident (in Germany)	Date of accident	Details of accident
A 44 freeway, near Geseke	Nov. 04, 1996	A tank truck combination skidded, tipped over and remained lying across the freeway. A car was unable to brake in time and collided with the trailer. Leaking gasoline ignited on the car engine. A large proportion of the load ran out and burned; the driver of the car died in the flames. The police suspects that the driver of the tank truck combination had fallen asleep at the steering wheel for a few seconds and came off the road on the right. When steered back in the opposite direction, the vehicle lost its stability due to the surging gasoline in the chambers of the tank.
A 19 freeway, near Röbel	Jan. 21, 1997	A truck drove into a tank truck combination in a roadworks area. The gas/air mixture in the empty chambers of the tank ignited, and there was an explosion. The tank truck combination and two other vehicles were gutted. The driver responsible for the accident and a car driver were killed.
Scharpzow, B 104	Jan. 16, 1998	At traffic lights by roadworks, an empty tank truck combination drove into a line of traffic and pushed two cars into a truck. The tank truck combination then skidded past the vehicles. The tank ripped open and the tank truck combination and front car caught fire. One of the occupants of the car was killed, two others were seriously injured.
A 81 freeway, near Zuffenhausen	Mar. 26, 1998	Due to the negligence of the driver, a tank truck combination crashed into a truck, skidded another 100 meters or so and came to a standstill across the road. At impact the tank ripped open and the truck tractor and leaking diesel caught fire. The vehicle was completely gutted. Part of the load seeped into the ground.

A 8 freeway, near Pforz- heim	Oct. 07, 1999	<p>Within a roadworks area with two very narrow lanes, a tank truck combination came off the road on the right on to the grass verge and drove into the end of a concrete barrier at the right-hand edge of the road, all due to the negligence of the driver. The tank truck combination then crashed into the concrete barrier dividing the road from the oncoming traffic on the left. The momentum of the crash pushed in part of the wall. The tank truck, the central chamber of which contained around 1,000 liters of light fuel oil, overturned and remained lying on the left. The empty tank trailer, which had not been cleaned, came to a standstill, jutting out into the oncoming traffic lane.</p> <p>Leaking fuel, igniting on the hot engine which could not be switched off, set the tank truck on fire and it was completely gutted. The light fuel oil contained in the tank created a jet flame. The driver was able to get out of his cab through the shattered windscreen and tried to put out the fire with a fire extinguisher.</p>
A 73 freeway, near Bamberg	Dec. 07, 1999	For unknown reasons, a tank truck combination loaded with roughly 28 tonnes of distilled grain spirit came off the road on the right. The tank trailer tore away from the truck tractor, damaged the fuel tank and fell over in the ditch on the right. Around 500 liters of its load escaped. The truck tractor caught fire.
A 5 freeway, near Freiburg- Süd	Mar. 10, 2000	Coming from a parking lot, the driver of an articulated truck cut right into the inside lane with his vehicle, presumably without using the full length of the acceleration lane and without sufficient speed. A tank truck combination, which was overtaken by a car on a level with the parking lot, crashed, probably without braking, head-on into the articulated truck which skidded into the central crash barrier. The tank truck combination, loaded with around 30 tonnes of gasoline and diesel, skidded and came off the road on the right. The vehicle knocked over some trees, and the load immediately went up in flames. It is still unclear why the driver of the tank truck combination did not react on time and brake.
way, near Öhringen	Apr. 19, 2000	For unknown reasons, a tank truck combination loaded with roughly 35 cubic meters of gasoline and diesel drove into a slower truck driving in front of it. After the collision, the cab of the tank truck combination caught fire. The driver was able to get out and tried in vain to put out the fire. The fire spread to the tank trailer and the tank truck combination was completely gutted.
A 9 freeway, near Homers- dorf	Aug. 02, 2000	<p>Presumably due to a burst tire, a tank truck combination loaded with 30 cubic meters of diesel skidded, overturned and slid right over the three lanes. The load leaked and caught fire. A large proportion of the diesel which escaped – around 25 cubic meters and some of it burning – ran into the drains and a rainwater collecting tank.</p> <p>The driver escaped from the vehicle and ran over the highway with his clothes on fire. Other road users were able to smother these flames with a blanket. Two days after the accident he died from his serious injuries.</p>
A 1 freeway, near Hamn/Burgka- men	Oct. 10, 2000	For unknown reasons, a tank truck combination loaded with 28 cubic meters of gasoline and five cubic meters of diesel drove on to the left-hand lane while being overtaken by a van. It is possible that the driver had fallen asleep at the steering wheel. After colliding, both vehicles skidded. The six-chamber tank truck combination broke through the right-hand crash barrier, drove on to the embankment and tipped over. The vehicle caught fire and was completely gutted. Part of the load which escaped seeped into the ground and ran into the drains. The 41-year old driver was killed in his cab by the fire.

REVIEW OF TRUCK AND DOG TRAILER OPERATIONS OVER 42.5 TONNES GROSS VEHICLE MASS

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ABSTRACT

In Australia, the national Road Transport Reform (Mass and Loading) Regulations 1995 only allow operation of truck and dog trailers to 42.5 tonnes gross combination mass (GCM) and a mass ratio of 1:1. However, many jurisdictions allow higher mass limits/mass ratios and operation up to 50 tonnes GCM on their general road network.

The Australian National Road Transport Commission (NRTC) has proposed a scheme (based on a Performance Based Standards (PBS) approach) which provides specific dimensional limits for truck-trailer combinations above 42.5 tonnes and up to 50 tonnes GCM with the options of providing separate evidence that showed compliance with a performance-based formula or a full PBS assessment.

While the most common usage for truck-trailers in Australia appears to be for transportation of quarry products, there is a growing application in logging and other sectors (such as fuel transportation) where stability factors are of concern.

The objective of the NRTC review was to develop appropriate performance-based controls for the design and operation of truck trailers at higher mass limits to deliver a consistent road safety performance. A national policy is considered desirable to:

- *improve consistency;*
- *introduce a common set of conditions;*
- *facilitate cross jurisdiction operations even though the percentage of interstate operation is expected to remain low; and*
- *ensure a consistent on-road safety performance for the truck trailer fleet.*

This paper will outline the various approaches taken in developing the operational conditions for general access operation of truck and dog trailers above 42.5 tonnes GCM and the process in obtaining agreement by operators, regulators and truck/trailer manufacturers to the proposal.

The policy development has not been completed and will need to align with the Performance Measures and Standards developed as part of the NRTC's Performance Based Standards (PBS) Project. The views expressed in the paper are those of the authors.

INTRODUCTION

The range and variety of regulatory frameworks under which Australia's road transport industry was required to operate was contrary to the very idea of competitiveness. This was recognised by the Special Premiers Conference in 1991 which led to the establishment of the National Road Transport Commission (NRTC). Initially, the Commission was set up with responsibility only for Heavy Vehicles (defined as greater than 4.5 tonnes Gross Vehicle Mass (GVM)). This responsibility was expanded in 1992 to also include Light Vehicles.

The mission of the National Road Transport Commission (NRTC) is:

To contribute to Australia's economic, social and environmental future by playing the lead role in developing and co-ordinating road transport reform in Australia.

Working in close partnership with the road freight transport and road passenger transport industries, governments and their agencies, police, community interest groups, unions and other organisations, the NRTC aims to develop and implement policies, practices and laws that:

- make road transport and road use more innovative, efficient and safe;
- introduce greater national transport uniformity and consistency;
- reduce the environmental impact of road transport; and
- reduce the costs of administration of road transport.

BACKGROUND

The national Road Transport Reform (Mass and Loading) Regulations 1995¹ limit the total mass of a combination (other than a road train or B-double) including load to 42.5 tonnes and also require that the loaded mass of the dog or pig trailer must not exceed the loaded mass of the towing vehicle, ie a 1:1 mass ratio. These limits are less, in most cases, than the sum of axle mass limits for the six, seven and eight axle truck and trailer combinations. Additionally, the general access overall length limit for these combinations is 19m. Figure 1 shows a typical three axle truck and four axle dog trailer with maximum overall length of 19m.

Operators in a variety of industry sectors, including quarry operations, sand, gravel, grain cartage and brick cartage, have expressed strong interest in using truck and dog trailer combinations at higher gross mass limits to improve productivity. Figure 2 shows a typical three axle truck and three axle dog trailer used in quarry operations.

Although most jurisdictions have implemented the national Mass & Loading Regulations, some jurisdictions allow truck trailers to operate at mass limits higher than 42.5 tonnes under special conditions that were developed by jurisdictions in conjunction with the NRTC in 1996-97 (Refer Annex A).

REVIEW OBJECTIVE

The review of the Truck and Dog Trailer mass limits is an initiative of the NRTC's Second Heavy Vehicle Reform Package, recognising that various schemes for these combinations already exist in most jurisdictions.

The objective of the review is to develop appropriate performance-based controls for the design and operation of truck trailers at higher mass limits to deliver a consistent road safety performance.

A national policy is considered desirable to:

- improve consistency;
- introduce a common set of conditions;
- facilitate cross jurisdiction operations even though the percentage of interstate operation is expected to remain low; and
- ensure a consistent on-road safety performance for the truck trailer fleet.

INITIAL POLICY DEVELOPMENT

In 1996, the Roads and Traffic Authority of New South Wales (RTA) commissioned ARRB Transport Research Limited (ARRB) to investigate the effects of higher mass (greater than 42.5 tonnes) on a range of vehicle safety and performance issues for truck and dog trailer combinations.

The testing conducted by ARRB covered dynamic stability, braking, high speed off-tracking and other criteria for vehicles with different wheelbases, axles and suspensions. On-road trials under permit conditions were also conducted for evaluation in New South Wales and Victoria by the RTA and VicRoads in conjunction with their respective operators and industry groups.

Based on the field testing of truck and dog trailer combinations in 1996 with gross combination mass (GCM) above 42.5 tonnes and on-road trials conducted under permit in Victoria and New South Wales (NRTC 1997), a preliminary set of general operating conditions as outlined in Annex A covering suspension type, axle spacing and vehicle capability were developed for the operation of truck-trailers above 42.5 tonnes GCM.

Figure 3 shows the four axle truck and three axle dog trailer testing conducted by ARRB Transport Research at Armidale, Australia in 1996.

These conditions were based on the results of testing by ARRB (NRTC 1997) that indicated that increasing the mass ratio from 1:1 to 1:1.4 is unlikely to compromise road safety for vehicles fitted with air suspensions. This means from a stability point of view, a three axle truck and a four axle trailer could have a gross combination mass limit of 54 tonnes. However, from a bridge perspective, the general access limit in Australia is generally 50 tonnes. Each of the State and Territory jurisdictions currently allow truck-trailers to operate above 42.5 tonnes as shown in Table 1 subject to special conditions. The mass increases do not exceed existing legal axle mass limits².

INDUSTRY CONCERNS

In July 1998, the Australian Trucking Association (ATA) expressed concern over the prospect of higher mass limits for six and seven axle truck-trailer combinations. This concern emanated principally from:

- computer simulation work, indicating stability problems for certain truck-trailer combinations (particularly with high centres of gravity and short truck wheelbases);

¹ Available on NRTC Website: www.nrtc.gov.au

² Steer Axle 6t, Tandem Axle Dual Tyred 16.5t (17t Road Friendly Suspension), Triaxle Dual Tyred 20t (22.5t Road Friendly Suspension)

- anecdotal reports of stability problems with these higher mass combinations; and
- the potential fleet conversion from more stable vehicles, eg 6 axle articulated vehicles to truck-trailer combinations (because of the higher payload).

As a result, the NRTC contracted Roaduser International Pty Ltd (now Roaduser Systems Pty Ltd) to undertake an assessment. Roaduser Systems interacted with a small group of jurisdictional representatives in undertaking its contract and the analysis involved the application of performance criteria for on-road stability, based on computer simulations.

The Roaduser Systems report (NRTC 1999), indicated in part, that the stability performance of truck-trailer combinations is dependent on the Centre of Gravity (CoG) of the trailer (predominantly), longitudinal dimensions and the low coupling rear overhang.

Additionally, the report indicated that:

- truck-trailer combinations with high CoG remain of concern;
- acceptable stability can be achieved by adjusting critical design dimensions;
- the most common usage for truck-trailers appears to be for transportation of quarry products and these seem to possess adequate stability (low CoG);
- only limited details on accidents involving truck-trailers at the higher mass limits were found; and
- there is no evidence of significant fleet conversion at this stage.

A range of options on how to proceed were considered by the NRTC. The NRTC was attracted to controlling CoG (or a surrogate) through regulation, coupled with the possible provision of a Technical Guide prepared by the regulators in conjunction with industry. The surrogate selected was the centroid of the load space as detailed in Annex B.

This approach was considered suitable to deliver an operating regime that maintains productivity improvements and a comparable stability performance to that of the existing fleet. However, a final position was dependent on consultation with the jurisdictions and industry.

It appears that the stability problems raised by the ATA may occur with the less common Truck and Dog combinations with a high CoG when loaded. Moreover, the analysis suggests that design relationships are available to adjust critical dimensions to ensure that a consistent level of on-road stability is achieved.

PROPOSED APPROACH

In February 2000, the NRTC proposed a three level scheme (based on a Performance Based Standards (PBS) approach) which provided specific dimensional limits for truck-trailer combinations above 42.5 tonnes and up to 50 tonnes with the options of providing separate evidence that showed compliance with a performance-based formula or a full PBS assessment.

A performance-based simulation analysis was conducted by the consultant (mainly in relation to stability of the combination (Load Transfer Ratio)) to derive control conditions and performance formulae for truck and dog trailer combinations operating above 42.5 tonnes GCM and up to 50 tonnes GCM.

For combinations operating at GCM's of 42.5 tonnes (the current legal maximum mass limit) or below, no change in existing conditions was considered necessary.

However, for combinations operating above 42.5 tonnes GCM and up to 50 tonnes GCM, a three level approach for approval of their operation was proposed as shown in Table 2.

Although there was general support for the three level approach concern was expressed by jurisdictions and industry in that:

- the core dimensions in Level 1 were too onerous and alternatives should be considered;
- a significant percentage of existing vehicles would be outside the core dimensions in Level 1, particularly for the three axle dog trailer;
- on-road safety problems are not being experienced to warrant such a restriction of the existing fleet; and
- trailer load height may be difficult to enforce and/or prosecute because some loads (eg gravel) do not have a level upper surface.

Jurisdictions were then requested to:

- consult further with industry on the proposed 3 level approach;
- quantify how many vehicles in the existing fleet may not comply; and
- provide information to justify alternative core dimensions.

Feedback from this round of consultations with jurisdictions resulted in advice to the NRTC that it should consider dividing Level 1 into a further two levels with appropriate conditions to cover:

- truck and trailer combinations that transport quarry and other related 'high density' commodities/products ie, low Centre of Gravity (COG) vehicle combinations; and
- all other commodities/products transported by truck and trailer combinations.

Overall, the formal responses received indicated general acceptance of the three level concept even though many considered it overly complex, cumbersome from an enforcement viewpoint and emphasised that many vehicles in the existing fleet may not comply especially with the Level 1 dimension limits. Additionally, some debate occurred on the selection of 0.8 as the value for Load Transfer Ratio (LTR) in the assessments and performance formulae (Refer Annex D).

A summary of the comments and views received on the revised Truck and Dog Trailer Proposal indicated that:

- there appears to be little evidence of a large scale problem;
- some outliers can occur based on short wheelbase truck;
- appropriate conditions that are not too complex for on-road enforcement are required;
- policy needs to be relatively simple to be enforceable in on-road situations;
- operators and industry know suitable configurations for their tasks;
- stability can be achieved by good design and proper selection of equipment;
- correct loading/unloading sequence based on industry best practice is important;
- generally air suspension drive axles have better roll stiffness than mechanical drive axles. (However, there are large variations between makes and types.); and
- four axle trailers handle better overall, they have better braking and a longer wheelbase than a three axle trailer.

REVISED PROPOSAL - CIRCA 2001

The revised proposal (See Annex B Part A, B and C) now caters for the two scenarios at the Level 1 dimensional limits:

Level 1	any commodity; and
Level 1A	commodities with a density greater than 1 tonne/m ³ for 3 axle trailers and 0.45 tonne/m ³ for 4 axle trailers.

As with the original submission, the revised dimensional requirements are a coarse selection generally based on the simulation work and are within the range of key variables used in the performance formulae.

Although this revised proposal retains some of the complexity of the initial submission (ie in relation to trailer load height, the new 'high density' provision should be more aligned with the existing dimensions for the relatively low COG 'high density' product quarry) fleet.

To clarify the range and type of commodities covered by the proposed new Level 1A, a specific guideline (for both operators and enforcement staff) listing examples of the densities of various commodities may be necessary for the implementation stage of this policy proposal. An example of such a document is at Annex E.

As with the initial submission, the use of the Level 2 performance formulae and the Level 3 full PBS assessment still allows a more sophisticated alternative process if a specific truck and trailer combination does not meet the Level 1 criteria.

If an existing truck and trailer combination does not comply and if vehicle modification action on the combination is not financially feasible or practical, grandfathering options may need to be considered as part of the implementation phase.

BASIC PREMISE FOR PERFORMANCE FORMULAE - LOAD TRANSFER RATIO

The load transfer ratio (LTR) is defined as the proportion of vertical load imposed on the tyres on one side of a vehicle unit that is transferred to the other side of the vehicle during a standard lane change manoeuvre (NRTC 2002). When the load transfer ratio reaches a value of 1, rollover occurs.

The policy development work to date by Roaduser Systems has been based on a LTR of 0.8. However, as provided at Annex D, there are different viewpoints on the appropriate value for LTR for Truck Trailer combinations.

The NRTC Performance Based Standards (PBS) project has been developing performance measures and standards over the last few years and indications are that the Load Transfer Ratio may not be selected as a PBS measure in the final analysis. In the report *Performance Characteristics of the Australian Heavy Vehicle Fleet* (NRTC 2002) LTR is shown as being highly correlated with static rollover threshold and rearward amplification. Additionally, the in-field determination of LTR is apparently costly and complicated and maybe prohibitive at this stage.

ADDITIONAL CONDITIONS FOR TRUCK TRAILER COMBINATIONS

In December 2000, following representations regarding a non-air RFS from Industry and VicRoads, additional research work was commissioned by the NRTC to determine the suitability of a non-air drive axle road friendly suspension in a Truck and Dog Trailer combination. Roaduser Systems conducted the work and they were also required to detail the values of the significant suspension characteristic values used in previous research and project reports for the NRTC on Truck and Dog Trailers.

The Roaduser Systems report for the NRTC, 'Performance Criteria for Non-Air Road Friendly Suspensions (RFS) in Truck and Dog Trailer Operations above 42.5 tonnes and up to 50 tonnes' of February 2001 (unpublished)³ found that:

- the influence of the type of drive axle (ie, air or non-air) on the dynamic stability of the trailer is minimal; and
- the combination of the roll stiffness and roll centre height of the trailer suspension is a significant parameter for the overall combinations dynamic stability and performance.

Based on this report and from a performance based standards aspect, additional conditions covering trailer suspension characteristics are considered warranted for both Level 1 and 2 situations. The proposed condition is:

The 'Total Roll Stiffness per axle' and 'Roll Centre Height' values for the Trailer suspension should satisfy the requirements of Figure C1 in Annex C Part A (Figure C1 is based on Figure 14 in the Roaduser Systems report). Basically the intersection point of the Roll Centre Height and Roll Stiffness should be in the Acceptable Zone of the chart.

Also, the *Coupling Rear Overhang (CROH)* dimension condition was expanded to include '35% of Truck Wheelbase' as an additional alternative limit to the 1.8m CROH requirement.

INDUSTRY INVOLVEMENT IN POLICY DEVELOPMENT AND CONSULTATION

A number of workshops and teleconferences with industry representatives and jurisdiction representatives have been held at each stage of the development of the policy. Transport media has also been utilised to broaden the coverage of the policy proposals.

During the course of the policy development, some operators have started to use the performance formulae to check their existing fleets and in developing their replacement fleet vehicle specifications.

REGULATORY IMPACT STATEMENT FOR TRUCK TRAILER COMBINATION POLICY

A Regulatory Impact Statement (RIS) for the proposal is being prepared (NRTC 2001). In Australia, a RIS must be developed for all proposals for regulatory changes that are to be submitted to the Transport Ministers for vote. Proposals for regulatory change are 'measures which, when implemented, would encourage or force businesses or individuals to pursue their interests in ways they would not otherwise have done'. The RIS considers alternatives and conducts a cost benefit assessment.

The objective of the Truck and Dog Trailer RIS is to:

- provide a clear definition of the problem;
- evaluate the need for government action in relation to Truck Trailer combinations operating at mass limits above the existing statutory limit of 42.5 tonne - specifically, the need to develop appropriate controls (using a performance based approach) for the design and operation of truck trailers at higher mass limits to deliver a consistent road safety performance; and
- assess the relative costs and benefits of the proposed policy.

The RIS evaluates the impact, cost and benefits of the NRTC proposal compared to the existing permit schemes operating currently in jurisdictions. Over ten years, the proposal would result in additional costs of \$18.7* million relative to the base case. The cost impact would be \$2.5* million on an annualised basis.

Table 3 shows the initial summary of impacts of the regulatory proposal outlined in the draft RIS:

The regulatory proposal would need to reduce the number of truck-trailer related fatalities by approximately 1.5 annually (or by 10%) in order to cover its costs⁴. While the potential of the regulatory proposal to achieve this safety improvement is conjectural, its costs of \$2.5 million annually are relatively low and would be at least partially ameliorated for affected operators by a suitable grandfather provision. The alternative, of body specific

³ Prepared by Brendan Coleman, Peter Sweatman and Scott McFarlane of Roaduser Systems Pty Ltd.

⁴ The average cost per fatal accident is \$1.7 m* (BTE 2000).

* Australian Dollars

mass limits, has costs nearly four times those of the regulatory proposal which would be unlikely to be offset by savings in accident costs. It would also have a more significant environmental impact than the regulatory proposal.

While expected to reduce truck-trailer fleet productivity (albeit by a small margin over all), the regulatory proposal responds to professional advice and industry concern that these vehicles may be unstable in some configurations and under some loading conditions.

The RIS concludes that assessment of the regulatory proposal is influenced by the degree to which operators and manufacturers will incur suspension related costs. Were those costs found not to be necessary (because in-service and on market suspensions were compliant with the requirements of the regulatory proposal), the regulatory proposal would appear on balance to be worthwhile given (1) its low costs; (2) the finding that most truck-trailers will not be affected; and (3) the greater assurance it would provide as to the safety of the truck-trailer fleet.

The possibility that operators and manufacturers may incur suspension related costs would make the assessment more marginal given the higher costs, the wider range of operators expected to be affected, and the uncertainty associated with the suspension-related cost estimates. Annual costs would increase from \$2.5* million to between \$3.6* million and \$3.9* million, and the breakeven reduction in fatalities attributable to truck trailers would need to increase to between 13% and 17% before the costs of the regulatory proposal could be covered.

AXLE MASS SPACING SCHEDULE (BRIDGE FORMULAE) IMPACT

General access vehicles are permitted to travel over virtually all parts of the road network. The axle spacing mass schedule (ASMS)(Bridge Formulae) for general access vehicles is based on the formulae:

- up to 42.5 tonnes ($L = 10$ metres), $M = 3L + 12.5$ tonnes; and
- from 42.5 tonnes to 50 tonnes, a formula of $M = L + 32.5$ tonnes.

where M tonnes is the mass allowed on all axle groups within L metres (measured to the outer axes of any two groups).

Truck trailer manufacturers and operators cannot in some jurisdictions achieve the full 50 tonnes (ie meet $L + 32.5$) within an overall length limit of 19m due to construction allowances for front and rear overhangs. Although this project has not addressed this issue, it is considered an issue for individual jurisdictions to decide whether truck trailers that are at maximum extreme axle spacing (ie 16.5m in the 19m overall length) to have 50 tonnes GCM instead of the 49.5 tonnes as per the current ASMS or allow the extra 0.5 tonne as a special truck-trailer allowance. This would give vehicle designers and manufacturers additional flexibility and achieve the maximum general access mass limit within the 19m overall length limit.

Some vehicle designers/manufacturers have recently had the opportunity to present their viewpoint regarding a relaxation or adjustment of the Bridge Formulae for Truck and Dog Trailers to the Austroads⁵ Bridge Committee.

FUTURE ACTION

Pending the outcomes of the performance measures and the values of the performance standards from the NRTC Performance Based Standards project, the truck and dog trailer policy will need to be further developed to enable a proposal to be prepared for submission to the Australian Transport Council.

The NRTC is committed to developing a consistent policy to address the potential stability problems for certain Truck-Trailer combinations and to reduce the uncertainty for industry in relation to Truck and Dog Trailer operation above 42.5 tonnes in jurisdictions. Accordingly, to enable further development of the proposal for submission to ATC, the following is still to be resolved:

- the suitability of the NRTC proposal and the relevant package of conditions;
- the likely enforcement issues related to in-field assessment of the 'trailer load height' measurement;
- a possible clearer definition for the commodities/products covered by Level 1A;
- adjustment of the axle spacing mass schedule to enable full achievement of the 50 tonnes GCM; and
- finalisation of the Regulatory Impact Statement.

This case study raised several issues relevant to a PBS approach, including the need for staged approaches, the difficulty in removing productivity gains on the grounds of safety risk and the potential complexity of PBS approaches.

CONCLUSION

The regulatory proposal is an early initiative in the NRTC's performance-based standards program which aims to improve the performance, safety and productivity of the national heavy vehicle fleet. In this particular initiative, the productivity gains of earlier decisions by jurisdictions to grant higher mass limits for truck-trailers are likely to

⁵ Austroads is the national association of Australian and New Zealand road transport and traffic authorities.

be pulled back in order to secure greater assurance of safety. A relatively small proportion of the truck-trailer fleet is expected to be affected.

Most vehicles have dimensions and load characteristics that will allow them to retain those productivity gains while still operating safely. The overall safety of the Australian truck-trailer fleet should be improved as a consequence of the regulatory proposal.

This project has provided a useful example of the potential for a Performance Based Standards approach. In part, it suggests that traditional assessment techniques have potential shortcomings that may now be better addressed through a complete PBS approach. The opportunity may also arise for further research on developing an appropriate low cost, in-field test method of determining Load Transfer Ratio for vehicle combinations.

ANNEXES

- A. Proposed Operating Conditions 1996-97
- B. Proposal and Additional Design Requirements 1998
- C. Proposed Operating Conditions 2000/2001
- D. Load Transfer Ratio Issues
- E. Draft Typical Commodity Densities Document

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TABLES & FIGURES

Table 1 - Truck-Trailer Mass Limits

Jurisdiction	3-axle truck		4-axle truck	
	3 axle dog trailer	4-axle dog trailer	3-axle dog trailer	4-axle dog trailer
New South Wales	48	50	50	50
Victoria	45	50	45	Not approved
Queensland	45	50	45	50
South Australia	45	49.5	45	50
Western Australia	48	55.5 ^a	53 ^a	60.5 ^a
Tasmania	48	55.5 ^a	53 ^a	60.5 ^a
Northern Territory	48	55.5 ^a	53 ^a	60.5 ^a
Australian Capital Territory	48	50	45	50

^a Concessional loading, sum of axle limits and restricted access schemes

Table 2 - GCM 42.5 tonnes to 50 tonnes

Level 1	The existing or proposed combination is required to meet the specific dimension limits and a general set of control conditions **derived from the performance-based simulation analysis.
Level 2	If an existing or proposed combination is outside the proposed performance-based controls in Level 1 above, then an assessment against the performance formulae** developed as part of this review is required.
Level 3	If an existing or proposed combination does not meet the performance formulae in Level 2 above, then a full performance-based assessment is required.

** The control conditions and performance formulae in Levels 1 and 2 cover a range of key dimensional variables.
See Annex C, Part B and C

Table 3 – RIS Benefits/Costs

Jurisdiction	Agency costs	Road wear costs	Vehicle certification costs	Vehicle operating costs	Total costs	Annualised costs
	(\$m)*	(\$m)*	(\$m)*	(\$m)*	(\$m)*	(\$m)*
TOTAL	2.447	-1.264	1.012	16.531	18.726	2.544

* Australian Dollars



Figure 1 – Typical Three Axle Truck and Four Axle Dog Trailer
Maximum Overall Length 19 m



Figure 2 – Typical Three Axle Truck and Three Axle Dog Trailer
Maximum Overall Length 19m



Figure 3 – Four Axle Truck and Three Axle Dog Trailer Testing, Armidale, Australia, 1996, ARRB Transport Research

ANNEX A

Proposal Circa 1996-97

PROPOSED OPERATING CONDITIONS FOR TRUCK AND DOG TRAILER COMBINATIONS GREATER THAN 42.5 TONNES AND UP TO 50 TONNES GROSS COMBINATION MASS (GENERAL ACCESS)

1. GROSS MASS LIMIT

Maximum gross mass for truck and dog trailer combinations operating without route restrictions.

3 Axle Dog Trailer		
(a)	3 axle rigid truck towing a 3 axle trailer (provided extreme axle spacing of trailer is at least 3.8m)	= 45t
(b)	4 axle rigid truck towing a 3 axle trailer (provided extreme axle spacing of trailer is at least 3.8m)	= 45t
(c)	3 axle rigid truck towing a 3 axle trailer (provided extreme axle spacing of trailer is at least 4.3m)	= 48t
4 Axle Dog Trailer		
(a)	3 axle rigid truck towing a 4 axle trailer.	= 50t
(b)	4 axle rigid truck towing a 4 axle trailer.	= 50t

2. DIMENSION LIMITS

Dimension limits are the same as for other general access combinations:

- length of combination not to exceed 19.0 metres.
- height not to exceed 4.3 metres.
- width not to exceed 2.5 metres.

3. VEHICLE CAPABILITY AND EQUIPMENT RATINGS

All vehicles, components and couplings used in the truck and dog trailer combination must have a manufacturer's rating and GCM appropriate to the maximum gross mass of the combination. (See Note: 1)

4. AXLE SPACING/MASS SCHEDULE

All truck and dog trailer combinations exceeding 42.5 tonnes GCM must comply with the 1994 Austroads recommended axle spacing/mass schedule (bridge formulae) for general access vehicles, ie '3L + 12.5' tonnes to 10m spacing (42.5 tonnes) and then 'L + 32.5' tonnes to 17.5m (50 tonnes).

5. SUSPENSIONS

Truck and trailers must be fitted with air suspension (except for the steer axle(s) of the truck).

6. MASS RATIO

The loaded mass of the dog trailer must not exceed the loaded mass of the truck by more than 25%, ie mass ratio of 1:1.25 under any loading condition.

Note: 1 Although a preference for a requirement that combinations should be capable of maintaining 70km/h on a 1% grade has been raised by jurisdictions, the requirement could be covered by ensuring that the combination has a suitable GCM rating by the manufacturer for the combination/load to be carried.

ANNEX B

Proposal Circa 1998

ADDITIONAL DESIGN REQUIREMENTS/INITIAL PERFORMANCE FORMULAE GENERAL ACCESS TRUCK AND TRAILER COMBINATIONS – 19M LONG

50 tonnes Combinations R12-T22

Trailer(&Truck) CoG Height	Requirement
Less than 2m	No additional requirements
Greater than 2m and less than 2.2m	$(TkWb + TrWb) + 2 \geq \text{to CROH} + 3.4\text{m}$
Greater than 2.2m	$TrWb + 1.24 TkWb - 3 \text{ CROH} - 14.1 TrCoG \geq -25.0$

48 tonnes Combinations R12-T12

Trailer(&Truck) CoG Height	Requirement
Less than 1.9m	No additional requirements
Greater than 1.9m and less than 2m	$TrWb \geq \text{to CROH} + 2.8\text{m}$
Greater than 2m	$4.8TrWb + 2.0 TkWb - 4.95 \text{ CROH} - 19.0 TrCoG \geq -15.1$

45 tonnes Combinations R12-T12

Trailer(&Truck) CoG Height	Requirement
Less than 1.9m	No additional requirements
Greater than 1.9m and less than 2m	$TrWb \geq \text{to CROH} + 2.4\text{m}$
Greater than 2m	$4.8TrWb + 2.0 TkWb - 4.95 \text{ CROH} - 19.0 TrCoG \geq -16.8$

CROH = Coupling Rear Overhang

TrWb = Trailer Wheelbase

TkWb = Truck Wheelbase

TrCoG = Trailer Centre of Gravity

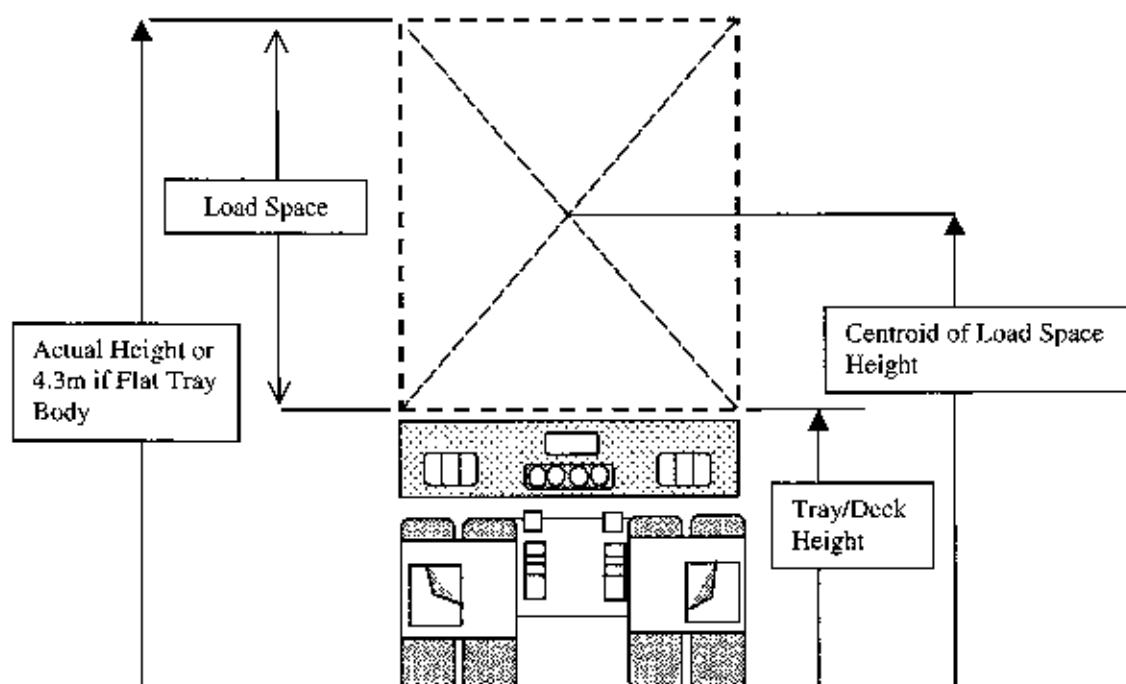
ANNEX C

DESIGN CONDITIONS GENERAL ACCESS TRUCK AND TRAILER COMBINATIONS – 19M LONG

Centroid Height Of Trailer (Centre of Gravity Control)

The centroid height from ground level of the imaginary or actual load space must be below 2.2m for all combinations

50 tonnes	Combinations	R12-T22
48 tonnes	Combinations	R12-T12
45 tonnes	Combinations	R12-T12



CONFIGURATIONS OUTSIDE ABOVE NOTICE DIMENSIONS

If a proposed configuration for a Truck and Dog Trailer is outside the above dimensions then an Engineering Certificate covering a dynamic assessment to show the LTR is below 0.8 will be required.

ANNEX C PART A

Proposal Circa 2000/2001

TRUCK AND DOG TRAILER – MASS LIMITS REVIEW (OPERATIONS OVER 42.5 TONNES)

Revised Proposal

The revised proposal now caters for two scenarios at the Level 1 dimensional limits:

- Level 1 any commodity; and
- Level 1A commodities with a density greater than 1 tonne/m³ for 3 axle trailers and 0.45 tonne/m³ for 4 axle trailers

Level 1 & 1A

	3 axle Truck and 3 axle Trailer 45t	3 axle Truck and 3 axle Trailer 48t	4 axle Truck and 3 axle Trailer 50t	3 & 4 axle Truck and 4 axle Trailer 50t
Truck Wheelbase (TkWB)	4.3m	4.3m		4.3m
Trailer Wheelbase (TrWB)	3.2m	3.7m		7 4.5m

Level 1

(Any commodity)

Trailer Load Height limit	3m (evidence that load is below height limit required)	3.8m
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Level 1A

(No overall height limit* depending on Density of load)

No overall Trailer Load Height limit* if density of load	7 t/m ³ (Evidence required of load density (dry, wet))	7 0.45t/m ³
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* Other than normal 4.3m legal maximum overall vehicle height limit

General Conditions

Coupling Rear Overhang	1.8m or 35% of TkWB (if 35% of TkWB is >1.8m)
Overall Dimension Limits	Length 19.0m, Width 2.5m, Height 4.3m (unless indicated differently)
Suspension	Truck and trailer must be fitted with a Road Friendly Suspension (except for steer axle(s) of truck). Also, see required additional Trailer Suspension Characteristics.
Mass Ratio Limit	The loaded mass of the dog trailer under any loading condition must not exceed the loaded mass of the truck by more than the following ratio/percentage. <ul style="list-style-type: none"> 45 tonnes 1:1 or 0%; 48 tonnes 1:1.15 or 15%; and 50 tonnes 1:1.25 or 25%.
Axle Spacing Mass Schedule (ASMS)	Compliance with the applicable Austroads Axle Spacing Mass Schedule is required for the combination.
Vehicle Capability, Braking and Equipment Ratings	All vehicles, components and couplings used in the truck and dog trailer combination must have a manufacturer's rating and GCM appropriate to the maximum gross mass of the combination and axle loads.

Trailer Load Height	<p>* The distance from ground level to the top of the load in/on the trailer.</p> <p>Trailer Load Height excludes all load restraint systems/attachments, eg load tarps/covers. Applies when combination load is between 42.5 tonnes and 45, 48 or 50 tonnes as applicable. (normal 4.3m maximum overall vehicle height applies)</p>
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Note 1: The control conditions and performance formulae are based on the assumption that the loading condition of the combination is such that the truck has maximum allowable axle loads.

Additional Condition

Trailer Suspension Characteristics

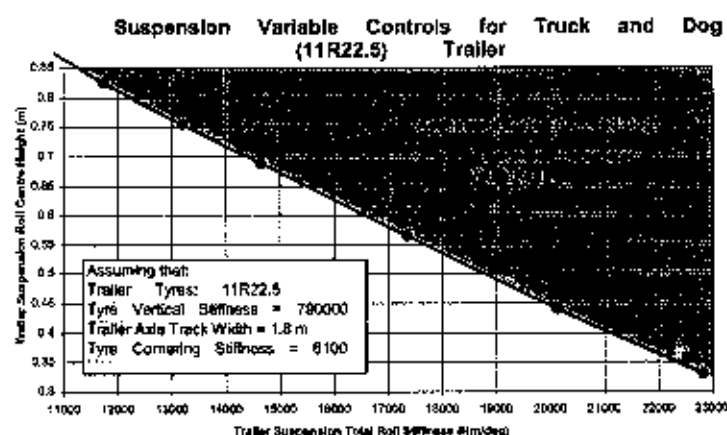


Figure C1 – Trailer Suspension Characteristics

Examples:

1. if trailer suspension Roll Centre Height is 0.36m, then Roll Stiffness should be greater than 22000 Nm/deg
2. if trailer suspension Roll Centre Height is 0.57m, then Roll Stiffness should be greater than 17000 Nm/deg

ANNEX C PART B

CONFIGURATIONS OUTSIDE ABOVE NOTICE/GAZETTE DIMENSIONS

As an alternative, if a proposed or existing configuration for a Truck and Dog Trailer is outside any of the Level 1 dimensions then either:

- Level 2 - evidence is required that the combination meets the performance formulae developed by Roaduser Systems as outlined at Annex C, Part C; or
- Level 3 - a full performance-based assessment of the proposed combination is required using 0.8 as the value for Load Transfer Ratio (LTR) or the performance measures and values developed and selected in the NRTC PBS Project.

ANNEX C PART C

LEVEL 2

PERFORMANCE FORMULAE

The performance formulae for the 3 axle truck and 3 or 4 axle trailer combinations below are detailed in the reports NRTC Performance-Based Controls for Truck and Dog Trailer Combinations, May 1999 and Roaduser Systems Truck and Dog Combination Issues, December 1999(NRTC unpublished) and cover the listed range of key vehicle dimensional variables.

3 Axle Truck and 3 or 4 axle trailers

Extract from Performance-Based Controls for Truck and Dog Trailer Combinations, May 1999.

8.1 Performance Formulae

8.1.1 Key Variables

Trailer COG height affects the tendency of the trailer to rollover in response to lateral acceleration. It also affects the tendency of the trailer to damp out lateral oscillations. It can also affect the rearward amplification mechanism, making the trailer lateral accelerations greater than the truck lateral accelerations.

Trailer wheelbase affects the rearward amplification mechanism and can affect the ability of the trailer to damp out lateral oscillations.

Truck wheelbase affects the rearward amplification mechanism.

Coupling rear overhang affects the rearward amplification mechanism.

8.1.2 Formulae

Performance formulae have been developed using the above variables, and are designed to ensure that the LTR will be no greater than 0.80. Allowance has been made for the fact that, although the most important variables are explicitly present in the formulae, other factors do have some influence on LTR.

The performance formula for 3-axle dogs is valid for the following ranges of key variables:

- Truck wheelbase: 4.1 m* - 5.3 m
- Coupling rear overhang: 1.5 m to 2.1 m
- Dolly drawbar length: 3.0 m to 4.5 m
- Trailer wheelbase: 3.2 m* to 5.0 m
- Trailer COG height: 1.6 m to 2.4 m.

The performance formula for 4-axle dogs is valid for the following ranges of key variables:

- Truck wheelbase: 4.1 m* - 6.1 m
- Coupling rear overhang: 1.6 m to 2.6 m
- Dolly drawbar length: 3.0 m to 4.5 m
- Trailer wheelbase: 4.6 m to 7.0 m
- Trailer COG height: 1.6 m to 2.4 m.

*Extended range for application of performance formulae.

Within these ranges of key variables, the performance formulae are able to reproduce the simulation results to an accuracy of better than 1 % on average, and the largest error relative to the simulations is 6 %.

The performance formulae assume that:

- The truck and trailer have air suspension; air suspensions can vary significantly and average value were assumed
- LTR varies linearly with each of truck wheelbase, trailer wheelbase, coupling rear overhang and trailer COG height
- Uncontrolled influences, including variations in the truck air suspension, may increase LTR by a total of 0.05.

8.1.2.1 Four Axle Dogs

The most general form of the performance formula for the truck and dog with 4-axle dog trailer at 50.0 tonnes GCM is given by:

$$\text{TrWB} + 1.24 \text{ TkWB} - 3 \text{ CROH} - 14.1 \text{ TrCOG} > - 25.0 \quad \{2\}$$

where	CROH	=	coupling rear overhang (m)
	TrCOG	=	trailer centre of gravity height (m)
	TrWB	=	trailer wheelbase (m)
	TkWB	=	truck wheelbase (m)

and an alternative form of the performance formula, where the trailer COG term is not explicit, is given by:

$$\text{TrWB} + 1.24 \text{ TkWB} - 3 \text{ CROH} > \text{TDS} \quad \{3\}$$

where TDS is a **Truck and Dog Stability** parameter which is a function of trailer COG height, as given in Table 4.

Table 4 Values for truck and dog stability parameter TDS (4-axle dogs at 50.0 tonnes GCM)

Trailer COG Height	Typical Corresponding Body Type	TDS
1.7	Tipper	-1.1
1.8	Low-COG Tanker	0.3
2.2	Conventional Tanker	6.0
2.4	General Freight	8.8

8.1.2.2 Three-Axle Dogs

The most general form of the performance formula for the truck and dog with 3-axle dog trailer at 48.0 tonnes GCM is given by:

$$4.8\text{TrWB} + 2.0\text{TkWB} - 4.95\text{CROH} - 19.0\text{TrCOG} > -15.1 \quad \{4\}$$

where CROH = truck rear overhang (m)
TrCOG = trailer centre of gravity height (m)
TrWB = trailer wheelbase (m)
TkWB = truck wheelbase (m)

And the analogous formula for the truck and dog with 3-axle dog trailer at 45.0 tonnes GCM is given by:

$$4.8\text{TrWB} + 2.0\text{TkWB} - 4.95\text{CROH} - 19.0\text{TrCOG} > -16.8 \quad \{5\}$$

An alternative form of the 48.0 tonne 3-axle dog performance formula, where the trailer COG term is not explicit, is given by:

$$4.8\text{TrWB} + 2.0\text{TkWB} - 4.95\text{CROH} > \text{TDS} \quad \{6\}$$

where TDS is a truck and dog stability parameter which is a function of trailer COG height, as given in Table 5.

Table 5 Values for truck and dog stability parameter TDS (3-axle dogs at 48.0 tonnes GCM)

Trailer COG Height	Typical Corresponding Body Type	TDS
1.7	Tipper	17.2
1.8	Low-COG Tanker	19.1
2.2	Conventional Tanker	26.7
2.4	General Freight	30.5

And an alternative form of the 45.0 tonne 3-axle dog performance formula, where the trailer COG term is not explicit, is given by:

$$4.8\text{TrWB} + 2.0\text{TkWB} - 4.95\text{CROH} > \text{TDS} \quad \{7\}$$

where TDS is a truck and dog stability parameter which is a function of trailer COG height, as given in Table 6.

Table 6 Values for truck and dog stability parameter TDS (3-axle dogs at 45.0 tonnes GCM)

Trailer COG Height	Typical Corresponding Body Type	TDS
1.7	Tipper	15.5
1.8	Low-COG Tanker	17.4
2.2	Conventional Tanker	25.0
2.4	General Freight	28.8

4 Axle Truck and 3 or 4 Axle Trailers

Although specific performance formulae have not been developed by Roaduser Systems for the 4 axle truck combinations, the 3 axle truck performance formulae above could also be applied to 4 axle truck combinations.

- For R22T22 use R12T22 formulae
- For R22T12 use R12T12 (48 tonne) formulae

Note: The performance formulae may give a more conservative outcome, ie more restrictive than if a specific performance formulae were developed for the 4 axle truck combination.

Reference: Roaduser Systems Report: 00-583-01 Truck and Dog Combination Issues: Twin Steer Hauling Units, 4 February 2000 (unpublished by NRTC).

ANNEX D

LOAD TRANSFER RATIO

The following extracts from the referenced reports illustrates some of the various views on LTR over the life of the project.

Extracts from pages 19 and 21 of Performance-Based Controls for Truck and Dog Trailer Combinations, May 1999. (NRTC 1999).

LTR is the ultimate stability performance measure for truck and dog combinations because, in addition to combining the influences of static roll stability and rearward amplification in one measure, it concentrates on the tendency of the trailer to roll over, and this is the greatest risk for these combinations in dynamic situations.

The Load Transfer Ratio (LTR) is a meaningful measure of the dynamic stability of truck and dog combinations, along with other multi-articulated configurations. It combines notions of the inherent resistance of the trailer to roll over with notions of the tendency of the vehicle combination to "crack the whip" and thereby amplify the lateral acceleration occurring in the trailer during emergency avoidance manoeuvres.

It is known that LTR values vary widely between vehicle configurations with the same body type (and therefore similar COG height); for maximum-weight, medium-COG vehicles, such LTR values range from 0.5 to values close to 1.0. For a given configuration, LTR is significantly affected by COG height and other dimensions. For vehicle configurations whose rear unit (not roll-coupled to its towing unit) is relatively short, COG height increases can readily take the LTR to a value of 1.0. Such vehicles cluster at the LTR value of 1.0 because it is not possible for LTR to exceed 1.0, even though some of these LTR = 1.0 vehicles are still "worse" than others.

The load transfer ratio is one of the few engineering performance measures which has been linked to accident risks. Such relationships have been developed in the US, where both the quantity and quality of truck accident data, pertaining to fatal crashes, is sufficient to support the necessary analysis.

Research by UMTRI (Francher, Campbell, Blower 1989) found increased fatal rollover accident involvement for doubles operating on interstates and rural primaries when the combination vehicles' rearward amplification exceeded a value of approximately 1.6. During the FHWA Comprehensive Truck Size and Weight Study, the UMTRI relationship between accident involvement ratio and rearward amplification was converted to a relationship between accident involvement ratio and LTR. This was done because LTR is an absolute measure of dynamic stability, while rearward amplification is an intermediate measure with regard to dynamic stability (rearward amplification has had significant use historically in the US because it is relatively easy to measure in the field, while LTR is difficult to measure).

Figure 1 shows the relationship between relative risk of being involved in a fatal rollover accident and the LTR of the vehicle, based on the UMTRI rearward amplification relationship (Francher, Campbell, Blower 1989). This shows that accident risk increases significantly when LTR exceeds a value of approximately 0.80. The vehicles involved in the UMTRI study were 5-axle short doubles combinations comprising a 2-axle prime mover, single axle semi-trailer and two-axle full trailer, with GCM up to 36.3 tonnes. This relationship gives credence to the idea that an LTR target of 0.6 is conservative, and that a target of approximately 0.8 will protect against increased accident risks related to dynamic instability.

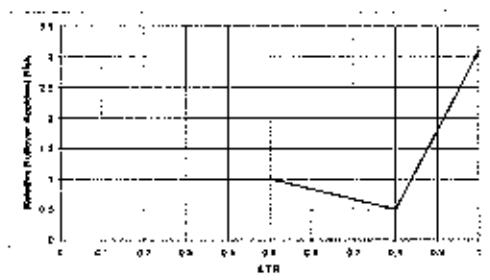


Figure 1D - Relative accident risk versus Load Transfer Ratio

Extracts from pages 34 and 35 of NRTC (2001)

Load transfer ratio is a well-established performance measure that is highly correlated with both static roll stability and rearward amplification within a given class of vehicle configuration (Winkler and Bogard, 1993; McFarlane et al, 1997; Mueller, de Pont and Baas, 1999; Winkler et al, 2000).

Load transfer ratio can have a value in the range 0 to 1. Lower values indicate comparatively better performance; high values imply high probabilities of rollover.

The same method for evaluating rearward amplification using the standard SAE lane change is used to determine load transfer ratio. However, it is important to note that unlike static rollover threshold and rearward amplification, load transfer ratio has never been measured⁶ (McFarlane et al, 1997) and it is calculated using computer-based simulation. Nevertheless, it has been recognized as a useful means of distinguishing between the performance of roll coupled and non-roll-coupled multi-unit vehicles (Winkler et al, 2000).

For load transfer ratios much above 0.6 most trucks are highly susceptible to rolling over (Clarke and Wiggers, 1998). However, according to Sweatman, Woodroffe and Blow (1998), "there appears to be some evidence that accident risks begin to increase for load transfer ratio values above 0.75 and are significantly higher 'above' 1.0".

The generally accepted performance level for load transfer ratio is 0.6, which is supported by the work of Mueller, de Pont and Baas (1999) and recommended by Land Transport Authority (2000) and Austroads (2000). However, for logging trucks in New Zealand a performance level of 0.75 has been chosen and is used in combination with a slightly relaxed performance requirement for static roll threshold of 0.3g (Land Transport Safety Authority, 1991; Land Transport Safety Authority, 1997).

Extract from page 3 of Truck and Dog Trailer Combinations Issues, Roaduser Systems Pty Ltd, December 1999, (NRTC unpublished)).

It is true that, for a specific vehicle configuration, there is a strong correlation between LTR, RA and SRS. Determination of this correlation for truck-trailers with 3-axle and 4-axle dog trailers was not included in the brief for the subject report (NRTC 1999). This may be a useful exercise, although it would be necessary to specify both LTR and RA in order to calculate the equivalent SRS.

Considering the SRS information that is now available for the subject report, it could be noted that:

- For a 3-axle dog of moderate COG height (2.0 metres) and benchmark dimensions, its LTR of 0.80 corresponds to a SRS of 0.34 g
- For a 4-axle dog of moderate COG height (2.0 metres) and benchmark dimensions, its LTR of 0.67 corresponds to a SRS of 0.46 g.

The amenability of LTR to measurement in the field is a relevant issue in the application of PBS. Rearward amplification may be measured in a reasonably straightforward manner; this could be combined with a measured SRS value to estimate the LTR, provided that a correlation equation (between LTR, RA and SRS) is available for the specific vehicle configuration.

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⁶ This should not be interpreted to mean that it cannot be measured but that the cost and time required to develop and test the necessary equipment is at present prohibitive.

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ANNEX E

TYPICAL COMMODITY DENSITIES (kg/m³) ⁷

(Extract)

The purpose of these tables is to reconfirm and set out density tables that are typical and representative of selected common loads. As the development of the policy framework moves towards performance based standards, an understanding of basic issues such as load densities will help to assist operators and enforcement people identify the products that have a greater likelihood of causing a load to become overweight or reduce the rollover threshold.

An example of Loose Dry Bulk Commodities

Fertilisers	kg/m ³
Ammonium Nitrate.....	721 - 993
Ammonium Sulphate.....	930 - 1010
Gypsum (granulated).....	913 - 960
Gypsum.....	1100
Mono Ammonium Phosphate.....	975 - 1030
Potassium Chloride.....	1150
Sulphate of Ammonia (granulated)	860 - 914
Superphosphate & or Lime.....	1180 - 1300
Triple Superphosphate.....	1240
Urea.....	550 - 820
Grain & Produce	kg/m ³
Barley (feed).....	380
Barley (malting).....	360
Cotton Seed (meal).....	560 - 640
Distillers Grains (wet).....	640 - 800
Linseed.....	800 - 929
Maize.....	752 - 760
Oats.....	480 - 650
Rice (hulled).....	720 - 784
Sugar (powdered).....	800 - 960
Sugar (raw).....	975 - 1030
Wheat.....	720 - 768
Quarry Materials	kg/m ³
Aggregate (dry).....	1700
Aggregate (wet).....	2000
Bulking Rubble.....	1300 - 1500
Cement (bulk).....	1200
Clay (moist).....	2000
Coal (hard).....	720 - 900
Coke.....	481 - 561
Concrete (ready mix).....	1800 - 2000
Gypsum.....	1100
Sand (dry).....	1600
Sand (wet).....	2100
Slurry.....	993

⁷ Prepared by Ian Wright & Associates, for the National Road Transport Commission, March 2001

INSTRUMENTED VEHICLE AND ITS USE FOR CALIBRATION OF WIM-SYSTEMS

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ABSTRACT

As a vehicle drives on a road the instantaneous wheel load is not steady, eg 10 tons but it varies because of the unevenness of the road. This varying axle load is often called as dynamic axle load. All weigh-in-motion (WIM) systems must be calibrated at site. The instrumented vehicle measures instantaneous wheel loads and thus seem to be an excellent tool for calibration of WIM-systems. The instrumented vehicle of VTT was used in calibration measurements at WAVE tests near Lulea in Northern Sweden and near Metz in France. The use of instrumented vehicle was more complicated than assumed. It is not only question of the accuracy of the measurement system nor the accuracy of matching the measurements at the vehicle and at WIM-systems but also a problem: what is really expected from the calibration. The (good) results and problems are presented in this paper.

1. INTRODUCTION

As a heavy vehicle drives on the road the axle load is not a steady, let say nominal 100 kN, but it may vary up to ± 15 (or more) percent even on a good road. The basic reason is the unevenness of the road. The amount of dynamic loading depends on the quality of the suspension and on the speed of the vehicle.

The main movements of a vehicle are (Figure 1):

- body bounce, which means pitching and bouncing of the vehicle body, natural frequency usually 1.2 to 3 Hz, lower limit corresponding good up to date construction of suspension and upper limit poor old stylish suspension
- axle hop, natural frequency usually around 10 Hz.

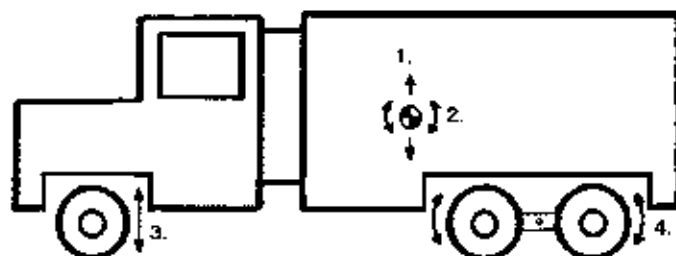


Figure 1 Dynamic movements of vehicle; body bounce (1), body pitch (2) and axle hop (3) and tandem pitch (4)

Weigh-in-motion (WIM) systems measure the instantaneous dynamic axle loads of the passing vehicle. The main interest is, however, to know the static axle loads of the passing vehicles and based on these the gross weight of the vehicle. WIM-systems are installed on even, straight road sections in order to minimize dynamic loadings and therefore there is very little rolling and other movements. In certain cases harsh weather conditions, for instance wind or ice on pavement may excite other dynamic movements.

There are several sources of errors in WIM-systems and therefore the calibration of each WIM-system is very necessary (see for instance Huhtala & al 2000). That is usually done using test vehicles and/or vehicles from the

traffic flow which are measured statically after passing the WIM-system. As the instantaneous dynamic axle load can be measured by an instrumented vehicle it is very tempting to use it in the calibration of WIM-systems.

2. DYNAMIC AXLE LOAD MEASUREMENT SYSTEM

Three main principles of measuring dynamic axle loads are:

- the change of the distance between the axle and the road surface is measured (flattening of the tyre),
- instrumented rim,
- bending of axles or shear strains in the axle are measured with strain gauges.

The first one is no more used. An instrumented rim is good but expensive, which means that dynamic loadings can be measured simultaneously only at one wheel. Because of its slightly greater weight and dimensions it may also change the dynamic properties of the suspension and moves tyres outwards. The measurement of bending of the axles or shear strains in the axle is cheap with strain gauges and all axles can be measured simultaneously. The authors have instrumented with strain gauges an own and a rented vehicle and have used two other in an earlier research project. Thus this paper is limited only to dynamic loading measurements with strain gauges.

The principle of measuring dynamic wheel loads is implemented by strain gauging the axle housing. Strain gauges are used to measure bending moment of the axle or shear strain on the sides of the axle. An inertial force component is measured with an accelerometer. Figure 2 shows the arrangement as bending is measured. If shear strains are measured the strain gauges are at the sides of the axle (not shown in Figure 2).

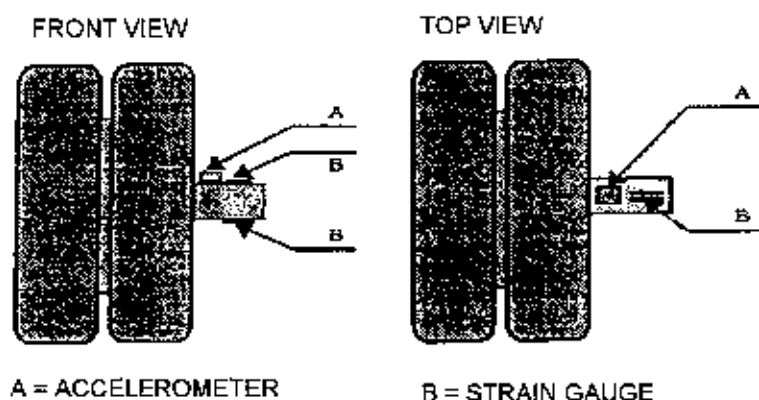


Figure 2. The locations of the sensors on the axle of vehicle

The location of strain gauges is critical especially in complicated suspension systems. Particularly the location of shear gauges demand special attention, experience and judgement.

The vertical wheel load measurement method is based on equilibrium of the moments acting on the mass outboard of the measuring point on the axle (place of the strain gauges). On straight road angles of the axle, caused by axle roll, are assumed to be small (there is no curvature on test section). Secondly, distances from wheel load centroid and mass centroid to strain gauges are equal. With these assumptions the vertical wheel load can be resolved in equation:

$$F_w = F_s + F_d + ma$$

where:

F_w Wheel load applied to the pavement

F_s Static wheel load

F_d Dynamic wheel load (deviation from the static wheel load) measured by the strain gauges

m Mass outboard of the strain gauges

a Vertical acceleration of the outboard mass measured by the accelerometer

The data acquisition system of VTT consists of amplifiers for the strain gauges and an A/D converter mounted in industrial PC. Strain amplifiers carry out signal conditioning for the strain gauge measurement. Accelerometers have a built-in signal conditioning. In addition, digital filtering is used to remove unwanted components. In order to correct phase shift due to filtering, signals are filtered twice (time-reversed signal on the second time). Typical sampling rate used is 1000 Hz. Cut-off frequency for the digital filter is 50 Hz.

VTT has made the first dynamic wheel load measurements in 1987. The instrumented vehicle has been used mainly at Vintaa Test Site and on certain WIM research. The system was improved for measurements at WAVE. The measurement system measures the bending of axle. For many practical reasons the result is the deviation from the static situation. The load is added to the platform of the vehicle stepwise and the wheel load is measured under each wheel. After loading the vehicle is unloaded and although there are steel suspensions no hysteresis has been seen.

Even if the road is relatively even without any sudden unevennesses the effect of the mass outboard of the strain gauges should be taken into account. The system was calibrated at the Helsinki University of Technology where each wheel was set on a shaker table one axle at the time. The shaker tables were vibrated from 2 to 16 Hz at 2 mm amplitude (peak to peak) and 0.5 - 3.5 Hz at 15 mm amplitude. The wheel loads were measured in the vehicle and in the shaker table system as well as the acceleration by accelerometers. The effective outboard mass of each axle system could be calculated and were used in the measurement system. As the vehicle after WAVE was calibrated also real road profiles were used in calibration.

Thus the calibration of the dynamic loading system is based on the equation presented earlier:

F_s = (static wheel load) is measured by static axle weighing pad before or after each measurement series (usually empty, half-loaded or full-loaded).

F_d = (dynamic wheel load) is the deviation from the static wheel load measured by the strain gauges in the axle. This calibration is made as the vehicle is stepwise loaded and unloaded. It is done basically once in a season.

M = (mass outboard of the strain gauges) is defined on the shaker table as a mass (pseudomass) which gives minimum error between the measured values at the shaker table force measurements and dynamic load measurements in the vehicle. The measurements are not very sensitive for the change of the mass (pseudomass).

a = (vertical acceleration of the outboard mass) is measured with the accelerometer.

Dynamic calibration is not only important for the calibration of the system but it may reveal problems in the incorrect location of strain gauges or hysteresis in suspension components which may cause considerable errors in measurements.

VTT has instrumented later in autumn 1998 another vehicle, two-axle tractor with a semi-trailer with tridem axles. It has good quality air suspensions and thus its suspension is much better than that of the vehicle used in WAVE which has old steel suspensions. The newly instrumented vehicle is used in an international research project (Huhtala & al 1999). In principle the system is the same but shear strain gauges were used. They are less sensitive for transverse lateral forces but their place must be more carefully selected. Because WIM-stations are on straight parts of roads this is not very important. The principle of measurement and calibration is the same as described earlier.

The basic principle of using instrumented vehicle in calibration of WIM-systems is the following: the instantaneous dynamic wheel load is compared to the WIM-system reading. Dynamic wheel load measurements are matched to the WIM-systems by using an electric eye to detect reflective tapes glued across road lane. Tapes have been fixed every ten meters in order to ensure exact measurements. A special software is used to link data collected from several sources mentioned above and ensure ready-made data for later analysis.

3. CALIBRATION OF WIM SYSTEMS WITH AN INSTRUMENTED VEHICLE

3.1 General

WAVE (weigh-in-motion of axes and vehicles in Europe) is a fourth Framework Programme of the European Commission. Work package 3.2 is handling the calibration of WIM-systems (Huhtala & al 2000). It included tests near Lulea in Northern Sweden and near Metz in France. The instrumented vehicle of VTT was used at both test sites and most results in this paper are based on those two tests.

The instrumented vehicle used at the Lulea and Metz test is owned and instrumented by VTT. It is old but a common three axle rigid lorry in Finland, SISU. The second and third axles establish together a tandem axle. It has traditional steel springs on the front and tandem axle. Both axles of the tandem axle have a mechanical connection and it is possible to lift up the third axle. Due to mechanical connection on tandem axle, axle masses are shared 55 % for the second axle and 45 % for the third axle. Tyre size for each wheel is 10R20 and there are dual tyres on the tandem axle.

Instrumented vehicle has been used earlier in UK and France at MS-WIM arrays within the OECD/DIVINE project in Abingdon (UK) with a 2-axle instrumented vehicle of the TRL, and in Trappes (RN10, France) with the 5/6-axle instrumented vehicle (semi-trailer) of the CNRC (Jacob & Dolcemascolo 1995, Huhtala and Jacob, 1995). No detailed reports have been published on those calibrations.

3.2 Measurements at Lulea

The measurements were made at Lulea on 9 - 10 June 1997. Five WIM-systems participated in the test but only one could be used for calibration with the instrumented vehicle. One selfcalibrating system had not yet enough material for reliable measurements, one used two sensors and data could not be attained separately from them and one was not yet ready for measurements. The test lasted longer than one year but because of economical reasons the instrumented vehicle could not be used later on the test site. The instrumented vehicle was used on a good quality bending plate WIM system. An example of the results is presented in Figure 3.

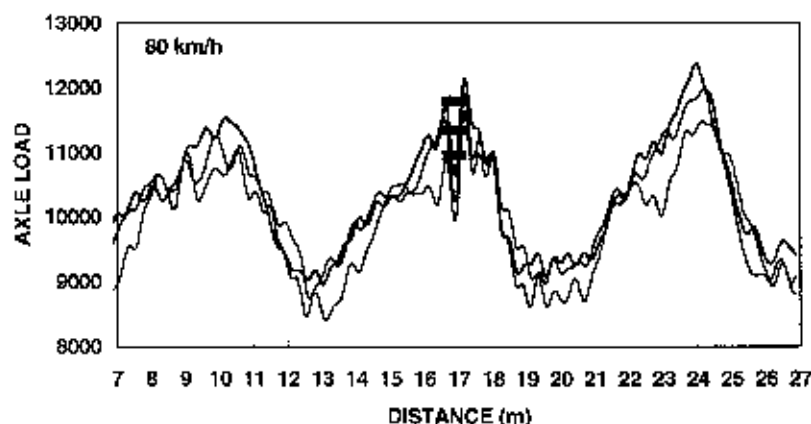


Figure 3. Dynamic axle loads of the driving axle over one WIM-system, three vehicle passes.

The horizontal bars show the position and the length of the bending plate and the axle loads measured by the WIM-system. The order of the corresponding WIM and dynamic axle loads are the same but the peak values of dynamic load measurements are smaller. However, if dynamic load on the bending plate is smoothed or the peak will not be taken into account, dynamic axle loads and WIM results fit very well. Perhaps the negative peak is typical and is taken into account in the calibration.

Please note, too that the tyre imprint has a length of about 0.3 meters.

Figure 4 shows dynamic steering axle loads at four speeds: 50, 60, 70 and 80 km/h. Because the speeds are different there is no repetitive dynamic loading. However, the small unevenness caused by the bending plate put all the dynamic axle loads in phase, which later dispenses. The same has been found in computer simulations (Huhtala & al 1992, Huhtala & al 1993, Huhtala & al 1994).

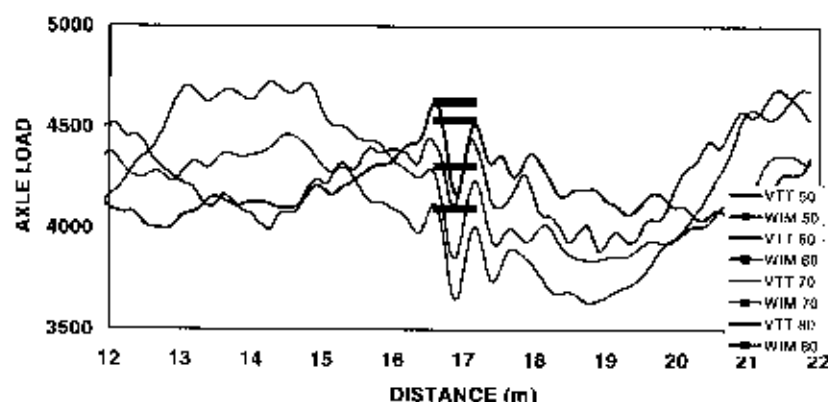


Figure 4. Dynamic axle loads of steering axle and corresponding WIM-measurements

As the horizontal bars which represent the WIM readings are compared to dynamic loads measured by the vehicle the values fit reasonably well in the same way as in Figure 4.

The same phenomenon is found in Figure 5 (driving tandem axle) and in carrying bogie axle (no Figure here).

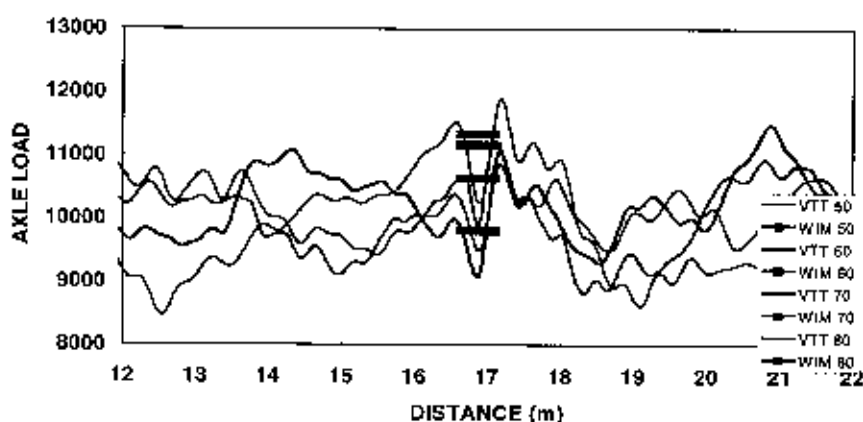


Figure 5. Dynamic axle loads of driving axle and corresponding WIM-measurements

This phenomenon may increase accuracy of a bending plate WIM-system. The WIM-system is calibrated at the site which may take into account how well the plate is embedded to the pavement and thus what is its effect on calibration. However, this phenomenon should be studied further with field tests and simulations.

All the points available from the data are presented in Figure 6. The axle load measured by the instrumented vehicle is on the abscissa and corresponding axle loads measured by the WIM-system are on the ordinate. The first group of points on the left (mainly between 2000 - 4000 kg) are front axle loads or empty bogie axle, the group on the right (mainly 9000 - 11000 kg) are the driving axle of the tandem axle and the group in the middle corresponding carrying (bogie) axle of the tandem axle. The driving axle carries about 55% of the whole tandem axle load in this vehicle type. The vehicle was driven with full load and empty and therefore there are also very low values for tandem axles. They are more scattered because suspensions are designed for full load and thus they do not work as well with small load.

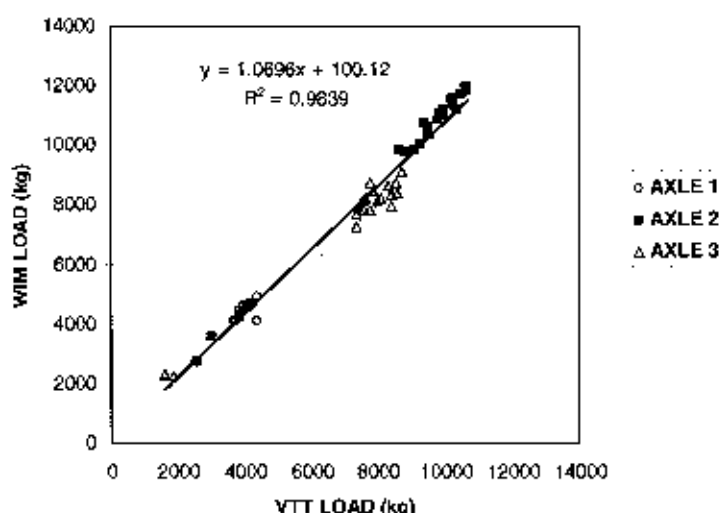


Figure 6. Dynamic axle loads measured in the vehicle (VTT) and by a WIM-system

Dynamic loads are different because four speeds were used. The points from the front axle and the points from the driving axle are reasonably nicely on a straight line but those from the running axle (second tandem axle) are more scattered and the measured axle loads are smaller. It may be due to the short recovery time of the sensor for tandem axle especially in this case as the distance between the tandem axles is only 1.20 m and thus the WIM-sensors give smaller axle load values for the latter axle. That axle can be lifted if the vehicle is empty or near empty. The more complicated mechanical system makes its dynamic properties worse than that of the first tandem axle.

The regression line deviates slightly from the 45-degree line and its coefficient is 1.07 or there is a systematic error of 7 %. The fit is quite good as the r^2 - value is 0.984. Regression lines were calculated for each axle and the coefficients are 1.104 for front axle, 1.109 for first tandem axle and 1.035 for second tandem axle.

Load coefficients (load from WIM divided by load with VTT measurement) 1.106, 1.111, 1.065 correspondingly and 1.094 for all.

The scatter in Figure 6 is due to both the inaccuracy of the WIM-system and the inaccuracy of the dynamic axle load measurement. There are no means to know which part is due to inaccuracy of the WIM-system and which part is due to inaccuracy of the dynamic wheel load measurements.

As is seen in Figures 3 - 6 the dynamic axle load on the plate changes and is equivocal. Thus it is easily understandable that the points are not on the 45-degree line. It could be possible to define that kind of algorithm used in the dynamic axle load measurement or the length or the point which is taken as the load on the WIM-plate would be defined in order to get best fit to 45-degree line. It might be of little use because it probably depends on the site and on the tyre types of the vehicle. Thus the deviation from the 45-degree line can be taken as error of neither WIM nor dynamic vehicle measurement systems.

3.3 Measurements at Metz

Multiple sensor WIM was used at Metz or all 18 strip sensors were identical. Sensor coefficients were defined for each bar and each pass (all passes were used except with bump or empty vehicle) as value measured by the instrumented vehicle divided by the value measured by the WIM-sensor. The mean values of coefficients by axles are presented in Figure 7.

It can be seen from Figure 7 that the coefficients of the first axle are clearly greater than one (mean value 1.15) or dynamic axle load measurements give greater values than sensors. The values of the second tandem axle (axle 3)

are also greater than one (mean value 1.13) and the values of the first tandem axle (driving axle) are smaller than one (mean value 0.94).

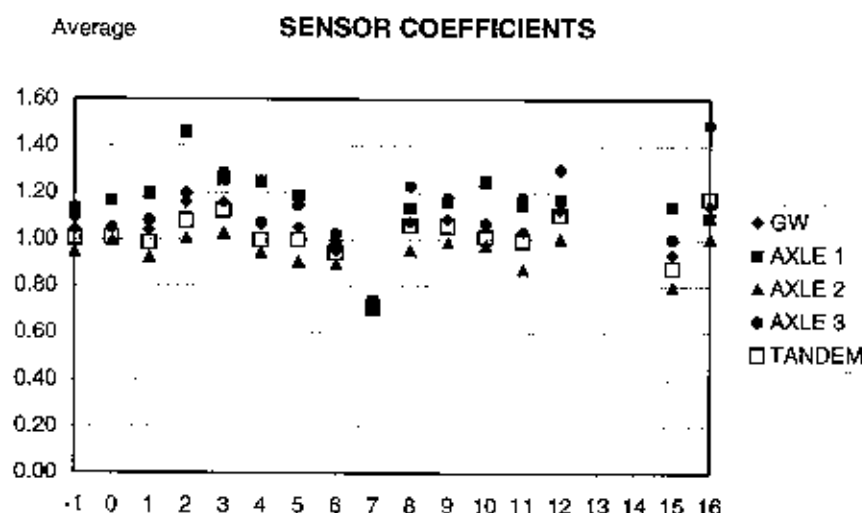


Figure 7. Sensor coefficients axle by axle, mean value of all passes but as the bump was used.

The mean value of the axles in tandem is presented as squares in Figure 7. They are relatively close to one or the high values of third axle and low values of the second axle compensate each other. The GW in Figure 7 is the mean value of all coefficients.

4. DISCUSSION

An instrumented vehicle measures the instantaneous wheel load as the vehicle runs over a WIM-system. Thus dynamic loadings can be eliminated as the real wheel load is known when the vehicle passes over a WIM-sensor and real loads can be used in calibration. For this reason an instrumented vehicle seems to be an ideal tool for calibration of WIM-systems.

There are, however, certain difficulties. Dynamic wheel load changes even within the time as the wheel is on a bending plate. Strip sensors are more difficult because they are narrow in the direction of vehicle movement and the wheel load is determined by integrating stress values as the wheel runs over the sensor. Dynamic loading changes during this time but the algorithm for integration is in "a black box" and the procedure is not known for outsiders.

Because of technical reasons the results of only one WIM-system could be compared to the results of the instrumented vehicle at Luleå. Relations between the axle loads measured by the bending plate WIM and by the instrumented vehicle or load coefficients (Figure 6) were the same for the first axle and second axle (1.106 and 1.111). They were slightly smaller for the third axle (1.065), which may be due to the short recovery time of the sensor for tandem axle. That may be more important in this case because the distance between the tandem axles is only 1.20 m and thus the WIM-sensors may give smaller axle load values for the latter axle. The second axle can be lifted if the vehicle is empty or near empty and as full load it carries about 45% of the load. The more complicated mechanical system makes its dynamic properties worse than that of the first tandem axle. However, the behaviour of tandem axles may also explain the difference because also in some other measurements at CET the second axle within the tandem axle was underestimated by other WIM-systems, too (Hallström & al 1999).

Bending plate WIM measurements give about 9% greater values than the instrumented vehicle based on the load coefficients and 7% greater if taken from the regression coefficients (Figure 6). The instrumented vehicle measures the real dynamic wheel load, which is much smaller in the middle of the plate than before or after because of the dynamic forces (Figure 3-5) and also much smaller than the static weight of the axle. The manufacturer does not necessarily know it and it is not important for him because the WIM-system is calibrated with test vehicles in order to compensate that difference.

The instrumented vehicle used the mean value within 150 mm and not the peak value. If the exact value in the middle of the plate had been used the difference would have been even greater.

Even the bending plate was installed very well and was at the same level as the pavement surface, it excited dynamic forces and got them in the same phase (Figures 4 and 5) or in very good spatial repeatability. Even only one vehicle was used here different speeds (50, 60, 70 and 80 km/h) simulated the effect of different vehicles as the dynamic properties are time (frequency) related. This phenomenon probably increases the accuracy of a bending plate WIM-system. The idea of using small exact unevenness to excite spatial repeatability has been presented in an earlier paper (Huhtala & al 1992), which was based on computer simulations but never studied further.

The corresponding sensor coefficients are defined at Metz for each bar as the value measured by the instrumented vehicle is divided by the value measured by the WIM-sensor. These values are inverses to the load coefficients used at Lulea. The sensor coefficients are presented in Figure 7. The mean value from 18 sensors is for the first axle 1.15, the second axle 0.94 and the third axle 1.13. Thus results of the first and third axles are similar but that of the second axle is much smaller. The first and third axles are underestimated and the second overestimated. The overall sensor coefficient is 1.07, which means that loads are underestimated by 7%. Standard deviations are 2 to 6 times greater than at the bending plate at Lulea. That difference may be due to the general accuracy and quality of the sensors but some effect may come from the algorithm, which was used in measurements.

It is logical to use load coefficients as the axle load from the instrumented vehicle is taken as a "true" value, which is compared to the measured values by WIM-systems. As the instrumented vehicle is used for calibration, the inverse or the sensor coefficient is more practical because it is the value with which the value from a sensor must be multiplied in order to get the correct value. In order to make the comparison easier the load coefficients are converted to sensor coefficients, 0.90, 0.90 and 0.94 correspondingly for the first, second and third axle. These numbers mean that the WIM systems gave smaller load than the instrumented vehicle.

The dynamic force was the mean load within 150 mm. The length of the tyre imprint depends on the tyre size, load and tyre inflation pressure, 300 mm can be taken as a rough estimate. Thus instrumented vehicle measured only the center part of the imprint and thus gave greater load than the WIM-system. The algorithms in the instrumented vehicle and in the WIM-systems should be "coordinated" but that is impossible because those algorithms are in secret "black boxes".

If exactly same length will be used the accuracy of matching measurements to the WIM-station should be exact. The maximum error in these measurements was in principle 22 mm at the speed of 80 km/h plus the inaccuracy in the installation of the reflective tapes which caused some problems at Metz.

The bending plate WIM-system does not see the difference between tyre types and thus the results are logical as the results from the first and second axles are similar and there may be good reason for the slightly smaller value from the third axle. The situation for a strip sensor is different because the stress values from the sensors are integrated both in longitudinal and transverse direction. Because each WIM-system has its own secret black box it is not known if the width of the tyre imprint has any effect to the results. The easiest explanation is that the difference between the first and the second axle is due to the difference in tyres (single and dual tyres) but it does not explain the difference between the second and third axle. The behaviour of tandem axles can explain a part (but not all) of the difference within the tandem axles because in some measurements at Lulea the second axle was underestimated by the WIM-systems (Hallstrom & al 1999). The results from Metz were analysed by the French WAVE colleagues and VTT provided only the results from the dynamic force measurements and thus the authors could not analyse the results further (all the available results are in reference: Huhtala et al. 2000).

The instrumented vehicle measures the instantaneous wheel load. WIM-systems are calibrated with passing test vehicles in order to get the static load of those vehicles. The WIM-system measures the instantaneous wheel load, which is not as great as the static load but may be smaller or greater because of dynamic effects. That

instantaneous load is then multiplied by a calibration factor (or some more complicated method is used) in order to get the static wheel loads of the test vehicles. Thus it is clear that the wheel loads measured by the instrumented vehicle are not necessarily the same as the result from a WIM-system.

If only one axle of the instrumented vehicle is used in the calibration of the WIM-system this is not a problem because in that case the instantaneous load is exactly the same as used in the calibration of the WIM-system. That is a good research approach but not possible for real calibration because several types of loadings are necessary for effective and reliable calibration.

The instrumented vehicle is first calibrated statically and then dynamically on shaker table. The dynamic calibration takes into account the pseudomass of the suspension but is not absolutely necessary because the road where WIM-system is used is relatively even and vertical accelerations are small. Dynamic calibrations on shaker table are, however, important as the quality control of the measurement system. They may reveal for instance hysteresis in the systems which cannot be tolerated. The steel suspension of the instrumented vehicle used at WAVE had no hysteresis. The suspensions of vehicles to be instrumented should be simple. If the suspension is complicated it may be difficult to find a good, suitable place for strain gauges. VTT instrumented a new tractor-semitrailer vehicle with air suspensions. The tractor presented no problem but an important hysteresis was found in the trailer suspension and it was not used for measurements. The reason may be the complicated structure with many joints and possibly rubber or plastic parts. Unfortunately it was not possible for us to analyse the problem further.

This instrumented vehicle used at WAVE is good but not ideal for this purpose because it is old and has relatively narrow tyres, which were used commonly at that time. VTT has later instrumented all axles of a new and modern vehicle (truck and semitrailer) and has now more experience in instrumented vehicles.

The instrumentation itself is not expensive but selecting the best position for sensors requires knowledge, experience and judgement. The cost of a good vehicle is usually much more important (rental or buying).

5. CONCLUSIONS AND RECOMMENDATIONS

Even if the WIM-systems themselves were perfect calibration is needed because of site dynamics and local traffic conditions. Each new WIM installation will have different characteristics, which will cause vehicles to behave in different ways. In order to minimize the effects of these factors on the accuracy of the WIM measurements each new site requires calibration. In addition to local differences in site dynamics the types of traffic will affect the measurements. In particular the normal types of suspension and loading will, combined with the site dynamics, require unique calibration figures for the local conditions.

An instrumented vehicle measures continuously the dynamic wheel load and thus might be ideal for calibration of Weigh-in-Motion systems. The main conclusions from the research made by VTT can be presented as follows:

1. Dynamic Axle Loads in all axles of a vehicle can be measured with strain gauges and accelerometers with good accuracy, relatively easily and at low cost: the cost of the vehicle is more crucial. Selecting proper position of strain gauges and accelerometers needs knowledge, experience and good judgement.
2. The instrumented vehicle must be calibrated statically and on shaker table dynamically. As the vertical accelerations on an even road are not important dynamic calibration is not essential but it may reveal hysteresis and other problems. Dynamic calibration is especially important if complicated air suspensions are used.
3. The exact match of dynamic wheel loads to WIM-stations is important and careful measurements of the locations is vital.
4. The aim of WIM-systems is to get static wheel or axle loads and based on them the gross weight can be calculated. In reality the WIM-system measures the dynamic wheel load on the sensor but it is calibrated with test vehicles to get the static load which may be considerably different from the dynamic load measured by the WIM-system. The instrumented vehicle measures basically the same dynamic load

which may be also different from the static load. Thus the value of dynamic loads measured by instrumented vehicle may be doubtful as a calibrating instrument but, however, useful as a research tool.

5. The instrumented vehicle measures the instantaneous wheel load. WIM-stations measure the passing axle or wheel loads. If a bending plate is used the maximum axle wheel load of the passing wheel is probably measured by the WIM-system and it can be compared to the maximum wheel load on the bending plate measured by the instrumented vehicle. If a strip sensor is used the tyre/sensor contact stress is measured and integrated. The exact dynamic load during the vehicle pass over the narrow sensor is probably not the correct way but which is the best way to coordinate the algorithms of black boxes which should be "opened". Further research is needed.
6. It must be noted that the wheel load on the bending plate or during the pass over the strip sensor is not constant but changes all the time because of dynamic loading. There is not an unequivocal instant wheel load for a WIM-sensor.
7. Spatial repeatability changed over the bending plate because of the small roughness caused by the bending plate. That probably made the results better but further research is needed (the same has been seen in earlier computer simulations made by the authors).
8. If an instrumented vehicle is used for calibration it should also be used as the main test vehicle in the calibration of WIM-systems in order to get all benefit from the measurements of dynamic loadings.
9. The use of instrumented vehicle for research is strongly recommended. It is also a promising tool for calibration but further research is needed and thus no exact recommendations can be given how it should be used for calibration.
10. The standard deviation of the dynamic load measurements will likely be about 1-2% if a new and good vehicle is used and the instrumentation and the calibration are made properly. That standard deviation includes the scatter both in dynamic force measurement in the vehicle and at the good quality WIM-system.

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EVALUATION OF THE EFFECTS OF HEAVY VEHICLES ON BRIDGES FATIGUE

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ABSTRACT

This paper presents and compares bridge lifetimes in fatigue for several heavy traffics, using traffic data recorded by Weigh-In-Motion (WIM) systems, and computed with CASTOR-LCPC software. CASTOR-LCPC was developed to compute traffic load effects on bridges, in order to assess structural design, to calibrate load models to be used in bridge codes, or to evaluate fatigue damages and lifetimes. Local effects (such as stresses) vs time, are computed using multi-lane traffic data files and theoretical or experimental influence surfaces. Among others, CASTOR-LCPC provides a "rain-flow" histogram of the stress cycles, used to calculate fatigue lifetimes, once combined with Miner's law. One detail of a common bridge structure sensitive to fatigue is considered in this paper: the welded joint between a vertical stiffener and the main girder upper or lower flanges, of composite (steel-concrete) bridges. Seven existing French bridges are considered as examples. Three traffics were chosen, with different vehicle flows and mean loads, measured by WIM systems on trafficked motorways in 1997 and 2001: one traffic recorded in 1986 on the A6 motorway at Auxerre, and which was used for the Eurocode "Traffic loads on road bridges" (EN 1991-3) is also used for comparison.

1. INTRODUCTION

Fatigue is a progressive deterioration of a structure by crack growth, due to a series of stress variations (cycles) resulting from the application of repeated loads, such as induced in bridge components under traffic loads and heavy vehicle crossings. The stress amplitudes are much lower than the failure strength of the material, and are experienced very often or even continuously during the structure lifetime. Fatigue damage is a function of the magnitude and frequency of load effect cycles, as well as of the fatigue strength or behaviour of a structural detail. The fatigue strength depends on the geometry of the detail and the parameters and quality of the welds (Bathias and Bailon, 1997).

In the mid-80's, extensive traffic data were collected in France and in a few other European countries (Bruls, Jacob, Sedlacek, 1989), including WIM (weigh in motion) data, to get figures of the real traffic loads applied on the existing bridges, and to be applied on future bridges. These data, recorded by WIM systems on motorways, heavily trafficked highways or secondary roads, were used for the calibration of load models and fatigue loads (Kretz and Jacob, 1991; Jacob and Kretz, 1996) of the Eurocode EN1991-3 (CEN, 1994) in the late 80's and early 90's. Most of the fatigue lifetime calculations were carried out for composite bridges (Jacob, 1989), using the most sensitive details of the steel structures, i.e. the welds between upper and lower flanges of main girders and the vertical stiffeners of the girders' web. These details are sensitive to vehicle weights and dimensions, and traffic parameters such as vehicle headings, crossings or take over, because they result from longitudinal bending moments coupled with transverse influence lines. Local effects, such as welds between longitudinal stiffeners and deck plate of steel orthotropic decks, widely investigated in ECSC research projects (Bignonnet et al., 1990a & b), are not considered here, because the lifetimes are mostly governed by the number of axles and the axle load distribution.

Ten years later, the freight traffic throughout Europe increased a lot, the truck and tire characteristics changed quickly, e.g. with the increase of wide tires and tridem axles on semi-trailers, the development of air suspensions, a better management of heavy vehicle fleets leading to a reduction of empty lorries travelling on the roads, etc. Also, the WIM systems and their calibration were improved (COST323 2002; Jacob 1999; WAVE 2001), and thus WIM

data became more reliable and accurate. Therefore, it seems necessary to re-assess the dynamic effects of heavy vehicles on bridges for fatigue, the safety of bridges in fatigue and the calibration of the fatigue load models.

New traffic (WIM) data were recorded on a few French motorways in 1997 and 2001, in order to be compared to those of 1987, and to analyse the evolution in traffic and lorry aggressiveness for bridges in fatigue. Attention is paid to the influence of individual truck characteristics and traffic parameters.

2. TRAFFICS AND WIM DATA

Traffics

Two highway traffics were chosen to be recorded during one whole week at least, and used for fatigue lifetime computations. Two older highway traffics were chosen in addition.

Each traffic will be apply on each bridges, without any correlation with the real traffic supported by the considered bridge. Indeed, the objective is not to check a particular bridge project, but to verify that some critical details regarding to fatigue in the bridges comply with best practice codes or eurocodes.

Traffic loads were recorded on the four lanes (both directions) of the A9 motorway (Nîmes-Montpellier-Perpignan) in October 2001, close to the Spanish border, by the concessionary company Autoroutes du Sud de la France (ASF). During the same period, the traffic was also recorded on the two lanes of each direction on the A5 motorway (Paris-Troyes-Langres) in Eastern France, by the concessionary company Société des Autoroutes Paris-Rhin-Rhône (SAPRR). The third traffic was recorded in 1997 on the A31 (Luxembourg-Metz-Nancy-Langres) motorway at Butgnéville, in Eastern France between Nancy and Langres, on the two lanes of both directions.

The traffic of the A6 motorway (Paris-Lyon) was recorded in 1986 at Auxerre, and was widely used for the studies and calibration of the Eurocode EN1991-3. Two lanes of each direction were recorded, but for the 2-lane bridges, only one direction was used (instead of the two slow lanes).

Generally, lorries and cars were recorded, but for bridge loading and fatigue purposes, only lorries with a gross weight above 3.5t are considered. Some coherence tests were carried out on the traffic files, in order to eliminate outliers (for speed, weights, axle spacing, etc...), so that CASTOR-LCPC could compute correct values. The transverse location of the lorries within each traffic lane was not measured by the WIM systems, but simulated afterwards by a truncated normal distribution (so that no wheel could be out of the lane).

The main traffic statistics are shown in Table 1. Among the recent traffics, the traffic on A9 motorway has a lorry flow twice higher than the A5 traffic, which is itself twice than the A31 traffic. The distribution by type of vehicles is very similar for the three recent traffics (A9, A5 and A31), if classified either by number of axles, or by class of lorry (1 = 2-axle rigid, 2 = 3 and 4-axle rigid, 3 = articulated tractor and semi-trailer, 4 = drawbar lorries). More than 75% of the lorries have 5 axles, and almost 80% are tractors with semi-trailer (4, 5 or 6 axles). The second and third largest classes are class 4 (10 to 15%) and class 2 (7 to 9%). The percentage in class 4 highly depends on the location, because drawbar lorries are more common in Germany and northern Europe than in France and Spain. Therefore, this percentage reaches 15% on the A31 motorway, which is a main route from Benelux and Germany to south of France, Italy and Spain, while it drops down to 10% on the A9 motorway near Spain.

The 15 year old traffic of the A6 motorway shows several patterns, which reveals an important time-evolution of the types of lorries used in France and Europe. In 1986, the 5-axles lorries were less than half of the population (43%), and the class 3 contained only 62%. The class 1 contained one fourth of the population. Moreover, at that time, approximately half of the articulated tractors with semi-trailers only had 4 axles, with a tandem axle (and twin wheels) under the semi-trailer, while now almost 85% of these articulated lorries have 5 axles and a tridem axle under the semi-trailer (with wide tires).

It is also well known that the proportion of air suspension highly increased since 15 years, and now it is likely that at least 50 to 60% of the lorries are equipped with such "road-friendly" suspensions.

WIM data

The traffic of the A6 motorway was one of the heaviest traffic in France in 1986, and one of the heaviest recorded in Europe at that time. Therefore, it was the traffic which was mainly used to calibrate the traffic load models of the Eurocode EN 1991-3 (extreme loads). The gross weight distribution always has a tri-modal shape: (i) a first mode is concentrated just above 3.5 t, which contains overloaded vans and small 2-axle lorries, (ii) a second mode is located between 18 and 25 t, which consists of overloaded 2-axle lorries, and fully loaded class 2 lorries, and partially loaded lorries of classes 3 and 4; this mode is the most scattered, (iii) the third mode is concentrated around the highest current legal limit (44 t), while the upper tail of the distribution extends until 55 or 60 t (overloaded lorries and special permits); it contains fully (or over-) loaded classes 3 and 4 lorries.

In 1986, on the A6 motorway, the two upper modes were not well distinguished and the upper tail was longer, even if the legal limit at that time was only 38 t. This is explained by two reasons: (i) the dynamic impact factors were higher with the mechanical suspensions than with the air suspensions, which leads to more scattered impact forces and higher values, (ii) the WIM systems were much less accurate than now, and thus again the scattering was higher and the maxima over-estimated. This overestimation was considered in the studies of the Eurocode, and the extreme load effects calculated were reduced by app. 10%.

The mean gross vehicle weight is 29.5 t on the A9, 29.9 t on the A5, 32.6 t on the A31 and 32.4 t on the A6. Because the A6 traffic was the heaviest measured in 1986, and loads were slightly over-estimated, it confirmed the general trend of a significant increase of the gross weights over the 15-years period.

Axles weight distribution presents only one peak, which is always centred around 6 t, and a rather constant shape over time and space. The distribution is more scattered and presents a longer upper tail (until 20 t) for the old traffic of the A6 for the reasons explained above.

Tandem and tridem axles distributions are always bimodal whatever the traffic. Indeed, one peak is related to empty lorries and the second one to fully loaded lorries. A5 traffic shows a very high proportion of axle groups in the first mode, which is in agreement with the proportion of partially loaded lorries - the 2nd mode of the gross weight distribution also exceeds 50% of the population -.

Length distributions reveals an evolution of large lorry design with time, e.g. an increase of the articulated tractor with semi-trailer length from 11 m in 1986 to 12.5 m in 1997 and 2001 (the higher value for the A9 traffic is due to the fact that the total vehicle length is considered instead of the spacing between the front and the rear axles). These class 3 vehicles are almost standardised, and thus the scattering of their length distribution is very low. In 1986, the higher proportion of class 1 vehicles induced a higher proportion of shorter lorries, with an axle spacing from front to rear between 2 and 6 m.

Higher the traffic density, lower the scattering of the vehicle heading distribution.

Table 1- Statistics associated to the chosen traffics

Highway	A9	A5	A31	A6 (Aux86)
Trucks flow / day	8 183	4 583	2 101	5 503
Load flow (t/day)	241 390	136 868	68 578	178 360
Percentage of each type of truck (Nb of axes)	<p>1-axis: 75% 2-axes: 1% 3-axes: 9% 4-axes: 12%</p>	<p>1-axis: 76% 2-axes: 2% 3-axes: 1% 4-axes: 14%</p>	<p>1-axis: 70% 2-axes: 1% 3-axes: 8% 4-axes: 10%</p>	<p>1-axis: 43% 2-axes: 22% 3-axes: 3% 4-axes: 31%</p>
Percentage of each class of truck class 1: 2-axes rigid vehicles class 2: >2-axes rigid vehicles class 3: articulated vehicles class 4: draw bar vehicles	<p>class 1: 10% class 2: 2% class 3: 79% class 4: 9%</p>	<p>class 1: 12% class 2: 1% class 3: 90% class 4: 7%</p>	<p>class 1: 25% class 2: 2% class 3: 62% class 4: 11%</p>	<p>class 1: 25% class 2: 2% class 3: 62% class 4: 11%</p>
Distribution of Gross Vehicles Weight	<p>Gross Vehicles Weight distribution on A9 highway class interval: 1 ton</p>	<p>Gross Vehicles Weight distribution on A5 highway class interval: 1 ton</p>	<p>Gross Vehicles Weight distribution on A31 highway class interval: 1 ton</p>	<p>Gross Vehicles Weight distribution on A6 highway class interval: 1 ton</p>
Distribution of Axes weights	<p>Axes Weight distribution on A9 highway class interval: 0.5 ton</p>	<p>Axes Weight distribution on A5 highway class interval: 0.5 ton</p>	<p>Axes Weight distribution on A31 highway class interval: 0.5 ton</p>	<p>Axes Weight distribution on A6 highway class interval: 0.5 ton</p>

Distribution of tandem axle weights	<p>Tandem Axle Weights distribution on A8 Highway</p> <p>class interval: 1 ton</p>	<p>Tandem Axle Weights distribution on A5 Highway</p> <p>class interval: 1 ton</p>	<p>Tandem Axle Weights distribution on A31 Highway</p> <p>class interval: 1 ton</p>	
Distribution of tridem axle weights	<p>Tridem Axle Weights distribution on A8 Highway</p> <p>class interval: 1 ton</p>	<p>Tridem Axle Weights distribution on A5 Highway</p> <p>class interval: 1 ton</p>	<p>Tridem Axle Weights distribution on A31 Highway</p> <p>class interval: 1 ton</p>	
Distribution of heavy vehicles length	<p>Heavy vehicles length distribution on A8 Highway</p> <p>class interval: 0.5m</p>	<p>Heavy vehicles length distribution on A5 Highway</p> <p>class interval: 0.5m</p>	<p>Heavy vehicles length distribution on A31 Highway</p> <p>class interval: 0.5m</p>	<p>Heavy vehicles length distribution on A8 Highway</p> <p>class interval: 0.5m</p>
Distribution of distance interval between consecutive heavy vehicles	<p>Interval between vehicles on a bridge, A8 highway</p> <p>Percentage of all vehicles: 72%</p>	<p>Interval between vehicles on a bridge, A5 highway</p> <p>Percentage of all vehicles: 59%</p>	<p>Interval between vehicles on a bridge, A31 highway</p> <p>Percentage of all vehicles: 52%</p>	

3. BRIDGES AND DETAILS

Bridges

Seven existing composite bridges, all located in France, are chosen : Libourne, Auxerre, Beaucaire, Joigny, Layrac, Millau and St Denis. All of them but St Denis are two main girder bridges with 2 or 3 traffic lanes, while the last one has four main girders and four traffic lanes. Auxerre and Joigny are simple supported spans, while the other bridges have 4 to 6 continuous spans.

The details considered are the welds between the vertical girder web's stiffener and the lower (at mid span) or upper (on piers) beam flanges. The strains in these details are induced by the general bending of the main girders. The influence lines of the stresses induced by these bending moments for each bridge are given in Figure 1, with the following indications: name of the bridge, "m" for bending moment effect, a number which represents the section of the detail (generally at mid-span or on a pier), and the letter "i" for lower flange or "s" for upper flange; e.g. "Layrac m3i" means effect in the section n°3 of Layrac bridge, in the lower flange (thus at mid span, here of the second span). The ordinate y (in MPa/kN) gives the stress induced in the detail by a load of 1 kN applied at the abscissa x, right on the main girder. Then, influence surfaces were build using linear transverse influence lines, with an ordinate 1 on the considered main girder, and 0 on the other one (for 2 girder bridges).

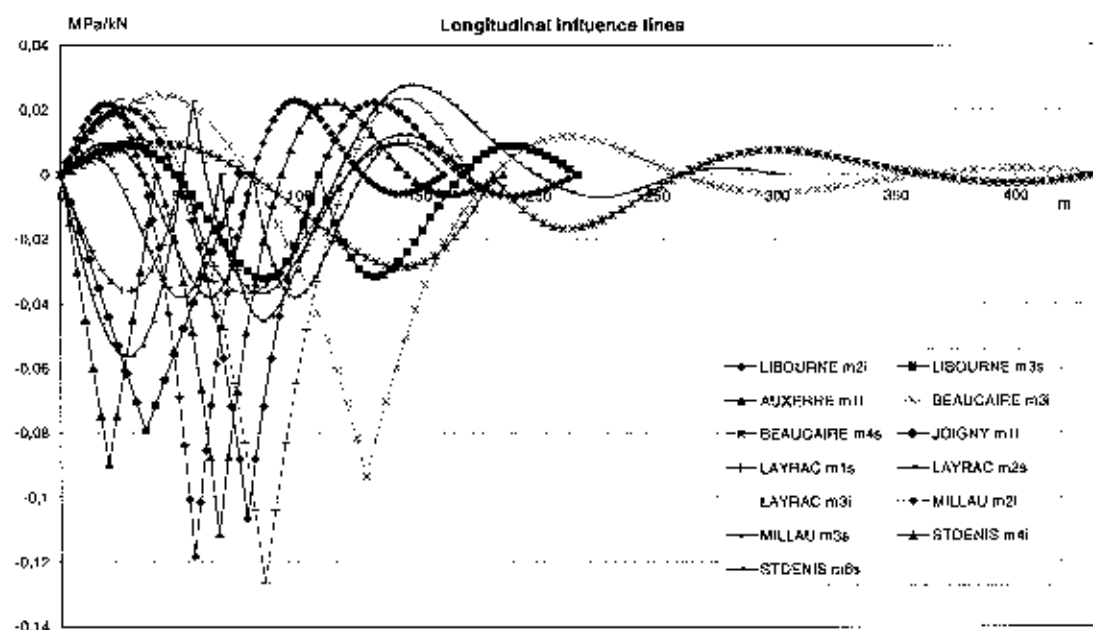


Figure 1 - Longitudinal influence lines of the chosen bridges and details

The fatigue strengths of the details, expressed by their class of fatigue, i.e. the ordinate of the log-log S-N curves for $N = 2 \cdot 10^6$ cycles, are presented in Table 2.

Table 2 - S-N classes of the details

Bridge/section	Class (Eurocode 3)	Thickness (mm)	Corrected class	Bridge/section	Class (Eurocode 3)	Thickness (mm)	Corrected class
Auxerre m1i	71	70	56	Libourne m3s	-	-	50
Beaucaire m3i	71	30	68	Layrac m1s	50	70	40
Beaucaire m4s	-	-	50	Layrac m2s	50	30	48
Joigny m1i	71	90	50	Layrac m3i	71	26	71
Millau m3s	45	80	34	St Denis m6s	50	45	45
Millau m2i	-	-	50	St Denis m4i	71	30	68
Libourne m2i	-	-	50				

The class, proposed by the Eurocode 3 (Steel constructions) for a given geometry of the weld (join) and the common welding process, is corrected with respect to the thickness of the main steel plate (here the girder flange). The multiplicative factor is $\sqrt[4]{25/t}$, where t is the plate thickness in mm; thicker the plate, lower the class and fatigue strength. In some cases, the data on fatigue strength were missing, and the corrected classes were chosen in agreement with the other ones.

4. SIMULATION SOFTWARE "CASTOR-LCPC"

CASTOR-LCPC is a software developed by the LCPC in 1988 (Eynard and Jacob, 1989), which simulates any traffic flowing on a bridge, described by a set of influence lines or surfaces. The inputs are a traffic data file, i.e. a series of vehicles described by sets of axles, with axle load and spacing, time of passage, traffic lane and lateral position, and the influence surfaces of the considered load effects. The software simulates the passage of the vehicles with respect to their lateral position, speed and headings, and calculates for any given time step, the time history of the load effects (e.g. total load on a span, bending moments, shearing forces, bearing reactions, or stresses) induced by the traffic loads (see Figure 2). The time history of these load effects is not stored, but statistics are computed on real time, and histograms are provided: (i) load effect distribution (computed values), (ii) local minima and maxima, (iii) level crossings, (iv) "rain-flow" cycles, the counting method used for fatigue calculation by Miner's law. CASTOR-LCPC was widely used to produce the target values of load effects (Flint and Jacob, 1996) used for the calibration of the Eurocode "Traffic Loads on Road Bridges" (EN 1991-3). CASTOR-LCPC automatically proceeds fatigue calculations, using the S-N curves of the given details, the "rain-flow" histograms, and the Miner's law.

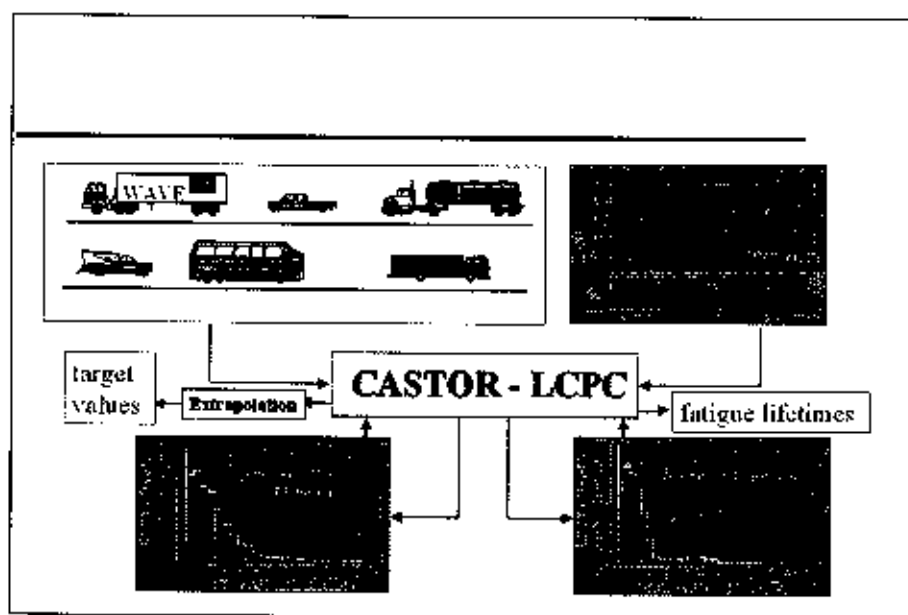


Figure 2 : CASTOR-LCPC software - general flow chart

5. RESULTS

Simulations were carried out with the 13 influence lines (and surfaces) of the 7 bridges described in section 3, for each of the three motorway traffics : A9, A5, and A31. The results are compared to those obtained with the traffic of the A6 motorway (1986) for the same influence lines. The calculations with the A6 traffic on the two-lane bridges were carried out using two lanes in one direction, i.e. a slow lane and a fast lane, because the aim was to be representative of a highway traffic. But the calculations with the A9, A5, and A31 traffics were carried out using the two slow lanes in each direction for the two-lane bridges. Therefore, for the two-lane bridges, the A6 traffic was not that aggressive that it could be.

Rain-flow histograms

Rain-flow histograms for each influence line, with the three traffics of A9, A5 and A31 are given in

Figure 3.

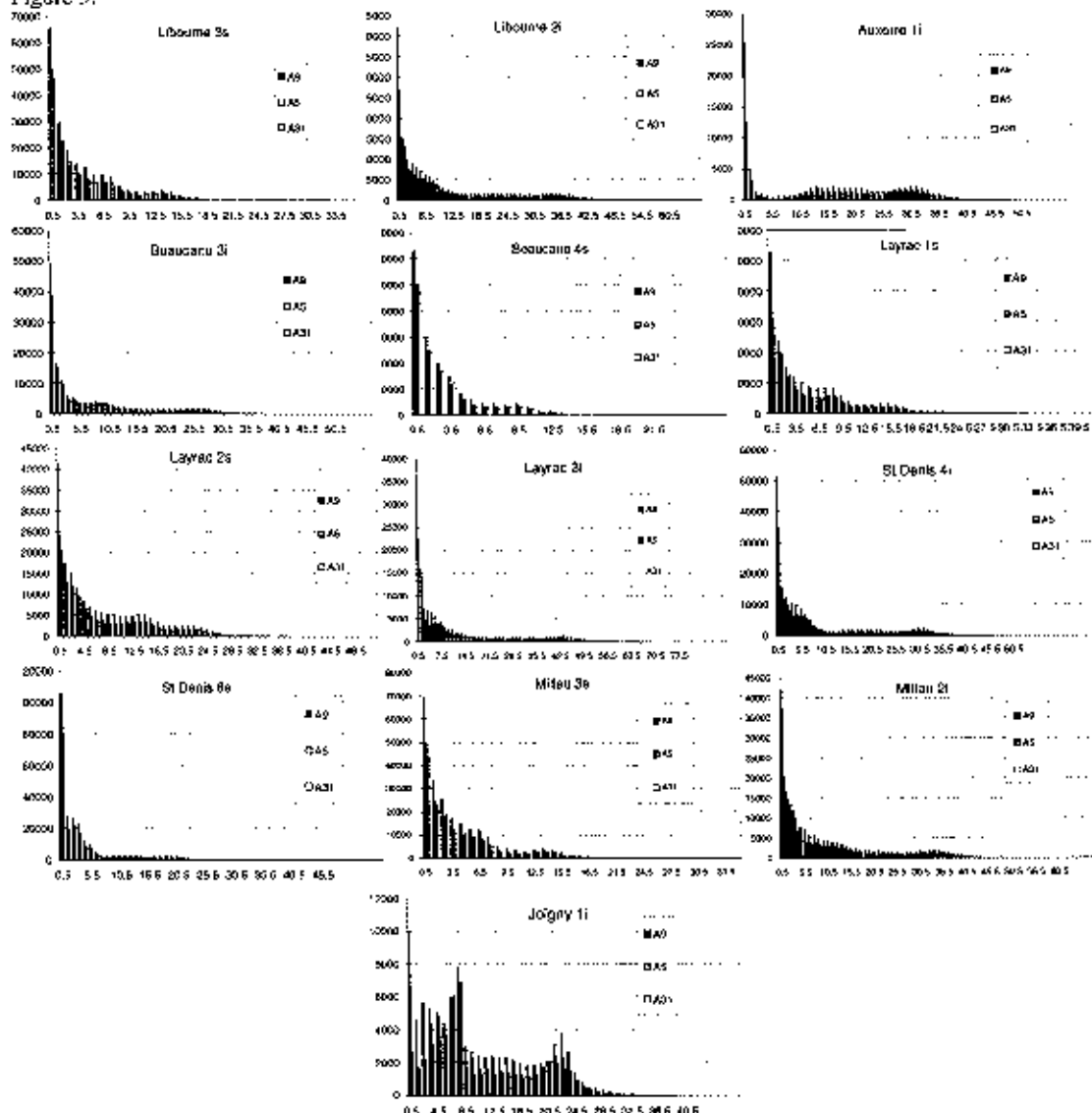


Figure 3 - Rainflow histograms for each influence line

The shape of the histograms are rather independent on the traffic, but the number of cycles in each class and the number of non empty classes vary according to the traffic density, and the vehicle loads. However, the shape of the histograms depends on the influence line. For the simple supported spans (Auxerre and Joigny), the rain-flow histograms show tri-modal distributions, as the gross vehicle weights distribution, because most of the lorries (passing alone on the bridge) induce one stress cycle per vehicle, with an amplitude proportional to its weight. For the continuous span bridges, the rain-flow histograms are smoother, but a second mode is mostly visible, which corresponds to the third gross weight mode.

Lifetimes

The computed lifetimes for each influence line and traffic are given in Table 3. It should be underlined, that these lifetimes are rather conventional, according to the crude hypotheses of the Miner's model. Only the order of magnitude should be considered, in a logarithmic scale. It means that lifetimes under 50 years are much too short for an acceptable bridge design, lifetimes between 50 and a few hundreds years are either acceptable or good, and over 1000 years it means either that the detail is not exposed to any significant fatigue damage, or that the traffic is

not aggressive for this detail. The most interesting is to compare the lifetimes for the same detail and different traffics, or for the same traffic and different details, i.e. to assess the relative aggressiveness of the traffic or the sensitivity of the details.

For four details (Libourne 2i, Auxerre 1i, Layrac 3i and Millau 2i) the lifetimes are unacceptable under each of the considered traffics. However, it doesn't mean that these bridges are unsafe or poorly designed, while the real traffics carried by them are much lighter than those on the motorways (they all carry secondary roads). Only St Denis' bridge carries a motorway (A86, an urban circular motorway around and outside Paris), with a traffic as dense as a motorway, but a high proportion of personal cars. For that bridge, only the traffic of the A5 for "St Denis 6s" leads to a too short lifetime. That is investigated below. The Beaucaire's bridge carries a main highway, but it is clearly well designed for heavy traffics.

Among all the traffics, the A6 is generally the most aggressive, whatever the detail, even if only one slow lane was used for the two-lane bridges. That is because of the heavier weights recorded and the non linearity of the fatigue damage with respect to the stress amplitude (a power 3 to 5 appears in the relationship between the stress amplitude and the damage). Thus, for example with the power 5 (mostly used), an increase of 15% in a lorry weight (in fact stress amplitude) will double the damage! According to the remark done in section 2 about the relative inaccuracy of the old traffic data (A6, 1986) and the overestimation of the upper tail of the GW distribution, we may assume that the fatigue lifetimes are underestimated by 30 to 50%. If so, the A6 old traffic would be approximately as aggressive as the recent A9 traffic.

Table 3 : Lifetimes (years)

	A9	A5	A31	A6 (1986)		A9	A5	A31	6 (1986)
Libourne 3s	1169	1646	1976	682	Layrac 3i	18	27	29	13
Libourne 2i	9	14	15	8	St Denis 4i	93	144	165	100
Auxerre 1i	35	41	54	23	St Denis 6s	86	33	155	85
Beaucaire 3i	247	372	421	100	Millau 3s	90	154	176	90
Beaucaire 4s	49217	53739	122032	358800	Millau 2i	11	17	18	9
Layrac 1s	125	183	209	70	Joigny 1i	108	165	188	85
Layrac 2s	58	84	92	40					

Results for A9, A5, and A31 show, as expected, that heavier (and denser) the traffic lower the lifetime. The case of St Denis 6s, where the A5 traffic is twice more aggressive than the other ones is not so clear. We performed some detailed analysis to try to explain this unexpected result. The assumption was that, according to the influence line shape, with two negative peaks at the abscissa 50 and 80 m, the detail could be more sensitive to a traffic where a high proportion of vehicle headings would be close to 30 m. But nothing confirmed such a specific characteristic of the A5 traffic... A more detailed investigation would be needed to clarify this phenomenon.

The traffic of the A31, which has a high proportion of heavy lorries in the third mode (around 40 t), but a density half than the A5 traffic, has the same aggressiveness for the details of the two-girder bridges with a sharp influence line (Libourne 2i, Layrac 3i, Millau 2i). For such sharp influence lines, the individual lorry weights mainly govern the fatigue lifetime. These details are also those with the shortest lifetimes.

The lifetimes under the traffics of the A9 and A5 motorways, with similar gross weight distributions, but a ratio of 1.8/1 for the lorry density, have generally a ratio of 1/1.5. This is due to a significant reduction of the lorry headings when the density increases, and therefore, more lorries are passing on the bridge together in a queue, being partially on positive and partially on negative parts of the influence line, which reduce the overall effect. Thus, less stress cycles and lower amplitudes are experienced than for the same number of vehicles passing alone.

6. CONCLUSIONS

This study confirms the importance of an accurate and periodical survey of traffic loads on bridges, using WIM systems, to assess and verify the lifetimes of structures exposed to heavy traffics. Moreover, sensitivity studies may be carried out using traffic data of various types of roads and highways, and bridge influence surfaces of different shapes, to identify the most critical situations.

Steel-concrete composite bridges, which are very common in Europe, and their most sensitive details in fatigue, i.e. the weld between vertical stiffeners and main girder flanges, in which stresses cycles are induced by girder bending, may be highly penalised by an increase of the individual lorry loads; that is above all true for the mid-span section of the 2nd span of 3 or more span bridges, and more generally for any details with a sharp longitudinal influence line. This conclusion will have to be considered in the future evolution of lorry design and in the weight limits. The tendency over the last 20 years in Europe, was a continuous increase of the maximum allowed gross weights (from 36 t until 44 t, and even more in northern Europe). It should be underlined that, because of the non linearity of the fatigue damage with respect to the loads, an increase of 15% of a gross vehicle weight may lead to double the damage, and thus reduce the lifetime of the structure by a factor 2 !

Conversely, the quick increase of the heavy traffic flow seems, at least for long and continuous span bridges, not to reduce the bridge lifetime proportionally, as expected. In our examples, an increase of the lorry density by a factor 1.8 only lead to a reduction of the lifetime by a factor 1.5. When the vehicle headings become smaller than the bridge length, several lorries pass simultaneously on the bridge being in the same traffic lane, and thus their effects are not fully cumulated, and even sometime are reduced, because of the changes of the influence line ordinate sign.

For bridge safety, it is important to divide the loads between more lighter lorries, than to concentrate them on heavier lorries. That is may be neither the main tendency, nor the wish of the transport companies, but the cost of the freight transport should likely also account for the bridge design and maintenance...

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COMPARATIVE PERFORMANCE OF SEMI-TRAILER STEERING SYSTEMS

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ABSTRACT

A yaw-roll model of a tractor semi-trailer is used to compare the performance of various trailer axle steering systems fitted to a standard length semi-trailer. The model is highly adaptable and incorporates effects such as non-linear tyres and lateral load transfer, which are important in heavy vehicle handling. Within the model a simple driver sub-model is used to steer the vehicle down the desired path at a desired velocity.

Results from simulations of the various steering systems are presented. Common manoeuvres are performed including low-speed corners, low-speed roundabouts, high-speed lane changes and high-speed circles. Performance is characterised by a set of performance measures, which include the vehicle's ability to follow the desired path, its handling characteristics, lateral tyre forces, load transfer and rearward amplification. Reasons for the differences in performance of the various systems are discussed and recommendations are made to improve the performance where necessary.

1. INTRODUCTION

In recent years a number of systems have been developed which allow the rear axles of semi-trailers to be steered. By steering the rear axles such systems aim to improve the low speed manoeuvrability of the vehicle as well as reduce tyre scrub. This is important for transporting goods in urban areas where vehicles need to negotiate sharp corners and small roundabouts.

Current semi-trailer steering systems can be grouped into one of three types; namely self-steering systems, command steer systems and pivotal bogie systems. Each type uses a different strategy to determine the angles of the axles. Self-steering systems steer the trailer axles in relation to the lateral tyre forces, command steer systems steer in relation to the articulation angle between the tractor and trailer and pivotal bogie systems steer in relation to angle of the rear bogie assembly. Hence although all can be classified as semi-trailer steering systems they are fundamentally different. Details of each type of steering system are presented along with the governing equations in Section 2.

To date, only a limited number of papers have been published on the performance of semi-trailer steering systems. In the 1980's LeBlanc, El-Gindy and Woodroffe undertook research into self-steering axles and their use on rigid body trucks and C-type dollies [1] and [2]. Command steer systems have been studied by Jones and Wright [3] and Sankar, Rakhja and Piche [4]. Most recently, the performance of pivotal bogie systems have been investigated by Henderson [5] and Prem [6]. The studies above have generally concentrated on one particular type of steering system and how it performs under certain circumstances.

The aim of this paper is to compare the performance of the three types of semi-trailer steering systems. The comparison is made using an adaptable simulation model, capable of modelling different steering systems fitted to the same standard length tri-axle semi-trailer. The model and its main components are described in detail in Section 3.

To quantify the high and low speed performance of the steering systems a comprehensive set of performance measures were used. The measures are introduced in Section 4 and tabulated in Table 1. The measures are mainly based on those proposed for Performance Based Standards (PBS) legislation in Australia.

The results from the comparative study are presented and discussed Section 5. The discussion includes how the results from the steering semi-trailers compare to the fixed baseline trailer as well as how the results compare to the proposed PBS limits. Reasons for differences in performance are given.

Finally, conclusions are drawn in Section 6 regarding how the various semi-trailer steering systems compare along with their potential advantages and limitations. Future work required in this area is also described, including validation of the model and comparison of the simulation results with field tests data.

2. DESCRIPTION OF SEMI-TRAILER STEERING SYSTEMS

2.1. Self-Steering Systems

Self-steering systems are the most widely used type of semi-trailer steering system, mainly due to their simplicity and low cost. In typical systems the rearmost fixed axle in the tri-axle group is replaced with a self-steering axle. The most popular type of self-steering axle is an "automotive style" self-steering axle. This is similar in design to a conventional steering axle but with a positive trail. Instead of being controlled by a steering box, the steering of the axle is typically controlled by a preloaded spring and damper attached to the trailing arm. The purpose of the spring-damper is to help centre the axle, offset the effects of unbalanced braking and provide lateral forces at low angles of steer necessary to prevent instability. Mechanisms are usually incorporated into the designs to lock the axle when travelling in reverse.

When a semi-trailer fitted with a self-steering axle transverses a low-speed corner the wheels on the self-steering axle align with the direction of travel. This reduces the lateral forces acting on the self-steering axle. Lateral forces on the other axles are also reduced because the self-steering axle decreases the magnitude of any "locked-in" lateral forces associated with the axle group. Hence the self-steering axle is generally beneficial to tyre wear on all axles.

A self-steering axle also improves the low-speed cornering and manoeuvrability of a vehicle. Since the self-steering axle generates little lateral force when cornering it can be neglected. This reduces the effective wheelbase of the semi-trailer to that of an equivalent fixed tandem semi-trailer (i.e. a semi-trailer without the self-steering axle).

The degree to which a self-steering axle steers is determined by the balance of moments about its king pins. For normal running it can be assumed that longitudinal tyre forces are negligible and thus the tyres are subjected to lateral forces only. LeBlanc et al. [2] showed that self-steering axles generally have a non-linear relationship between the steer angle and the lateral tyre force, as shown in Figure 1. This relationship can be represented by the following equation;

$$\delta = \begin{cases} \frac{F_y}{K_1} & , |F_y| < F_{yc} \\ \frac{F_y}{K_2} + \frac{F_{yc}}{K_2} \left(1 - \frac{K_2}{K_1} \right) & , |F_y| > F_{yc} \end{cases} \quad \dots(1)$$

where

δ = Steering angle [rad]

F_y = Total lateral tyre force acting on axle [N]

F_{yc} = Centring force [N]

K_1, K_2 = Axle Cornering Stiffnesses [N/rad]

Hence a self-steering axle is relatively stiff in yaw at low steer angles but becomes relatively compliant at larger steer angles.

2.2 Command Steer Systems

In a typical command steer system the rear two fixed axles in the tri-axle semi-trailer group are replaced with conventional style steering axles. These axles are made to steer in relation to the articulation angle between the tractor and semi-trailer. The articulation angle is sensed using either electronics or a special ballrace attached to the trailer side of the fifth wheel. This angle then transmitted to the steering axles electronically or via mechanical or hydraulic linkages.

The equations that govern the behaviour of a command steer system can be obtained by considering the vehicle in a low speed turn, as shown in Figure 2. In order to have no sideslip the rearmost wheels need to be steered so that their normals pass through the same centre as the front trailer axle. This eliminates the lateral tyre forces on the rear axles and eliminates locked in forces in the group, both of which are beneficial to tyre wear.

The following relationships between the articulation angle and steer angles can be obtained from the geometry of the system:

$$\begin{aligned}\delta_m &= \tan^{-1} \left(\frac{d \cdot \sin \Gamma}{b \cdot \cos \Gamma - a} \right) \\ \delta_r &= \tan^{-1} (2 \cdot \tan(\delta_m))\end{aligned}\quad \dots(2)$$

where;

- Γ = Articulation angle [rad]
- δ_m, δ_r = Middle axle and rear axle steer angles [rad]
- a = 5th wheel to rear tractor axle distance [m]
- b = 5th wheel pin to front trailer axle distance [m]
- d = Spacing between trailer axles [m]

Note that the relationships are non-linear. However, if small angles are assumed they can be linearized giving the following relationships:

$$\begin{aligned}\delta_m &= \frac{d}{b-a} \times \Gamma \\ \delta_r &= 2 \cdot \delta_m\end{aligned}\quad \dots(3)$$

Thus the steering angles are directly proportional to the articulation angle. The small angle approximation is valid for most vehicle operating conditions. However, on very tight radii turns the angles become significant and thus a system designed with a linear relationship does not quite achieve the desired steering angle. This results in small lateral forces being applied to the steering axles and a slight decrease in the cornering radius.

Like the self-steering system, the command steer system also effectively reduces the wheelbase of the semi-trailer. Since both rear axles generate no lateral forces in a turn they can be neglected and the wheelbase of the semi-trailer is reduced to that of an equivalent fixed single axle semi-trailer, as indicated on Figure 2.

2.3 Pivotal Bogie Systems

Pivotal bogie systems have been used for some time in the heavy haulage industry as a means of steering extremely long semi-trailers. It is only recently however that this principle has been applied to normal length semi-trailers.

In a pivotal bogie system a ballrace-mounted tri-axle bogie assembly replaces the fixed-axle trailer group. The ballrace allows this assembly to yaw freely relative to the trailer chassis. The bogie consists of a fixed front axle and two steered rear axles, which steer in relation to the angle between the bogie and the trailer chassis. Thus the axles progressively steer as the angle between the bogie and the trailer increases, bringing the bogie back inline with the trailer chassis.

The governing equations for a pivotal bogie system can be derived in a similar manner to the command steer equations. Figure 3 shows a simplified representation of a pivotal bogie system in a curve. To minimise the lateral tyre forces and turning radius the rearmost wheels must again steer such that their normals pass through the instantaneous turning centre. The geometry dictates the following relationships:

$$\delta_m = \tan^{-1} \left(\frac{2d \sin \beta \cos \beta}{b \cos \beta - a + 2(d-c) \sin^2 \beta} \right) \quad \dots(4)$$

$$\delta_r = \tan^{-1}(2 \tan(\delta_m))$$

where:

β = Bogie angle [rad]

c = Trailer centre axle to ballrace distance [m]

Again, the relationships are non-linear due to the bogie articulation and steer angles. Assuming small angles results in the following linearized equations:

$$\delta_m = \frac{2d}{b-a} \times \beta \quad \dots(5)$$

$$\delta_r = 2 \delta_m$$

Comparison of equation (5) with equation (3) reveals that the relationship is the same as a command steer system but with half the gain. Thus the small angle approximation and fifth wheel offset associated with the command steer system produce similar errors in a pivotal bogie system.

By steering all of the axles in the tri group a pivotal bogie system is able to greatly reduce the effective wheelbase of the semi-trailer. For such a system the wheelbase is approximately half of the distance from the fifth wheel to the bogie ballrace, as indicated on Figure 3. Thus the effective wheelbase is approximately half that of a standard fixed tri-axle semi-trailer.

3. SIMULATION MODEL

In order to compare the performance of the various steering systems a roll-yaw model was developed in Simulink. The model consists of a single drive tractor unit coupled to a triaxle semi-trailer. The equations of motion for the system are similar to those used in the UMTRI yaw-roll model [7]. Parameters for the model were chosen to represent a typical European articulated tanker vehicle that complies with current UK legislation [8]. Critical weights and dimensions are shown on Figure 4.

The simulation model has yet to be fully validated although preliminary results from the model were compared with those from other yaw-roll models. Such models have been found to quite accurately represent real vehicles in basic handling manoeuvres. Full validation of the model will be achieved in the near future by comparing simulated results to those from field tests. This is the subject of ongoing work.

The main difference between this model and a conventional articulated vehicle model is the way in which the semi-trailer axle group is modelled. In most vehicle models the axles are non-steerable and connected directly to the trailer chassis. However in this model the axles are steerable and connected to a ballrace mounted bogie assembly.

The steering input to the trailer axles can be varied to suit the type of steering system. For a fixed axle system the steering input is zero and the axles are locked. For a self-steering system the input is the axle's lateral force. For a command steer system the input is the semi-trailer articulation angle. Finally, for a pivotal bogie system the input is the bogie articulation angle.

In a similar manner the bogie can be locked or released to suit the type of steering system. For fixed, self-steering and command steer systems the bogie is locked to the trailer chassis. For a pivotal bogie system the bogie is

released. Thus by choosing the steering input and locking or releasing the bogie the steering systems investigated in this paper can be modelled.

A simplified block diagram of the model is given in Figure 5. Each of the main elements are discussed in the sections below.

3.1 Driver Sub-Model

The driver sub-model determines the front axle steering angle required to make the vehicle move along the desired path. The sub-model uses a preview controller that steers in proportion to the lateral offset between the desired path and the vehicle's current heading at a user defined preview distance from the front axle L , as shown in Figure 6. Thus;

$$\delta_1 = K_1 \cdot x \quad \dots(6)$$

where

δ_1 = Front axle steer angle [rad]

x = Lateral offset [m]

K_1 = User selectable gain

The desired path can be selected from a list of standard manoeuvres or input directly by the user. Standard manoeuvres include constant radius circles, 90-degree bends and SAE lane changes [9].

3.2 Semi-Trailer Steer Angle Sub-Model

The semi-trailer steer angle sub-model outputs the steer angle of each of the steered semi-trailer axles. The block essentially applies equation (1), (2) or (4) depending on which type of steering is selected; self-steer, command steer or pivotal bogie.

3.3 Static Vertical Load Sub-Model

The static vertical load sub-model is used to determine the static vertical load on each axle based on the mass and geometry information for each body input by the user. In the model it is assumed that there is perfect loadsharing between the triaxle trailer group so that each axle has the same static vertical load.

3.4 Lateral Load Transfer Sub-Model

The lateral load transfer sub-model calculates the additional load transferred to each wheel when the vehicle is subjected to a lateral acceleration. The model initially determines the roll angle of the tractor and trailer body based on the roll stiffness of each axle and the compliance of the chassis' and fifth wheel. The lateral load transfer for each axle is then computed. The variation in roll stiffness due to the tractor and bogie articulation angle is taken into account.

The results from the lateral load transfer sub-model are added to the static vertical loads to yield the total vertical load on each wheel of the vehicle.

3.5 Tyre Force Sub-Model

The tyre force sub-model determines the lateral force generated by each wheel based on the vertical load and slip angle. The model is based on an empirical "brush" model with a parabolic pressure distribution and is similar to the models used by Billing [10] and Radt and Milliken [11]. The model, however, takes into account the non-linear relationship between vertical tyre load and lateral stiffness as described by Fancher et al. in [12]. In the model the lateral tyre force is described by the equation;

$$F_y = \begin{cases} F_z \cdot \bar{C} \cdot \delta - \frac{F_z \cdot \bar{C}^2}{3\mu} \delta^2 + \frac{F_z \cdot \bar{C}^3}{27\mu^2} \delta^3 & , \quad \delta < \frac{3\mu}{\bar{C}} \\ \mu F_z & , \quad \text{elsewhere} \end{cases} \quad \dots(7)$$

where

F_y = Lateral tyre force [N]

F_z = Vertical tyre force [N]

μ = Coefficient of friction

C_1 = Tyre cornering coefficient [rad^{-1}]

C_2 = Tyre curvature coefficient [$(\text{N}\cdot\text{rad})^{-1}$]

$\bar{C} = C_1 + C_2 F_z$ [rad^{-1}]

3.6 Vehicle Dynamics Sub-Model

The vehicle dynamics sub-model calculates the displacements, velocities and accelerations of the vehicle's constituent bodies, namely the tractor, semi-trailer and bogie. Application of Newton's Laws of motion to the bodies yields nine differential equations. A further four differential equations are obtained by considering the force balance at the fifth wheel and bogie ballrace. These equations are solved simultaneously to determine the body accelerations and coupling forces. Body displacements and velocities are then calculated by integration of the body accelerations.

Four test vehicles were constructed using the model. The first represented a typical articulated vehicle with fixed semi-trailer axles, which was used as a baseline for comparison purposes. The other test vehicles contained one of each of the semi-trailer steering systems in question.

4. PERFORMANCE MEASURES

In order to compare the performance of the various steering systems a set of performance measures was defined. These measures were used to judge how well each semi-trailer steering system performed relative to a standard fixed semi-trailer and the other steering systems.

A comprehensive set of performance measures for heavy vehicles was compiled by Australia's National Road Transport Commission (NRTC) as part of their performance based standards project [13]. These have recently been reviewed and updated following a review of the characteristics of Australia's heavy vehicle fleet [14]. It was decided to use the NRTC measures as a basis for this study. Only measures deemed relevant to trailer steering systems were selected.

In addition to the NRTC measures, two measures that are applicable to tractor/semi-trailer units operating in the UK were defined. The measures were the maximum swept path width and peak trailer lateral tyre force associated with a vehicle travelling in a small radius circle at low speed. These measures were deemed important for vehicles that have to manoeuvre around tight roundabouts.

The complete set of performance measures used in this study is shown in Table 1.

5. RESULTS

For each test vehicle, simulations were conducted of the vehicle undergoing the six manoeuvres outlined in Table 1. Results from these simulations were used to determine the twelve performance measures for each vehicle. These are shown in absolute terms in Table 2 and relative to the fixed-axle trailer vehicle in Table 3. Each measure is discussed in detail below.

5.1 Low-Speed Corner Swept Path Width (SPW_{90})

As expected, all steering systems reduced the width of the swept path of the vehicle when travelling around a low-speed corner. Hence an articulated vehicle with a semi-trailer steering system can better negotiate tighter corners and narrower roads compared to a conventional articulated vehicle. This is evident in Figure 7a, which shows the path followed by each test vehicle's trailer group. A narrower swept path is one of the major benefits of installing a trailer steering system.

The narrower swept path opens up the possibility of using the vehicle on narrower roads. An indication of the suitability of a vehicle to a particular road type can be gained by comparing its swept path to the NRTC proposed performance level in Table 1. For travel on arterial roads SPW_{90} must be less than 7.4 m. All vehicles, including the fixed semi-trailer, passed the arterial road criterion (Table 2). For travel on local roads SPW_{90} must be less than 5.0 m. Only those vehicles with semi-trailer steering systems passed this criterion. Thus fitting of semi-trailer steering systems could allow articulated vehicles to be used on local roads as well as arterials.

The relative ability of each of the semi-trailer steering systems to reduce SPW_{90} can be obtained by comparing the values in Table 3. The best performing system was found to be the pivotal bogie system, which reduced SPW_{90} by 27%. For the pivotal bogie vehicle the swept path was only 1.3m wider than the vehicle, indicating excellent tracking. The superior tracking performance of the pivotal bogie system is primarily a result of its effective wheelbase being approximately half that of the fixed semi-trailer.

The command steer system reduced SPW_{90} by 11%. In a corner the command steer system steers two of the axles to reduce the effective wheelbase of the trailer to that of an equivalent single. The least effective system was the self-steer system, which reduced SPW_{90} by only 6%. The reduction in performance is due to steering only one of the axles in the group and hence only reducing the wheelbase of the trailer to that of an equivalent tandem. To improve tracking two self-steering axles could be fitted to a trailer. However, this is seldom done in practice because it can lead to stability and tracking problems at high speed.

5.2 Tail Swing (TS)

All steering systems investigated increased the amount of entry tail swing compared to a fixed semi-trailer. The order of performance was found to be opposite to that of SPW_{90} . The best performing steering system was the self-steer system while the worst was the pivotal bogie, as shown in Table 3. Entry tail swing of the pivotal bogie was above the proposed NRTC limit as indicated in Table 2.

The increased entry tail swing associated with steering semi-trailers is due to their greater effective rear overhang. All steering systems reduce the effective wheelbase of the semi-trailer to reduce the amount the vehicle cuts in on a corner. However, reducing the effective wheelbase increases the effective rear overhang, producing more tail swing.

The fixed semi-trailer, self-steer and command steer systems exhibited no tail swing on exit. In contrast the exit tail swing of the pivotal bogie semi-trailer was above the proposed NRTC limit. Such tail swing may prove to be a problem since drivers are not used to taking it into account when exiting a corner.

The difference in exit tail swing performance is due to differences in the paths the vehicles follow through the corner. The fixed, self-steer and command steer semi-trailers remain inside of the path of the tractor throughout the turn. The pivotal bogie semi-trailer, however, overshoots the path of the tractor on exit, as highlighted in Figure 7a. This overshoot causes the tail to swing out.

Tail swing performance of any of the steering systems can be improved by moving the trailer group rearwards to reduce the amount of rear overhang. However, this will also lead to an increase in SPW_{90} and possible uneven loading distribution between axle groups. Thus the location of the trailer group has to be chosen to produce the best compromise between tail swing, SPW_{90} and load distribution. The ideal location will vary depending on the type of steering system used.

5.3 Steer Tyre Friction Demand (STFD)

Steer tyre friction demand was found to reduce with the addition of a semi-trailer steering system, as shown in Table 3. The order of performance was similar to SPW_{90} , with the best performing system the pivotal bogie and the worst the self-steer axle. All systems had STFD values well below the NRTC proposed level in Table 1 and therefore this area is not of real concern for the type of articulated vehicle simulated.

5.4 Low-Speed Corner Lateral Tyre Force (LTF_{90})

All steering systems investigated were effective at reducing the lateral tyre forces generated by the drive and trailer wheels when travelling around a low-speed corner. LTF_{90} gives a relative indication of the amount of tyre wear due to lateral sliding and associated surface damage to pavements due to shear. Lower lateral tyre forces reduce tyre wear and road damage, which are major benefits of installing a semi-trailer steering system.

The most effective systems at reducing lateral tyre forces were the command steer and pivotal bogie systems. Both produced similar results, reducing drive LTF_{90} values by more than 60% and trailer group LTF_{90} values by 50%, as shown in Table 3. This represents a substantial reduction in tyre and road wear. The reduction in lateral tyre forces on the trailer group is due to the steering systems aligning all trailer tyres with the direction of travel of the vehicle. The reduction in lateral forces on the drive axle is due to lower lateral forces being transferred from the semi-trailer to the tractor through the fifth wheel.

The self-steering axle was not as effective at reducing lateral tyre forces. Nonetheless, it still reduced drive and trailer LTF_{90} values by more than 20%. The decrease in performance compared to other systems is due to only steering one of the trailer axles. Opposing "locked in" lateral forces are still developed by the two front trailer axles which leads to higher trailer and drive axle lateral forces.

5.5 Low-Speed Circle Swept Path Width (SPW_{360})

The width of the swept path of the vehicle when travelling around a low-speed circle gives an indication of the vehicle's ability to negotiate roundabouts. This measure differs from SPW_{90} because the increased time in the circle allows the vehicle to obtain steady state or equilibrium conditions. A plot of the path followed by each of the trailers compared to the desired path is shown in Figure 7b.

As expected, most vehicles had a greater swept path width when travelling in a circle compared to a corner, as seen by comparing Figures 7a and 7b. Relative performance compared to the fixed-axle trailer, however, was similar, as shown in Table 3. The exception was the pivotal bogie system, which had the same SPW_{90} and SPW_{360} . By contrast to the other systems, the pivotal bogie system's swept path width reached a peak near the start of the turn and reduced the further it travelled. Thus the maximum swept path width is independent on the length of curve traversed in a circle.

The level proposed for the swept path width in a corner was used as the acceptable swept path width in a circle, namely 5.0m for local roads. Only the command steer and pivotal bogie systems were found to pass this criterion. Hence it may be necessary to fit articulated vehicles with such semi-trailer steering systems if they are to be used on local roads containing tight roundabouts.

5.6 Low-Speed Circle Lateral Tyre Force (LTF_{360})

All vehicles had LTF_{360} values that were slightly higher than their corresponding LTF_{90} values, as shown in Table 2. This indicates that the steady state lateral tyre forces developed when traversing a circle are generally higher than the initial transient forces encountered on entry.

Although LTF_{360} values were higher, their level relative to the fixed-axle trailer was similar, as shown in Table 3. The pivotal bogie and command steer systems still produced the lowest forces. The self-steer system did not reduce forces by the same extent but still showed marked improvement over the fixed-axle trailer.

5.7 Static Rollover Threshold (SRT)

All vehicles were found to have the same SRT. Hence SRT does not appear to be affected by the use of a semi-trailer steering system. Similar results would be expected if tilt table tests were conducted on representative vehicles since the steering systems do not significantly alter the location of the centre of gravity or the suspension roll characteristics. The SRT was well above the proposed NRTC level of 0.35g.

5.8 Rearward Amplification (RA)

The level of rearward amplification was found to vary depending on the type of semi-trailer steering system used. Two of the three systems increased the level of rearward amplification, as shown in Table 2. The self-steering system, however, was found to have the same rearward amplification as the fixed-axle trailer. This is due to the self-steering axle being relatively stiff in yaw at low steer angles, allowing both systems to generate similar lateral tyre forces in high-speed manoeuvres.

The command steer system was found to have 20% more rearward amplification than the self-steer system while the pivotal bogie system had 30% more, as shown in Table 3. Both of these systems significantly reduce the lateral tyre forces which react inertial loads, hence they increase the lateral acceleration of the trailer unit in high-speed manoeuvres.

Although the command steer and pivotal bogie systems produced higher RA values, the values were still well below the proposed NRTC criterion in Table 1. Thus the increased rearward amplification should be acceptable. Note however, that the SAE lane change manoeuvre used to assess the level of rearward amplification is a relatively mild manoeuvre to perform with a tractor/semi-trailer vehicle. If more drastic manoeuvres are performed with such vehicles the additional rearward amplification could lead to premature rollover.

One way of limiting excessive rearward amplification is to prevent the trailer axles from steering at high-speeds, so the vehicle behaves like a fixed-axle trailer. Some commercially available command steer and pivotal bogie systems employ this strategy.

5.9 High-Speed Transient Offtracking (TO)

All steering systems tested had higher levels of transient offtracking than the fixed-axle trailer. A good indication of the relative performance of each system can be seen in the high-speed lane change paths in Figure 8.

By far the worst performing system was the pivotal bogie. This showed twice the amount of high-speed offtracking as the fixed-axle trailer. Motion of the bogie caused the trailer to overshoot when coming back in line with the tractor after a high-speed lane change.

The command steer and self-steer systems were found to have similar levels of performance, with the command steer system being slightly superior. Transient offtracking levels were 20% and 30% of that of the fixed-axle trailer, as shown in Table 3.

In all cases the level of transient offtracking was well below the NRTC proposed criterion in Table 1. Hence although the transient offtracking is worse it should be acceptable. Like rearward amplification, the increase in transient offtracking can be removed by locking the steering systems at high-speed.

5.10 Load Transfer Ratio (LTR)

LTR was found to vary with the type of steering in a similar manner to rearward amplification. The self-steering system performed the same as the fixed-axle trailer whilst the command steer and pivotal bogies were progressively worse. Compared to the proposed criterion, however, the load transfer ratios were quite similar and relatively low. Hence the load transfer ratios should be acceptable.

5.11 High-Speed Steady State Offtracking (SSO)

As with transient offtracking, all steering systems produced higher levels of high-speed steady state offtracking than the fixed-axle trailer. The command steer system was found to perform best, tracking just 10 mm outside of the fixed-axle trailer as shown in Table 3. The self-steer and pivotal bogie systems had similar results, both tracking 70 mm outside of the fixed-axle trailer. In absolute terms the small increase in offtracking due to the trailer steering system is insignificant. Hence high-speed steady state offtracking of steered semi-trailers is acceptable.

Note that all vehicles, including the fixed-axle trailer, failed the proposed NRTC criterion. This indicates that the proposed criterion may be too stringent for the type of vehicle modelled.

5.12 Handling Quality (HQ)

The handling quality refers to the way a vehicle feels to the driver. A driver may find it difficult to control a vehicle with a significantly different handling quality to a conventional vehicle since it responds in an unexpected way to steering inputs.

Handling quality is usually evaluated using a tractor handling diagram, as shown in Figure 9. The plot shows the lateral acceleration vs. $\delta - L/R$ relationship for the vehicle travelling at constant speed in a diminishing radius circle. The yaw stability of the vehicle can be evaluated by comparing the slope of the handling line to the critical understeer gradient K_{uscr} (indicated by the slope of the black line on the diagram). When the slope of handling curve reaches the critical value the vehicle exhibits yaw instability and becomes directionally unstable. It is desirable for a heavy vehicle not to become directionally unstable before reaching its rollover threshold. Points of yaw instability and rollover have been marked on the diagram.

From Figure 9 it is clear that the fixed, command steer and pivotal bogie vehicles have similar handling qualities while the self-steer vehicle is quite different.

The slopes of the command steer and pivotal bogie curves indicate that both vehicles in turn are slightly more understeer at low lateral acceleration levels and more oversteer at high acceleration levels compared to the fixed-axle trailer. The understeer gradient of the pivotal bogie vehicle reaches the critical level K_{uscr} before reaching the SRT. Hence it becomes directionally unstable before rolling over in contrast to the fixed-axle trailer and command steer vehicles.

The self-steer vehicle starts out being more understeer than the fixed-axle trailer. The understeer gradient stays relatively constant until the lateral acceleration reaches the point where the stiffness of the self-steering axle changes. The vehicle then suddenly becomes more understeer. As lateral acceleration levels increase above this point it becomes progressively more oversteer. The vehicle understeer coefficient reaches the critical level long before the vehicle reaches the static rollover threshold. Hence not only does it become directionally unstable but it does so long before the other vehicles.

In order to quantify the handling performance of a vehicle the NRTC had proposed using a "three point" method. This proposal is currently under review. The method involves looking at the slope of the curve or lateral acceleration at three critical points as shown in Table 1. The results of applying this method are presented in Table 2. All vehicles passed the three point criterion, even the self-steering vehicle with its vastly different handling characteristics. Thus it appears as if the proposed method may not easily indicate if a vehicle handles differently. A better, somewhat qualitative, indication is gained by looking at the complete handling diagram.

6. CONCLUSIONS

Overall, semi-trailer steering systems were found to generally improve the low-speed performance of articulated vehicles. The main advantages of such systems are that they substantially reduce the vehicle's swept path width and lateral tyre forces. As a result semi-trailers fitted with these systems are generally more manoeuvrable, able to access more of the road network, have lower tyre wear and do less damage to the road surface whilst turning compared to conventional fixed-axle semi-trailers. The only disadvantage found at low-speed was an increase in the amount of tail swing.

The advantages gained at low-speed by semi-trailer steering systems are partially offset by poorer high-speed performance. Such systems generally increased rearward amplification and transient offtracking, which in the worst case could lead to high-speed stability problems. These areas can be improved, however, by locking the steering mechanisms at high speeds.

The semi-trailer steering systems were found to have little effect on the remaining performance measures. These included the static rollover threshold, load transfer ratio and high-speed steady state offtracking. With the exception of the self-steering system, trailer steering also had little effect on the handling of the vehicle.

Comparatively, no semi-trailer steering system was found to perform better than the others in all areas. The pivotal bogie system generally showed the best low-speed performance due to its ability to steer all trailer axles. However, it also had the worst high-speed performance and tail swing. The self-steer system performed best at high-speeds but did not improve load speed performance by such an extent and greatly influenced vehicle handling. The command steer system generally performed at a level between the other two, showing that it may provide the optimal balance between high and low speed performance.

It is important to note that the comparison has been made using the same standard semi-trailer configuration, based on the type of articulated vehicle currently allowed under U.K legislation. As a result parameters such as the trailer length, axle group location and weight distribution were not varied. It is likely that changing these parameters, so that they are optimal for the type of steering system employed, could further improve both high and low speed performance and lead to safer and more economic articulated vehicles. The introduction of performance based vehicle legislation, such as that being proposed in Australia, would allow steering trailers optimised in this way to operate.

6.1 Future Work

The work described in this paper forms the first stage of a larger project concerned with semi-trailer steering systems. The next phase of the project is to validate and calibrate the model, as outlined in Section 3. Further field trials will then be conducted to verify that the main claims made in this paper are correct.

As described above, it may be possible to further improve the performance of steering trailers by changing parameters such as the trailer length, axle group location and weight distribution. More work is required to determine what effect these parameters have and to what extent the performance of the vehicle can be optimised.

The use of active steering systems may also improve performance, especially in those areas that are degraded or not affected by the current passive semi-trailer steering systems. One strategy may involve steering the trailer axles in the same direction as the tractor steer axle at high-speed, similar to four-wheel steering automobiles. It is proposed to investigate various active strategies and verify their performance on a full sized vehicle in the near future.

It is important to note that this study has just compared the performance of the various semi-trailer steering systems. A number of other factors need to be considered when deciding which system is 'best' for a certain application. Such factors include weight, capital and operating costs, durability and adaptability. Whilst this is outside the scope of this project it may be the subject of future work.

7. ACKNOWLEDGEMENTS

The author would like to acknowledge the following organisations and individuals whose help with this project has been invaluable:

- The Cambridge Vehicle Dynamics Consortium (www.cvdc.org)
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9. TABLES AND FIGURES

Table 1- Summary of Performance Measures

	PERFORMANCE MEASURE	SHORT NAME	MANOEUVER	DEFINITION	PROPOSED PERFORMANCE LEVEL	NRTC PERFORMANCE LEVEL
1*	Low-Speed Corner Swept Path Width	SPW ₉₀	Centre of steer axle to follow path on straight approaches to a 11.25 m radius 90° circular arc. Vehicle speed is 10 km/h.	Maximum width of the path swept by the vehicle as it traverses the corner.	5 m for local roads.. 7.4 m for arterial roads	
2*	Tail Swing	TS	Same as above	Maximum lateral distance the rear of the vehicle travels outside of the front wheel path.	< 0.35 m on both approaches.	
3*	Steer Tyre Friction Demand	STFD	Same as above	$F_{y1} / (F_{y1} + F_{peak}) \times 100$	< 80% of the maximum available tyre/road friction.	
4*	Low-Speed Corner Lateral Tyre Force	LTF ₉₀	Same as above	Driver: $\text{Max}(F_{y2})$ Trailer: $\text{Max}(F_{y3} , F_{y4}, F_{y5})$	-	
5	Low-Speed Circle Swept Path Width	SPW ₃₆₀	Centre of steer axle to follow path on straight approaches to a 11.25 m radius 360° circular arc. Vehicle speed is 10 km/h.	Maximum width of the path swept by the vehicle as it traverses the circle.	-	
6	Low-Speed Circle Lateral Tyre Force	LTF ₃₆₀	Same as above	Driver: $\text{Max}(F_{y2})$ Trailer: $\text{Max}(F_{y3} , F_{y4}, F_{y5})$	-	
7*	Static Rollover Threshold	SRT	Centre of steer axle to follow a 100 m radius circular path, test speed slowly increased from 60 km/h until rollover occurs	$\text{Max}(V^2 / Rg)$	> 0.35 g	
8*	Rearward Amplification	RA	Prescribed-path lane-change manoeuvre as defined in SAE J2179 (Society of Automotive Engineers, 1993b). Vehicle speed is 88 km/h.	$\text{Max}(A_{y \text{ trailer}} / A_{y \text{ front axle}})$	< 5.7 x SRT	
9*	High-Speed Transient Offtracking	TO	Same as above	Maximum lateral distance the rear axle tracks outside the path of the front axle	< 0.8 m	
10*	Load Transfer Ratio	LTR	Same as above	$\text{abs}(\text{sum}(F_{y1} - F_{y7}) / \text{sum}(F_{y1} + F_{y7}))$ for the trailer unit	< 0.6	
11*	High-Speed Steady State Offtracking	SSO	Centre of steer axle to follow a 393m radius circular path, test speed 100km/h.	Maximum lateral distance the rear axle tracks outside the path of the front axle	< 0.5 m	
12*	Handling Quality (Note: Still Under Review)	HQ	As specified in El-Gindy, Woodroffe and White (1991). Vehicle speed of 100 km/h.	Defined by 3 points on handling curve: 1 st point @ 0.15 g 2 nd point @ over/understeer transition 3 rd point @ 0.3 g	Required values at 3 points: 1 st point: $0.5 < K_u < 2 \text{ deg/g}$ 2 nd point: $A_u > 0.2g$ 3 rd point: $K_u > K_{ur}$	

Note: * denotes performance measures obtained from [13] and [14].

Table 2- Absolute Performance Measures

	PERFORMANCE MEASURE	FIXED-AXLE TRAILER	SELF-STEER SYSTEM	COMMAND STEER SYSTEM	PIVOTAL BOGIE SYSTEM	PROPOSED NRTC LEVEL
	Low-Speed Corner Swept Path Width (SPW _{low})	5.2 m	4.9 m	4.6 m	3.8 m	<5 m
	Tail Swing (TS)	Ent. 0.08 m Exit none	Ent. 0.11 m Exit none	Ent. 0.27 m Exit none	Ent. 0.68 m Exit 0.39 m	Ent. < 0.35 m Exit < 0.35 m
	Steer Tyre Friction Demand (STFD)	20.4 %	18.4 %	15.1 %	14.3 %	< 80 %
	Low-Speed Corner Lateral Tyre Force (LTF _{low})	Drive 8.4 kN Trailer 18.5 kN	Drive 6.4 kN Trailer 14.4 kN	Drive 3.4 kN Trailer 8.8 kN	Drive 2.7 kN Trailer 9.1 kN	-
	Low-Speed Circle Swept Path Width (SPW _{low})	6.3 m	5.9 m	5.0 m	3.8 m	<5 m
	Low-Speed Circle Lateral Tyre Force (LTF _{low})	Drive 8.5 kN Trailer 22.0 kN	Drive 6.6 kN Trailer 18.0 kN	Drive 3.4 kN Trailer 12.3 kN	Drive 3.0 kN Trailer 10.4 kN	-
	Static Rollover Threshold (SRT)	0.42 g	0.42 g	0.42 g	0.42 g	> 0.35 g
	Rearward Amplification (RA)	1.07	1.07	1.24	1.38	< 2.4
	High-Speed Transient Offtracking (TO)	0.26 m	0.34 m	0.31 m	0.53 m	< 0.8 m
0	Load Transfer Ratio (LTR)	0.30	0.30	0.34	0.39	< 0.6
1	High-Speed Steady State Offtracking (SSO)	0.53 m	0.60 m	0.54 m	0.60 m	< 0.5 m
2	Handling Quality (HQ)	Pass/Pass/Pass	Pass/Pass/Pass	Pass/Pass/Pass	Pass/Pass/Pass	-

Note: Shaded cells show values that fail the proposed NRTC criterion from [13] and [14].

Table 3- Normalized Results Relative to Fixed Trailer

	PERFORMANCE MEASURE	FIXED-AXLE TRAILER	SELF-STEER SYSTEM	COMMAND STEER SYSTEM	PIVOTAL ROGIE SYSTEM
	Low-Speed Corner Swept Path Width (SPW ₉₀)	1.00	0.94	0.89	0.73
	Tail Swing (TS)	Ent. 1.00 Exit none	Ent. 1.38 Exit none	Ent. 3.38 Exit none	Ent. 8.50 Exit >10
	Steer Tyre Friction Demand (STFD)	1.00	0.90	0.74	0.70
	Low-Speed Corner Lateral Tyre Force (LTF ₉₀)	Drive 1.00 Trailer 1.00	Drive 0.76 Trailer 0.94	Drive 0.40 Trailer 0.48	Drive 0.32 Trailer 0.49
	Low-Speed Circle Swept Path Width (SPW ₃₆₀)	1.00	0.94	0.79	0.60
	Low-Speed Circle Lateral Tyre Force (LTF ₃₆₀)	Drive 1.00 Trailer 1.00	Drive 0.78 Trailer 0.82	Drive 0.40 Trailer 0.56	Drive 0.35 Trailer 0.47
	Static Rollover Threshold (SRT)	1.00	1.00	1.00	1.00
	Rearward Amplification (RA)	1.00	1.00	1.16	1.29
	High-Speed Transient Offtracking (TO)	1.00	1.31	1.19	2.04
0	Load Transfer Ratio (LTR)	1.00	1.00	1.13	1.30
1	High-Speed Steady State Offtracking (SSO)	1.00	1.13	1.02	1.13
2	Handling Quality (HQ)	-	-	-	-

Note: Shaded cells show values that are worse than the fixed-axle trailer

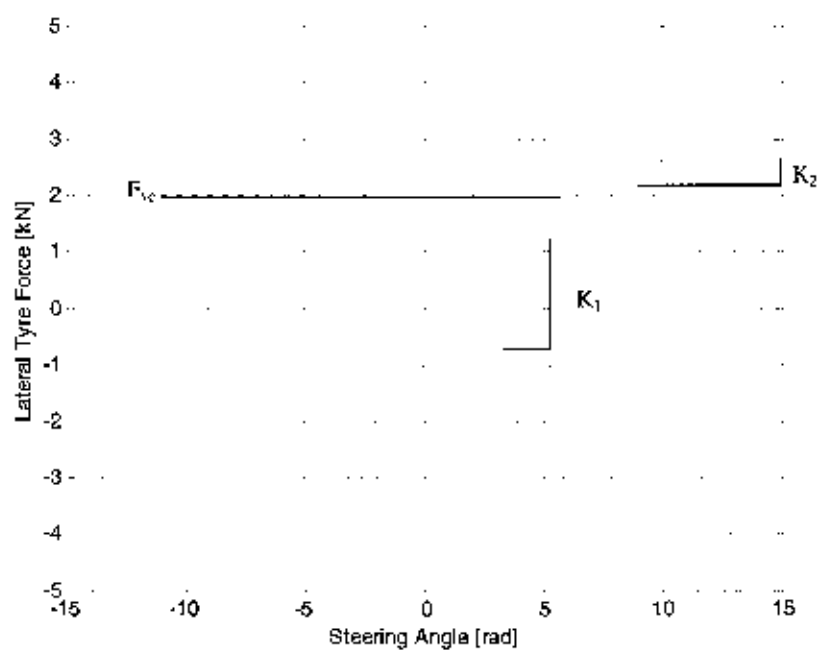


Figure 1 – Self Steering Axle Lateral Tyre Force vs Steer Angle Relationship [2]

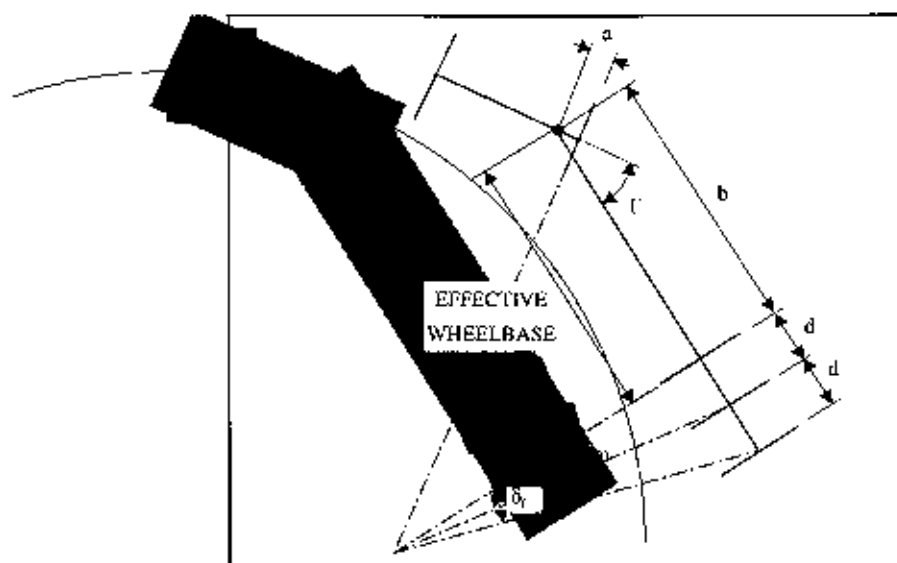


Figure 2— Command Steer Low Speed Geometry

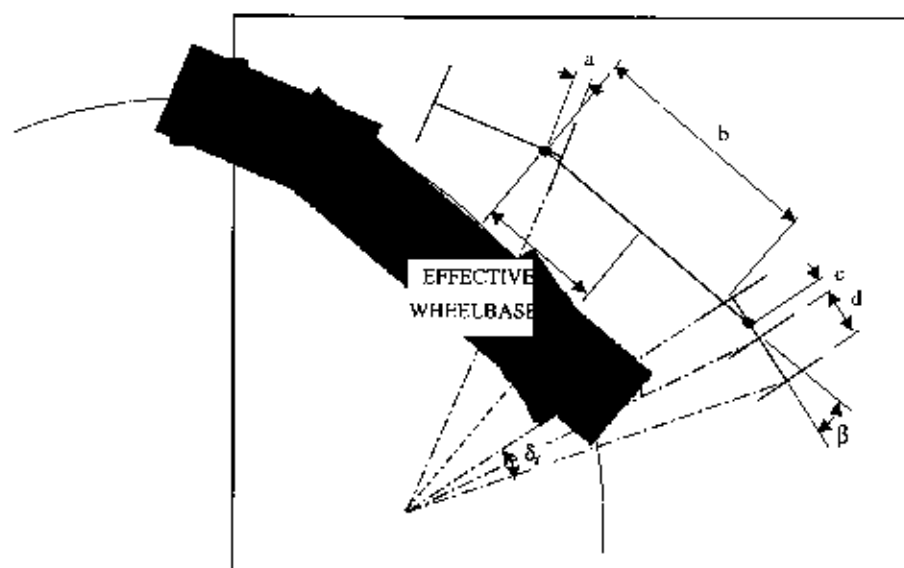


Figure 3—Pivotal Bogie Low Speed Geometry

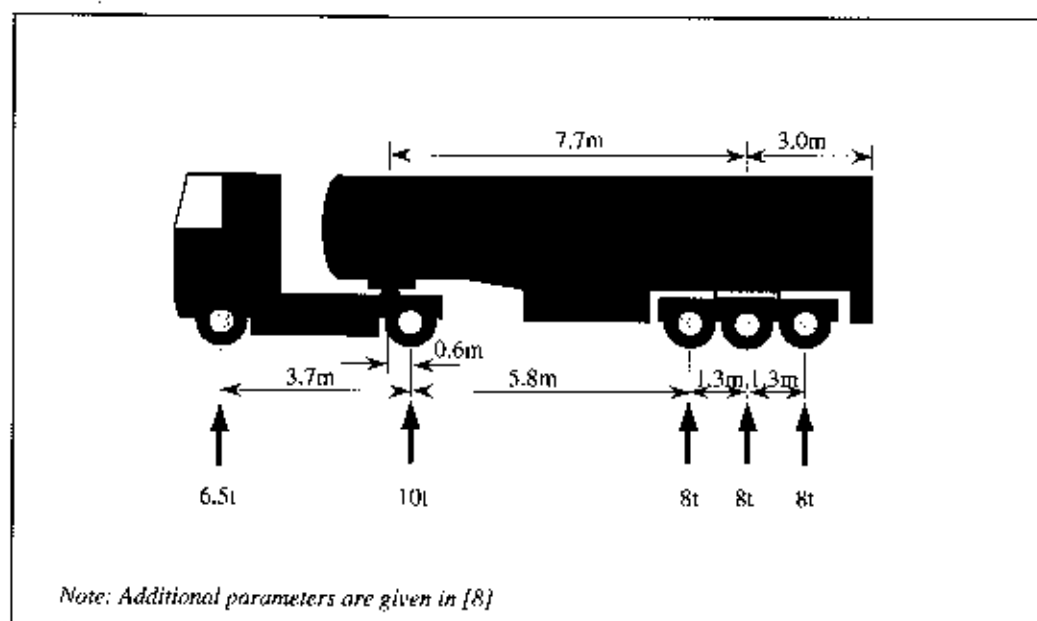


Figure 4—Simulation Model Weights and Dimensions

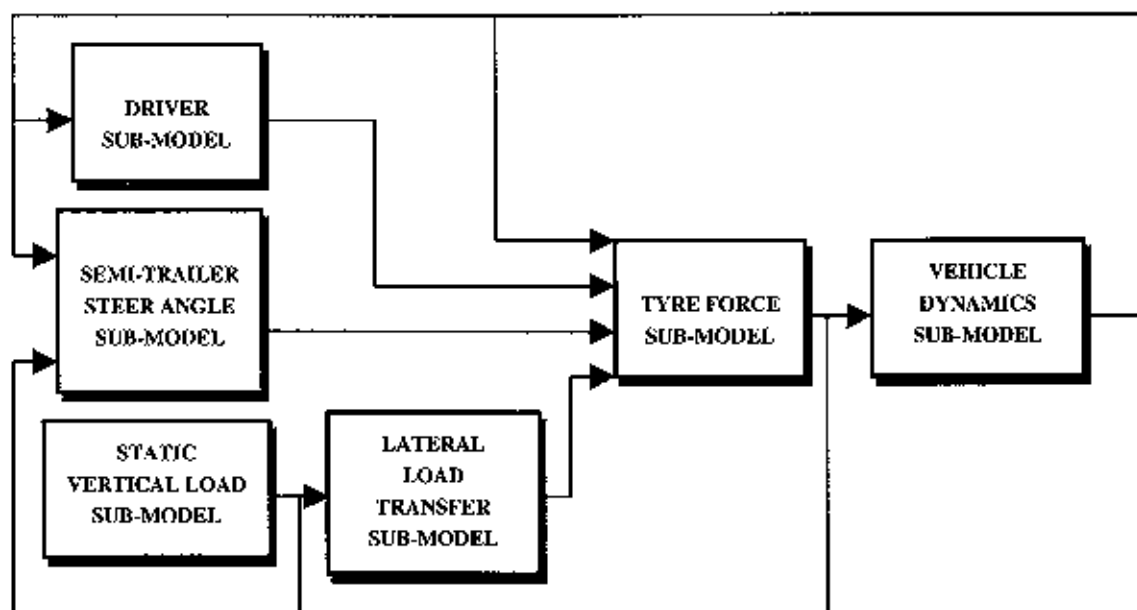


Figure 5- Simulation Program Block Diagram

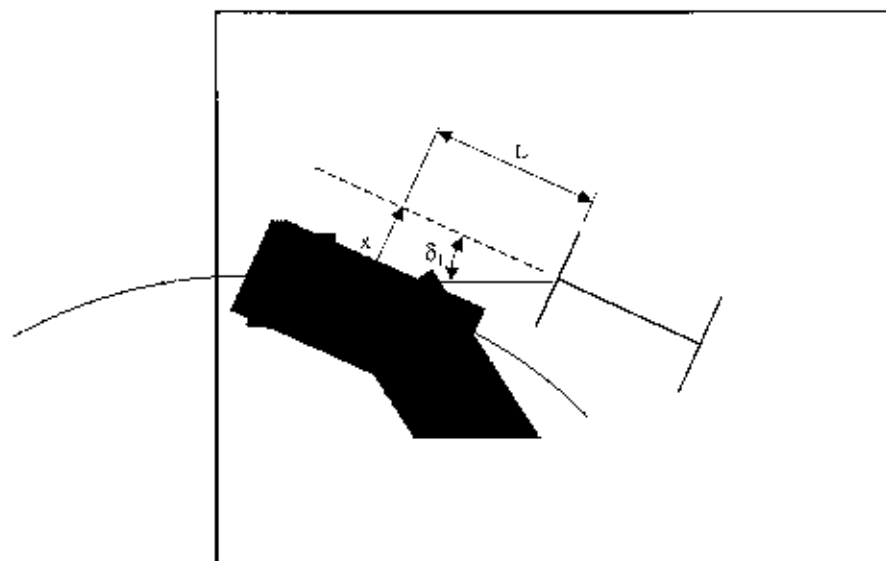
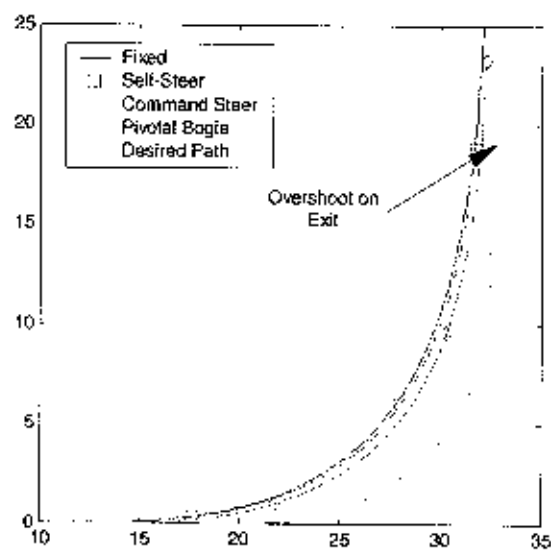
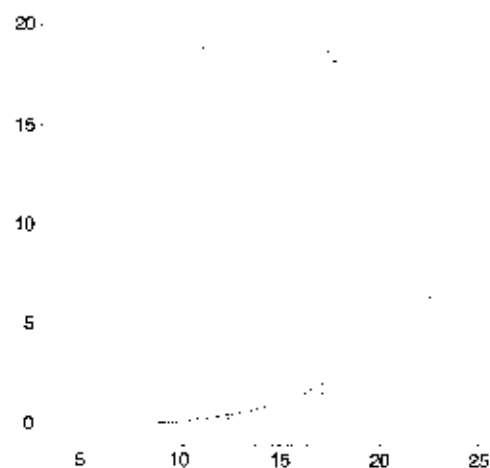


Figure 6- Driver Sub-Model



7a - Low-Speed Corner



7b - Low-Speed Circle

Figure 7 - Path Followed by Trailer Group in Low Speed Manoeuvres

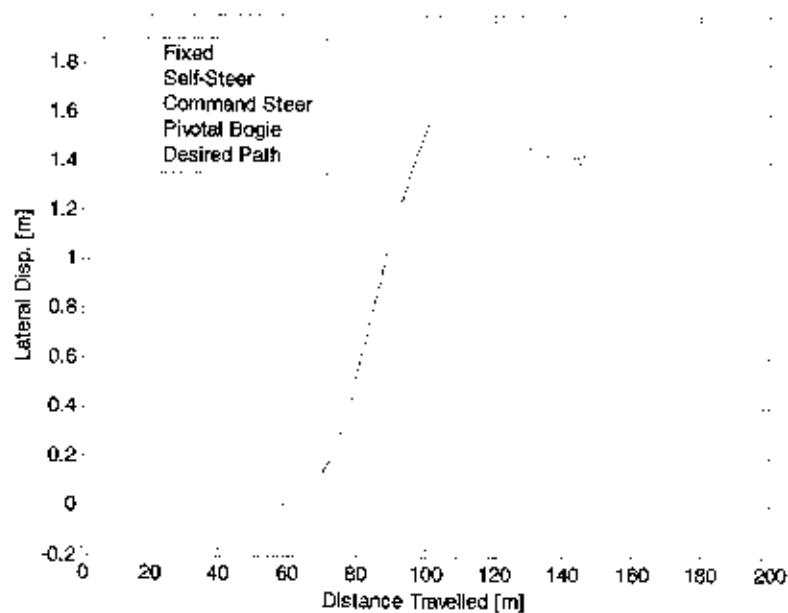


Figure 8 - Path Followed by Trailer Group in a High-Speed Lane Change

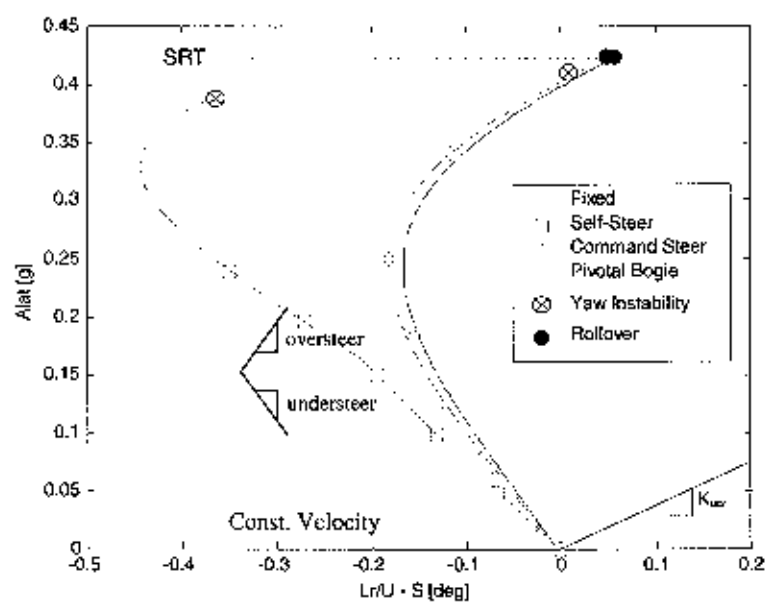


Figure 9 –Handling Diagram

INTRODUCING LONGER AND OR HEAVIER VEHICLE COMBINATIONS (LZV'S) IN THE NETHERLANDS, A LONG AND HEAVY PROCESS

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ABSTRACT

Introducing longer and/or heavier vehicle combinations in a densely populated country like the Netherlands showed to be a long and difficult task, as the political decisionmaking was troubled by aspects like safety, local politics and intermodal competition. It took some seven years before the first combination was on the road and running. Still there are only four combinations participating and operating. Now the first conclusions can be drawn. First it shows that due to the long list of requirements of participation, most interested companies do not apply or don't take the risk of participation. Second: in viewing just the use of containerised loads, heavier vehicle combinations provide relatively more efficiency-improvement in this test than longer ones. Next to that, already during the process of getting the combinations on the road, it was noticed that the consequences of introducing longer and or heavier vehicle combinations reach out much further than just the road and the driver, but to the complete logistic chain.

INTRODUCTION

The Dutch government started a project to get information regarding the problems and benefits in practice of longer and/or heavier vehicle combinations in the Netherlands. It took 7 years before the first combination was up and running on the streets and highways in the Netherlands. Due to political, emotional, legal and other barriers the project has been faced with a lot of setbacks and delay.

The Dutch government started thinking about the project due to a number of trends:

Rapid growth in road transport, a very densely populated country, an ever-increasing congestion problem and a fast growing largest port of the world. There was at first a request from the transport branch, which was criticized for being not realistic. It was followed by a request for shippers, transporters and the port of Rotterdam. This made the ministry, facing increasing problems, aware of the potential influence of this solution. There was a lot to investigate, though.

Items that were really demanding attention were safety, road wear, European Union legal barriers, international vehicle regulation and the comparison to other modalities as short sea shipping, rail and inland shipping.

The studies took about a year and some results to be mentioned were:

Longer vehicles don't need different traffic lights settings due to their long length;

Longer vehicles sometimes have trouble crossing rail-roads if they have fully automated traffic barriers;

The steepest railroad crossings provide probably extra trouble

If a combination has more than two turning points they become less stable when running in daily traffic and turning corners;

The heavier vehicles need an additional amount of power to keep up with traffic and to prevent traffic jams;

For safety reasons it is important to have additional attention for both vehicle and driver to reach a higher level of both active and passive safety.

After a period of two years the studies resulted in the decision that a pilotproject was worth trying and an extra set of requirements was formulated. A format for testing the vehicle combinations in the Netherlands was made up.

This comprised testing for a short while in a limited number on specific roads. A lot of additional demands were formulated and transporters were requested to apply. Then in 1997 the Parliament showed their concern for environment and the competition to other modalities by requesting the Ministry to add additional demands resulting in a limitation for a driving range of 50 kilometres and the demand that it should be allowed only to and from an intermodal terminal.

After the elections the aspect of safety was questioned again by the present Minister and a period of about a year passed by allowing the SWOV research institute to perform an additional study on safety. In the meantime transporters were withdrawing their application.

In spring 1999 the Minister decided the project could continue, if additional requirements were added to the already existing list.

After formulating another additional set of requirements (see appendix I for almost all of them,) it was decided to allow three categories: "Heavier" (60 tons, not exceeding permitted length), "Longer" (not exceeding permitted weights but allowing length to be 25, 25 meters) and longer and heavier (60 tons, length 25,25 meters).

Next step was to allow transporters to apply. This resulted in 25 applications, of which were 21 to be called realistic. After selecting the maximum of 15 transporters (the limited allowed number) in March 2000 we could start with the preparation of the real field test.

Aims of this test are multiple: we have had a lot of investigations done regarding competition to other modalities, environmental benefit, safety, et cetera, but we wanted to know the practical possibilities and bottlenecks of this kind of combinations in the Netherlands. Next to it we wanted to know the pros and cons for further policymaking regarding longer and/or heavier vehicle combinations and the effect in the logistic chain of these combinations.

The test was to be monitored very closely, related to the opposition we faced by a number of groups: the local authorities (afraid for their roads and safety), the government and Parliament (safety and environment, competition to other modalities), European Commission and other EU member states (competition) and organisations of road users (safety).

Aspects included in the monitoring are fuel efficiency, safety, operating efficiency, contribution to lower congestion, bottlenecks in operations and reactions of both drivers and operators. One aspect taken into account is the perception of the vehicle combinations by different parties. Therefore monitoring is not just focused on the "hard" indicators as fuel efficiency, but on the less quantitative aspects as felt safety, too.

Before the first fuel gauges were installed, it took some months after starting the combinations. Due to both problems on the side of exemptions (local and regional; governments) and by poor work on the side of the truck constructors and producers of semi-trailers and trailers, most of the applicants took some 7-14 months to get their combinations on the road. Even after 18 months after the first truck started we don't have results over a year of driving.

In the meantime the number of participants is down from 15 to 7, of which 4 are really on the road, one is on the edge of starting and two are reconsidering participation. The remaining participants have withdrawn themselves or are rejected by the selection committee.

Due to the limited number of participants the possibility of generating overall conclusions is very hard, and it looks that the monitoring is heading toward a group of case-studies instead of allowing general conclusions.

There are still some conclusions to be drawn, anyway.

First results of the monitoring show that the use of these combinations has consequences for all parties in the logistic chain and their partners: both shippers as receivers, both transporters as road managers, logistic managers, planners and drivers have to adjust to the larger scale. This is especially true for the "longer and heavier"

combinations, which demand for different planning techniques, more space for turning, special places for container loading, adjustments in software, more loading docks et cetera.

As of February 2002 the resulting benefits on fuel efficiency, congestion, et cetera are not yet clear, but trying to get the project running has provided us with a lot of knowledge regarding the vehicle combinations, the interaction with infrastructure, traffic and logistic flows.

First conclusions:

- Heavier vehicle combinations provide relatively more efficiency-improvement in this test than longer ones
- Driving on the non-expressway could lead to road blockings in heavy traffic;
- Combinations are not just used for shuttling to and from intermodal terminals;
- The consequences of introducing longer and/or heavier vehicle combinations reach out much further than just the road and the driver, but in the complete logistic chain;
- Introducing longer and/or heavier vehicle combinations demands for more close study before allowing them in densely populated areas with a highly developed logistic infrastructure;
- Allowing transporters to operate longer and/or heavier vehicle combinations under a lot of additional demands diminishes the participation. Reconsidering the present additional demands for participants will be necessary if the test is called a success and a next phase will be decided for.

APPENDIX 1

Concept-conditions for the tests with longer and/or heavier lorries.

1. General conditions

In the period starting 1 December 2000 until 1 June 2003 a limited number of tests will be taken with combinations that are:

- A. Heavier than allowed by the Dutch law, being a Gross Combination Weight of 50.000 kilograms
- B. Longer than allowed by the Dutch law, being 18,75 meter
- C. Combinations being longer and heavier than allowed by the Dutch law, meaning longer than 18,75 meter and heavier than 50.000 kilograms

A maximum number of 15 participants can participate, each of them with only one combination on a maximum of three destinations. The number of participants for category A is limited by 4.

On weekdays between 6.00 am and 10.00 am and when road conditions are dangerous, the combinations are not allowed to be operated.

Combinations carrying dangerous goods or tank containers are excluded from the tests

A firm participating in the tests is obligated to fill in a form regarding the driven trips, which includes:

1. The driven hauls of that day;
2. the carried containers or swap bodies;
3. the weight of the carried goods;
4. the fuel consumption
5. The experienced bottle-necks and special experiences

The participating firm is obligated to send in the forms at least every month to the ministry of transport. The firms are obligated to provide comparable figures on the fuel consumption of the present situation of carrying.

2. Conditions for the routes

1. The hauls have to be part of intermodal transport and therefore has a starting point or destination a terminal for combined transport.
2. The hauls have a maximum length of 50 km. For the Rotterdam harbour area, the Waalhaven will be the point of orientation;
3. The routes consist mainly of roads with a separate infrastructure for cyclists and other slow traffic. A maximum of 5 km of the roads of every route does not have a separate infrastructure.
4. No living or shopping area may be part of the route;
5. The connecting point of an expressway has to have a hard shoulder of at least 250 m.
6. In the routes for the combinations as stated by I.B or I.C, a railroad crossing is not allowed when trains are allowed to exceed the speed of 40 km/h.
7. In the routes for the combinations as stated by I.B or I.C., an overtaking prohibition is valid for all vehicles allowed to drive faster than 50 km/h.

3. Obligated characteristics of the combinations

- A. Combinations, described at I.A, can have a maximum combination weight of 60.000 kg.
- B. Combinations, described at I.B, can have a maximum combination length of 25,25 m
- C. Combinations, described at I.C, can have a maximum combination weight of 60.000 kg and a maximum length of 25,25 m

Each combination has to be tested by the Dutch government on compliance with the law and the additional technical requirements as stated below:

1. All vehicles being part of the combination have an ABS.
2. The combination is stable in all driving conditions
3. The area of view on the right side of the combination has a width of at least 5 m, while the combination is turning to the right in a turn with a 14,5 m maximum radius.
4. The combination has a maximum of two turning points
5. The side of the combination is covered with closed side panels, in compliance with 89/227
6. The tractor is provided with a front run under protection as described in ECE Regulation 93
7. The axles are provided with anti splash and spray devices, conform Guideline 89/297
8. The engine power is at least 5 kW per ton of the maximum Gross Combination Weight
9. The axle pressure does not exceed the maximum allowed on other combinations
10. The combination is provided a side marking conform ECE Regulation 104
11. The turning radius is at a maximum of 14,5 m outside, 8 m inside radius.

4 Demands for the driver

The driver has to have a valid drivers license, as well as all the papers and licenses as prescribed by the law.


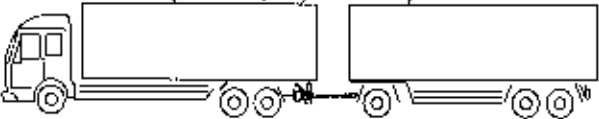

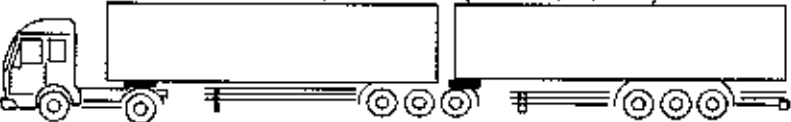

In addition to these demands:

1. The driver has at least 5 years of driving experience with lorries with a minimum gross vehicle weight;
2. The driver was not involved in any accident
3. The driver has successfully completed the certification for the driving of the longer and heavier lorries. Special attention is given to the awareness to other road users and the mentality of the driver.

All applicants participating in the tests have to pay all the costs that result from the application and the resulting actions, like the costs for the exemptions of every community.

TABLES & FIGURES

ALLOWED COMBINATIONS

Category Heavier	
H1	Tractor + semitrailer (60 tons; 16,50 m) 
H2	Truck + trailer (60 tons; 18,75 meter) 
Category Longer	
L1	Tractor + semitrailer + midaxletrailer (60 tons; 25,25 m) 
L2	Tractor and two semitrailers, "B-double" (60 tons; 25,25 m) 
L3	Truck + dolly + semitrailer (60 tons; 25,25 m) 

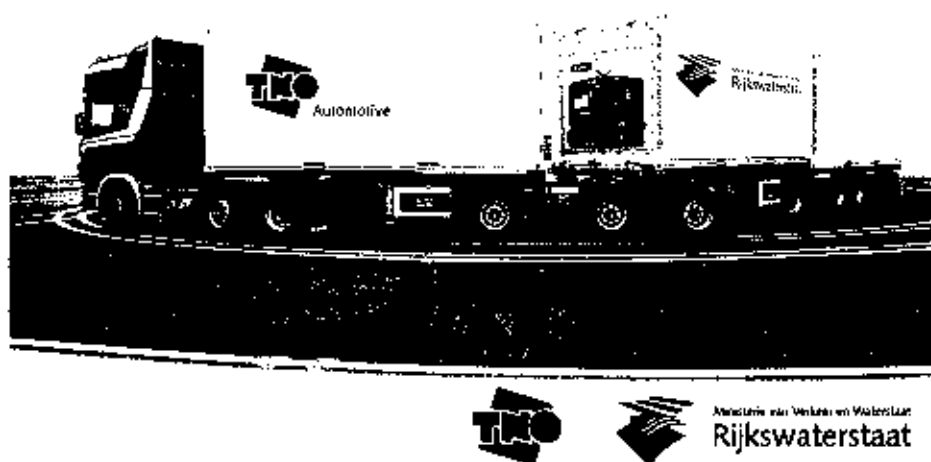


Figure 1 – The most inventive combination of this project (a special form of B-double), the combination of Overdevest, was used for the logo of this Congress

ON DEVELOPMENT OF THE SUPER-SINGLE DRIVE (GMD) TYRE

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ABSTRACT

Technical ingredients for replacing dual mounting of truck and bus tyres with the single wide base tyre (GMD: GREATEC Mega Drive) are presented. Aim of developing the technology is underlined by directing dramatic enhancement of tyre durability to advantages on economy, environment and vehicle utility: reduced rolling resistance to benefit energy efficiency, less tyre weight to save limited resources on earth, less tyre room occupied to enable new vehicle designs. Technical issues in the tread area were uneven growth due to wide sectional configuration and large stress concentration. While in the bead area, large strain induced by considerable deflection should effectively be minimized. To overcome the difficulties innovative structures were proposed. The results showed outstanding performance under critical testing conditions.

INTRODUCTION

As recognized widely in global industrial societies, proactive technology should be adopted on ecological and economical issues when developing new products. Heavy commercial vehicles such as trucks, trailers and buses have already started including the idea by aiming at: less energy consumption and less exhaust emission leading ultimately to hybrid and/or fuel cell engine systems, higher efficiency in transporting cargos. One of the crucial components of the vehicle, the tyre, should also respond to the requirements of the social needs; effort to fulfil simultaneously ecology- and economy-consciousness and high quality of tyre performance is needed. More specifically, lower rolling resistance, less weight (less amount of raw material used), and lower tyre noise and so on, should be pursued without sacrificing fundamental performance such as durability, tread wear and manoeuvrability.

No doubt considerable effort has been made to date in tyre design; economy-efficient and environmentally-correct materials are mostly used, various tyre designs of carcass configuration, structure and tread pattern, which effectively reduce rolling resistance and achieve weight reduction, are employed.

Now as a remaining but a very hopeful approach to meet the needs, we focus on the mounting system of dual tyres on drive and non-steer axles of the heavy vehicles, and intend to replace these tyres with single wide-base tyre of less volume, see Figure 1. This concept, named GREATEC Mega Drive (GMD), assumes considerable enhancement of durability for the single tyre. Following this strategy, if concentration of stress and strain can successfully be suppressed, lower rolling resistance can be obtained because the GMD uses less amount of hysteretic material (rubber).

On the premise that the GMD clears all requirements of the tyre performance, the concept could motivate innovative and flexible vehicle design. For instance, a wider room on the axle might enable installation of a high-tech engine/braking system, comfortable passenger cabin and large cargo space (see Fig.2). Even when designing a vehicle with narrower width, e.g. luxurious petit-bus, ample cabin space could be retained. Of course, the use of wider wheel indicates the possibility of equipping a wheel-in motor system.

This paper describes technical effort to search advanced structures for the wide base tyre with special emphasis on durability performance, on which the GMD concept is grounded.

TECHNICAL TASKS FOR DEVELOPING THE SUPER-SINGLE (GMD) TYRE

The GMD tyre has less volume enclosed by the tyre outer surface and the rim base than the sum of the dual tyres. It means that the nearly identical vertical load, the driving torque and the side force act on the smaller envelope causing larger deformation. To meet the nearly doubled loading, the air pressure could be raised. For example of the GMD concept, which replaces the dual 315/70R22.5 tyres with the single 495/45R22.5 GMD tyre, the vertical

load and the air pressure are specified in the European Tyre and Rim Technical Organisation (ETRTO) standards manual respectively as 56.84kN and 900kPa compared to 30.87kN and 850kPa of 315/70R22.5. Though the load is not exactly doubled in the standards, the GMD tyre is subjected to 84 percent more load than the single 315/70R22.5 with only 6 percent increase of the air pressure and 57 percent increase of the tyre width (overall diameter is virtually equal in the both tyres). In fact, the tyre deflection of 495/45R22.5 is larger than its counterpart by 51 percent. Thus, the ability to bear the increased load and the increased air pressure is demanded.

Tasks for belt package structure

In the tread area of the tyre, large deformation and heat generation are the main cause of belt failure and uneven growth. Conversely, the non-uniform growth of the tread is a cause of the heat generation and the stress concentration leading to the early failure, and a cause of the uneven wear as well. The growth is due to the pressurisation of the tyre and the creep behaviour under various service conditions.

In mechanical theory of the radial tyre, if all fibre-reinforced members are assumed to be inextensible, the belt package produces the tensional force by pressurisation in the equatorial (or circumferential) direction: the force is roughly proportional to the coordinate difference in the radial direction between the equilibrated carcass (or ply) without the belt and the original coordinate of the belt [1]. Therefore, the tyre with lower aspect ratio (tyre width divided by tyre height in the meridian section plane) has larger belt force produced by pressurisation and hence larger shear strain between belt layers than the tyre with higher aspect ratio. The GMD is obviously bound for the very low aspect ratio. In our knowledge of engineering practice, the mechanism of belt deformation as consulted with the developed belt forces is indeed of great significance [2,3].

The conventional belt package (Fig. 3) is characterised as the structure of multiple layers of composites: each layer is made of uni-directionally oriented yarned steel cords and coating rubber in between, the steel cord forms the angle of 15 to 60 degrees to the equator line, a couple of adjacent layers are crossed each other. Such a composite, known as the Fibre Reinforced Rubber, excels in deformability and damping characteristics especially when impact forces are transmitted from the ground. However, as the belt force develops by lowering the aspect ratio, the conventional belt indicates the unbalance of the growth across the tread (Fig.4). Consequently, a high concentration of the inter-laminar shear strain is generated at the belt edge.

To solve the problem, the rigidity in the circumferential direction needs to be increased dramatically. After a long course of research we came to the conclusion that the wavy steel cord coated by polymer (the WAVED), which has a number of regulated waves in the plane of the belt layer with its wave length λ and wave amplitude a , is suited and performs excellent. The WAVED belt layer behaves like a nonlinear spring. First, the concentration of shear strain due to the rigidity gap between adjacent conventional layers can effectively be relaxed by its spring-like movement. Further, the tread growth can be controlled directly and smoothly by setting appropriate λ and a . This idea is very useful and is protected by a series of international patents.

The typical WAVED belt structure currently used is a combination of the WAVED layers and the conventional layers (see Fig.3), where the belt tension in the WAVED layers is nearly comparable to that in the conventional layers. The current WAVED belt performs better than its counterpart (Fig.4 and Fig.5). However, as we go beyond the aspect ratio lower than 0.60, the current WAVED belt should be revised to achieve the uniform growth and the durability life equivalent to the aspect ratio 0.70.

Tasks for bead structure

The GMD strategy requires the bead also of a considerable progress. From the conventional dual tyres to the GMD, the load per tyre increases larger than the increase of the contact area with the ground and the increase of the air pressure. Therefore, the GMD tyre deflects 51 percent more, and a very large deformation develops in the sidewall to the bead area. As the deformation in the right-under-the-load section shows that the outer surface of the bead is compressed hard against the rim flange, a large shear strain builds up in the vicinity of the failure point.

On the other hand, fractography analysis concluded that circumferential shear is another cause of failure. This behaviour is apparent in the bead area corresponding to the edge of the contact patch (with the ground).

The conventional bead (Fig. 6, right) is the structure characterised in that: the radial carcass (ply) is turned up past the bead core (a wire bundle oriented to the circumferential direction), a couple of FRR layers and rubbers of high modulus are deployed to suppress the strains. Figure 6 shows that the durability deteriorates as the aspect ratio goes lower. The drastic drop below the aspect ratio 0.50 can be interpreted as firstly that the ability to bear the load (the load index) is actually raised in the ETRTO standards, consequently the strains relevant to failure

increase. Secondly, the circumferential shear deformation develops rapidly: the tyre sidewall tends to shear as its rigidity (effect of tension included) decreases relative to the belt, clearly large belt force builds up with the low aspect ratio. More straightforwardly for the GMD concept, the halved number of the sidewall causes serious difficulty. In general, the effect of the circumferential shear is further typified when the fore-and-aft forces act on the tyre.

Thus, the GMD strategy demands approximately the three times enhancement of the durability (Fig. 6).

RESEARCH METHODOLOGY

Research was conducted so as to get insight into important relations between: fractural behaviour and relevant strains, tyre deformation and strains, crack propagation and strains and temperature. The first relation was investigated by the failure analysis of the tested tyres, and synchronously a number of specimen tests were carried out in the laboratory to identify the failure mode and the influence of the stress and the temperature [4,5]. The strain was calculated using the finite element method (FEM), in which the nonlinear time-dependent analysis [6,7] with the scrupulous FE-model (Fig. 7) was conducted. Modelling of the WAVED belt was particularly elaborated. Temperature distribution was also assessed by the stress and the strain field using the FEM and was compared with experimental data. The second was carried out mainly by using the FEM, and was supplemented by experiments to prove hypotheses. The last owed to numerous specimen tests. Effects of material degradation due to heat build-up and oxidization were also considered.

For the purpose of verification, ample amounts of laboratory tests, drum tests and field tests of the GMD tyres were conducted.

ESTABLISHMENT OF THE GMD-TECHNOLOGY

As a consequence of the research, the correlation between the crack growth rate and the dominant strain for every failure mode, and the correlation between the crack growth rate and the temperature were found out quantitatively. In ensuing optimisation, the following structures were proposed.

New Waved belt

Figure 8 typically shows the family of the belt package (the New WAVED belt) proposed for the GMD tyre. Main points of alteration from the current WAVED belt are: the waved layers deployed on the inner side of the tyre, the widened waved belt, and the optimised rigidity of the conventional belt layers. The first and the third alteration are because the inter-laminar shear strain between the conventional belt layers reduces substantially. The second obviously suppresses the uneven growth due to pressurisation and the creep of the tyre. This was proven to be effective especially when the aspect ratio is below 0.70. The third is also directed for taking advantages in manoeuvrability and resistance to foreign object damage (FOD).

In Fig. 9, the Von Mises stress is depicted to show the tendency to creep. The New WAVED belt yields remarkable reduction of the stress, which can retard crack initiation and propagation.

Turn-In-Ply (TIP) bead

One candidate is obtained by laying reinforcements (FRR) correctly onto the conventional turn-up-carcase end. In fact, the reinforcements have significant effect on reducing the circumferential shear deformation and on dispersing the stress concentration right under the load.

Another is achieved by turning in the carcass around the bead core (Turn-In-Ply bead: TIP, see Fig.10). The rational is to move the expected point of failure (carcase end) away from the area of high gradient of deformation. Actually, the conventional bead suffers this point, hence needs protective layers additionally. The TIP bead excels in the moderate strain field, which can clearly be observed in Fig. 11. The figure schematically shows the principal strain under pressurised and loaded condition. The tyre is assumed to be exposed to the thermal aging, the process of keeping pressurised tyre in the environment of a high temperature, say approximately six days in a chamber of 80 degrees Celsius. Therefore, the creep behaviour is mainly simulated.

A significant reduction of the strain is observed in the proposed structures.

RESULTS AND DISCUSSIONS

The GMD concept was verified in the following for the 435/45R22.5 and the 495/45R22.5 tyres, which are the replacement of the dual mount of 275/70R22.5 and 315/70R22.5 respectively.

For the growth due to pressurisation, the GMD yields the growth ratio at the tread centre approximately 0.6 percent and at the maximum point 0.7 percent, the distribution of the growth is virtually flat, cf. Fig. 4. Still, under a typical service condition, the uniformity of the tread growth can be maintained within 0.1 percent of the difference between the tread centre and the maximum point. This differs from the growth of the current WAVED belt remarkably. The data was confirmed by good performance on wear.

As for the durability, the major challenge of the GMD development, the following results were obtained in drum tests. Belt durability test indicates for the New WAVED belt the index of 107 as referred to Fig. 5, which is far beyond the required level. For the bead durability, the reinforced conventional-type bead and the TIP bead show respectively the indexes of 98 and 110 as referred to Fig. 6. The proposed bead structures are satisfactory as compared with the tyre of the aspect ratio 0.70. Reliable durability was again confirmed in the critical drum test, in which the effect of material degradation was considered.

As can be expected from GMD's narrower tyre width and larger belt tension than the dual tyre system, the contact area with the ground under static condition reduces approximately by 13 percent. This leads directly to the increase of the contact pressure, and then to the concern for the damage to the road. However, the GMD tyre has the vertical spring constant 25 percent lower than its counterpart, which seems to represent dynamic behaviour correctly. This can generally be understood as follows: the deformation of the belt ring is determined by the balance of the belt and the sidewall rigidity (tensional force), the rigidity of the sidewall relatively decreases due to the reinforced New WAVED belt, the tyre tends to deflect eccentric with weakening the spring constant. As a consequence, the GMD tyre excels in the FOD test, where plunger head is pressed hard against the tyre tread to measure the stored energy to failure.

Advantages of the GMD concept were solidified as compared with the current dual mount in that: the rolling resistance reduced approximately by 10 percent excluding the effect of rubber properties, the weight reduction including the tyre and the rim was achieved by 80 to 110 kg per axle, the material used for building the tyre reduced by 20 to 25 percent, the space occupied in single tyre house reduced more than 15 cm.

Vehicle tests revealed that the manoeuvrability of the GMD tyre on both dry and wet surfaces was nearly equal to the dual tyres. The riding comfort, the performance of the wear, and the resistance to irregular wear were also equivalent.

CONCLUSION

Technological challenge initially posted for the GMD strategy was addressed. By suppressing the concentration of the stress and the strain, the durability of the GMD tyre was successfully enhanced. In other terms, the novel structures, the New WAVED belt and the TIP bead, were proposed and proven to be excellent. Indeed, all consequences owed to clear understanding of the physics and to the spirit of innovation.

The established technology could be helpful to evolve into further stages of tyre development.

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TABLES & FIGURES



Figure 1 – The super-single drive (GMD) tyre, right, and the conventional dual tyres.

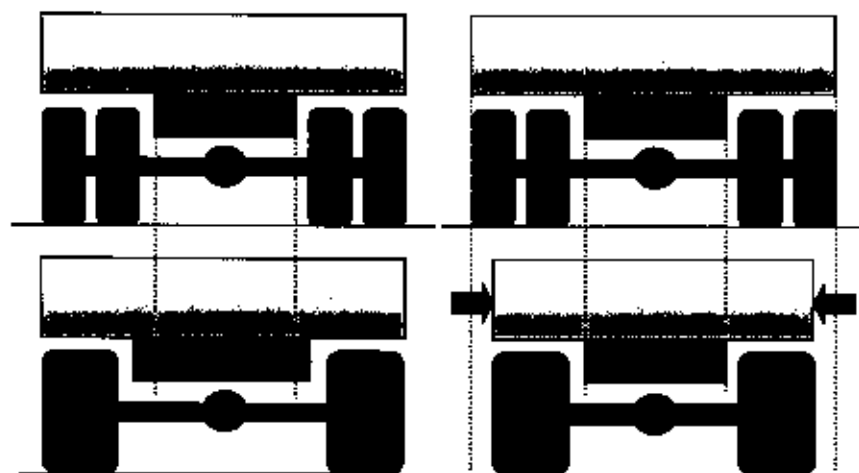
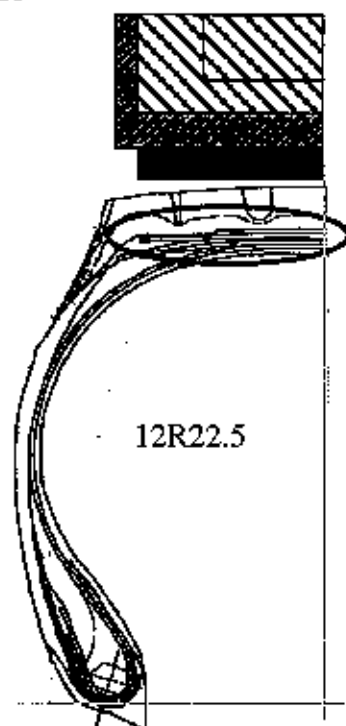


Figure 2 – Advantages of employing the GMD tyre in vehicle design.

CONVENTIONAL
BELT



WAVED (CURRENT)
BELT

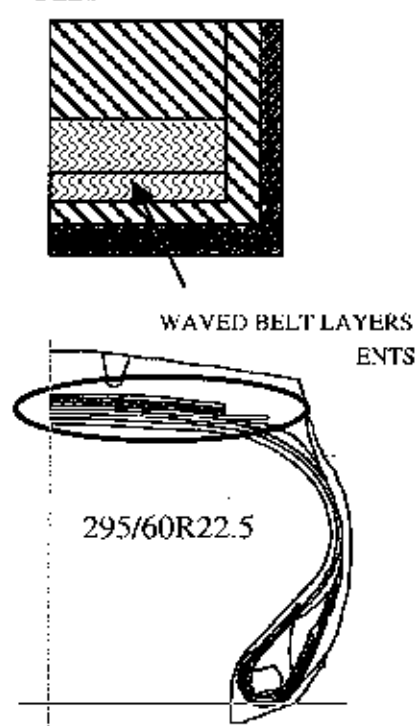


Figure 3 – Typical belt structures adapted to tyres for the truck and the bus.

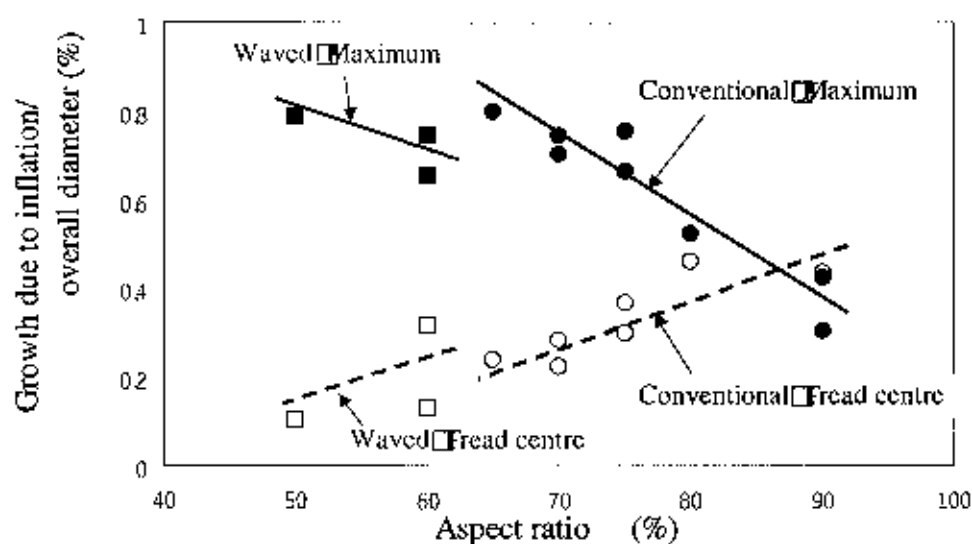


Figure 4 – Effect of the tyre aspect ratio on the tread growth by pressurisation.

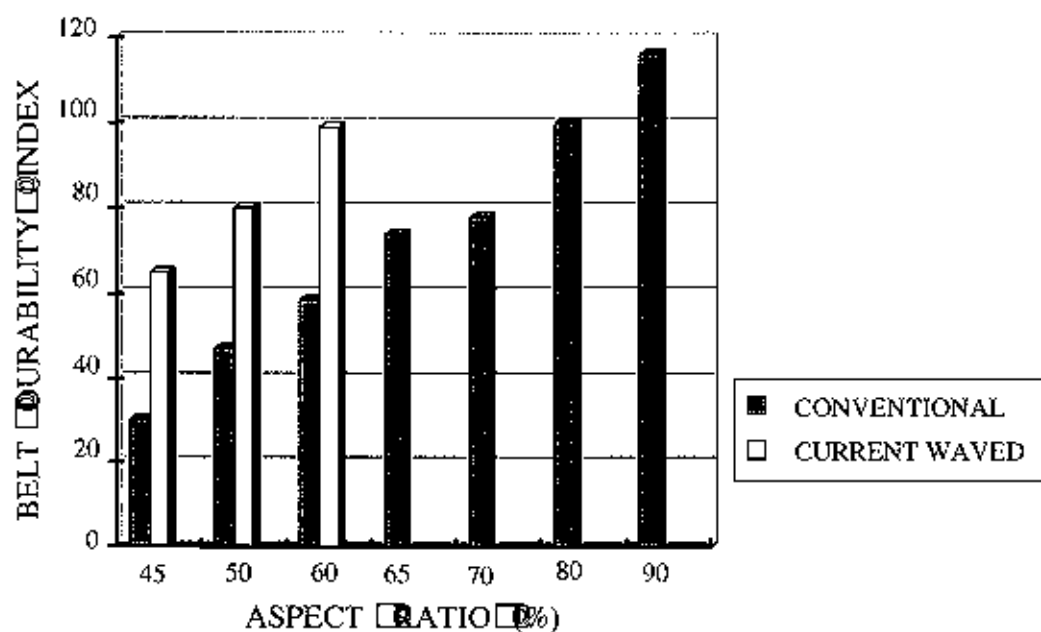


Figure 5 – Trend of the belt durability as a function of the aspect ratio.

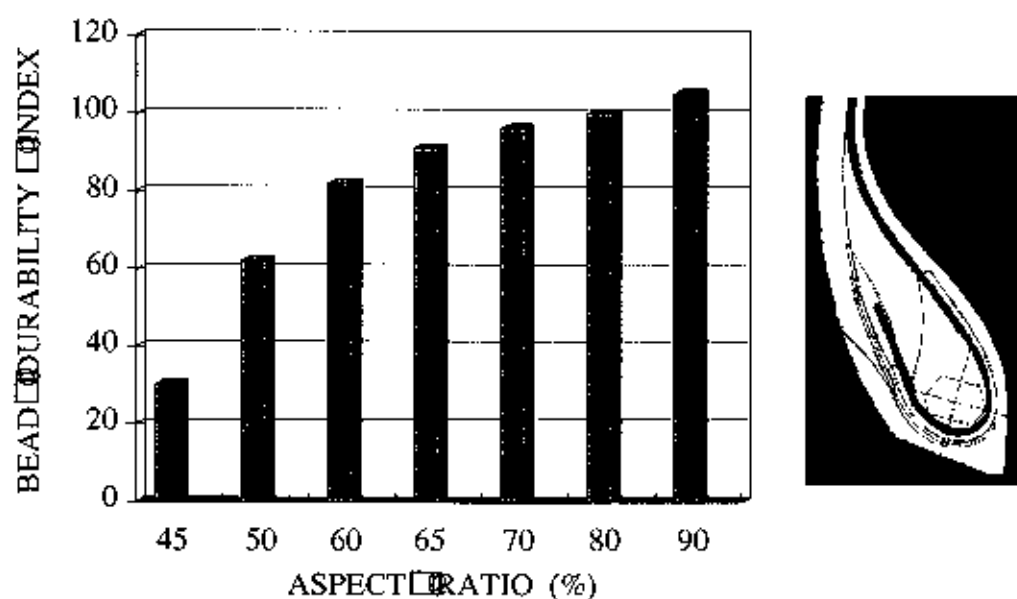


Figure 6 – Conventional head structure, right, and trend of the durability as a function of the aspect ratio.

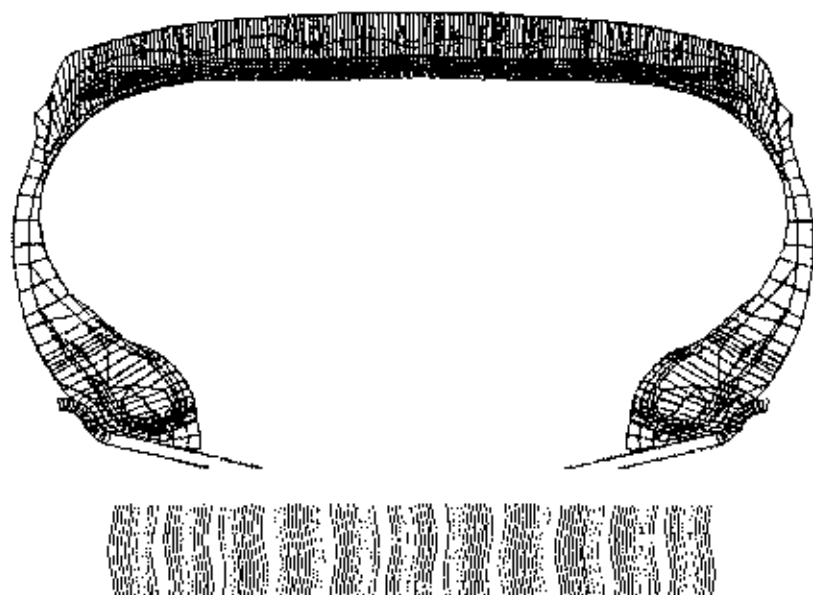
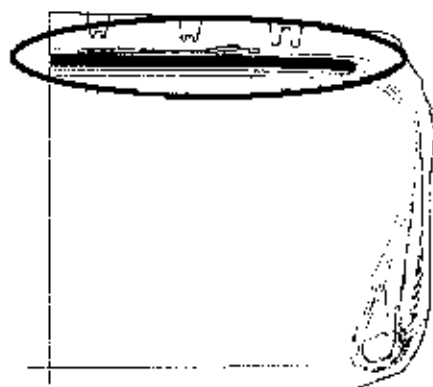
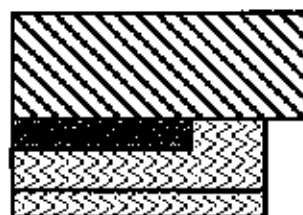
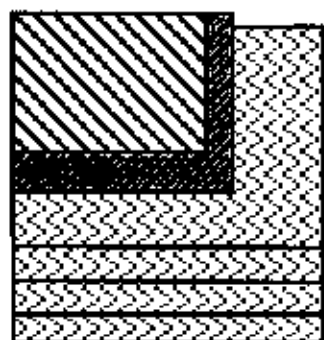


Figure 7 – Finite element mesh: a sectional view of the full three-dimensional FE model and a plane view of embedded WAVED cord bundles slightly out of phase each other.



GREATEC

435/45R22.5

495/45R22.5

435/50R19.5 etc.

Figure 8 – Proposed belt package: the New WAVED belt.

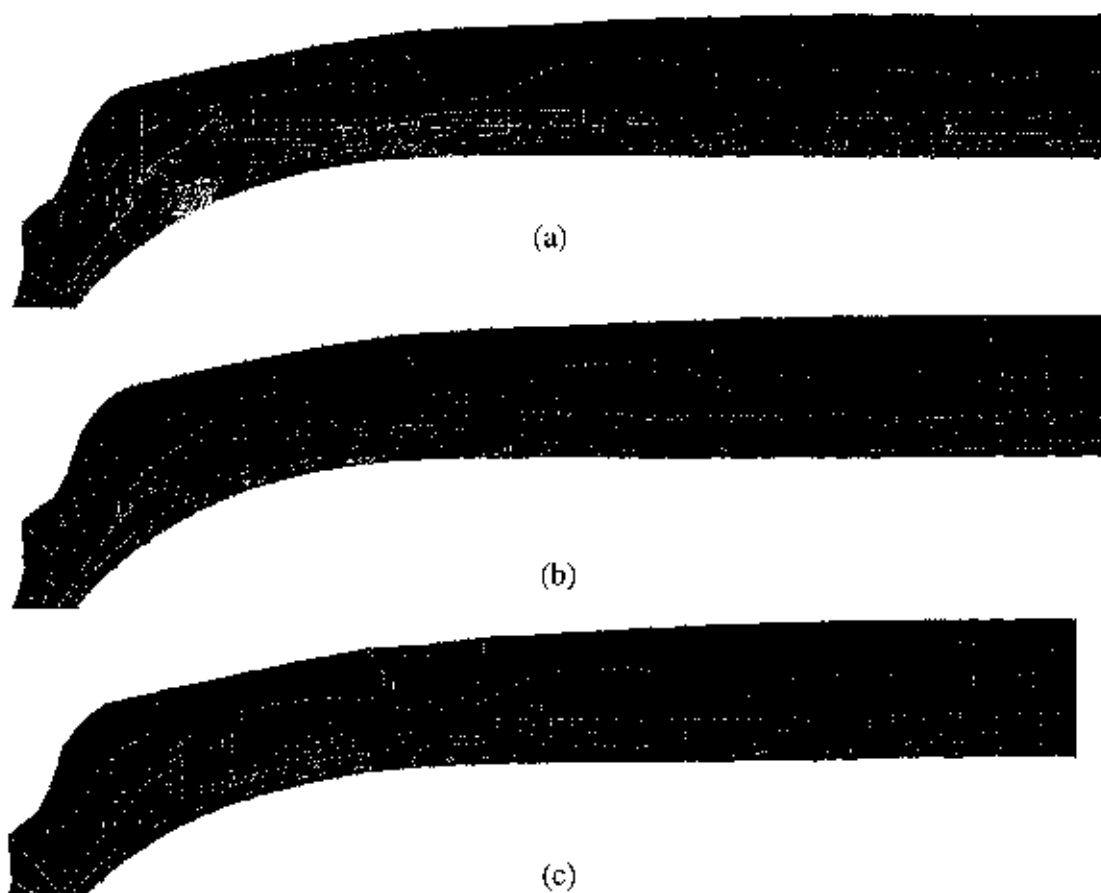
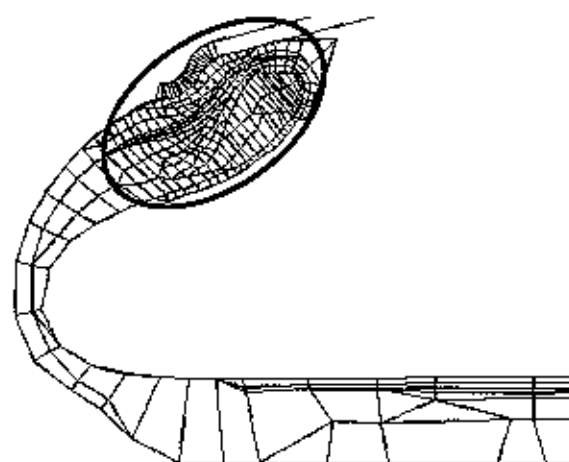


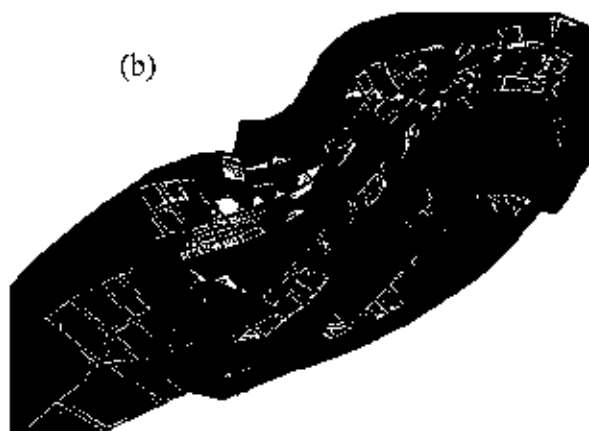
Figure 9 – Von Mises stress under pressurised condition: (a) the conventional belt, (b) the WAVED (current) belt, (c) the New WAVED belt.



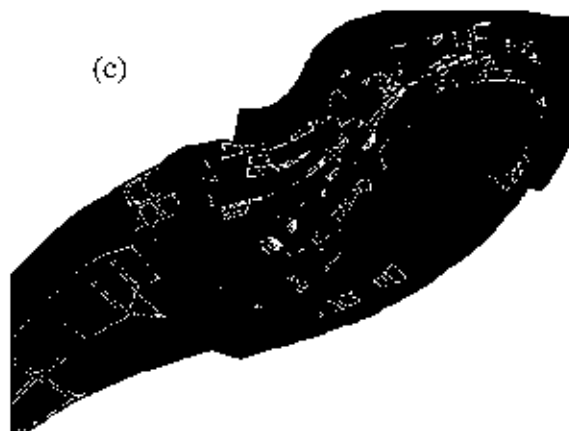
Figure 10 – An example of proposed bead structures: the TIP bead.



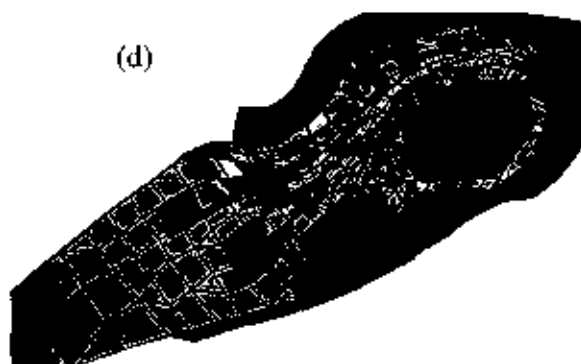
(a)



(b)



(c)



(d)

Figure 11 – Principal strain under loaded condition after thermal aging, 142 hours at 80 degrees Celsius: (a) deformed configuration in the right-under-the-load section, (b) the conventional bead, (c) the reinforced conventional-type bead, (d) the TIP bead.

DESIGN AND OPERATIONAL CONSIDERATIONS TO ACCOMMODATE LONG COMBINATION VEHICLES AND LOG HAUL TRUCKS ON RURAL HIGHWAYS IN ALBERTA, CANADA

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ABSTRACT

The purpose of this paper is to outline the geometric design features that have been developed to accommodate Long Combination Vehicles (LCVs) and log haul trucks on rural highways in Alberta, Canada. Vehicles longer than 25 m are referred to as LCVs and include the following vehicles: Triple Trailer combinations, 35 m in length; Rocky Mountain Doubles, 31 m in length; log haul trucks which can be up to 30.5 m in length, with a 9 m overhang and Turnpike Doubles, 38 m in length. The paper presents the vehicle dimensions, swept path, and off-tracking characteristics of each vehicle type. Geometric design features include intersections, ramps and centre-line spacing on two-lane highways. The paper also presents the criteria used to develop the LCV and log haul truck route networks for the province. The movement of over-dimensional equipment, machinery and pre-assembled components is accommodated on the high-wide-load (HWL) corridor. This 2100 km HWL corridor allows loads up to 9 m high and weights from 122 tonnes in the summer and 177 tonnes in the winter. Weights as high as 380 tonnes may be conveyed on condition that the size and space requirements of undercarriage wheel assemblies are met for critical bridge structures.

INTRODUCTION

Alberta Transportation (formerly Alberta Infrastructure) has been monitoring the operations of Long Combination Vehicles (LCVs) and log haul trucks since their introduction on Alberta highways in the late 1960s. In addition to almost 40 years of operational experience with LCVs, a high load corridor has also been operated since 1986. The high load corridor consists of designated Alberta highways, which have been specially designed or retrofitted to accommodate traffic that may be up to 9 m in height (Morrall et al, 1989, 1992). The transport weights allowed on the high load corridor range from 122 tonnes in the summer to 177 tonnes in the winter. Weights as high as 380 tonnes may be conveyed providing size and spacing requirements of undercarriage wheel assemblies are met for critical bridge structures. The three main LCV configurations are the Rocky Mountain Double, Turnpike Double, and Triple Trailer Combination. The design characteristics of LCVs are described in detail later in the paper, however they are defined as vehicles in excess of 25 m in overall length. These vehicles are only allowed to use the public highway system under special permit, which restrict the routes and impose other conditions. Log haul trucks also operate under permit. These log haul vehicles are 30.5 m in maximum overall length and 80% of all log haul traffic occurs in the winter months. It is noted that three-quarters of the Canadian LCV road network are in Alberta and Saskatchewan. With respect to uniform weights and dimensions on a nation-wide basis it is noted that in 1988 all Canadian provinces and territories signed a national Memorandum of Understanding (MoU) which established national standards for truck weights and dimensions. Under the MoU a maximum truck length of 23 m and a maximum gross vehicle weight of 62.5 tonnes were adopted as national standards, however, road authorities were permitted to exceed these values within their own boundaries. Provinces in western Canada allowed a double trailer length of 25 m. Five jurisdictions, Alberta, Saskatchewan, Manitoba, Quebec, and the Northwest Territories have also developed permit systems which allow various truck configurations that exceed the dimensions in the MoU. The permit systems regulating these long combination vehicles vary according to jurisdiction. In Alberta, double trailer units (such as the Super-B Train) up to 25 m in overall length are allowed to travel throughout the rural primary highway system without restriction.

LCV research to date has focused on operational characteristics and highway geometric design improvements to accommodate large trucks. Harkey et al. (1996) has documented several of the operational characteristics of LCVs believed to have an impact on transportation safety and their relationship to geometric design. These included off-tracking, stability, speed, acceleration, braking and stopping distance. Harwood et al. (1999a) identified four specific roadway design elements whose current geometrics may not be adequate to accommodate larger trucks. Those four elements included horizontal curves, curb radii at intersections, curb radii for ramp terminals, and horizontal curves in freeway on-and-off ramps. Harwood's paper demonstrated that substantial costs could be required to accommodate LCVs on the existing roadway system. The costs were sensitive to the type of truck and extent of the roadway system considered.

Harwood et al. (1999b) examined the distribution of the dimensions of roadway elements critical to accommodation of LCVs. These included horizontal curves and grades on mainline roadways, horizontal curves on interchange ramps, and curb return radii for at-grade ramp terminals and intersections. These distributions were critical to assess the adequacy of current roadways to accommodate LCVs and determine the associated roadway reconstruction cost to accommodate larger trucks.

Sanderson (1996) has documented the need for cost-effective guidelines to enhance truck safety on Canadian highways. While the focus of his research was on trucks up to 25 m in length it has implications for LCVs. The main cost effective recommendations include the following:

- For intersection maneuverability, longer trucks should be used as the design vehicle if they comprise more than 20% of the turning volume.
- Adequate sight distances at railway grade crossings are costly and physically difficult to provide for large trucks.
- The minimum radius of curvature for freeway off-ramps should be increased given the low rollover threshold of large trucks and resultant collisions.

1. DESIGN VEHICLES, DIMENSIONS, TURNING TEMPLATES, AND ACCELERATION CHARACTERISTICS

The basic design vehicle dimensions permitted for the various oversized design vehicles are shown in Fig. 1. Although some vehicles in these groups may be smaller, these dimensions are used for design purposes.

Vehicle turning templates have been developed as a design tool to facilitate intersection layout and design of off-road facilities. These templates are produced for various radii to simulate maneuvers that may be made at various speeds. The "medium" turning templates are generally used for design of highway intersections. The load outswing of the log haul truck is a unique feature of this vehicle configuration that requires special treatment in geometric design of log haul intersections. The outswing is due to the long overhang as well as a special telescoping mechanism in the body of the trailer unit. The telescoping mechanism is required to allow the articulated vehicle to turn while carrying a fixed (non-articulated) load. Because of the telescoping mechanism the load of a log haul truck has a much wider sweep than a conventional truck with the same overhang.

Data on "acceleration from stop" has been recorded for various vehicle types and is illustrated in Fig. 2. This information is used to determine the design intersection sight distance requirements.

2. LCV ROUTES

Alberta Transportation has developed an LCV network (length of 4600 km) on the primary highway system as shown in Figure 3. To date, Triple Trailer Combinations and Turnpike Doubles operate under specific permit conditions that include:

1. Limited to multi-lane highways with four or more driving lanes.
2. Not allowed to operate on statutory holidays or weekends.

3. Not allowed to operate during adverse weather conditions or when the highway is icy or heavily snow covered.
4. No entrance to or exit from Highway 2 may be made except at interchanges, rest area turnouts, or where acceleration or deceleration lanes are provided.
5. Access to and egress from Red Deer is via four-lane roadways only.
6. Where routes fall within a city boundary, the operation of over-length combination units is controlled by the city.
7. Age, experience, and performance of driver.
8. Aspects of vehicle performance such as minimum speed.

The use of LCVs on two-lane highways in Alberta has recently been evaluated. Barton and Morraff (1998) have set forth traffic volume criteria for LCV use on two-lane highways as shown in Table 1. Maximum traffic volumes for a given percent passing zones with no passing lanes are shown in Table 1. The maximum volumes shown ensure a net passing opportunity (NPO) of 30% or greater. The traffic volume criteria become more stringent as the percentage of passing zones is reduced. For example, the maximum traffic volume below which Rocky Mountain Doubles may operate on a conventional two-lane highway is 425 veh/h for a road section with 100% passing zones. On the same highway, with a 2 km passing lane every 10 km, Rocky Mountain Doubles could operate at traffic levels up to 734 veh/h and still ensure a net passing opportunity of 30%. This indicates that the addition of passing/climbing lanes can be very effective in improving opportunities for passing, virtually doubling the traffic volume level under which LCVs can operate. Assuming rolling terrain with ideal geometrics, a traffic stream consisting of 15% trucks and 3% recreational vehicles, a traffic volume of 437 veh/h with a 60/40 traffic split defines the limiting traffic volume for level of service C. The maximum volume up to which either of the LCVs qualify to operate on a conventional two-lane highway, so as to ensure a 30% net passing opportunity, is 425 veh/h. Therefore, the traffic volume criteria established from a passing operations point of view are similar to the desired level of service, which is level of service C, for a two-lane highway in Alberta.

3. LOG HAUL ROUTES

Routes for log haul trucks in Alberta are quite changeable due to the nature of the forestry/lumber/pulp mill industries. Hauls that utilize the special log haul truck involve hauling tree length logs from the cut areas to the various mills that process the lumber. The source of logs varies from year to year and may be impacted by external factors such as forest fires and/or unseasonably warm winters. The changeable nature of log haul routes presents a challenge for the highway designer. It is difficult to justify a major capital cost for a short term or temporary haul, however, it is appropriate to provide a fully functional and safe treatment at more permanent locations such as accesses to mill sites.

4. LCV SPECIAL DESIGN CONSIDERATIONS

Most of the special design considerations for LCVs relate to at-grade intersections or interchanges. Most of the geometric features and traffic operations characteristics on Alberta's rural highway system are suitable for high speed operation of LCVs.

Where LCVs are allowed to enter an undivided highway by means of a left turn at an at-grade intersection, the design sight distance requirements are theoretically longer than they would be for any other design vehicle. This is due to the greater overall length of LCVs. Although the intersection sight distance model indicates that LCVs may require a sight distance of up to 600 m on undivided highways at a design speed of 110 km/h, observations of gap acceptance in the field indicate that 500 m is acceptable and safe.

Where a rural divided highway is being designed to suit LCV operation, the safe access and egress to the highway is again the primary geometric design consideration. Normally in Alberta, divided highway networks are constructed as limited access "expressways" with the ultimate goal of upgrading to "freeway" standards as required in the future to meet traffic demands. Consequently, there are many at-grade intersections on high speed divided highway facilities. Where these intersections are intended to allow LCV crossings, the most practical way

to accommodate this operation (with minimal disruption to through traffic) is through the use of a wide median, which will allow refuge for all vehicle types. This can be done at a relatively low cost at the time of initial construction and may also be done as a retrofit on existing divided facilities as required. For example, the centreline to centreline spacing was increased from 30.7 m to 70.0 m to accommodate LCVs at the Lakeside Road intersection on the Trans-Canada Highway.

Where LCVs are provided access to divided facilities through interchanges, the layout of all interchange components should be compatible with LCV turning templates and sight distance requirements.

5. LOG HAUL TRUCK SPECIAL DESIGN CONSIDERATIONS

As mentioned previously (in Section 2), the load outswing of a log haul truck is unconventional. This unexpected characteristic is potentially hazardous to other road users especially at locations where sharp turns are made such as at intersections. A series of design layouts have been developed to provide the protection needed for the various movements required. In all cases, these intersection design layouts are aimed to prevent encroachment of load outswing onto adjacent lane during deceleration and turning movements.

Additional measures taken to accommodate log haul include providing longer sight distance at intersections and providing additional signage to advise motorists of the presence of special oversized vehicles. Roadside turnouts are constructed to accommodate drivers to stop, rest and use the facilities, and to provide for vehicle inspection and the tying down of loads. Where log haul routes cross divided highways, wide medians are desirable to allow vehicle refuge in the median.

6. LCVS AND PASSING OPERATIONS

The Transportation Association of Canada (TAC) has also investigated the effect of vehicle length on two-lane, two-way roads in Canada (TAC Committee Report 1991). One objective of the study was to determine if current passing sight distances and pavement marking practices are adequate for vehicle lengths of 23 m and 25 m. A key finding was that design passing sight distance (PSD) requirements increase as the length of the impeding vehicle increases. However, the design PSD requirements match well with the standard set by TAC with an operating speed of 80 km/h and passing maneuvers involving vehicles up to 25 m and a 10 km/h differential between the design speed and operating speed. At speeds of 90 or 100 km/h and passing maneuvers involving 23 m or longer vehicles a higher design PSD than that indicated by TAC is required. A conclusion of the TAC technical committee (1991) was "that is not obvious or clear nor is the data available to suggest that a serious problem exists with the performance of TAC's present barrier line roadway marking and the passing of long (25 m) vehicles." The TAC study did not consider LCVs.

In a recent study of the need to enhance truck safety, Sanderson (1996) has recommended that "minimum passing zone lengths should be increased to at least 300 m from the current 100 m, particularly where legislation requires that a driver not drive to the left of a solid center line." When considering fatal and incapacitating collisions, 6% or more are related to passing maneuvers. Sanderson (1996) also recommended that cost-effective guidelines should also be developed to indicate when the minimum length should be increased due to increased volumes of trucks in the traffic stream.

Field observations were made in Alberta (Barton and Morral 1998) to establish driver behavior when overtaking 30 m Rocky Mountain Doubles. The mean gap size accepted by motorists for overtaking was 17 seconds when impeded by a passenger car, compared to 39 seconds when impeded by a 30 m Rocky Mountain Double. The "impeding" vehicle described here is a vehicle travelling in the same direction as the subject motorist's i.e. not an on-coming vehicle. About 70% of the drivers impeded by the passenger car accepted a gap size of 25 seconds. Compared to this, 70% of drivers accepted a 50 second gap size when impeded by a Rocky Mountain Double. In addition to the actual overtaking time, the "indecision time" or the perception-reaction time involved in the overtaking maneuvers was analyzed. "Indecision time" is the time interval between overtaking vehicle being able to overtake, and commencing to overtake. The field data indicated a mean indecision time of about 3 seconds when overtaking a passenger car, and 8 seconds when overtaking the Rocky Mountain Double.

7. SUMMARY

Alberta Transportation has been monitoring the operations of LCVs and log haul trucks for more than 30 years. Many roadway geometric design enhancements such as intersection layouts, vehicle turning templates, acceleration and deceleration characteristics, sight distances, and median spacing and opening have been developed to ensure safe and efficient movements of LCVs and log haul trucks. LCV and log haul routes will continue to expand in Alberta. In this regard, on going monitoring, analysis and research are necessary to validate and enhance LCV and Log Haul geometric design parameters and values.

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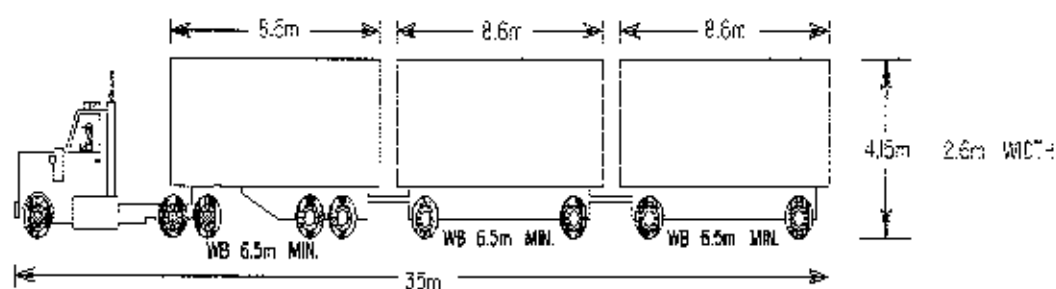
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TABLES & FIGURES

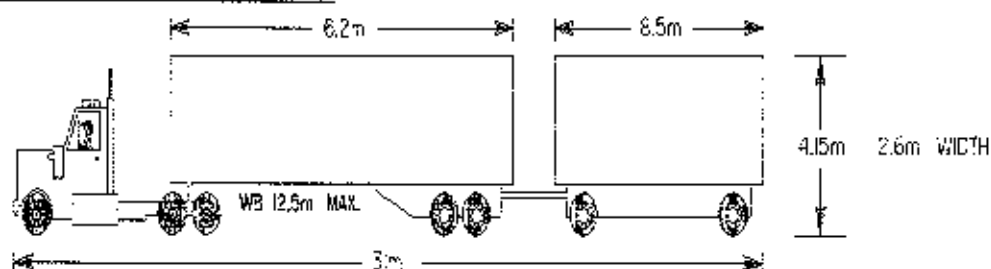
Table J – Maximum Traffic Volume Criteria for LCV Use on Two-Lane Highways

Percent Zones	Passing	Maximum Two-Way Traffic Level (veh/h) Criteria With No Passing Lanes				Maximum Two-Way Traffic Level (veh/h) Criteria With 20% Passing Lanes			
		Rocky Double	Mountain	Turnpike Triple Trailer	Double and Triple Trailer	Rocky Double	Mountain	Turnpike and Triple Trailer	Double and Triple Trailer
10%		-		-		-		-	
20%		-		-		166		149	
30%		-		-		309		277	
40%		102		91		411		368	
50%		180		162		490		438	
60%		245		219		554		496	
70%		299		268		608		545	
80%		346		310		655		587	
90%		388		347		697		624	
100%		425		381		734		658	

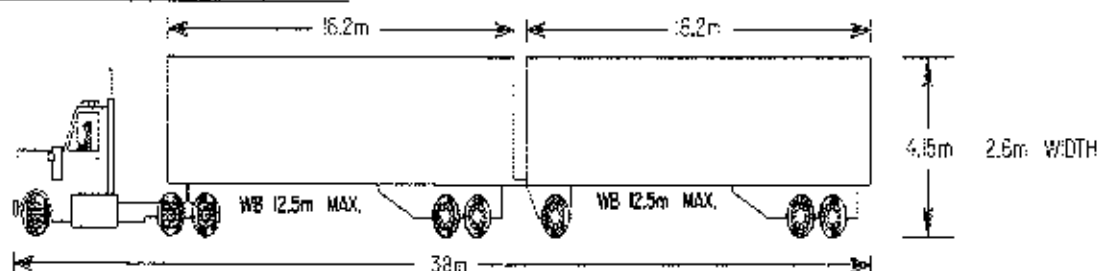
TRIPLE TRAILER COMBINATION (WB-33)



ROCKY MOUNTAIN DOUBLE⁺ (WB-28)



TURNPIKE DOUBLE⁺ (WB-36)



ALBERTA LOG HAUL TRUCK**

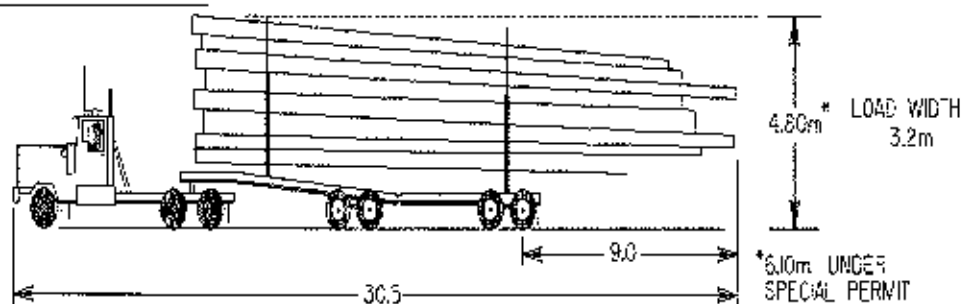


Figure 1 – Design Vehicle Dimensions

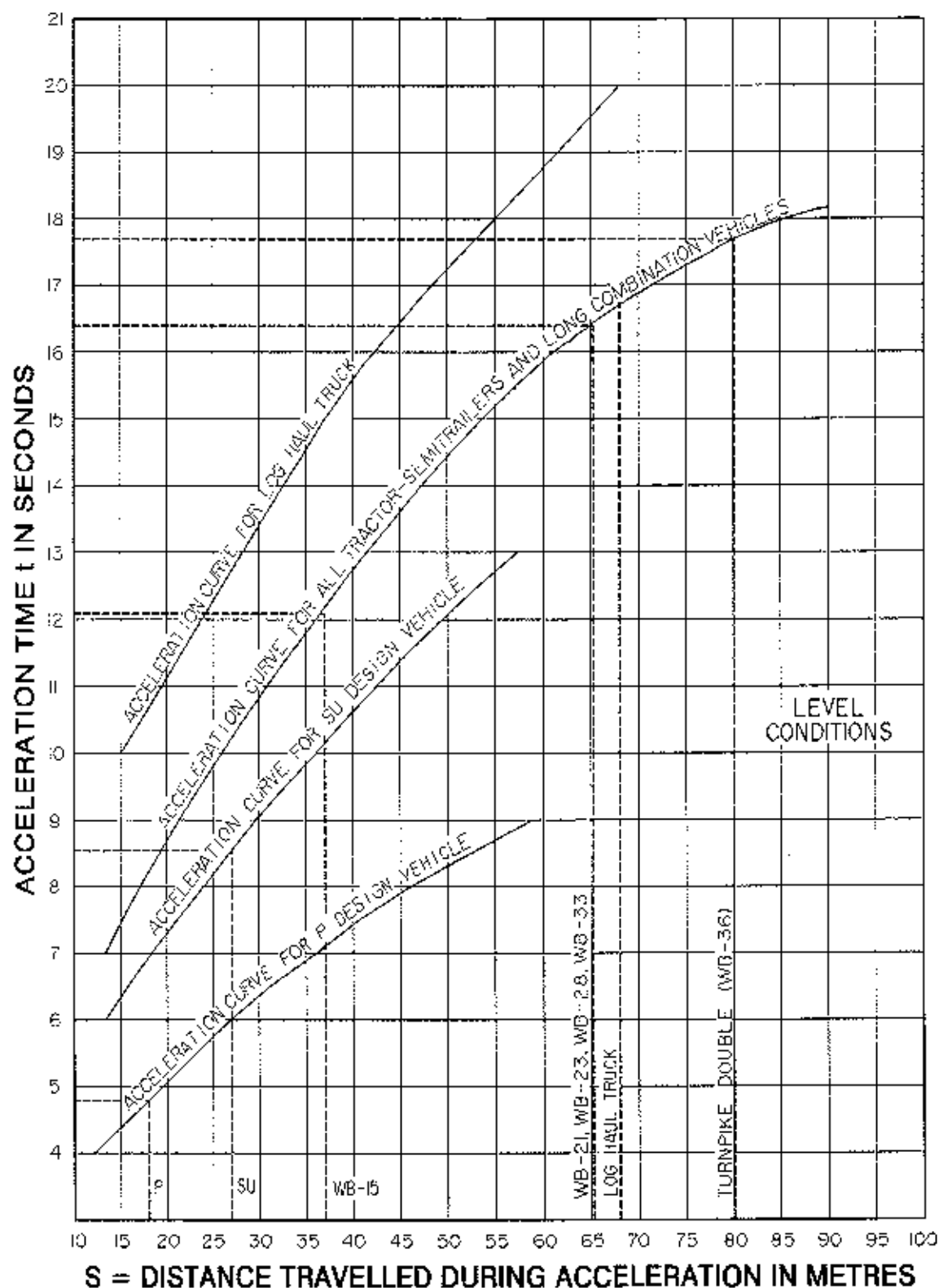


Figure 2 - Data on Acceleration from Stop

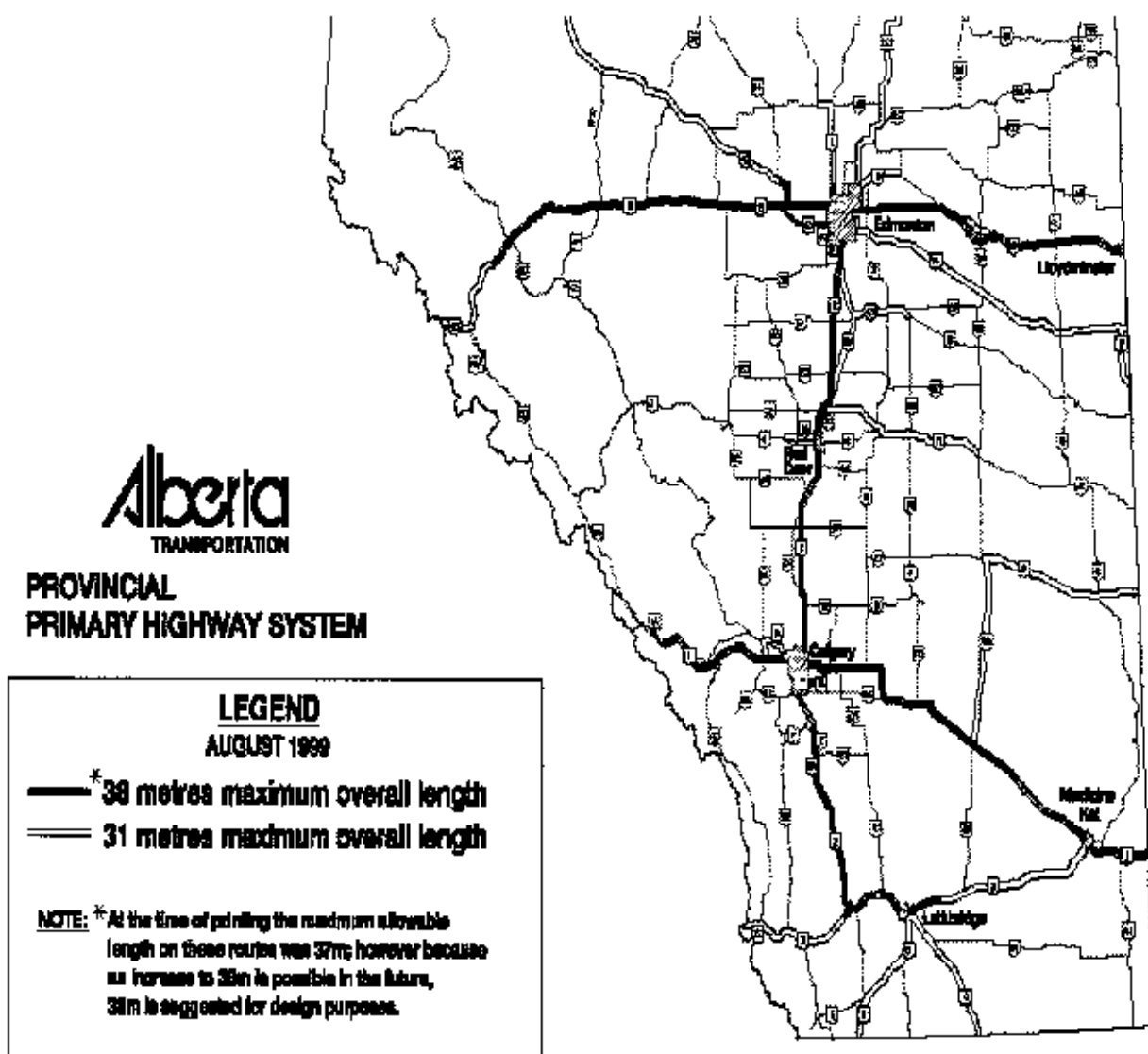


Figure 3 – Alberta's LCV Network on the Primary Highway System

NORDIC VS. CENTRAL EUROPEAN VEHICLE CONFIGURATION; FUEL ECONOMY, EMISSIONS, VEHICLE OPERATING COSTS AND ROAD WEAR

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ABSTRACT

The paper includes, firstly, description of the Finnish Vehicle Motion Simulator, VEMOSIM, its principles, input and output data and analysis output data. Secondly, the paper gives among other the following results:

*The so-called Nordic Vehicle Configuration in Finland, Sweden and Norway (truck + trailer, gross mass 60 t and maximum length 25.25 m) is much more effective measured in the transport power (tkm/h) and energy consumption (fuel, l/100 tkm) and much more environmental friendly (emission amounts, g/tkm) than the Central European vehicle combinations (gross mass 40 t, maximum length 18.75 m). The payload of the Nordic vehicle configuration is approximately 42 t, but the one of the Central European is only approximately 25 t. Though the fuel consumption per the traffic product unit (vehicle*km) increases with the mass, but calculated per the transport product unit (ton*kilometer) it decreases remarkably as well as the emissions.*

These results and many more have been obtained by using the VEMOSIM together with digital road and street data, DIGIROAD. The VEMOSIM results have been validated with field measurements.

The VEMOSIM-DIGIROAD system will be the future tool for analysing effectively, fast and economically the effects of road and traffic conditions, traffic management, alternative routes, vehicle and engine characteristics, driving technique, etc. on the energy consumption, emission amounts, etc. of transport and traffic systems.

*The VEMOSIM is based on vehicle dynamics - on one of the basic laws of physics, namely Force = Mass * Acceleration ($F = m*a$).*

In the case of road traffic, input data of the VEMOSIM includes three data categories: 1) engine and vehicle data, 2) road data and 3) driving patterns. The principal engine data consist of engine maps for fuel consumption and different emission components.

INTRODUCTION

This paper deals with the comparison of the Nordic vehicle configuration (later NVC) and the Central European vehicle configuration (later CEVC). The comparison is made so that the comparison quantities are produced for the NVC and CEVC in the same conditions (on the same road sections in Finland).

The tool used in the calculations is a vehicle motion simulator VEMOSIM based on the dynamics. The VEMOSIM outputs directly the used time, the fuel consumption and the emissions by components for any vehicle type. The economical impacts are produced indirectly by using VEMOSIM results and unit prices.

In this case VEMOSIM it has been applied for the Nordic vehicle configuration and the Central European vehicle configuration on the selected road sections. A short description on VEMOSIM is given in Appendix of this paper.

There are two kinds of comparison quantities; physical and monetary. The both are related to the traffic product (vehicle kilometers) and to the transport product (ton kilometers). The physical quantities are: the used time, the fuel consumption, the emissions by components and the equivalent single axle loads (ESAL). The monetary quantities are: the variable and fixed vehicle operating costs.

The road sections included in the simulations represent two types of topography in Finland: flat (in Western coast of Finland) and hilly (in the Central Finland). The lengths of the road sections used in the simulations are 176 km and 142 km.

VEHICLE DIMENSIONS AND MASSES IN THE EU AND SOME OTHER COUNTRIES

Heavy duty vehicles for goods transport are different in the European countries. When there is question of long haulage transportation, in general, goods are not transported by single unit trucks (without trailers), but the transportation occurs with vehicle combinations. This means that trucks are equipped either with semi-trailers or with trailers. However, the regulations concerning the masses and dimensions of these vehicles in national transportation vary from country to country.

Vehicle dimensions

The dimensions are more homogeneous, because they have been harmonized within the EU with few exceptions, but the masses can be decided at the national level. A very common length for an articulated vehicle (truck + semi-trailer) is 16.5 m and for a road train (truck + trailer) is 18.75 m. In Finland the maximum length for all road trains is 25.25 m as well as in Sweden for module combinations. If the road train in Sweden is not a module combination, the maximum length is 24 m. In Norway (not a member state of the EU) the maximum length for a timber vehicle combination was earlier 22 m, but since 2002 Norway has adopted the Swedish system.

Maximum gross mass of a vehicle combination

The maximum gross mass of a vehicle combination is in general 40 tons in the Central European countries. Before the year of 2000 in Switzerland (not a member state of the EU) this was 28 t. In Denmark the maximum gross mass of a vehicle combination is 48 t and in the Netherlands 50 t as well as in Norway until 2002. In Finland the maximum gross mass for a road train is 60 t, if the combinations has at least 7 axles, 53 t with 6 axles at least, 44 t with 5 axles at least and 36 tons with 4 axles at least. For an articulated vehicle the maximum gross mass is 48 t.

In Sweden there is no separation between an articulated vehicle and a road train. The maximum gross mass is 60 t, but naturally the number of axles restricts that as well as in Finland. Since 2002 Norway has adopted the Swedish practice.

Nordic vehicle configuration

The solution utilized in Finland and Sweden is called here a **NORDIC VEHICLE CONFIGURATION (NVC)**. Today the Netherlands and Belgium have adopted this concept for test purposes during three years. After this period we can see whether this will remain permanent in those countries. The proposal to adapt the NVC is under consideration in Norway.

The main idea in the Nordic Vehicle Configuration is to save energy, reduce emissions and decrease transportation costs. Also the road wear will be reduced, while a reduced vehicle fleet with increased loads can take care of the same transportation product (ton*kilometers) distributed on an increased number of rear axles, but a decreased number of equivalent axles.

The solution utilised in the Central European countries is called here a **CENTRAL EUROPEAN VEHICLE CONFIGURATION (CEVC)**.

TYPE VEHICLES, TRANSPORTATION ROUTES AND DRIVING TECHNIQUE IN THE CASE STUDY

In this presentation some cases are studied. The fuel consumption and emissions have been analyzed by VEMOSIM, which is a computer simulation system for the motion of any road and rail vehicle. The simulation method has been validated by several field tests.

The calculations concerning the vehicle operation costs are based on the current cost and price level in Finland, and the road wear survey is based on the AASHO road tests started in the early 1960's in USA and continued later all over the world

Type vehicles

In order the comparison between the different vehicle types would be possible the load space in all cases is assumed to be a sheeted body. The rated engine power varies according to the vehicle size (313 kW to 390 kW), but the engines are different versions of the same engine base model CUMMINS N14, which belongs to EURO 2 emission class.

A very common vehicle type in Central Europe is an articulated vehicle (truck + semi-trailer) The truck has 2 axles and the semi-trailer has 3 axles. The semi-trailer has a triple bogie with single wheels at each axle. In Finland the gross mass of this

vehicle combination can be 42 tons, but in Central Europe it is normally 40 tons. In this survey we select the gross mass of 40 tons for this vehicle type, and then the payload (load capacity) is 26 tons. The rated engine power is 313 kW (425 hp).

In those countries, where the gross mass is 48 to 50 tons a very common vehicle combination is a 3-axled truck with 3-axled trailer. In Finland the gross mass of this type vehicle is 53 tons. The payload is 37 tons. This is selected for the second type vehicle to be surveyed. The rated engine power is also 313 kW (425 hp).

In Finland and Sweden the most common combination is a 3-axled truck with a 4-axled trailer. Then the gross mass is 60 tons and the payload 42 tons. The rated engine power is 350 kW (475 hp). This is the third type vehicle.

So far the gross mass of 60 tons is the maximum, although the combination has more than 7 axles. This occurs in general in case of the length of 25.25 meters, because the turning rule requires 5 axles in the trailer. In order to utilize the full capacity of the axle masses the gross mass of this combination could be 68 tons (truck 26 tons and trailer 42 tons) and respectively the payload would be 48 tons. This is not yet legal, but it might be the next step in the future solutions. This is selected for the fourth type vehicle in this survey. The rated engine power is 390 kW (530 hp).

By using 4-axled or 5-axled trucks the number of axles in the combination can still be increased. With a 4-axled truck the combination has 9 axles and gross train mass could be 74 tons and the payload 53 tons. This is the fifth type vehicle and its rated engine power is 390 kW (530 hp).

The sixth type vehicle is a 5-axled truck (38 tons) and a 5-axled trailer (42 tons). The gross train mass is then 80 tons and the payload 59 tons. The rated engine power is 390 kW (530 hp), see table 1.

Transportation routes

Two test road sections have been selected. The first one is KOKKOLA-OULU on the coast of Western Finland, where the terrain is very flat, and the second one is TAMPERE-JYVÄSKYLÄ in the Central Finland, where the terrain is hilly. In this way the impacts of the terrain topography can be observed. Figure 1 shows the dependencies of the longitudinal gradient as the functions of the distances of the both test road sections.

Because the VEMOSIM is based on the laws of physics, its results are applicable on any route and road type in any country of the globe.

Driving technique

The legal speed for trucks is 80 km/h, but the speed limiter is set to approximately 90 km/h (89 km/h). The test drives are made here with two goal speeds 80 km/h and 90 km/h. On downward slopes, where the gravitation accelerates the vehicles the goal speed can be temporarily exceeded by 10 km/h before brakes are used. The accelerator pedal is then in the upward position and no fuel flow occurs. This is called "swinging".

The rated engine speed in all versions is 1900 r/min. A gear change down takes place, when the engine speed falls to 1100 r/min and up, when it reaches 1700 r/min. In changing down the whole step is used, but in changing up the splitter is used.

RESULTS OF THE CASE STUDY

Fuel consumption

When the vehicle size is increasing the fuel consumption calculated per driven distance unit [l/100 km] is also increasing. However, the net load is increasing also, and if the fuel consumption is calculated per transport product unit [ton kilometer], the fuel consumption is decreasing. In this survey there are two road sections. The one highway no 8 represents a flat terrain and the other highway no. 9 a hilly terrain. The results of the fuel consumption concerning direction A (north) are presented in Table 2.

Emissions

The emissions of nitrogen oxides, carbon monoxide, hydro carbons, particulate matters and carbon dioxide were also analyzed. The results concerning direction A (north) are presented in Table 3.

Vehicle operation costs

The vehicle operating costs can be divided into variable and fixed costs. The variable costs are:

- fuel costs
- lubricant costs
- repair & maintenance costs
- tyre costs

The fixed costs are:

- capital costs: depreciation and interest
- wages + overhead costs
- insurance costs
- motor vehicle tax
- administrative costs

Normally the vehicle operating costs [/km] or [/h] increase with the vehicle size. The purchasing price is higher (capital costs), and also wages, insurance costs and fixed taxes are higher. The variable operating costs [/km] increase also, and the most obvious example of this is the fuel consumption and thus the fuel costs. But there is also an indirect impact on other variable operating costs. All the factors that affect the fuel consumption affect the other cost components in the same way. In this study a linear relationship has been applied; so a certain relative change of the fuel consumption has been converted directly to the lubricant, repair & maintenance and tyre costs. This is called Wehner's principle.

In the examples presented here the driving product (mileage) per vehicle has been assumed to be 150 000 km/a and the operation time is respectively 3 000 h/a. The results are presented in Table 4 and in figures 2 and 3.

Impacts on road wear

Concerning the road wear attention must be paid to the number of equivalent axles in the vehicle combination and the net load size. The real axles of the vehicle are converted to the equivalent single axle loads (ESAL) according to the rules of the AASHO Road Test.

The unit of this quantity is a single axle of 10 tons with twin wheels, and the other types of axles or axle groups are converted to these units.

From the viewpoint of road wear the most "road friendly" vehicle is the one that has the maximum load per equivalent axle, or inversely when a certain amount of goods must be transported, the number of equivalent axles shall be minimized.

The characteristics of the type vehicles from the road wear viewpoint are seen in table 5.

CONCLUSIONS

General conclusion concerning the way of analysis

The Vehicle Motion Simulator VEMOSIM based on dynamics is an effective tool for analyzing impacts of the characteristics of the different vehicle configurations on the motion state, fuel consumption and emissions.

Conclusions concerning the Nordic Vehicle Configuration (NVC) vs. the Central European Vehicle Configuration (CEVC)

NVC is in many aspects superior to CEVC.

The transportation of goods in general by the Central European vehicle is approximately 30 % more expensive per the transport product unit [tkm] than by the Nordic vehicle.

The Central European articulated vehicle consumes approximately one quarter more fuel per the transport product unit [tkm] than the Nordic road train.

Respectively, carbon dioxide emissions are one quarter higher in the Central Europe than in the Nordic countries.

The Central European articulated vehicle generates approximately two thirds more nitrogen oxides per the transport product unit [tkm] than the Nordic road train.

The Central European articulated vehicle wears approximately two thirds more road pavement per the transport product unit [tkm] than the Nordic road train.

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8th World Congress on Intelligent Transport System, 01-04 October 2001, Sydney, Australia.

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FEATURES AND EFFECTS OF PLATOON DRIVING BASED ON VEHICLE MOTION SIMULATION

8th World Congress on Intelligent Transport System, 01-04 October 2001, Sydney, Australia.

Olavi H. Koskinen and Jussi Sauna-aho:

OPTIMAL ROAD JUNCTION FROM THE VIEWPOINT OF ENERGY SAVING, EMISSION REDUCTION AND VEHICLE OPERATING COSTS

12th Crc On-Road Vehicle Emissions Workshop, San Diego, Ca, April 15-17, 2002

Additionally several presentations at Finnish workshops and congresses as well as several articles in Finnish magazines.

TABLES & FIGURES

Table 1 Characteristics of the type vehicles studied

No.	Configuration	Number of axles			Gross mass	Payload	Rated power
		truck	trailer	tot.	t	t	kW
1	Truck + semi-trailer	2	3	5	40	26.3	313
2	Truck + trailer	3	3	6	53	36.5	313
3	Truck + trailer	3	4	7	60	41.7	350
4	Truck + trailer	3	5	8	68	48.3	390
5	Truck + trailer	4	5	9	74	52.9	390
6	Truck + trailer	5	5	10	80	58.9	390

Table 2. Fuel consumption of alternative vehicles by gross mass

GOAL SPEED 80 km/h						
GROSS MASS	PAYLOAD	ROAD	AVERAGE SPEED	FUEL CONSUMPTION		
t	t		km/h	l/100 km	l/100 tkm	
40	26.3	8A	80.09	38.93	1.48	
40	26.3	9A	79.10	41.33	1.57	
53	36.5	8A	79.85	44.73	1.23	
53	36.5	9A	77.16	47.84	1.31	
60	41.7	8A	79.87	49.74	1.19	
60	41.7	9A	77.27	53.15	1.27	
68	48.3	8A	79.75	52.44	1.09	
68	48.3	9A	76.54	57.03	1.18	
74	52.9	8A	79.56	55.46	1.05	
74	52.9	9A	75.44	60.34	1.14	
80	58.9	8A	79.37	58.54	.99	
80	58.9	9A	74.14	63.65	1.08	

GOAL SPEED 90 km/h

GROSS MASS	PAYLOAD	ROAD	AVERAGE SPEED	FUEL CONSUMPTION	
t	t		km/h	l/100 km	l/100 tkm
40	26.3	8A	89.93	42.59	1.62
40	26.3	9A	87.93	43.84	1.67
53	36.5	8A	89.48	47.92	1.31
53	36.5	9A	85.21	49.86	1.37
60	41.7	8A	89.57	52.35	1.26
60	41.7	9A	85.42	55.03	1.32
68	48.3	8A	89.31	55.81	1.16
68	48.3	9A	84.53	58.81	1.22
74	52.9	8A	89.00	58.82	1.11
74	52.9	9A	83.28	61.95	1.17
80	58.9	8A	88.75	61.82	1.05
80	58.9	9A	81.95	65.13	1.11

Table 3. Emissions of alternative vehicles by gross vehicle mass (GVM)

GOAL SPEED 80 km/h											
GVM	LOAD	ROAD	AVERAGE SPEED	NOx		CO		HC		PM	
t	t		km/h	g/km	g/tkm	g/km	g/tkm	g/km	g/tkm	g/km	g/tkm
40	26.3	8A	80.09	22.21	.845	1.24	.047	.35	.013	.11	.004
40	26.3	9A	79.10	18.79	.715	1.19	.045	.30	.011	.11	.004
53	36.5	8A	79.85	23.91	.655	1.50	.041	.36	.010	.13	.003
53	36.5	9A	77.16	20.83	.571	1.19	.033	.33	.009	.12	.003
60	41.7	8A	79.87	21.05	.505	.78	.019	.44	.011	.19	.005
60	41.7	9A	77.27	20.06	.481	.80	.019	.45	.011	.18	.004
68	48.3	8A	79.75	23.97	.496	1.72	.036	.49	.010	.17	.004
68	48.3	9A	76.54	21.30	.441	1.18	.024	.47	.010	.18	.004
74	52.9	8A	79.56	25.23	.477	1.86	.035	.52	.010	.18	.003
74	52.9	9A	75.44	22.14	.419	1.21	.023	.49	.009	.19	.004
80	58.9	8A	79.37	26.23	.445	1.96	.033	.54	.009	.19	.003
80	58.9	9A	74.14	23.11	.392	1.25	.021	.51	.009	.20	.003

GOAL SPEED 90 km/h											
GVM	LOAD	ROAD	AVERAGE SPEED	NOx		CO		HC		PM	
t	t		km/h	g/km	g/tkm	g/km	g/tkm	g/km	g/tkm	g/km	g/tkm
40	26.3	8A	89.93	25.53	.971	.88	.033	.37	.014	.10	.004
40	26.3	9A	87.93	20.68	.786	.83	.032	.32	.012	.11	.004
53	36.5	8A	89.48	27.32	.748	1.03	.028	.38	.011	.12	.003
53	36.5	9A	85.21	21.45	.588	.96	.026	.33	.009	.12	.003
60	41.7	8A	89.57	25.70	.616	.69	.017	.49	.012	.15	.004
60	41.7	9A	85.42	21.81	.523	.76	.018	.45	.011	.17	.004
68	48.3	8A	89.31	26.03	.539	1.15	.024	.53	.011	.16	.003
68	48.3	9A	84.53	22.04	.456	.98	.020	.49	.010	.18	.004
74	52.9	8A	89.00	26.75	.506	1.22	.023	.53	.010	.17	.003
74	52.9	9A	83.28	23.09	.437	.99	.019	.51	.010	.18	.003
80	58.9	8A	88.75	27.39	.465	1.26	.021	.57	.010	.18	.003
80	58.9	9A	81.95	24.11	.409	1.03	.017	.53	.009	.19	.003

Table 4. The vehicle operating costs for type vehicles are the following:

PRESENT VEHICLES											
	TYPE VEHICLE 1			TYPE VEHICLE 2			TYPE VEHICLE 3				
	40 t 5 axles			53 t 6 axles			60 t 7 axles				
	€/a	€/km	€/tkm	€/a	€/km	€/tkm	€/a	€/km	€/tkm		
FUEL	41625	.278	.0105	48361	.322	.0088	51912	.346	.0083		
LUBRICANT	1303	.009	.0003	1650	.011	.0003	1771	.012	.0003		
REPAIR&MAINT.	16413	.109	.0042	17620	.117	.0032	18913	.126	.0030		
TYRES	10068	.067	.0026	10004	.067	.0018	12691	.085	.0020		
VARIABLE	69409	.463	.0176	77636	.518	.0142	85288	.569	.0136		
DEPRECIATION	20038	.134	.0051	28028	.187	.0051	28988	.193	.0046		
INTEREST	6012	.040	.0015	8408	.056	.0015	8696	.058	.0014		
WAGES	45791	.305	.0116	48046	.320	.0088	48046	.320	.0077		
INSURANCES	4710	.031	.0012	6730	.045	.0012	6730	.045	.0011		

MOTOR VEH. TAX	1544	.010	.0004	2361	.016	.0004	2361	.016	.0004
FIXED	78095	.521	.0198	93573	.624	.0171	94822	.632	.0151
TOTAL	147505	.983	.0374	171209	1.141	.0313	180109	1.201	.0288

FUTURE VEHICLES

	TYPE VEHICLE 4			TYPE VEHICLE 5			TYPE VEHICLE 6		
	68 t 8 axles			74 t 9 axles			80 t 10 axles		
	€/a	€/km	€/tkm	€/a	€/km	€/tkm	€/a	€/km	€/tkm
FUEL	55678	.371	.0077	58409	.389	.0074	60675	.404	.0069
LUBRICANT	1900	.013	.0003	1993	.013	.0003	2071	.014	.0002
REPAIR&MAINT.	20286	.135	.0028	21280	.142	.0027	22106	.147	.0025
TYRES	15707	.105	.0022	17575	.117	.0022	19398	.129	.0022
VARIABLE	93570	.624	.0129	99258	.662	.0125	104249	.695	.0118
DEPRECIATION	29949	.200	.0041	32332	.216	.0041	34715	.231	.0039
INTEREST	8985	.060	.0012	9700	.065	.0012	10415	.069	.0012
WAGES	48046	.320	.0066	48046	.320	.0061	48046	.320	.0054
INSURANCES	6730	.045	.0009	6730	.045	.0008	6730	.045	.0008
MOTOR VEH. TAX	2361	.016	.0003	2906	.019	.0004	3451	.023	.0004
FIXED	96070	.640	.0133	99714	.665	.0126	103357	.689	.0117
TOTAL	189641	1.264	.0252	198971	1.326	.0251	207606	1.384	.0235

Table 5. Characteristics of the type vehicles from the road wear viewpoint

#	Gross mass t	Net load t	# axles	# equivalent axles	Net load/eq. axle t/eq. axle	Index
1	40	26.3	5	3.30	7.93	166
2	53	36.5	6	3.34	10.89	121
3	60	41.7	7	3.15	13.19	100
4	68	48.3	8	3.44	13.98	94
5	74	52.9	9	3.82	13.82	95
6	80	58.9	10	3.88	15.15	87

FIGURE 1. TEST ROAD SECTIONS: GRADIENT VS. DISTANCE

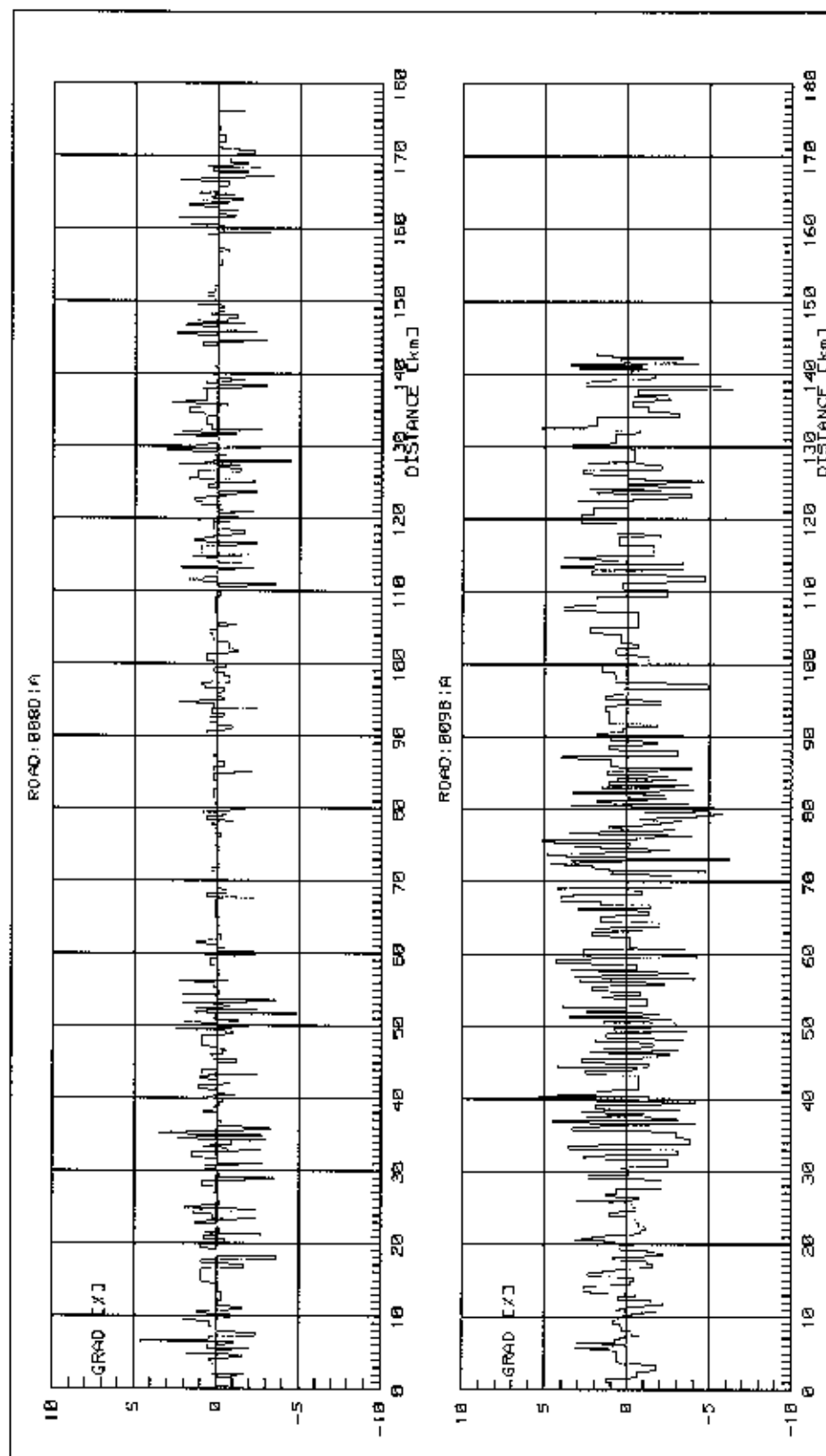


FIGURE 2. OPERATING COSTS VS. GROSS MASS OF DIFFERENT VEHICLE TYPES

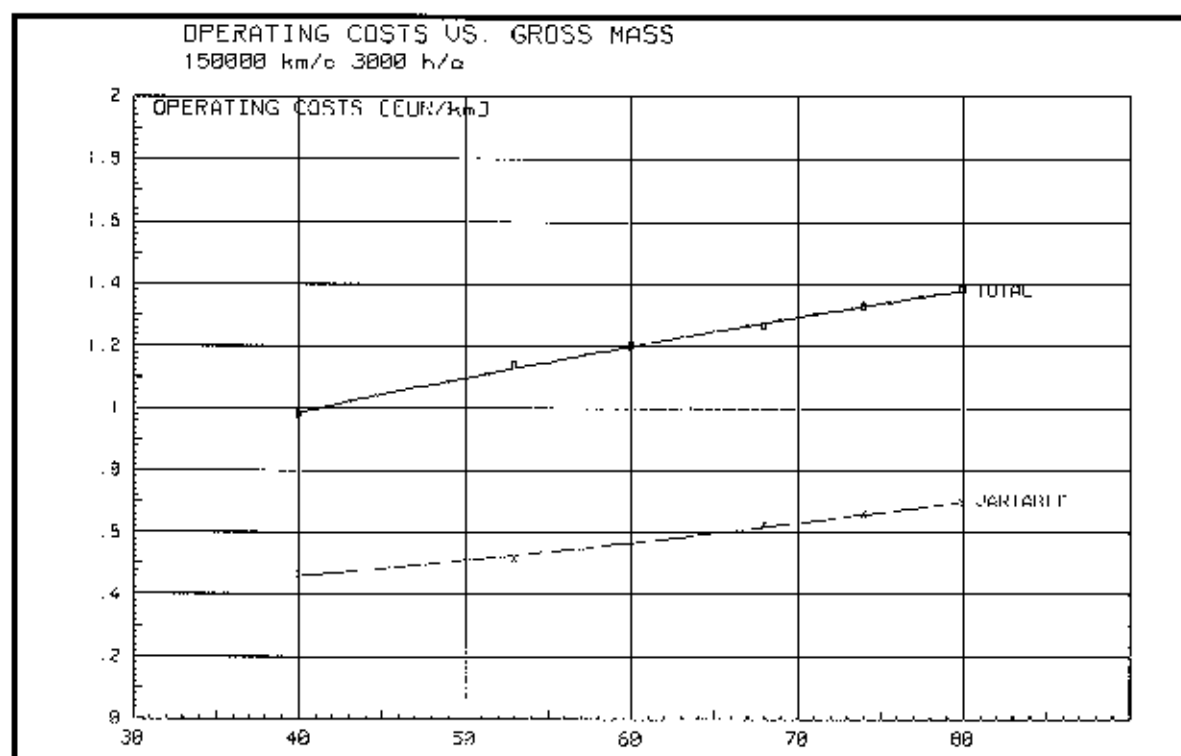
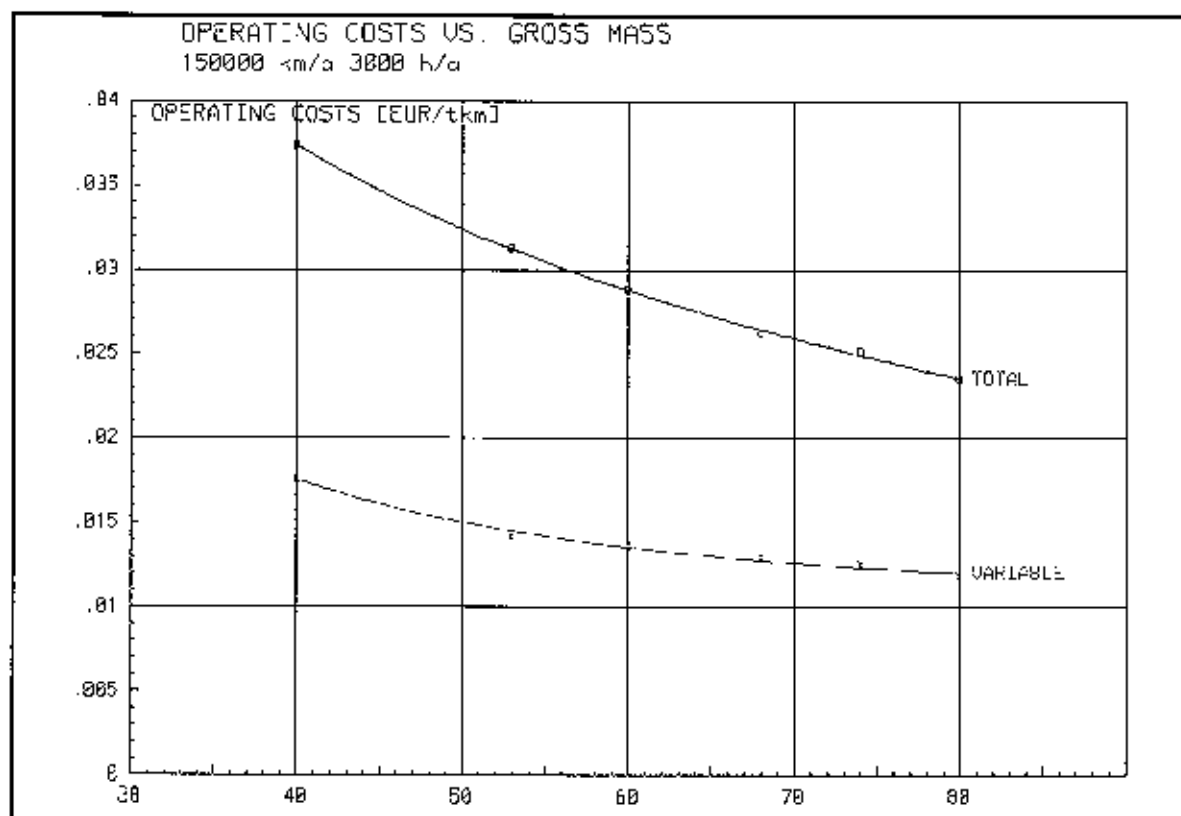


FIGURE 3. OPERATING COSTS PER TON KILOMETER VS. GROSS MASS OF DIFFERENT VEHICLE TYPES



Appendix

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VEHICLE MOTION SIMULATOR (VEMOSIM)

INTRODUCTION

The vehicle computer simulation system has been developed by Mr. Olavi H. Koskinen in the Ministry of Transport and Communications in Finland in order to simulate the motion of a real vehicle in normal road and traffic conditions. In that simulation process the vehicle can be driven either in undisturbed or disturbed traffic flow according to predefined driving techniques and rules (goal speed, schwung, use of gears etc). The simulation output can be continuous, so also the statistical analysis of the drive containing the elapsed time, proceeded distance, consumed fuel amount, use of gears, engine power, torque, engine load distribution etc.

SIMULATION SYSTEM IN DETAIL

The technical characteristics of the vehicle and road must be given as input data, before the simulation process can be started. The idea of the simulation is simply based upon the Newton's second law, which is expressed in the following differential equation:

$$m \, dv/dt = F_t - F_r$$

where:

- v = vehicle speed
- t = time
- m = vehicle mass
- F_t = traction force
- F_r = resistance force

The current resistance force (F_r) is determined by the drive resistance parameters of the vehicle (rolling and air resistance coefficients) and current longitudinal road gradient. The traction force (F_t) is regulated by using the accelerator and brake pedals and various gears.

The geometric data of the road are stored in a very compact mode as distance-altitude or respectively as distance-gradient coordinates. In the former case the rounding curves between them are determined by an effective iterative algorithm.

The vehicle data for the computer simulation are:

- drive resistance coefficients
- description of the power train
- description of the engine
 - maximum torque as a function of the engine speed
 - fuel flow rate as a function of the engine speed and torque
(engine map of fuel consumption)
- emission flow rates by components (NO_x , CO, HC, PM, CO_2) as functions of the engine speed and torque

The quantities, which are followed continuously in the standard output during the drive, are:

- time
- proceeded distance
- consumed fuel amount
- emission amounts if available and selected
- gear position
- current speed
- current engine speed
- longitudinal road gradient

- position of accelerator pedal
- current engine power
- engine load degree
- traction force
- resistance force
- brake force

In the graphical output it is possible to plot several different quantities. The standard graphical output contains, as a function of the distance, the current speed, the cumulative fuel amount or the current fuel consumption per distance unit [l/100 km] and the longitudinal road gradient. In addition, the locations of the gear changes are plotted in the figure; changes up and down with different symbols.

The drive can be analyzed. There are two options, short and long. The long analysis prints the following information:

- time of drive
- distance of drive
- average speed
- cumulative rotation angle of engine
- average engine speed
- loading distribution of engine
 - time distribution by engine speed and torque categories
 - distance distribution by engine speed and torque categories
- drive at full load (maximum torque): time and distance
- average power, torque and loading degree of engine
- fuel consumption
 - time, distance and fuel amount distribution by specific fuel consumption categories
- consumed fuel amount
- average fuel consumption per distance, per time and average specific fuel consumption
- emission amounts if available and selected for survey
- use of brakes: time and distance
- use of gears
 - drive at different gears: time, distance and average speed
 - number of gear changes: down, up, to neutral
- time and distance distribution by different speed and acceleration categories
- drive work: engine work, traction work, resistance work, brake work, acceleration work and average thermal efficiency

EMISSION SIMULATION

If emission maps of different pollutants are available, the impact of the environmental pollution can be studied by this simulation system in various road and traffic conditions. However, for the time being there is a lack of emission maps of engines and they are not easily available from the engine manufacturers.

Therefore the COST 346 project was started in 1999 from the initiative of Finland and supported by Sweden to produce engine maps including also emissions from heavy duty vehicles.

COMPUTER MODELING OF TRANSIT BUSES IN ASSESSING ROAD DAMAGING POTENTIAL

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ABSTRACT

Fully-loaded transit buses and coaches often exceed the maximum permissible axle weights allowed by the federal law on the United States Interstate Highway System. While it is widely understood that overweight trucks operating on the Interstate System contribute significantly to road wear, the extent of damage caused by overweight buses and coaches has not been fully investigated. This paper describes the results of the first phase of a study undertaken in order to determine the level of static and dynamic loads applied by buses and coaches to pavements and to assess their road damaging potential. Axle weight data were reviewed for a representative population of buses. A mathematical model of a transit bus equipped with an air spring suspension was derived. The model is employed in computer simulation of bus-road interaction to determine dynamic loads generated by a transit bus over a typical range of axle weights. Other parameters whose effects on bus wheel forces were studied included road roughness and vehicle speed. Preliminary results of the computer simulation for three levels of road roughness, three speeds, and three static axle loads are presented. In the next phase of the study, the results of the computer simulation will be combined with estimates of highway usage by buses and coaches as a percentage of average daily traffic to evaluate the extent of road wear attributable to buses and coaches.

INTRODUCTION

To protect the civil infrastructure of the United States Interstate System highways, the federal law limits the total weight as well as individual axle weights of the vehicle traveling on the Interstate roads and bridges. The maximum weights allowed on the Interstate System are 20,000 lbs (9091 kg) on a single axle, 34,000 lbs (15455 kg) on a tandem axle, and 80,000 lbs (36364 kg) gross vehicle weight. In addition, the vehicles must satisfy the Bridge Formula, which limits the weights on groups of axles in order to reduce the risk of damage to highway bridges. Transit buses and coaches, which often exceed the maximum permissible axle weights when fully loaded, have been temporarily exempted from the axle weight limitations. In studies of traffic-induced pavement wear, the primary focus has been on heavy trucks, while the impact of buses has received only sporadic attention [1]. A study has recently been undertaken to determine the extent of pavement wear caused by overweight buses and coaches. The results of the study will be taken into consideration to decide if the exemption from the axle weight limits for transit buses and coaches should be granted.

REVIEW OF BUS AXLE LOADS

The population of buses that are in service over the United States Interstate Highways includes a variety of vehicle types that range from modified minivans to large articulated transit coaches. The smaller paratransit vehicles and mid-size buses usually have static axle weights far below the single axle weight limits whereas large, heavy-duty buses and coaches often exceed axle weight limits when fully loaded. This study focuses on axle loads generated by heavy-duty transit vehicles that exceed 10 meters in length, including inter-city transit buses and intra-city coaches.

Inter-city transit buses are typically 12-meter long and are equipped with a single rear axle. The two-axle configuration results in high single axle loads at the rear, drive axle. Several municipalities operate articulated transit coaches, equipped with three axles, that can reach 18 meters in length. Intra-city, over-the-road, coaches are typically 14-meter long and are equipped with a tag axle. While these vehicles are usually larger and heavier than a conventional transit bus, the additional rear tag axle provides for a more even load distribution resulting in lower single axle loads. In general, bus manufacturers are challenged with trying to achieve a balance between a reduction in structural weight and improved structural durability while maintaining or even increasing passenger capacity. The weight problem is compounded by the addition of heavy components associated with advanced drivetrain technologies, alternative fuels, and passenger comfort and assist devices.

The axle weight data were collected from the Federal Transit Administration's Bus Testing and Research Center, operated by the Pennsylvania Transportation Institute. All new and modified transit bus models that are considered for purchasing with federal funds must be tested at the center. Almost 200 buses completed the test program since its inception in 1989. The population of tested vehicles includes virtually all models of transit buses that are currently in service in the United States, including alternative fuels, advanced designs, composite materials, and electric/hybrid-electric buses. An extensive amount of vehicle data have been compiled, including static axle weight measurements. The measurements are conducted for three loading conditions, representing actual operating conditions: curb weight (no passengers), seated load weight (seated passengers only), and gross vehicle weight (seated and standing passengers). Passenger loading is simulated using ballast of 68 kg for every seated and standing passenger position, including the driver.

Axle weight data were collected for thirty-eight 10-meter buses and thirty-four 12-meter buses tested during a period of 1990 to 2001. The results for front and rear axles of all vehicles in the two bus categories are presented in figures 1 and 2. The curves in figures 1 and 2 show the percentage of vehicles in the population of buses having axle loads equal to or less than the corresponding axle weight values. The six curves refer to front and rear axle weights at three load levels: curb weight (CW), seated load weight (SLW), and gross vehicle weight (GVW).

As can be seen from the plots, the front-axle weights for all buses in both 10-meter and 12-meter groups are well below the maximum single-axle weight limit of 20,000 lbs (9091 kg). However, the measured rear axle weights do exceed the limit in most of the 12-meter buses and also in a few of the 10-meter buses. Among the 10-meter buses, 3 vehicles or 8% of the population sampled exceed the single-axle weight limit in both "seated passengers only" (SLW) and in "fully loaded" (GVW) conditions. The axle overweight problem reaches alarming proportions in the 12-meter bus category, where actual rear-axle weights exceed the maximum weight limit in 83% of buses with seated passengers only and in 86% of buses under GVW condition. Table 1 summarizes the axle weight data and their compliance with the axle weight limits.

SIMULATION MODEL OF BUS DYNAMICS

In this section, a nonlinear dynamic model of a transit bus is developed to simulate the rear axle dynamic load. As shown in figure 3, the model comprises of 7 degrees of freedom: pitch ψ , roll ϕ and bounce z_m of the sprung mass, as well as roll ϕ_{n1} , ϕ_{n2} and bounce z_{n1} , z_{n2} of the front and rear unsprung masses, respectively. The equations of motion can be written as follow:

$$\begin{aligned} m_{n1}\ddot{z}_{n1} &= F_5 + F_6 - F_1 - F_2, \\ m_{n2}\ddot{z}_{n2} &= F_7 + F_8 - F_3 - F_4, \\ I_{nr1}\ddot{\phi}_{n1} &= l_{nr1}(F_6 + F_1 - F_3 - F_2), \\ I_{nr2}\ddot{\phi}_{n2} &= l_{nr2}(F_8 + F_3 - F_7 - F_4), \\ m\ddot{z}_m &= -(F_5 + F_6 + F_7 + F_8) + mg, \\ I_p\ddot{\phi} &= S_1(F_5 + F_7) - S_2(F_6 + F_8), \\ I_p\ddot{\psi} &= L_2(F_7 + F_8) - L_1(F_5 + F_6) \end{aligned} \quad (1)$$

where m_{n1} , m_{n2} are the masses of the front and rear axles, respectively; m is the sprung mass; I_{nr1} , I_{nr2} , and I_p are the roll moments of inertia of the front axle, the rear axle, and the sprung mass, respectively; I_p is the pitch moment of inertia of the sprung mass; l_{nr1} and l_{nr2} are the distances from the roll centers of the front and rear axles to one of

their suspension mounting points, respectively; L_1, L_2, S_1 , and S_2 indicate the position of the sprung mass' center of gravity with respect to the wheels; $F_1 \sim F_4$ are the tire forces; and $F_5 \sim F_8$ are the suspension forces.

The tires are simply modeled as point-contact springs. The tire forces are

$$\begin{aligned} F_1 &= k_{t1}(z_{m1} - S_1\phi_{n1} - u_{f1}) \\ F_2 &= k_{t2}(z_{m1} + S_2\phi_{n1} - u_{f2}) \\ F_3 &= k_{t3}(z_{m2} - S_1\phi_{n2} - u_{r1}) \\ F_4 &= k_{t4}(z_{m2} + S_2\phi_{n2} - u_{r2}) \end{aligned} \quad (2)$$

where k_{ti} 's are the tire spring constants, $i = f1, f2, r1, r2$; Subscripts $f1, f2, r1$, and $r2$ indicate front left, front right, rear left, and rear right, respectively; and $u_{f1}, u_{f2}, u_{r1}, u_{r2}$ are the input road profiles to the wheels.

The suspension forces consist of the restoring forces F_{si} 's and the damping forces F_{ci} 's, which are generated from air springs and shock absorbers, respectively.

$$\begin{aligned} F_5 &= F_{s1f1} + F_{c1f1} \\ F_6 &= F_{s1f2} + F_{c1f2} \\ F_7 &= F_{s2r1} + F_{c2r1} \\ F_8 &= F_{s2r2} + F_{c2r2} \end{aligned} \quad (3)$$

The damping forces are approximated by piecewise linear functions as

$$F_{ci} = b_i \dot{z}_{susp_i} + a_i, \quad i = f1, f2, r1, r2 \quad (4)$$

where, the suspension deflections are calculated from the following equations

$$\begin{aligned} z_{susp1} &= z_m - S_1\phi - L_1\psi - (z_{n1} - S_1\phi_{n1}) \\ z_{susp2} &= z_m + S_2\phi - L_1\psi - (z_{n1} + S_2\phi_{n1}) \\ z_{susp3} &= z_m - S_1\phi + L_2\psi - (z_{n2} - S_1\phi_{n2}) \\ z_{susp4} &= z_m + S_2\phi + L_2\psi - (z_{n2} + S_2\phi_{n2}) \end{aligned} \quad (5)$$

The coefficients b_i and a_i in Equation (4) depend on the suspension deflections' time derivatives [2]. For front suspension, $i = f1, f2$,

If $z_{susp_i} > 0.2$ m/s, then $b_i = 3.4$ (KN-s/m), $a_i = 1.4$ (KN)
 If $0 \text{ m/s} < z_{susp_i} < 0.2 \text{ m/s}$, then $b_i = 7$ (KN-s/m), $a_i = 0$ (KN)
 If $-0.3 \text{ m/s} < z_{susp_i} < 0 \text{ m/s}$, then $b_i = 29.4$ (KN-s/m), $a_i = 0$ (KN)
 If $z_{susp_i} < -0.3$ m/s, then $b_i = 11.8$ (KN-s/m), $a_i = -8.82$ (KN)

For rear suspension, $i = r1, r2$,

If $z_{susp_i} > 0.2$ m/s, then $b_i = 3.9$ (KN-s/m), $a_i = 3.22$ (KN)
 If $0 \text{ m/s} < z_{susp_i} < 0.2 \text{ m/s}$, then $b_i = 16.1$ (KN-s/m), $a_i = 0$ (KN)
 If $-0.3 \text{ m/s} < z_{susp_i} < 0 \text{ m/s}$, then $b_i = 40$ (KN-s/m), $a_i = 0$ (KN)
 If $z_{susp_i} < -0.3$ m/s, then $b_i = 22.4$ (KN-s/m), $a_i = -12$ (KN)

Transit buses are commonly equipped with pneumatic suspensions because of their practical advantages, such as road friendliness, as well as a constant suspension frequency and ride height regardless of the vehicle load. A number of air spring models have been developed. Some of them are focused on thermodynamic characteristics of the air suspension, while others concentrate on the force-deflection relationship. These models are either linear or nonlinear, but always with constant model parameters. In other words, the influence of preload (or static load) on the stiffness and/or on the model parameters has not been fully addressed in the existing models.

From the measured data shown in figure 4, the air springs show nonlinear characteristics with respect to the deflection, often with an increased stiffness at larger deflections. Furthermore, the air springs become stiffer when static load increases. For instance, the approximate stiffness of one front axle air spring may vary from 21 KN/m to 75 KN/m when the loading condition varies from curb weight to fully loaded with passengers. Considering that the loading condition of the transit buses may vary significantly from trip to trip, it is necessary to take the preload factor into account for a more realistic model of the transit bus dynamics.

In this paper, to represent nonlinear force-deflection relationship as well as to account for the preload dependence, the air spring force is modeled as a fourth-order polynomial function. Since the bus modeled in this paper has four air bags per axle, the suspension spring forces are formulated as follow:

$$F_{ksi} = 2(k_{i0} + k_{i1}z_{susp} + k_{i2}z_{susp}^2 + k_{i3}z_{susp}^3 + k_{i4}z_{susp}^4) \quad i = f1, f2, r1, r2 \quad (6)$$

where the coefficients k_i depend on the loading condition

$$k_{ij} = a_{ij} + b_{ij} \cdot P_j \quad i = f1, f2, r1, r2 \quad j = 0, 1, \dots, 4 \quad (7)$$

where P_j is the static load on the spring, and the coefficients a_{ij} and b_{ij} are obtained from fitting the experimental data.

PRELIMINARY RESULTS

To assess the extent of road damage contributable to transit buses and coaches traveling on the Interstate System highways, the computer model of bus dynamics will be subjected to actual road profiles measured on interstate highways and covering a representative range of roughness. In the preliminary stages of the study reported in this paper, the road profile was generated artificially following the approach described in [3].

The road profile is modeled as a homogeneous Gaussian random process with zero mean and power spectral density, $S(k)$, defined as

$$S(k) = \begin{cases} S(k_0) \left(\frac{k}{k_0} \right)^{-n_1} & k \leq k_0 \\ S(k_0) \left(\frac{k}{k_0} \right)^{-n_2} & k > k_0 \end{cases} \quad (8)$$

where k is the wave number in cycles/m, $k_0 = 1/2\pi$ (cycles/m); n_1 and n_2 are the spectral slope coefficients. According to ISO classification of road types, $S(k_0)$ ranges from 2 to 2028, $n_1 = 2$, $n_2 = 1.5$. The single road profile is generated by

$$z(j) = \sum_{m=0}^{N-1} \sqrt{\frac{2\pi}{N\Delta}} S\left(\frac{2\pi m}{N\Delta}\right) e^{i(\theta_m - \frac{2\pi mj}{N})} \quad j = 0, 1, 2, \dots, N-1 \quad (9)$$

where Δ is the distance interval between successive ordinates of the profile and θ_m is a set of independent random phase angles uniformly distributed between 0 and 2π . The generated road profiles are shown for three different values of International Roughness Index (IRI) are shown in figure 5.

The bus model derived in the previous section was simulated using the three road profiles, three levels of axle weights, and three speeds. The output variable in the simulation was the Dynamic Load Coefficient (DLC) defined as the ratio of the standard deviation of dynamic wheel load over the mean wheel load. Figures 6, 7, and 8 show the values of DLC versus static axle load and speed for three levels of road roughness. As expected, the DLC increases with increasing speed and road roughness.

FURTHER RESEARCH

The computer model of a transit bus equipped with air spring suspension presented in this paper will provide a tool for evaluating dynamic wheel loads generated by transit buses traveling on interstate highways.

To evaluate the road damaging potential of transit buses, the pavement cost responsibility of buses will be determined and compared to the pavement cost responsibility of a 5-axle tractor-trailer combination. The pavement damage will be estimated using the equivalent standard axle load (ESAL) weighted by an average annual vehicle-miles traveled (VMT)[4]. Furthermore, the impact of future changes in bus design standards, such as new materials, alternative fuels, and new propulsion systems, including hybrid-electric and fuel cell buses, on highway wear will be evaluated.

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TABLES & FIGURES

Table 1 – Percentages of buses exceeding maximum allowable axle load.

		10-Meter Buses	12-Meter Buses
Number of Bus Models		38	32
Number of Sample Buses		38	34
Front Axle Overweight Percentage	CW	0%	0%
	SLW	0%	0%
	GVW	0%	0%
Rear Axle Overweight Percentage	CW	0%	23%
	SLW	8%	83%
	GVW	8%	86%

*CW – curb weight

*SLW – seated load weight

*GVW – gross vehicle weight

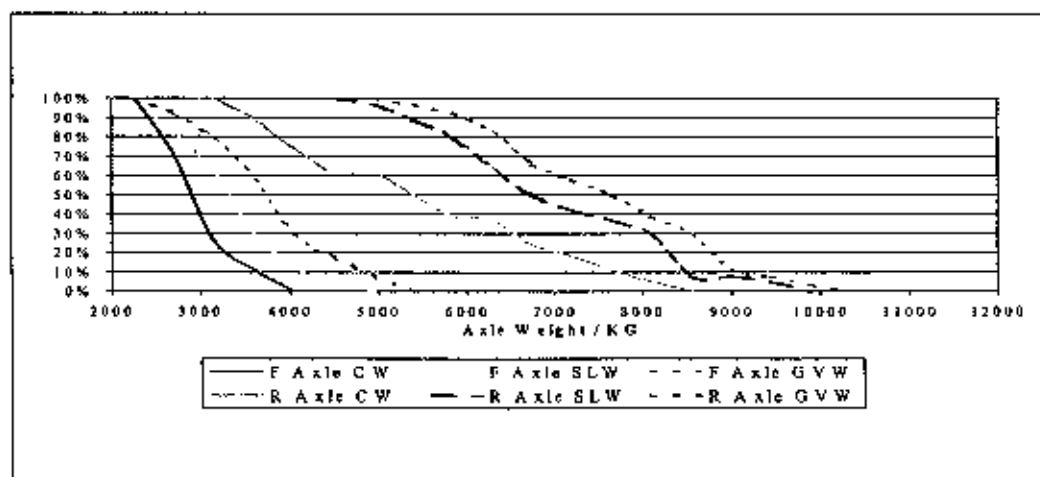


Figure 1. Statistical distribution of single-axle weights for 10-meter buses.

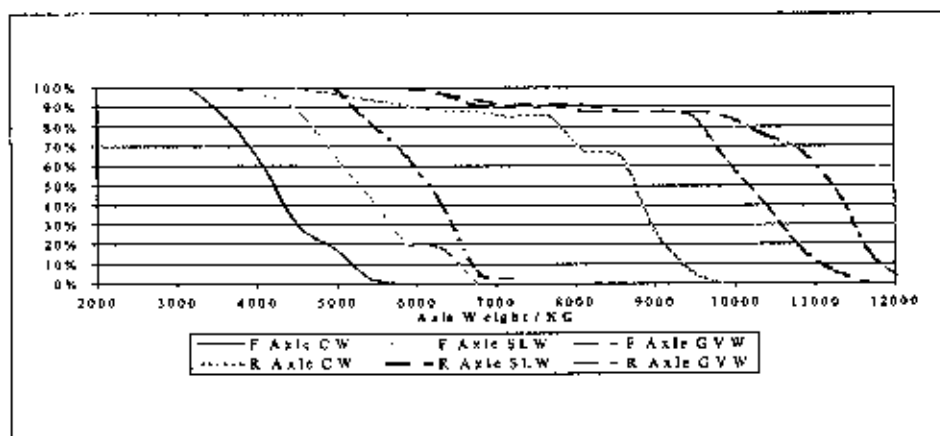


Figure 2. Statistical distribution of single-axle weight for 12-week buses.

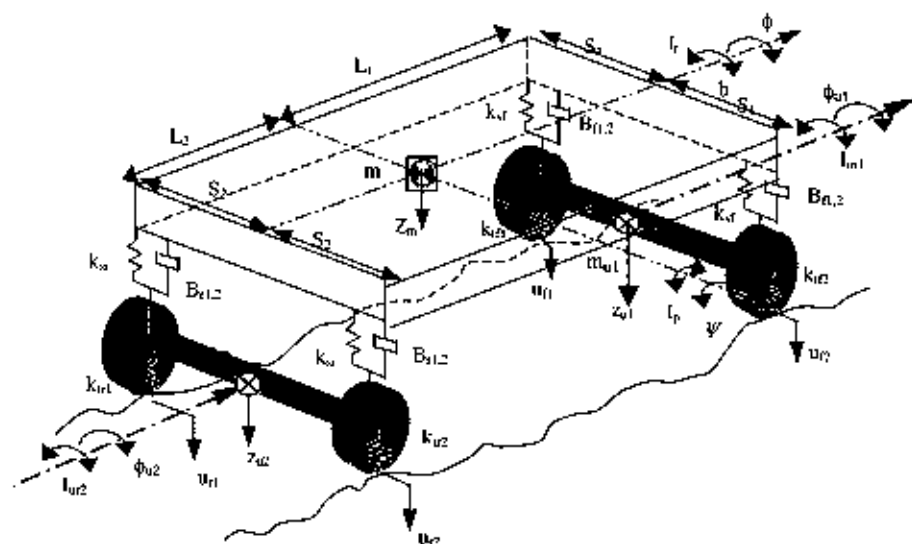
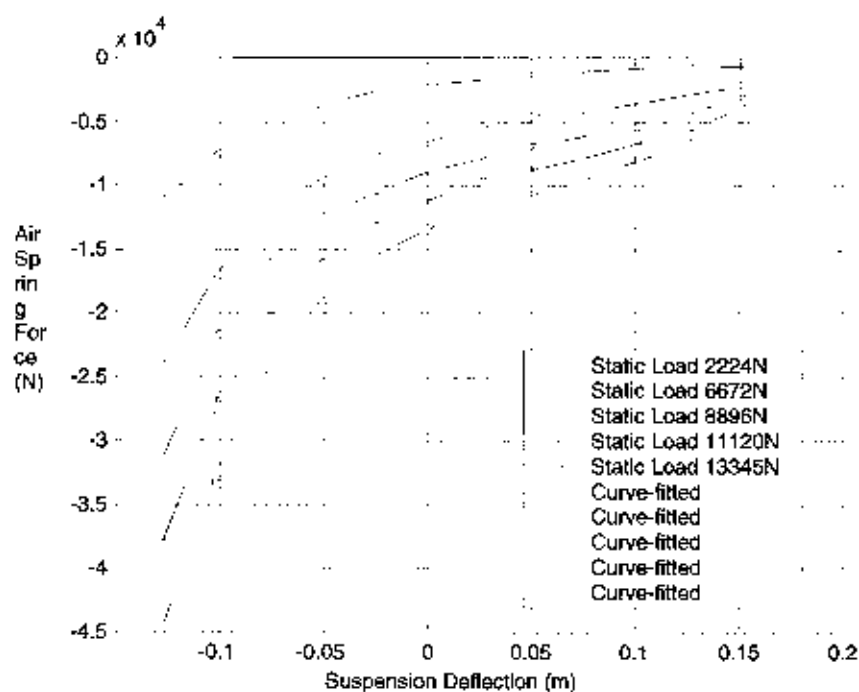
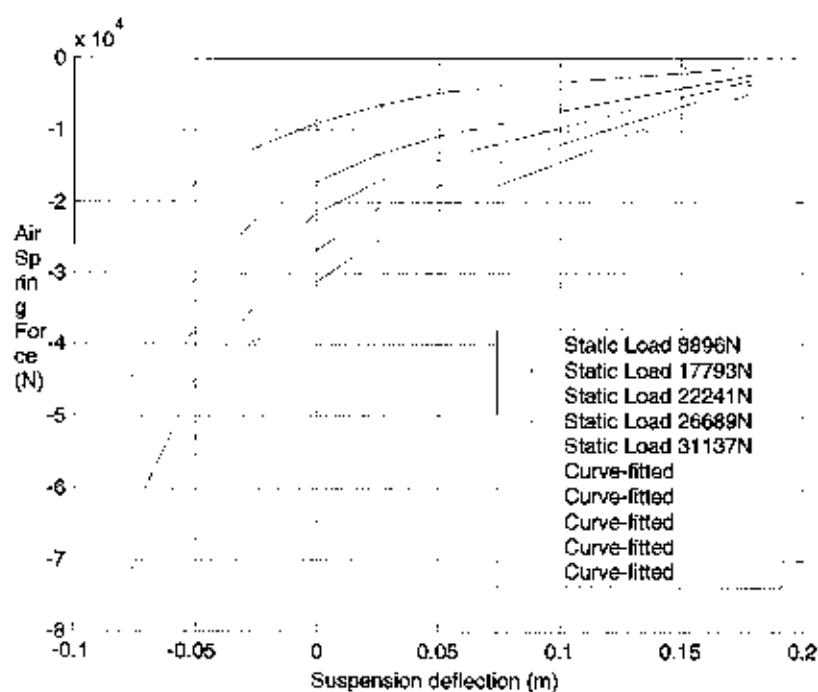


Figure 3 - The vehicle dynamic model.



(a)



(b)

Figure 4 - Modeling of nonlinear air spring force

- (a) Front Axle Air Spring;
- (b) Rear Axle Air Spring

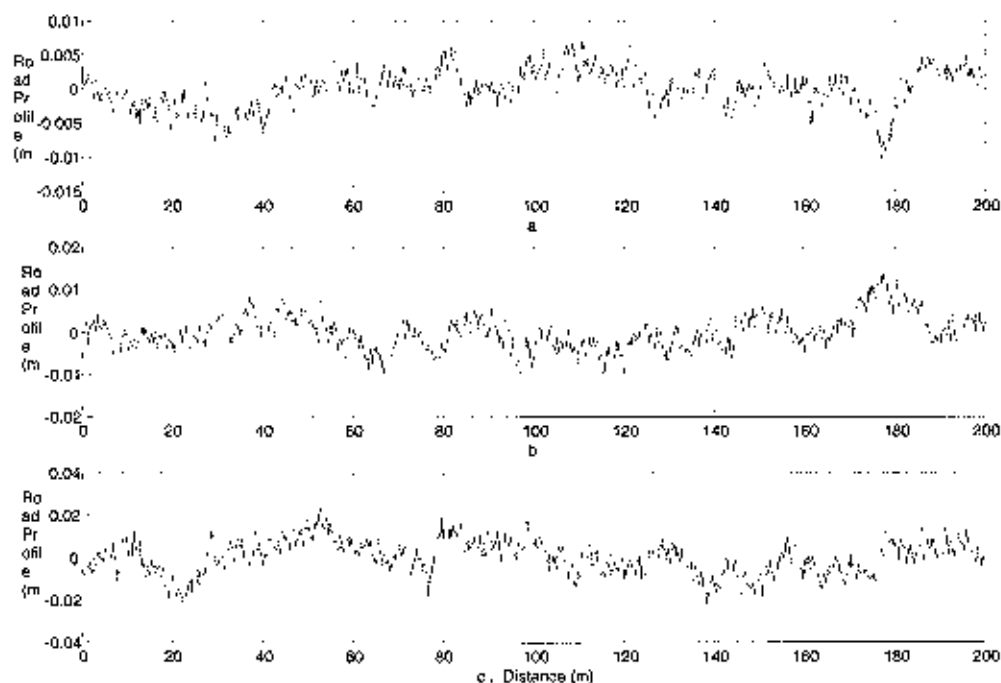


Figure 5 - Generated road profiles with IRI equal to 1.5 mm/m (a), 2.5 mm/m (b), and 5.1 mm/m (c).

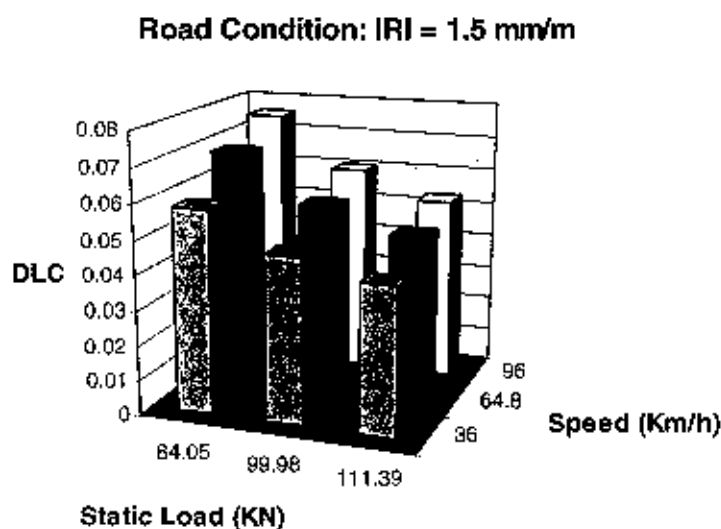


Figure 6 - DLC versus axle load and vehicle speed for IRI = 1.5 mm/m.

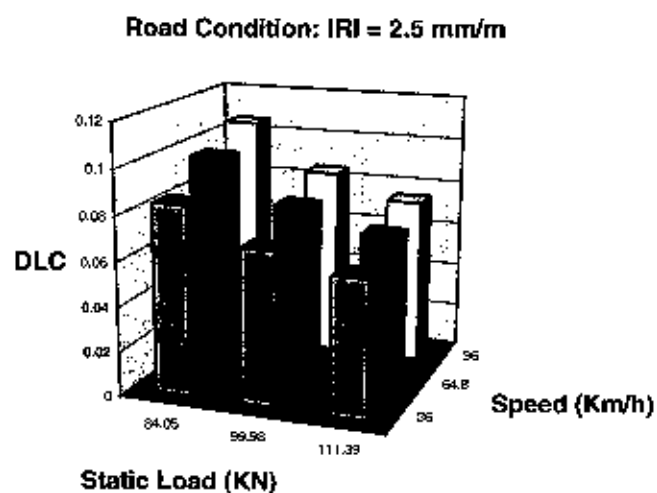


Figure 7 - DLC versus axle load and vehicle speed for IRI = 2.5 mm/m.

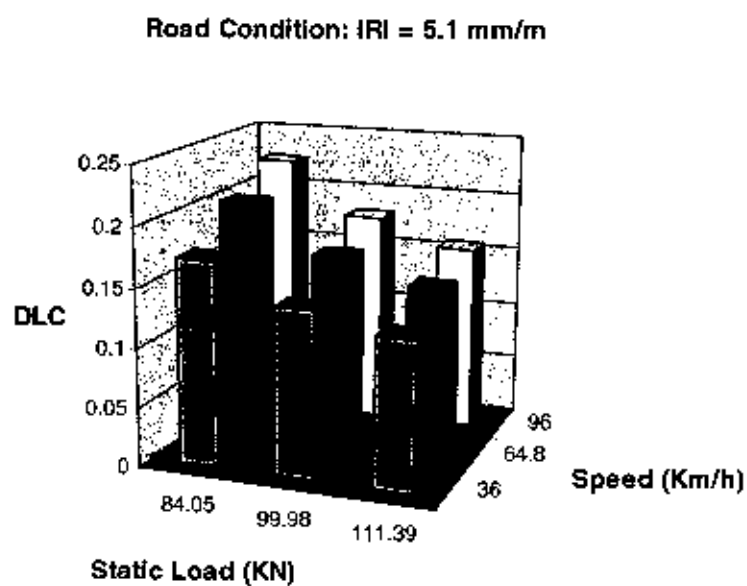


Figure 8 - DLC versus axle load and vehicle speed for IRI = 5.1 mm/m.

NEW PAVEMENT ROUGHNESS THRESHOLDS TO REDUCE DYNAMIC TRUCK LOADING

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ABSTRACT

In this paper, roughness threshold values and corresponding life extensions are determined using relative damage and reduction in pavement life concepts. Using the 4th power law, relative damages from the 95th percentile dynamic load at different roughness index values, and the corresponding percent reduction in life were calculated and plotted for 333 pavement sections. A newly developed roughness index called the dynamic load index (DLI) was used for this purpose. Estimates of pavement life extension resulting from smoothing its surface were then generated for different Remaining Service Life (RSL) values. The results were presented in tables showing the expected life extension for a range of RSL- and DLI- values. These tables would enable a highway agency to determine when a particular pavement needs to be smoothed to obtain a given (desired) life extension. The analysis was done for the three pavement types (rigid, flexible and composite). RSL-values were calculated for 805-m (0.5-mile) sections using actual distress growth over time. The results showed that for rigid pavements, 17 to 51% of sections would have life extensions of more than 3 years depending on roughness level. For composite pavements, none of the sections would have life extensions of 3 years or more. For flexible pavements, 9 to 34% of sections would have life extensions of more than 3 years depending on roughness level. These results indicate that preventive maintenance by smoothing action is best suited for rigid pavements.

INTRODUCTION

Roughness thresholds aimed at minimizing dynamic loads can play an important role in pavement management and preventive maintenance (PM) program. If a PM action, in the form of smoothing the pavement surface, is taken when the critical roughness level is reached it could extend the service life of the pavement by several years, since it will reduce roughness-generated dynamic loads. In this paper, a new roughness index called the Dynamic Load Index (DLI) described in Reference (1,2) was used to develop tables of predicted life extension for pavements with various Remaining Service Life (RSL) values. The analysis was based on mechanistic principles. Dynamic axle loads obtained from the TruckSim™ program were used to calculate the relative damage at different roughness (DLI) levels. The corresponding reduction in pavement life was then used to calculate the life extension that would be achieved, if a PM smoothing action were to be taken, at different RSL-values. These tables can be used to decide on preventive maintenance candidates for smoothing action.

DEVELOPMENT OF A DYNAMIC-LOAD-BASED ROUGHNESS INDEX

The Dynamic Load Index (DLI) is a profile-based index that represents dynamic truck-axle loading. The DLI is calculated as a weighted index of variances of the profile elevation in the frequency ranges 1.5-4 Hz and 8-15 Hz. The first frequency range corresponds to truck body bounce, while the second frequency range corresponds to axle bounce. DLI is calculated using the following equation (1,2):

$$DLI = \sqrt{V_1 + 14V_2} \quad (1)$$

where: V_1 is the variance in elevation of Profile 1 (unit: 10^{-2} in; 1 in = 25.4 mm); with Profile 1 containing only waves in the wavelength range of 6.7 to 17.9 m (22 to 59 ft), which corresponds to a frequency range of 1.5-4.0 Hz for a truck travelling at 96 km/hr (60 mph);

V_2 is the variance in elevation of profile 2 (unit: 10^{-2} in; 1 in = 25.4 mm); with Profile 2 containing only waves in the wavelength range of 1.8 and 3.3 m (6 to 11 ft), which corresponds to a frequency range of 8.0-15.0 Hz for a truck traveling at 96km/hr (60 mph).

The weighting factor of 14 in Equation (1) was determined by trying different values, plotting the variation of the Dynamic Load Coefficient (DLC) and 95th percentile dynamic load, respectively, with DLI for rigid, composite and flexible pavements and selecting the value which gave the overall highest R²-value for all pavement types. The analysis used 333 pavement sections representing a large range of ride quality index (RQI) values¹.

The DLC is an "average" measure of the magnitude of the dynamic variation of axle load over a given pavement surface profile, and is calculated as the ratio of the standard deviation of the dynamic load fluctuations over the static load. The DLC-value for a perfectly smooth pavement surface would theoretically be zero. DLC-values less than 8% indicate moderately smooth pavements, while DLC-values higher than 10% are considered to be indicative of moderately rough pavements, and DLC-values higher than 15% indicate very rough pavement surfaces (4). The 95th percentile axle load is an "extreme" measure of dynamic loading that is indicative of "hot" spots within the pavement surface.

The DLI was tested over a range of road profiles from in-service pavements, and it was found that for any particular value of ride quality index (RQI), the DLI can cover a wide range of values, and this variation in DLI was found to correlate very well with dynamic load, as predicted by a truck simulation program. This was not the case for the International Roughness Index (IRI), which gave a low coefficient of correlation with dynamic load for the same range of profiles. Therefore, the new index can differentiate between profiles that generate high dynamic loads and those having the same RQI but generating low dynamic loads. Most importantly, the use of the DLI index negates the need for running a truck simulation program. This makes it possible for a state highway agency to decide whether a particular pavement with a given surface profile needs smoothing (to extend its service life) based on the DLI-value.

The relationships between DLI and dynamic load were plotted for each pavement type. In this analysis, actual pavement surface profiles of the 333 (161-m or 0.1-mile) sections from 37 projects were used. The pavements included all types (rigid, flexible and composites) with age varying from zero to 39 years. Rigid pavements were mainly jointed reinforced pavements (JRC) with slab lengths ranging from 8.2 to 30.2 m (27 to 99 ft). Distress levels included the entire range from no distress to distress levels exceeding the threshold for rehabilitation. The average daily commercial traffic volume varied from 70 to 8,900, and the project lengths varied from 2.4 km (1.5 mi) to 26.7 km (16.6 mi). The profiles were input to the truck simulation program, TruckSim™; axle load time histories were generated from a typical 5-axle tractor-semi-trailer. From these dynamic axle-load profiles, DLC and 95th percentile axle loads were calculated and plotted against the corresponding DLI-values, as shown in Figure 1. The figure shows a very good relationship between DLI and the two measures of dynamic loading.

DEVELOPMENT OF DLI-THRESHOLDS AND RESULTANT LIFE EXTENSIONS

The damage induced in pavements by a dynamic load relative to static load can be expressed as the equivalent number of static load passes, using a power law:

$$\text{Relative Damage} = \left(\frac{L_{\text{dynamic}}}{L_{\text{static}}} \right)^n \quad (2)$$

where n is the damage exponent (typically, $n = 3-5$).

The theoretical percent reduction in pavement life can be calculated as (5):

$$R = \text{Percent Reduction in Pavement Life} = 100\% [1 - (\text{Relative Damage})^{-1}] \quad (3)$$

¹ The RQI is a ride number that was developed by the Michigan DOT in the 1970's. It is calculated from three PSD wavelength bands according to the equation shown below (3):

$$RQI = 3 \ln(\text{Var1}) + 6 \ln(\text{Var2}) + 9 \ln(\text{Var3})$$

where Var1 , Var2 and Var3 are variances for 7.6-15.2 m (25-50ft), 1.5-7.6m (5-25ft) and 0.6-1.5 m (2-5 ft) wavelengths, respectively. These wavelengths were found to correlate at 90 percent with subjective opinions obtained from a series of "psychometric" tests conducted on passenger car users.

Using the 4th power law, relative damages from the 95th percentile dynamic load at different DLI values, and the corresponding percent reduction in life were calculated and plotted in Figure 2 for each pavement type. The general equation for the curves relating relative damage to DLI can be written as:

$$y = a(DLI)^4 + b(DLI)^3 + c(DLI) + 1 \quad (4)$$

where y is the relative damage and a , b , and c are regression constants.

The corresponding R^2 -values were between 0.914 and 0.954, with the higher values being for rigid pavements. Table 1 summarizes the relevant regression parameters of the best-fit curves.

The life extension that can be achieved by smoothing a pavement section of a given remaining service life (RSL) can be determined as:

$$\text{Life extension} = (R - R_0) \text{RSL} \quad (5)$$

where R corresponds to the reduction in life at the current DLI value;

R_0 corresponds to the reduction in life at the DLI value immediately after smoothing.

The following example shows how to calculate life extension for given RSL and DLI values. The calculation of the life extension expected from smoothing a rigid pavement surface that has an RSL of 14 years and a DLI of 10 is as follows: From Figure 3 (a-2), the R -value corresponding to a DLI of 10 is 43.6%. The R_0 -value corresponding to DLI of 3 is 21.9%. Therefore, the life extension can be calculated as $LE = (0.473 - 0.218) \times 12 = 3.0$ years.

The analysis was done for rigid, flexible and composite pavements with RSL values ranging from 20 to 6 years. For the calculation of life extension, the DLI value corresponding to a pavement condition immediately after the smoothing action was determined to be 3 based on Figure 2. Tables 2 through 4 show the life extension that can be expected for a range of RSL- and DLI- values. If a minimum life extension of 3 years were to be adopted, then the shaded areas within each table would represent conditions where smoothing PM action is warranted. Note that the life extension calculations are based on a DLI-value of 3 (and not zero) after the smoothing action, since no pavement is perfectly smooth. This should translate in more realistic estimates shown in the tables. Also note that while the tables give life extension predictions of all possible combinations of RSL and DLI values, the combinations of high RSL and DLI values are less likely to occur in practice.

POTENTIAL LIFE EXTENSION FOR IN-SERVICE PAVEMENTS

Determining the remaining service life (RSL) of pavement sections at different roughness levels is important since it allows for determining the applicability of the life extension tables developed in this study in the context of the current MDOT pavement management system. Superimposing the predicted RSL-values from actual pavement performance on the life extension tables allows for determining the proportion of pavement sections that would be favourable candidates for PM smoothing action, i.e., those with a minimum RSL value to get the desired life extension of 3 years or more. For example, a rigid pavement section with a DLI of 10 needs to have a minimum RSL of 14 to yield a life extension of 3 years or more upon smoothing. The proportion of rigid pavement sections with DLI of 10 having RSL-values greater than 14 would determine the usefulness/applicability of any PM smoothing action.

A large data set from MDOT PMS system was used for this analysis. Remaining service lives were calculated for those 0.5-mile pavement sections that have DLI greater than 7. This was done using the Distress Index (DI) prediction model developed by MDOT. This model uses a logistic function having the following form,

$$DI(t) = \frac{(a+b)a}{a+b \times \exp(-rt)} - a \quad (6)$$

where t = age, and a , b and r are regression parameters.

Figure 3 illustrates how to calculate the RSL given past DI-values. The RSL is defined by MDOT as the number of years needed to reach the threshold DI-value of 50, from the current DI-value. The DI is defined as the sum of distress points corresponding to different distresses normalized to a unit pavement section of 161 m (0.1 mi). Individual distress points vary with distress type, level and extent, with their values calibrated to reflect MDOT's

pavement management practice. The DI threshold of 50 corresponds to the value at which rehabilitation should be scheduled. RSL distributions for pavement sections that have DLI-values between 7 and 11, and between 11 and 15 are shown in Figure 4 for each pavement type. The number of sections with a DLI greater than 15 was too small to show a reliable distribution of RSL-values.

Rigid Pavements

Figure 4 indicates that rigid pavements have the largest RSL-values. All rigid pavement sections analysed have positive RSL-values. About 29% of the rigid pavement sections that have DLI-values ranging from 7 to 11 have RSL-values greater than 13 years, which according to Table 2 is the minimum required RSL for getting a life extension of 3 years or more. Combining the information in Table 2 and Figure 4(a), one can determine that about 60% of these sections (17% of rigid pavement sections with DLI between 7 and 11) would have life extensions of more than 3 years if they were to be smoothed. On the other hand, 57% of the pavement sections with DLI-values ranging from 11 to 15 have RSL-values greater than 10 years (the minimum required RSL for getting a life extension of 3 years or more for DLI range 11-15), and about 89% of these sections (51% of rigid pavement sections with DLI between 11 and 15) would have life extensions of more than 3 years, i.e., they would be PM candidates.

Composite Pavements

According to Figure 4 (b), about 92% of composite pavement sections with DLI-values ranging from 7 to 11 have positive RSL-values, as compared to 67% with DLI-values ranging from 11 to 15. However, none of these pavement sections have RSL-values corresponding to a life extension of 3 years or more, as can be seen in Table 3. The maximum RSL for pavements with DLI between 7 and 11 is 9 years, which corresponds to a maximum life extension of 2.9 years. For pavement sections with DLI between 11 and 15, the maximum RSL is 6 years, which corresponds to a life extension of 2.5 years.

Flexible Pavements

For flexible pavements, more sections have negative RSL-values in the above DLI ranges, as can be seen in Figure 4(c). About 48% of sections with DLI-values ranging from 7 to 11 and 32% with DLI-values ranging from 11 to 15 have negative RSL-values. This means that these pavements sections already reached the DI threshold ($RSL=0$) before they reach the roughness level at which smoothing action would be needed. Nonetheless, about 14% of pavement sections with DLI between 7 and 11 have RSL-values greater than 12 years (the minimum required RSL for getting a life extension of 3 years or more for the DLI range 7-11). Combining the information in Table 4 and Figure 4(c), one can determine that about 62% of these sections (9% of all flexible pavement sections with a DLI between 7 and 11) would have life extension of 3 years or more. For sections with DLI between 11 and 15, about 39% have RSL-values greater than 9 years (the minimum required RSL for getting a life extension of 3 years or more for DLI range 11-15). Combining the information in Table 4 and Figure 4(c), one can determine that 88% of these sections (34% of all flexible pavements with DLI between 11 and 15) would have life extensions of 3 years or more, i.e., they would be PM candidates for smoothing action.

Summary

Figure 5 summarizes the relative distribution of life extension for each pavement type within the DLI ranges of interest. Figure 6 summarizes the distribution of pavement sections with different life extensions, as a percentage of the total population within each pavement type. The figures show that rigid pavements have the highest proportion of sections that would benefit from smoothing action, with about twenty percent of the total population having a potential life extension (LEX) of 3 years or greater. About ten percent of flexible pavement sections would benefit from such action, while no composite pavement section is expected to have a life extension of 3 years or more. However, more composite sections have positive RSL than flexible pavements. These results indicate that, in terms of life extension gain, preventive maintenance action in the form of smoothing the pavement surface is most suited for rigid pavements. It can be applied to flexible pavements with relatively less success. However, under the current MDOT pavement management system, it would appear that such smoothing actions might not be as useful for composite pavements.

CONCLUSION

In this paper, a newly developed dynamic load index (DLI) was used to determine roughness threshold values and corresponding life extensions based on relative damage and reduction in pavement life concepts. Using the 4th power law, relative damages from the 95th percentile dynamic load at different DLI values, and the corresponding percent reduction in life were calculated and plotted for 333 sections. Estimates of pavement life extension resulting from smoothing its surface were then generated for different Remaining Service Life (RSL) values. The results were presented in tables showing the expected life extension for a range of RSL- and DLI- values. These tables would enable a highway agency to determine when a particular pavement needs to be smoothed to obtain a given (desired) life extension. The analysis was done for the three pavement types (rigid, flexible and composite). Based on the results of this analysis, it can be stated that in terms of life extension gain the preventive maintenance action in the form of smoothing the pavement surface is most applicable to rigid pavements.

ACKNOWLEDGEMENT

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TABLES & FIGURES

Table 1- Regression Parameters for DLI-Relative Damage Relationships

Pavement Type	Regression Parameters				
	a	b	c	R ²	SE
Rigid	2.81E-4	-6.75E-3	1.16E-1	0.954	0.081
Composite	-2.52E-5	2.63E-3	5.31E-2	0.914	0.187
Flexible	2.67E-4	-5.81E-3	1.09E-1	0.932	0.145

Table 2- Life Extension for Different RSL and DLI-values for Rigid Pavements

DLI	RSL														
	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20
5.0	0.5	0.6	0.7	0.8	0.9	1.0	1.1	1.2	1.3	1.4	1.5	1.6	1.6	1.7	1.8
6.0	0.7	0.9	1.0	1.1	1.2	1.3	1.5	1.6	1.7	1.8	2.0	2.1	2.2	2.3	2.4
7.0	0.9	1.0	1.2	1.3	1.5	1.6	1.8	1.9	2.1	2.2	2.4	2.5	2.7	2.8	
8.0	1.0	1.2	1.4	1.6	1.7	1.9	2.1	2.3	2.4	2.6	2.8	2.9			
9.0	1.2	1.4	1.6	1.8	2.0	2.2	2.4	2.5	2.7	2.9					
10.0	1.3	1.5	1.7	2.0	2.2	2.4	2.6	2.8							
11.0	1.4	1.7	1.9	2.1	2.4	2.6	2.9								
12.0	1.6	1.8	2.1	2.3	2.6	2.8									
13.0	1.7	2.0	2.2	2.5	2.8										
14.0	1.8	2.1	2.4	2.7											
15.0	1.9	2.3	2.6	2.9											
16.0	2.1	2.4	2.8												
17.0	2.2	2.6	2.9												
18.0	2.3	2.7													
19.0	2.5	2.9													
20.0	2.6														

Table 3- Life Extension for Different RSL and DLI-values for Composite Pavements

DLI	RSL															
	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	
5.0	0.6	0.7	0.8	0.9	1.0	1.1	1.2	1.3	1.4	1.5	1.6	1.7	1.8	2.0	2.1	
6.0	0.9	1.0	1.2	1.3	1.5	1.6	1.7	1.9	2.0	2.2	2.3	2.5	2.6	2.8	2.9	
7.0	1.1	1.3	1.5	1.7	1.9	2.0	2.2	2.4	2.6	2.8	3.0	3.2	3.3	3.5	3.7	
8.0	1.3	1.6	1.8	2.0	2.2	2.5	2.7	2.9	3.1	3.3	3.5	3.7	3.9	4.1	4.2	
9.0	1.5	1.8	2.1	2.3	2.6	2.8	3.1	3.3	3.5	3.7	3.9	4.1	4.3	4.5	4.6	
10.0	1.7	2.0	2.3	2.6	2.9	3.2	3.5	3.7	3.9	4.1	4.3	4.5	4.7	4.9	5.0	
11.0	1.9	2.2	2.6	2.9	3.2	3.5	3.7	3.9	4.1	4.3	4.5	4.7	4.9	5.1	5.2	
12.0	2.1	2.4	2.8	3.1	3.4	3.7	3.9	4.1	4.3	4.5	4.7	4.9	5.1	5.3	5.4	
13.0	2.2	2.6	3.0	3.4	3.7	4.0	4.2	4.4	4.6	4.8	5.0	5.2	5.4	5.6	5.7	
14.0	2.4	2.8	3.2	3.6	4.0	4.3	4.5	4.7	4.9	5.1	5.3	5.5	5.7	5.9	6.0	
15.0	2.5	3.0	3.4	3.8	4.2	4.5	4.7	4.9	5.1	5.3	5.5	5.7	5.9	6.1	6.2	
16.0	2.7	3.2	3.6	4.1	4.5	4.8	5.0	5.2	5.4	5.6	5.8	6.0	6.2	6.4	6.5	
17.0	2.8	3.4	3.8	4.3	4.7	5.0	5.2	5.4	5.6	5.8	6.0	6.2	6.4	6.6	6.7	
18.0	2.9	3.6	4.0	4.5	4.9	5.2	5.4	5.6	5.8	6.0	6.2	6.4	6.6	6.8	6.9	
19.0	3.0	3.8	4.2	4.7	5.1	5.4	5.6	5.8	6.0	6.2	6.4	6.6	6.8	7.0	7.1	
20.0	3.1	4.0	4.4	4.9	5.3	5.6	5.8	6.0	6.2	6.4	6.6	6.8	7.0	7.2	7.3	

Table 4- Life Extension for Different RSL and DLI-values for Flexible Pavements

DLJ	RSL															
	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	
5.0	0.6	0.6	0.7	0.8	0.9	1.0	1.1	1.2	1.3	1.4	1.5	1.6	1.7	1.8	1.9	
6.0	0.8	0.9	1.0	1.1	1.3	1.4	1.5	1.6	1.8	1.9	2.0	2.1	2.3	2.4	2.5	
7.0	0.9	1.1	1.2	1.4	1.5	1.7	1.8	2.0	2.2	2.3	2.5	2.6	2.8	2.9	3.0	
8.0	1.1	1.3	1.4	1.6	1.8	2.0	2.2	2.3	2.5	2.7	2.9	3.1	3.2	3.4	3.5	
9.0	1.2	1.4	1.6	1.8	2.0	2.2	2.5	2.7	2.9	3.1	3.3	3.5	3.7	3.8	3.9	
10.0	1.4	1.6	1.8	2.1	2.3	2.5	2.7	3.0	3.2	3.4	3.6	3.9	4.1	4.2	4.3	
11.0	1.5	1.8	2.0	2.3	2.5	2.8	3.1	3.3	3.5	3.7	4.0	4.2	4.4	4.5	4.6	
12.0	1.6	1.9	2.2	2.5	2.7	3.0	3.3	3.5	3.7	3.9	4.2	4.4	4.6	4.7	4.8	
13.0	1.8	2.1	2.4	2.7	3.0	3.3	3.5	3.7	3.9	4.2	4.4	4.6	4.8	4.9	5.0	
14.0	1.9	2.2	2.6	2.9	3.2	3.5	3.7	3.9	4.2	4.4	4.6	4.8	5.0	5.1	5.2	
15.0	2.1	2.4	2.7	3.1	3.4	3.7	3.9	4.2	4.4	4.6	4.8	5.0	5.2	5.3	5.4	
16.0	2.2	2.6	2.9	3.3	3.6	3.9	4.2	4.4	4.6	4.8	5.0	5.2	5.4	5.5	5.6	
17.0	2.3	2.7	3.0	3.4	3.7	4.0	4.2	4.4	4.6	4.8	5.0	5.2	5.4	5.5	5.6	
18.0	2.5	2.9	3.2	3.6	3.9	4.2	4.4	4.6	4.8	5.0	5.2	5.4	5.6	5.7	5.8	
19.0	2.6	3.1	3.4	3.8	4.1	4.4	4.6	4.8	5.0	5.2	5.4	5.6	5.8	5.9	6.0	
20.0	2.7	3.2	3.5	4.0	4.3	4.6	4.8	5.0	5.2	5.4	5.6	5.8	6.0	6.1	6.2	

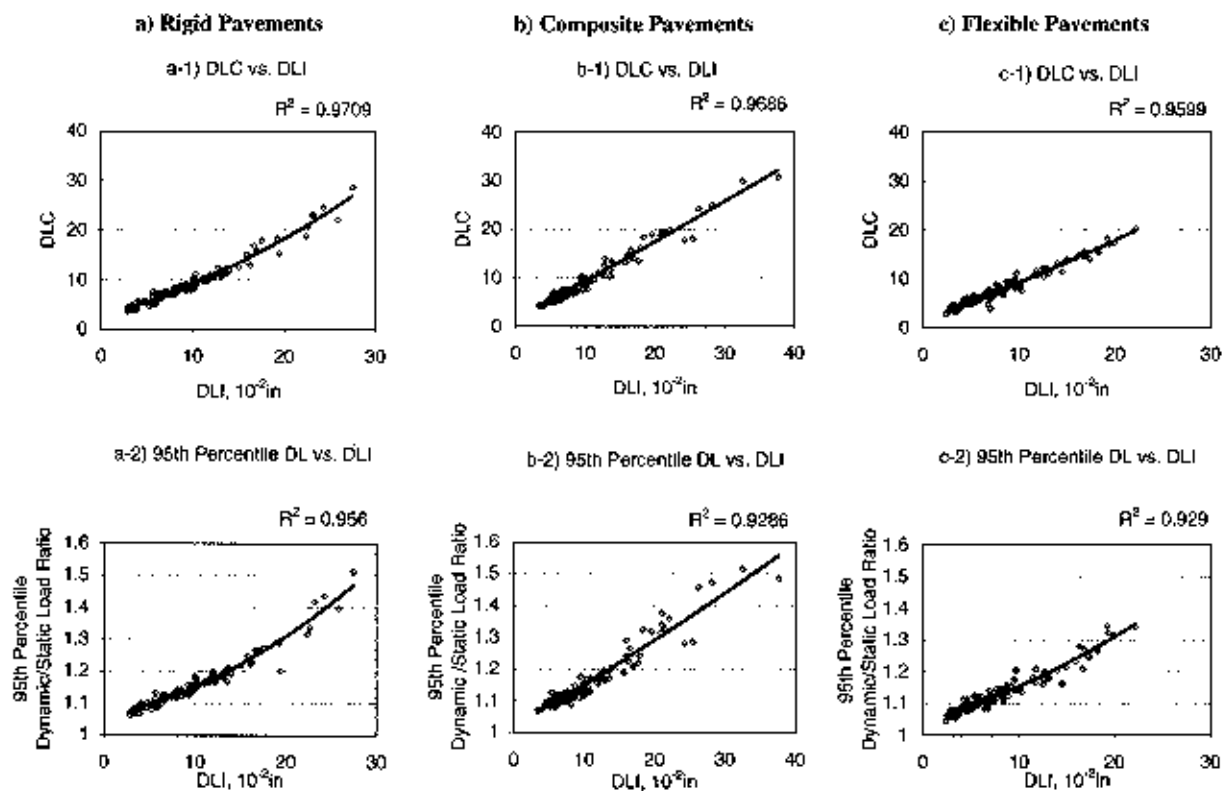


Figure 1 - Dynamic Truck-load Response vs. DLI for Each Pavement Type

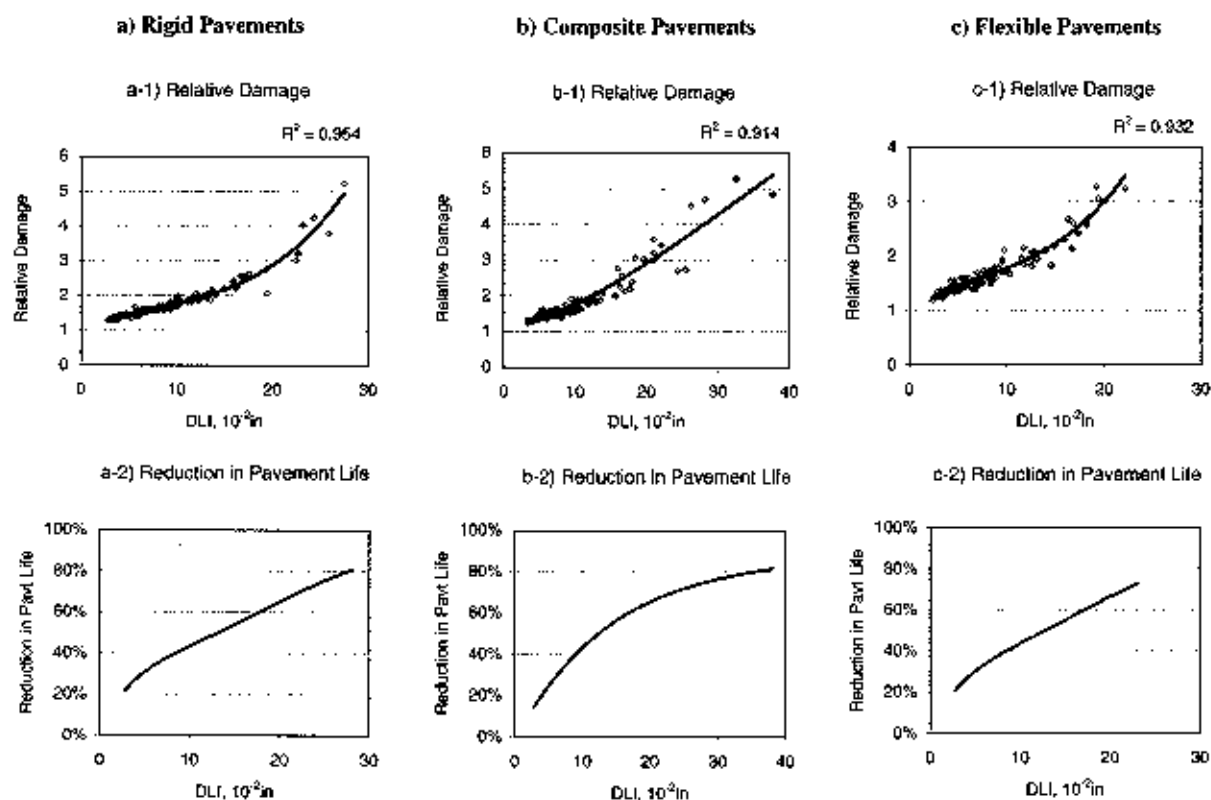


Figure 2 - Relative Damages from the 95th Percentile Dynamic Load as a function of DLI for Each Pavement Type

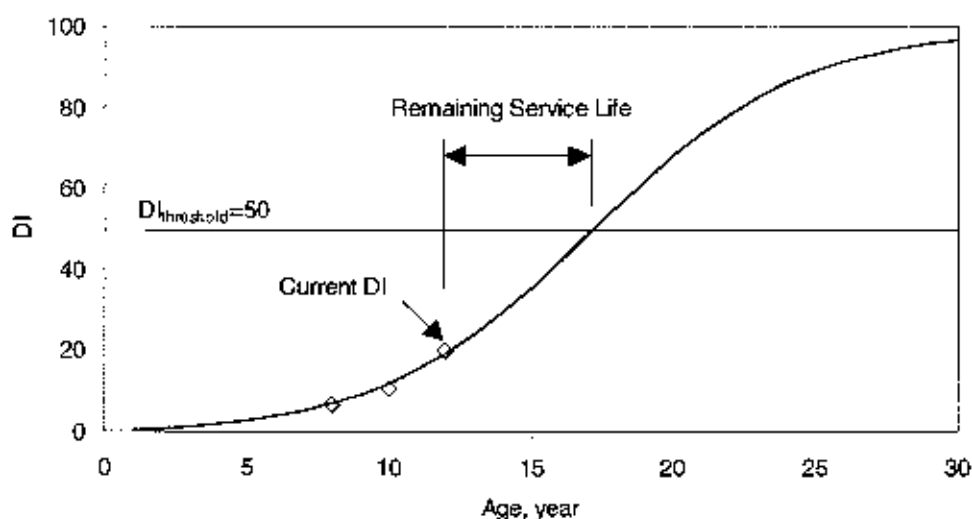


Figure 3 - Illustration of Remaining Service Life Calculation

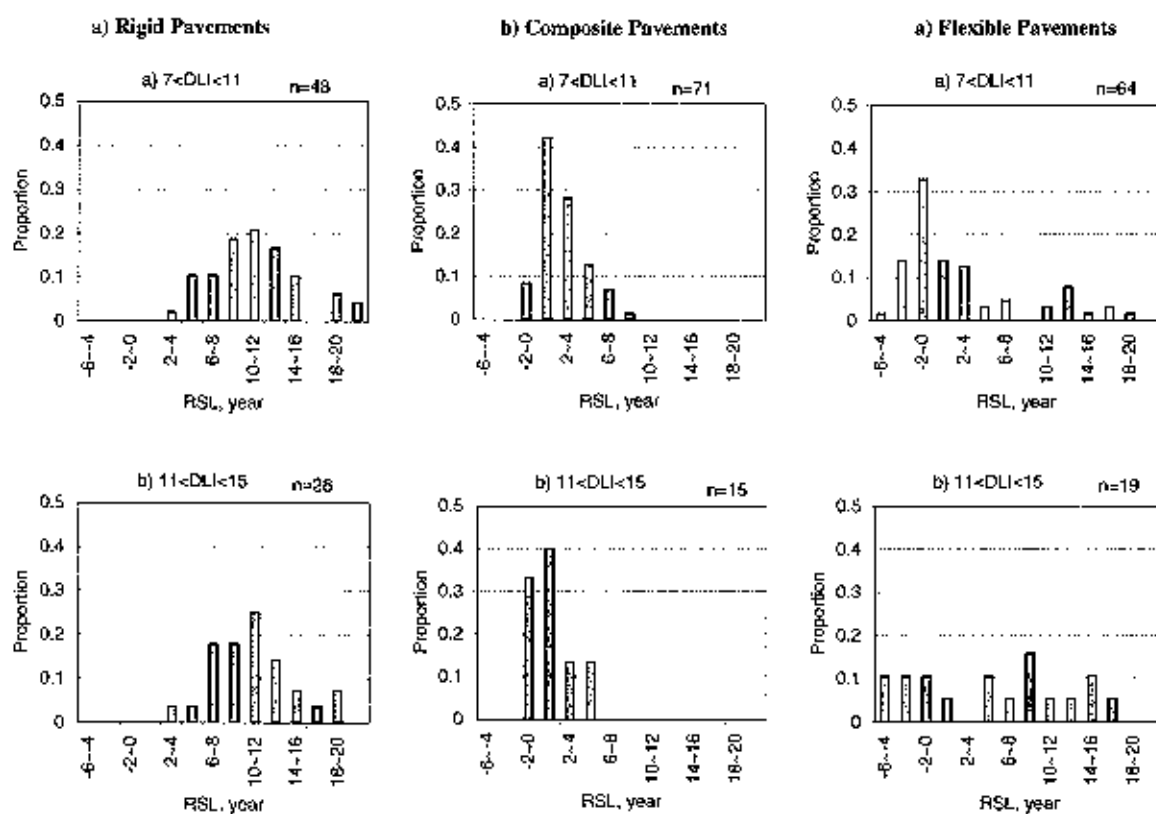


Figure 4 - Remaining Service Life Distributions for Each Pavement Type

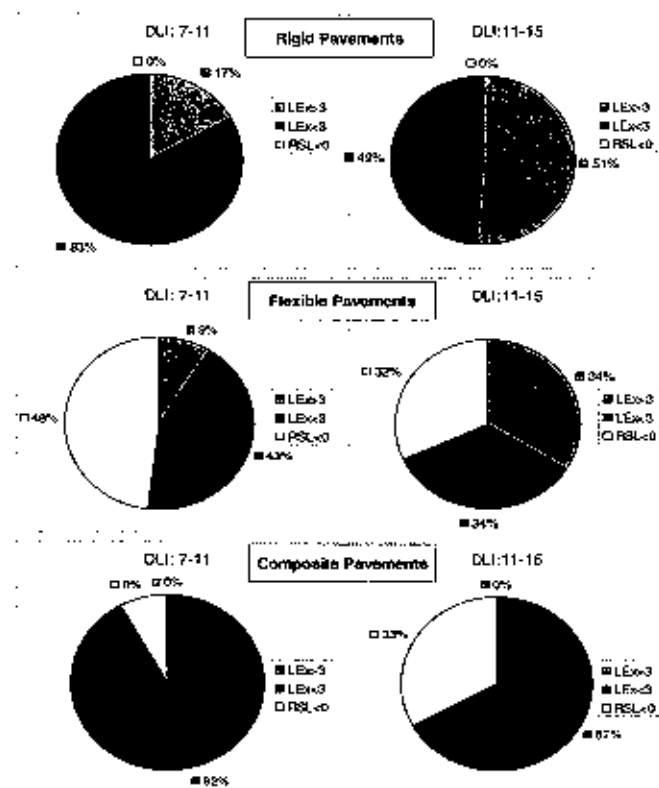


Figure 5 – Relative Distributions of Life Extension for Each Pavement Type

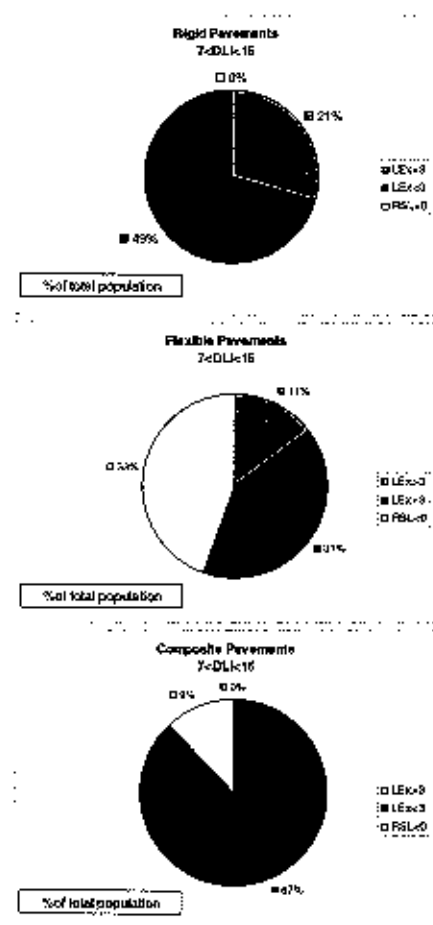


Figure 6 – Distributions of Pavement Sections with Different Life Extensions for Each Pavement Type

TRUCK TYRE WEAR ASSESSMENT AND PREDICTION

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ABSTRACT

Tyre wear is a complex phenomenon. It depends non-linearly on numerous parameters, like tyre compound and design, vehicle type and usage, road conditions and road surface characteristics, environmental conditions (e.g., temperature) and many others. Yet, tyre wear has many economic and ecological implications. The possibility to predict tyre wear is therefore of major importance to tyre manufacturers, fleet owners and governments.

Analogous observations can be made for road wear prediction due to the high road maintenance costs and traffic safety implications. Tyre and road wear are strongly related; the energy that wears the road is the energy that wears the tyre. There is therefore much to gain from an integrated approach to studying the mechanisms behind both wear phenomena.

Based on these observations, in 2000 we started the three-year 5th framework EU project TROWS (Tyre and ROad Wear and Slip assessment). The results include tools to analyse tyre wear and road polishing. These will be combined in a suitable wear prediction environment.

This paper focuses on the followed methods and results sofar of TROWS for truck tyre wear. Several types of tests were performed to obtain insight in the mechanisms behind the truck tyre wear phenomenon. These include material tests on tyre compounds, carousel tests with truck tyres, and tests with an articulated MAN truck on a public road course in Italy. Truck tyre wear mechanisms are presented and explained in relation to results from the performed tyre wear tests. Some modelling activities are discussed, as well as the wear prediction environment (Prolinx) that was developed to integrate the vast amount of test and modelling results from partners working at different locations with different tools. The relation between tyre and road wear is briefly touched upon. A discussion on the remaining challenges in predicting tyre wear concludes this paper.

INTRODUCTION

Any project on tread wear that includes modelling can be conceived as a two-part program. In the first part the forces and moments transferred in the tyre contact patch, the tyre-road contact area, are determined for selected operational and vehicle conditions. The second part is modelling what happens in the contact patch in terms of friction and tread wear behaviour due to the tyre loads that result from the selected operational conditions.

Part one may be complex but part two is significantly more difficult than part one. The main reason for this is that the tyre has to perform under a wide range of external world conditions. One does not have a controlled environment to conduct any model validation. Different pavements, different vehicle operations, different seasonal conditions (temperature, rainfall), different tread rubber compounds, all are responsible for this complexity of friction and wear behaviour in the contact patch.

An attempt to set up any mechanical model by running a simple road test after model building is bound to fail under more general applications. It may validate nicely for the situation at hand but at may fail to give any confirmation when run at a later time for a road test under different conditions. Unless the model is based on sound physical considerations on how external conditions influence the phenomena in the contact patch, it cannot predict tread wear given the wide range of operational conditions a tyre is subjected to during its live span.

The challenge is therefore to attempt to understand, as best as possible, the complexity of the phenomena involved; how do various tyre factors acting under a range of external factors influence tread wear. One of the demonstrated issues is the interaction of tyre factors with external factors. The outcome of the action of two or more factors is not a simple additive effect. The effect of any factor is a function of what other factors are operating and at what levels.

The EC 5th framework project TROWS (Tyre and Road Wear and Slip assessment) aims to gain insight into tyre wear and road polishing and the effects thereof on slip characteristics. Simulation models and test procedures are being developed that should allow tyre manufacturers to further improve the balance between tyre wear and other design factors, along with providing road manufacturers with improved design rules and maintenance plans for road surfaces.

The specific objective of the project is to make a significant step forwards in the prediction of tyre wear and road polishing, and to facilitate the wear prediction process using an integrated software environment for processing experimental and simulation results.

Veith ([1]) distinguishes several different kinds of (tire) wear mechanisms:

- adhesive wear – removal of material caused by high transient adhesion ('welding')
- abrasive wear – caused by cutting-rupture action of sharp angular asperities on the sliding counterface or as third bodies (particles)
- erosive wear – cutting-rupture action of particles in a liquid (fluid) stream
- corrosive wear – from direct chemical surface attack
- fatigue wear – caused by rapid or gradual material property changes that give rise to cracks and with their growth, a loss of material

Grosch ([2]) states that the wear of rubber is largely caused by fatigue.

One of the wear mechanisms studied in TROWS is that of truck tyres. This paper discusses the set-up and results of truck tyre wear tests in TROWS, and shows what tools are developed to be able to use the results in the prediction of truck tyre wear.

WEAR ASSESSMENT AND PREDICTION IN THE TROWS EC PROJECT

This section first gives a brief overview of the activities undertaken in the TROWS project, related to tyre wear. Next, the way tyre wear is assessed, is presented in greater detail. This is followed by an explanation of how the wear assessment results are used for the prediction of tyre wear, and how a truck model is essential to this prediction. The section ends with an introduction to the software application that was developed to manage test results and models for usage in the wear prediction process.

TROWS project summary

In TROWS, tyre wear is studied experimentally both under well-controlled laboratory test conditions and under real-life test conditions. Experiments are performed on three levels:

- Compound level. Three different tread compounds are tested on three different wear machines.
- Tyre level. One type of truck tyre is tested under controlled loading and road surface conditions on a so-called 'carousel'.
- Vehicle level. A truck-trailer combination and two passenger cars are driven on a well-characterised track of public roads in Italy.

Developed numerical models include vehicle models, models for the tyre structural behaviour and models to predict the (sliding) behaviour in the tyre-road contact area.

In the study of road polishing a similar subdivision is used for the experimental work: laboratory asphalt compound tests, carousel tests (with the road as main subject), and monitoring of the Italy public road track (i.e., collection of data on road lay-out, road surface changes, traffic density, etc.).

The results of TROWS will include experimental and numerical tools to analyse tyre wear and road polishing. These will be combined in a suitable software environment for wear prediction allowing tyre manufacturers to study more design alternatives at lower costs and thus to come closer to the most optimal compromise. Vehicle manufacturers will be able to improve their suspension design for tyre wear reduction for new vehicles by analysing changes in tyre loading due to changes in suspension compliance for a given vehicle usage profile. Road pavement and policy experts will be able to find a better compromise between skid resistance and other road characteristics (e.g. noise and traffic polishing effect) through better road layers techniques.

Overall approach to tyre wear assessment and prediction

A complex range of modelling and testing activities is covered in the project. This paper focuses on truck tyre wear, thus excluding the car tyre wear and road polishing aspects of TROWS.

Figure 1 shows the basic tyre wear approach adopted in TROWS. The left half of the figure shows the model-based approach, the right half the experiment-based approach. In both cases, vehicle usage profiles are necessary to characterise the typical usage of a vehicle, e.g. a passenger car in Northern Europe for shopping purposes is used differently than a truck in Southern Europe for professional freight transport. If sufficient data and knowledge is available, a tyre and vehicle model could be built to perform analyses of important manoeuvres leading to information on slip, stresses and deformations in the contact patch. Using these in a suitable wear law should give a wear rate for a specific manoeuvre and, in combination with the vehicle usage profile, a reasonable estimation of the total driven distance for a given minimum tread profile depth. On the other hand, if a vehicle is driven over a specific suitable reference course and the tyre wear is measured, this could be compared with the wear of a known tyre and then "scaled" to obtain a similar distance estimation for a given usage profile.

Obviously, both approaches have their pros and cons, allowing a more effective approach by combining them. The tyre wear prediction process requires the use of complex models, the handling of vast amounts of data, and the use of different kinds of software possibly running on different platforms. To manage and facilitate such a process, an efficient software environment has been developed (see Figure 2).

Tyre wear assessment

Tyre compound tests

Tyre compound tests were performed by rolling or sliding truck tyre tread compound specimen over road or road-like surfaces under laboratory conditions. For this, three different types of abrasion testing machines are used. These include two machines where specimen wheels are worn by rolling either over the side or over the outside of a disk of road material. A linear sliding machine is used to study the wear of a block of tread compound material sliding over a piece of material with road surface properties. To illustrate this laborious task an example of tests on one of the machines is given in Figure 3. The figure shows the well-known IAT100 abrasion tester, on which tread compound specimen wheels were tested, rolling a distance of 11,500 m at 23 km/h on two different surface types, at one slip angle (2°), and at loads varying from 80 to 140 N. The used surface types were chosen such that one of them was comparable to the road surface used in the carousel tests. For comparison, the other surface type was chosen well outside the range of surface types used in the carousel tests.

Carousel tests

Well-controlled carousel tests were performed, where eight truck tyres were mounted in total on a four-arm carousel (Figure 4). The main purpose of the carousel tests was to study road polishing. Yet, the mounted tyres could also be studied for tyre tread wear. For each of the tyres, load and slip angle were set within a range of 30 to 37.5 kN and -4 to $+7^\circ$, respectively. The carousel radius at the wheel position was 19 m. Two tests of 200,000 rotations were completed at 6.5 rotations per minute for the first 10,000 rotations and at 10 rotations per minute for the next 190,000 rotations. Tread depth was measured before the tests and tyre wear was studied after 10,000, 30,000, 50,000, 100,000, 180,000, and 200,000 rotations. Before the last 20,000 rotations the loading was increased significantly to increase the road polishing effects. This was done using a different set of tyres, because this severe testing ruins the tyres completely. Three different road surfaces were tested, where two surfaces were tested simultaneously by dividing the circular test track in two concentric parts.

Vehicle-on-the-road tests

Truck tyre wear under realistic conditions was studied by actually driving around on public road. This required hundreds of hours driving, especially for long lasting truck tyres. To obtain a suitable result with a reasonable effort a quite aggressive road course in Italy has been chosen. The selected road course, termed the Cisa course, is shown in Figure 5.

The Cisa testing track was chosen to represent typical European roads and traffic situations as well as several different types of road surfaces. The total length of the route, which is close to Parma, is 134 km and the estimated lap time was about 2 hours. Test series of 25,000 km were performed, one in the winter and one in the summer.

The Cisa track has been used for both the passenger car testing (two Peugeot 406 cars driving as a platoon) and the truck testing (see Figure 6). Tyre wear rates and wear patterns (i.e., wear rate across the tyre width) were monitored by measuring tread depths across the tyre width after each 5,000 km of driving. Figure 7 shows that, for truck rear (middle panel) and front tyres (right panel), tread depths were measured at six and four positions across

the tyre width, respectively. Tread depths at each position across the tyre were averaged for four positions along the tyre circumference (left panel).

The track was divided in eight sections for pavement characterisation and location referencing ([4]). It was characterised on a microtexture level (height < 0.2 mm, length < 0.5 mm), a macrotexture level (0.1 < height < 20 mm, 0.6 < length < 60 mm), and on a megatexture and unevenness level. Measurements included laboratory tests on asphalt compounds and road measurements using the SCRIM (micro/macrotexture), RUGO (macrotexture) and ARAN (Ruts, Curves, Unevenness, Slopes, Video) methods.

Wear prediction

Tyre wear prediction takes place on different levels. On the most detailed level, local rubber wear in the contact patch is calculated. This is done by calculating frictional power dissipated in the contact area and then determining the corresponding local wear by applying a frictional power-wear relation for the tread compound. The frictional power-wear law is determined experimentally from tread compound measurements. Frictional power in the contact area for different manoeuvres is calculated with a detailed FEM-like tyre model.

On a less detailed level, tyre wear for a complete tyre mounted on a truck, driving a particular route is calculated by assuming that tyre wear is proportional to the work done by the contact forces. Each section of a particular route can be assigned an 'wear aggressivity' value, where aggressivity is defined as the work per unit length and unit weight, averaged over that particular road segment. It must be noted that the aggressivity depends on vehicle type (total weight and weight distribution), type of wheel (drive or carried wheel, left or right wheel), road class (vehicles speed, minimal value of the bend's radius), and type of road element (straight section, bends, transitions). This means that, in order to calculate the aggressivity, a sufficiently detailed vehicle model is required. To be able to determine the corresponding wear value, an aggressivity-wear relationship has to be determined.

Truck modelling

The articulated MAN F2000 truck has been modelled using the multi-body simulation package ADAMS. The main modelling attention was on the front, rear and trailer suspensions. The front suspension comprises the rigid axle, the roll-bar and the steering system. Some truck testing has been done on a well-defined track for validation of the multi-body model. Two test series were performed: one with the traction unit alone and one with the trailer attached. The manoeuvres included acceleration and braking on a straight lane, circle tests ($R=44$ m, 60 km/h) and a double lane change, also at 60 km/h.

The Prolinx wear prediction environment

The wear prediction process consists of a combination of performing experiments, data processing, modelling, simulation and reporting. These tasks are performed by different partners all over Europe. Apart from the technical details, managing and executing the different steps in this project's technical process is quite a challenge. To be able to manage and reproduce the tyre and road wear prediction processes, a clear view is necessary on the following issues:

Responsibility: who performs which process (e.g. measurements, calculations, simulations etc.)?

Integration: how can we link and execute the various processes in a manageable way?

Data flow: which data is transferred between different processes?

Data consistency: is the generated or received data valid?

Reproducibility: can all results be reproduced unambiguously using a well-defined starting point?

Transferability: can the knowledge about a process be transferred efficiently to different people?

Prolinx was developed by TNO Automotive to facilitate defining and implementing the solutions to these questions. See Figure 2 for a glance on Prolinx' user interface. Using Prolinx, the partners can keep track of their sub-process, its related documentation, and the processes on which they depend. Prolinx displays the hierarchy and data flow of the wear prediction process, keeps track of the validity of exchanged data, and performs the necessary calculations to generate valid results, e.g. by calling programs like Matlab/Simulink or ADAMS. Prolinx has a generic set-up, so it can also be used for mathematical processes other than tyre or road wear prediction.

The software application thus serves as a guideline for the TROWS partners. Each partner knows exactly which steps to perform to generate their contribution to the wear prediction process. Once implemented in Prolinx, the wear prediction process can be reproduced unambiguously at any time using different parameters, conditions,

usage patterns, measurement data et cetera to perform wear prediction calculations for different types of tyres and road surfaces.

TRUCK TYRE WEAR MECHANISMS AND PRELIMINARY RESULTS

The objectives of the TROWS project for tyre wear prediction are limited to regular wear. Other types of wear, like polygonal wear ([3]) and heel-and-toe wear are considered out of the scope of the models constructed in TROWS. In this section, first a brief general overview is given of truck tyre wear phenomena. Next, preliminary results of the tests will be presented and, finally, it will be shown how the truck model is used in the prediction of tyre wear.

General truck tyre tread wear phenomena

Truck-type vehicles can be subdivided into several categories or segments for which the related tyre wear varies from low to severe. The lowest tread wear can be observed on long-distance transport vehicles on motorways or major roads. The most severe tread wear is found on vehicles used in extreme off-road conditions on all types of terrain (loose surfaces, unpaved roads, mud, grass, and sand). This typically holds for vehicles for military use. The truck-trailer combination studied in the TROWS project falls in the long haulage segment, which is close to the first category.

In general, the vehicles are used at maximum transport capacity, so that truck tyres always work at the maximum load. Figure 8 shows the load distribution on the axles of a typical 40-ton truck-trailer combination, the combination used in the TROWS project for the study of truck tyre wear. Note that the load is approximately equally distributed over the combined tractor and trailer axles (each 20 ton).

The studied truck typically operates on a motorway surface type. The tyres are usually fitted in the autumn and are used for one year. Typical wear rates for the front and rear axles of the truck are:

- *front axle*: from 30,000 km/mm tread depth loss to 15,000 km/mm tread depth loss
- *rear axle*: from 15,000 km/mm tread depth loss to 10,000 km/mm tread depth loss

Besides the difference in front (steered) and rear (driven) axle, wear strongly depends on axle load.

The following paragraphs present the general characteristics relevant to tyre wear for the different types of axles on the typical truck shown in Figure 8.

Front axle characteristics. The front axle is a free rolling steered axle, where only braking torques and slip forces are applied. The lateral acceleration is zero for most of the truck mileage. The outer shoulders of the front axle tyres wear significantly more than the inner shoulders. This is due to the positive camber angle of 0.6 to 0.7 degrees that is applied to both of the front axle tyres. The tread pattern is always formed by longitudinal grooves. The ribs are quite often covered by transversal narrow sipes to ensure a more even wear distribution. The left front axle tyre wears faster than the right front axle tyre, due to the combination of the applied camber angle and the profile of the road (road banking).

Rear axle characteristics. The rear (truck) axle is the driven axle. The wear of the rear axle tyres is higher than that of the front axle tyres, because of high accelerating and braking forces applied. Due to the flexion of the rear axle, the inner shoulders of the inner tyres on both sides wear significantly more than the inner shoulders of the outer tyres. The tread pattern is formed by blocks with deep longitudinal and transversal grooves. This leads to a better traction performance. The use of braking systems like retarders can apply a constant braking couple, thus improving the mileage in km/mm tread depth loss. Due to slippage of the backward part of the tread blocks, the rear axle tyres typically show so-called heel-and-toe wear effects. This means that the backward side of tread blocks is more worn than the forward part.

Trailer axle characteristics. The three axles on the trailer have a different behaviour depending on the type of the trailer settings. Yet, all three axles are generally subjected to a high level of lateral forces. On some trailers, the first axle can be lifted, which explains why the tyres on these axles can show less wear than those on the other trailer axles. The tyres on the second axle wear significantly less than those on the first and third axles. This is due to the fact that the second, central axle spins around its centre, and therefore lower lateral forces act on these tyres than on the tyres of the first and third axles. Due to the possibly high lateral forces, the tyres of the first and third trailer axle show a higher degree of chipping and chunking of the tyre rubber. The third axle of the trailer may be a

steering axle, which reduces lateral forces of the tyres and thus tyre tread wear. All the trailer tyres show rounded shoulders and ribs edge.

Preliminary results of truck tyre wear assessment in TROWS

Truck tyre wear mechanisms were studied in three ways:

- by tyre compound measurements
- by controlled carousel tests
- by driving on a well-defined track of public roads in Italy.

Only regular wear was studied. Other types of wear, like polygonal wear ([3]) and heel-and-toe wear were not taken into account. Preliminary results for each of the three wear studies are presented.

Tyre compound tests. Three different types of abrasion testing machines were used to study the wear of the truck tyre tread compound. With these machines, similar compound material was tested under different local abrasive regimes. It was found that comparison between the various results could be done by determining the wear (as mass loss per unit contact area) as a function of frictional power per unit contact area. Figure 9 shows the combined results of the tests on the three different machines, as obtained for two different types of truck front tyres and two different types of truck rear tyres. At least two observations can be made from this figure. Firstly, it can be seen that the wear – frictional power relation is non-linear. Secondly, comparison of the left and right panel shows that the wear of the compound of the truck front tyre is similar to that of the compound of the rear tyres at lower values of frictional power, but is about 50% at higher frictional power values.

Carousel tests. Figure 10 shows two tyres after wear tests on the carousel, for two different combinations of applied load and slip angle. The left panel shows a commonly observed phenomenon for the carousel tests: a tyre that has caught up worn rubber from other tyres in the same track or from previous rotations. This testing artefact prohibits the construction of realistic slip-load-wear relationships to be used in tyre wear prediction.

The right panel in Figure 10 shows a tyre that was realistically worn. Although only the minority of the tyres used on the carousel showed realistic wear behaviour, several satisfactory slip-load-wear relationships could be constructed for them.

Driving on well-defined track of public roads.

Figure 11 shows the results of the tests on the Cisa track in terms of tread depth loss (in mm) as a function of position across the tyre width. The figure shows a very high level of tread wear rate. The average truck tyre wear rate on the Cisa track was about twice as high as that observed in the carousel tests and about four times higher than reference levels. The wear profiles (wear rate across the tyre width) can be seen to be similar for the winter (blue lines) and the summer tests (red lines).

Figure 12 shows a comparison of the observed summer (red) and winter (blue) 'mileage' in km/mm tread depth loss. It can be seen from this figure that the mileage was higher in the summer than in the winter tests. In other words, the tyres have worn more in the summer than in the winter. The figure also shows that the left front tyre is more worn than the right front tyre. As explained in the section on the general truck tyre tread wear phenomena, this is due to the combination of the applied camber angle and the profile of the road (road banking). That section also explains the observation that the internal rear tyres wear more than the external rear tyres, as well as the observation from Figure 11 that the front tyres wear more on the sides than in the middle. Similarly, pairs of left or right rear tyres wear more on the outside than in the middle, as can be seen from this figure.

Truck model validation results

Measured and computed eigenfrequencies up to 2 Hz were found to be within 15% accuracy and the time responses for the manoeuvres were also represented sufficiently well. The maximum lateral acceleration during the double lane change was approximately 3 m/s^2 whereas the lateral acceleration measured on the truck while driving on the Cisa track was under 2 m/s^2 for 98% of the whole on-the-road test. The model therefore seems sufficiently accurate for the calculation of hub forces as input of further tyre analyses.

CONCLUDING REMARKS AND FUTURE WORK

One of the goals of the TROWS project is to be able to predict tyre tread wear. The approach has been presented in this paper as well as some measurements on truck tyres and their tread compounds. Also the vehicle modelling has been touched upon briefly.

The tests on the Cisa track showed very high wear rates compared to the wear rates found in the carousel tests and those found in reference tests. This means that the Cisa track can be regarded as much more aggressive to the tyres than the average track to be expected in Italy. This is corroborated by the results of the passenger car tyre tests on the Cisa track. They also showed much higher tread wear rates than found on average public roads.

The usage of the current multi-body vehicle models built in ADAMS require a large computer time. By using the TNO Automotive tool ADVANCE this computation time can be reduced significantly. ADVANCE is a tool especially developed to assist in the design of vehicle intelligent systems and even contains capabilities for real-time simulations (see Figure 13). It is a toolbox built on top of Matlab/Simulink, allowing full Matlab/Simulink functionality, and contains predefined powertrain and chassis component models, controllers and driver models. The chassis module contains among others wheel suspension models, steering system and tyre models. The modelling capabilities are therefore sufficient to build efficient truck and passenger car models that fits perfectly in the Prolinx environment.

The tyre modelling activities undertaken in TROWS have not been presented in this paper. The current status is that suitable models for the dynamic tyre behaviour as well as a contact patch discretisation have been realised and validated as far as possible. The current attention is on the physical laws to be used in the contact patch analyses. A crucial law is the compound wear law. In this paper some first results for a potentially promising relation between frictional power and mass loss per unit area has been shown. The main focus of the last project year will be to bring all models and data together to relate the various aspects efficiently in the Prolinx environment. This will allow an effective evaluation of the different concept laws for the different experiments performed. If a reasonable correlation is found, the Prolinx environment will be used for a further fine-tuning of the different building blocks of the wear prediction process.

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FIGURES

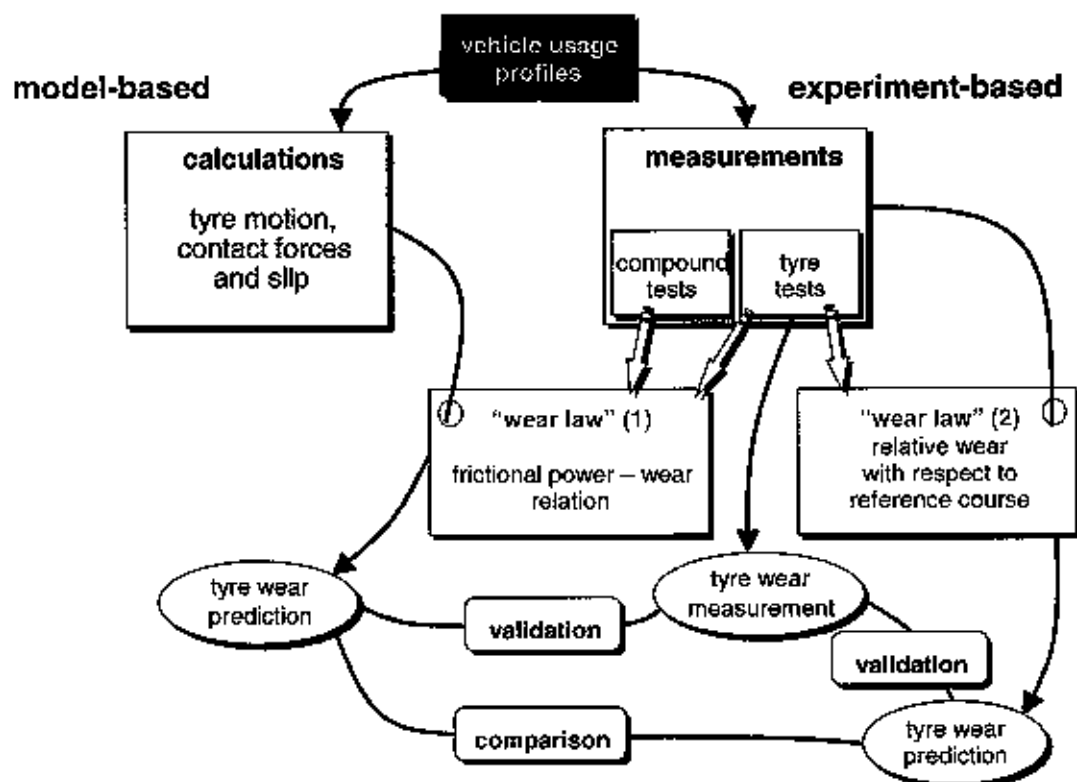


Figure 1 – TROWS workflow for tyre wear assessment, modelling and prediction.

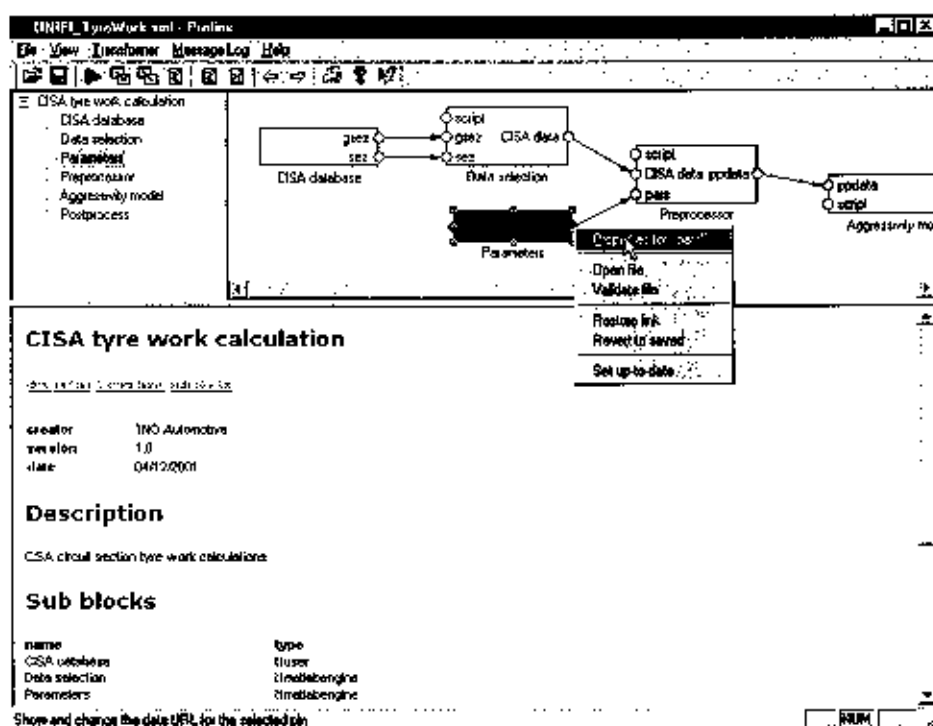


Figure 2 – A view on the Prolix environment developed to predict tyre and road wear in a structured and well-documented fashion.

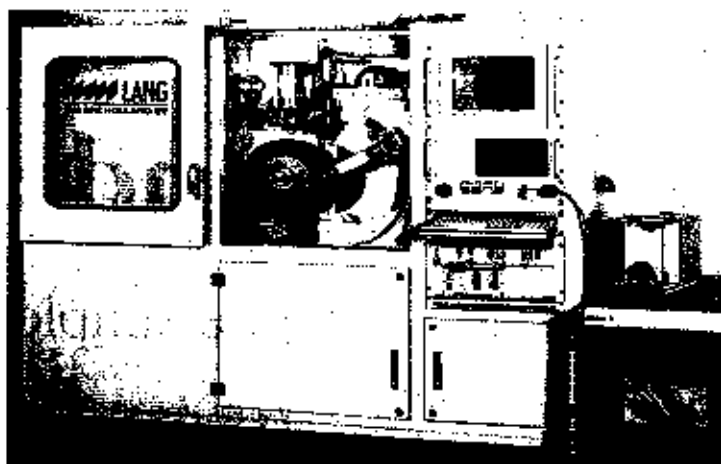


Figure 3 – The LAT100 compound testing machine.



Figure 4 – Carousel test set-up (4 arms of 19 meters long; two truck tyres mounted at each arm).

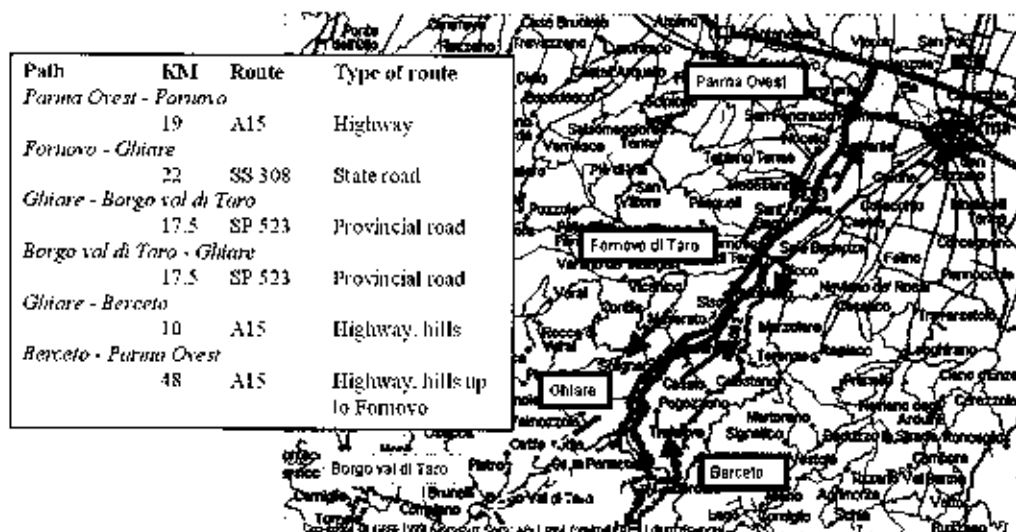


Figure 5 – TROWS tyre wear testing course.



Figure 6 - TROWS test vehicles: MAN articulated truck and Peugeot 406.



Figure 7 - Truck rear (middle panel) and front tyres (right panel) tread depth were measured at six and four positions across the tyre width, respectively. Tread depth at each position across the tyre were averaged for four positions along the tyre circumference (left panel).

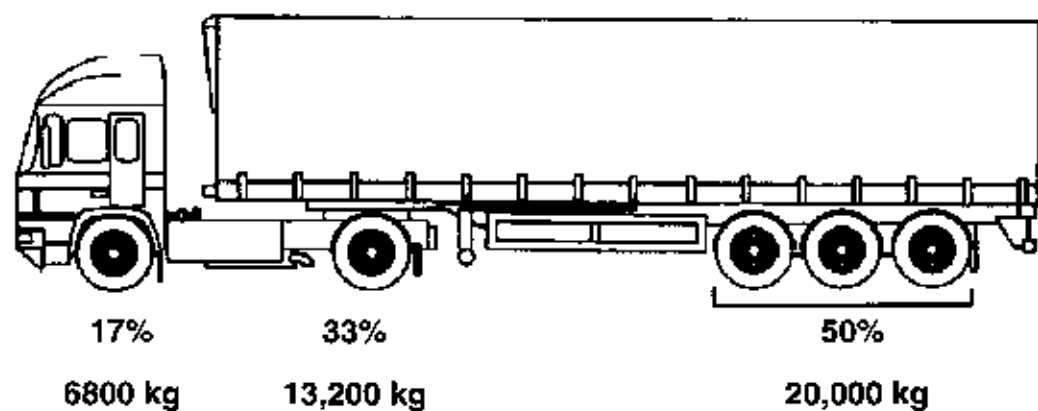


Figure 8 - Load distribution on a typical 40 ton truck.

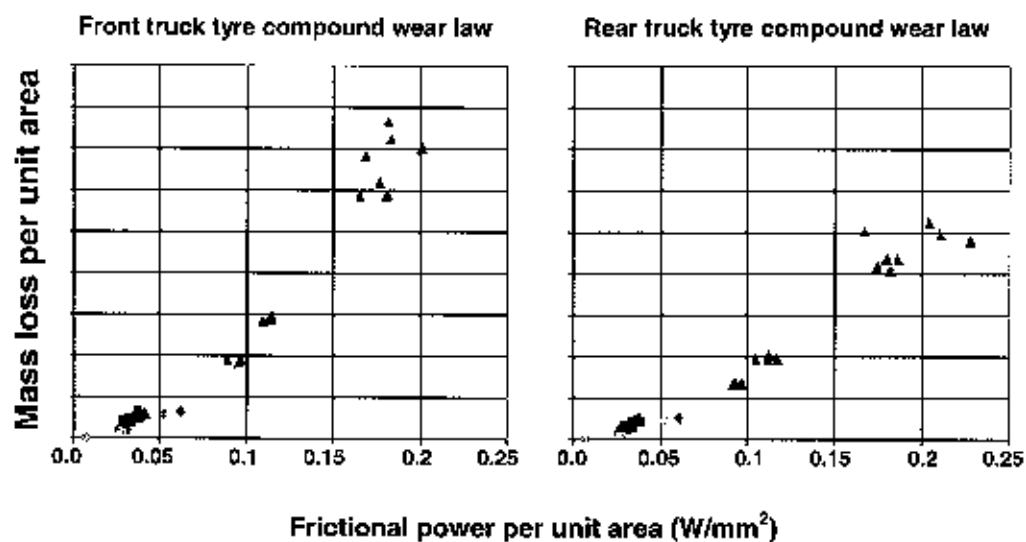


Figure 9 – Wear laws for front (left panel) and rear (right panel) truck tyre treadwear compound.

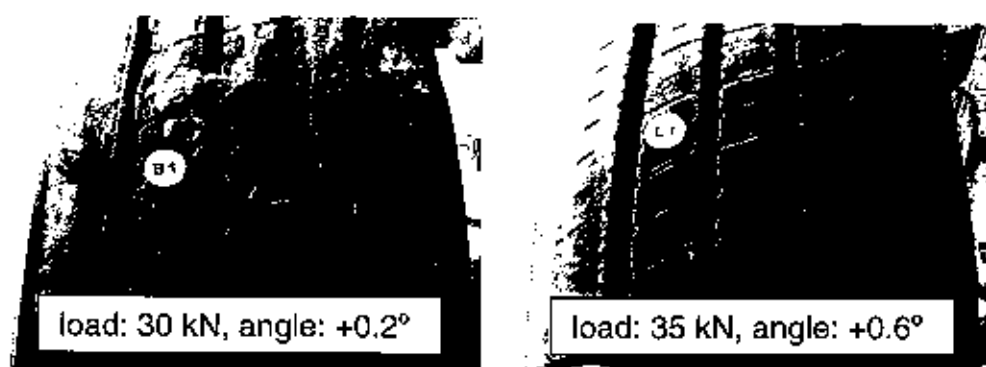


Figure 10 - Truck tyres after carousel tests for different applied loads and slip angles may have become worthless for wear measurements due to clotting of debris (left panel) or have worn in a realistic fashion (right panel).

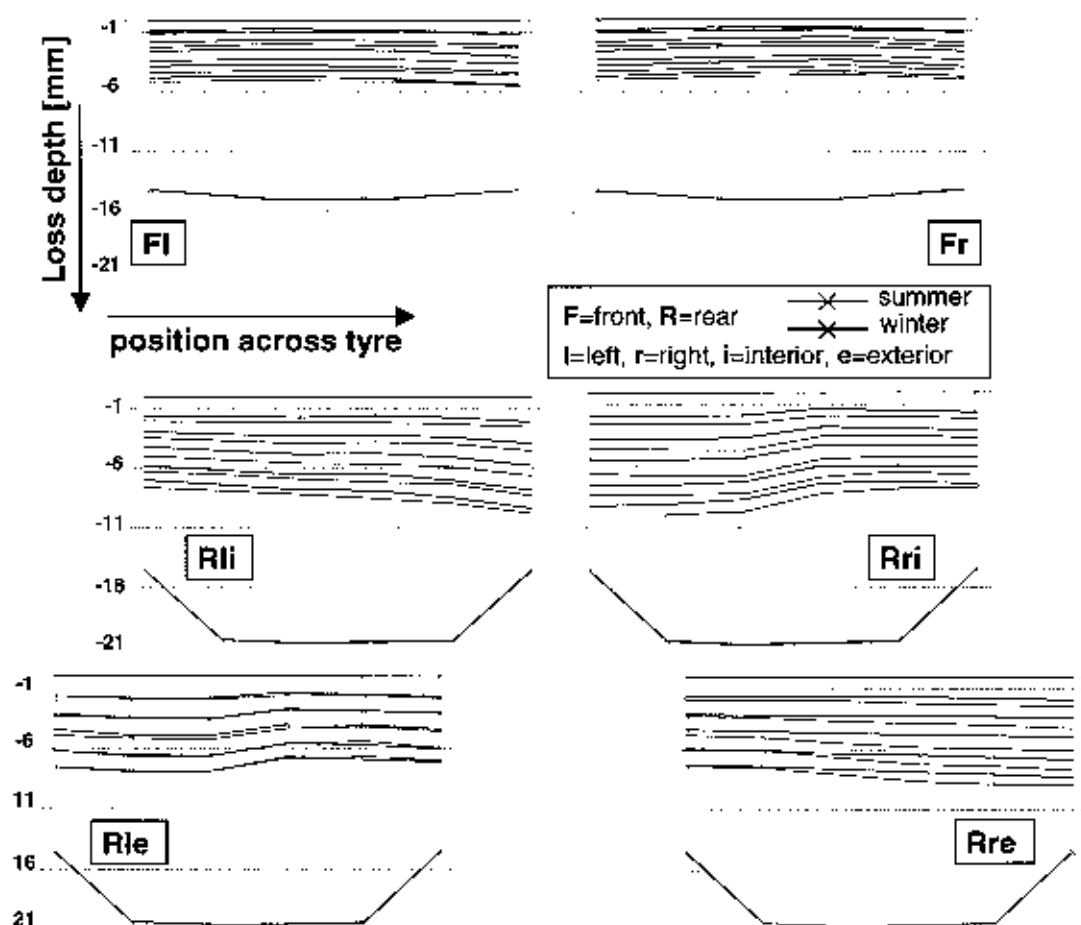


Figure 11 – Tread loss depths across the tyre width for all truck tyres as measured in summer test (red lines) and in winter tests (blue lines). Lower lines represent maximum possible depth loss before the tyre getting slick.

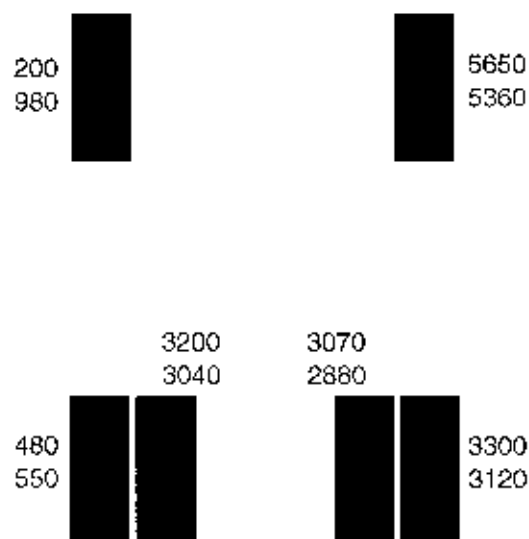


Figure 12 – Comparison of summer (red) and winter (blue) 'mileage' (km/mm tread depth loss) observed from the truck tyre wear tests on the Cisa track.

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In order to obtain a better insight in why expansion joints fail, dynamic loads on the road near expansion joints in bridges and dynamic responses in the construction of modular expansion joints are measured and evaluated. For this purpose, a test vehicle has passed these expansion joints at various velocities. Both dynamic load of the vehicle and strains in the expansion joints were measured.

Modular expansion joints are applied in many recently built bridges. These expansion joints form the connection between abutment and bridge, see figure 1. In this figure, an overview is given of the main construction parts establishing a modular expansion joint. The steel lamellas (center beams) transfer the traffic load to the cross beams, which are steel beams mounted in the bridge and in the abutment.

A commonly used design method for modular expansion joints in bridges is based on the calculation of stresses under quasi static loading. These stresses are multiplied by a dynamic amplification factor, which takes into account the roughness of the pavement near the expansion joint, and by a dynamic amplification factor, which takes into account the interaction between the time history loading and the dynamic response of the expansion joint, for which damping and eigen frequency of the expansion joint are important. In this paper, the first factor is referred to as dynamic amplification factor for loading, the second as dynamic amplification factor for interaction.

For current practice, Eurocode 1 part 3 (1991) prescribes a magnitude of 1.3 for the dynamic amplification factor for load. The dynamic amplification for interaction should be determined using equations prescribed in the Eurocode. Most manufacturers of expansion joints make such a design of the joint that the dynamic amplification factor is approximately 1. The total dynamic amplification is thus $1 \times 1.3 = 1.3$ for most designs.

In many bridges in The Netherlands, cross beams of modular expansion joints collapse long before the calculated life time, see figure 2. This is possibly caused by a dynamic increment in real bridges that exceeds the assumed magnitude in the design. Therefore, dynamic amplification factors were measured and analyzed with measurements for two bridges in The Netherlands, the bridge in motorway xx over xx near Zaltbommel: the "Martinus Nijhoffbrug" and the bridge in motorway xx over xx near Grubbenvorst: the "Noorderbrug". For these bridges, a dynamic amplification (load x interaction) of 1.4 is used in the design. This paper describes the measurements and analyses and the resulting dynamic responses.

An instrumented vehicle with two axles, weighing approximately 18 tons, crossed the expansion joints, see figure 4. Accelerometers were attached to the axles and chassis of this vehicle in order to determine the dynamic wheel load through a mass-spring-damper model. The vehicle model consists of two unsprung elements (front and rear axle) and two sprung chassis parts (front and rear) connected with a rotational joint to allow chassis roll. The model parameters are listed in table I. The input for the model are the eight measured accelerometer signals. The

output are the four wheel loads. The lateral position of the test vehicle corresponded to the lateral position of passages of normal traffic. A variation in lateral position of approximately 100 mm occurred for low velocities. For the measurements, strain gauges were attached on the under-surface of the cross beams of the expansion joint, see figure 3.

Measurements took place at a velocity of 5 km/h to determine the static response, and at a velocity of 50, 70 and 85 km/h to determine the dynamic response at various velocities. All measurements were repeated three times. During the measurements, the bridges were closed for other traffic.

The dynamic amplification factors for an expansion joint is determined in two steps: first, strains in the expansion joints are measured under a static loading. Strains divided by static loading result in the static transfer. Second, strains are measured under dynamic loading. The dynamic transfer is calculated as the dynamic strains divided by dynamic loading. The dynamic amplification factor for interaction is determined as dynamic transfer divided by static transfer. The dynamic amplification factor for loading is determined as the dynamic peak load divided by the static load.

DYNAMIC LOADING

Figure 5 shows accelerations measured at a velocity of 70 km/h for the "Noorderbrug". The upper picture shows accelerations at the axles while the lower picture shows accelerations at the chassis. The dynamic load is calculated from these accelerations by using a model of a two-mass-spring system per quarter of the vehicle, see [1]. The dynamic loads at the point where the vehicle crosses the expansion joint for the three measurements are shown in figure 6. As for this velocity, the repeated measurements for all velocities showed an extremely good similarity.

The dynamic amplification factor for loading is defined as the maximum dynamic load effect divided by the static load effect. The dynamic amplification factors for loading for various velocities for the vehicle used are shown in table II. Dynamic amplification factors for the bridge "Martinus Nijhoffbrug" are larger than for the bridge "Noorderbrug". The unevenness and roughness of the pavement near the expansion joint have significant influence on this dynamic amplification factor.

For calculation of the dynamic amplification factor for interaction, the load should be determined at the moment at which a strain gauge signal is maximal. While a small difference in time exists between the maximum signals for different strain gauges, the load differs for all strain gauges. Figure 7 shows the load for a measurement with velocity 70 km/h for the bridge "Noorderbrug". Dots indicate loads at the moment that a strain gauge signal was maximal.

The loads at the moment that the signal of strain gauge 2 is maximal are given in figure 8. It is shown that the three different measurements with the same velocity result in an almost equal load.

MEASURED STRAINS

Figure 9 gives an example of the strains measured at the heaviest loaded crossbeam of the bridge "Martinus Nijhoffbrug". The top picture shows results at a velocity of 5 km/h, the middle picture at a velocity of 85 km/h. For every measurement, the maximum strain and the range in strain (difference between maximum and minimum strain) are determined. Maximum strains for all measurements in the heaviest loaded crossbeam are shown in figure 10. Strains measured in cross beams 1 to 8 (numbers are referred to figure 2) are of the same magnitude, strains measured in cross beams 9 to 12 are a factor 5 to 10 lower. Figure 10 shows that the three different measurements with the same velocity result in an almost equal strain. In this figure, strains decrease for increasing velocity. However, this was not found for all cross beams.

At the bridge "Noorderbrug", strains vary slightly for the different measurements at low velocities. This is caused by a small change in lateral position of the vehicle, due to which some cross beams are loaded slightly heavier and others slightly lighter; the average strains measured at all cross beams show no significant difference between the measurements.

The eigenfrequencies and the damping of the crossbeams are determined using the strain gauge signal, see the bottom picture in figure 9. The eigenfrequency measured at the bridge "Martinus Nijhoffbrug" was $f = 139$ Hz. The damping ratio is defined as the ratio between actual damping and critical damping. This damping ratio is $\phi = 5.0\%$ for the bridge "Martinus Nijhoffbrug". For the bridge "Noorderbrug", eigenfrequency and damping ratio are $f = 132$ Hz and $\phi = 2.3\%$, respectively. Measured damping ratios are much lower than used in the design calculation for these expansion joints ($\phi = 10\%$), which means that much more load cycles with low load occur than assumed in the design. The lower damping ratios are lower than expected, probably because the damping decreases during lifetime.

INTERACTION BETWEEN LOADING AND DYNAMIC RESPONSE OF JOINT

The dynamic amplification factor for interaction indicates the change in stresses at a change in velocity when loading remains constant. It is calculated as the ratio between dynamic transfer (strains divided by load under dynamic loading) and static transfer (strains divided by load under static loading). The dynamic amplification factors for interaction at various velocities presented here are the average of the three measurements.

Dynamic amplification factors for interaction for loading by the front axle are shown in figure 11 for the bridge "Martinus Nijhoffbrug". Maximum strains measured in cross beams 9 to 12 have a magnitude of 10 times lower than the heaviest loaded cross beam. Although the dynamic amplification factors for interaction for these strain gauges are relatively high, the resulting dynamic strains remain low. Therefore, the dynamic amplification factors for interaction for these cross beams are of less significance and are not shown in figure 11. The weighing average of the dynamic amplification factors for interaction is defined as the sum of the products of dynamic increment factors and strains for all strain gauges divided by the sum of the strains of all strain gauges. The weighing averages for maximum strains and range in strains at loading by the front or the rear axles for all velocities are listed in table III.

The maximum dynamic amplification factor for interaction for strains in cross beam 5 was 1.96 for range in strains, which occurred for loading by the front axle at a velocity of 85 km/h. This means that the maximum strain for a dynamic load is nearly twice the strain for a static load, when the load magnitude on the expansion joint remains equal.

Dynamic amplification factors for range in strains at loading by the front axle are shown in figure 12 for the bridge "Noorderbrug". Maximum strains measured in cross beams 5 and 6 have a magnitude of 3 times lower than the heaviest loaded cross beam. The weighing averages for maximum strains and range in strains at loading by the front or the rear axles for all velocities are listed in table IV.

The maximum dynamic amplification factor for interaction for strains in cross beam 3 was 1.57 for range in strains for loading by the front axle at a velocity of 85 km/h.

FATIGUE LIVES FOR ORIGINAL DESIGN AND USING NEW PARAMETERS

The highest measured amplification factors for loading and for interaction were 1.7 and 1.9 respectively. However the moment that the strain gauge gave a maximum signal did not correspond to the moment at which the peak in the load occurred. The product of the actual dynamic load times the amplification factor for interaction was maximal 1.57 but for most cross beams this product was approximately 1.0. In the design, 1.4 was assumed for the product of the two amplification factors. The measured damping ratio was 2.3% and 5%, while 10% was assumed in the design.

Fatigue calculations have been carried out with different values for damping and products of the dynamic amplification factors. In these calculations, the load cycles of the cross-beams were determined for loading by 1000 axles. Equivalent static axle loadings were calculated using the magnitude of the load cycles and the gradient of the logarithmic s-N curve of 3. Results are shown in table V. The equivalent axle loadings, fatigue damage and fatigue lifetime for calculation with a damping and a dynamic amplification factor used in the original design is used as a base. It is shown that lower damping and higher dynamic amplification factor leads to a much lower calculated life time.

CONCLUSIONS AND RECOMMENDATIONS

With the measurements carried out, a good insight has been obtained in dynamic impact loads on the road near expansion joints in bridges and dynamic responses in the construction of modular expansion joints.

All measurements at a specific velocity were repeated three times. Both strain gauge signals and dynamic loads showed extremely good similarity between these three measurements, indicating that the followed measuring procedure is a reliable method for obtaining information on dynamic amplification factors for traffic loading and dynamic amplification factors for interaction between time history loading and dynamic response of the cross beams in expansion joints. Dynamic amplification factors for loading and for interaction vary widely for different velocities and different cross beams. The highest measured dynamic amplification factors for loading and for interaction were 1.7 and 1.9, respectively. The highest difference in measured strains between static loading and dynamic loading was xx, while 1.4 was assumed in the design. The measured damping ratio were 2.3% and 1.5%, while 10% was assumed in the design.

The dynamic amplification factors for loading and for interaction and the damping play a large role in the fatigue life of the expansion joints. A higher occurring dynamic amplification factor (1.8 instead of 1.4) leads to higher amplitudes in the response functions and result in a calculated fatigue life that is half the originally determined fatigue life of the structure. A lower occurring damping ratio (2.5% instead of 10%) leads to more load cycles and a higher dynamic amplification factor for interaction and results in a calculated fatigue life that is 40% of the originally determined fatigue life of the structure. A combination of higher occurring dynamic amplification factor and lower damping leads to a calculated fatigue life that is 20% of the originally determined fatigue life of the structure.

The dynamic loading is a combination of axle load and dynamic effects. Different vehicles with different axle loads may cause other dynamic amplification factors. The realistic dynamic amplification factor is defined as the maximum probabilistic dynamic load divided by the maximum probabilistic static load. For the maximum probabilistic static load, reference is made to the paper [2]. It is recommended to carry out measurements with other vehicles to obtain a complete range of dynamic amplification factors for realistic traffic, with which it is possible to obtain a maximum probabilistic dynamic load.

The dynamic increment factor depends on the geometry of the expansion joint and on dynamic loading. It is recommended to carry out measurements on other bridges to obtain a complete range of appearing dynamic increment factors.

In designs, it is recommended to take into account the actual dynamic interaction between loading and response. A damping ratio of 10% is not appropriate for life time calculations, because the damping decreases during lifetime.

Although only based on these two measurements carried out, the assumed dynamic amplification factor for loading in Eurocode 3 of 1.3 seems to be too low.

REFERENCES

1. J. Maljaars, P.H. Waarts, R.B.J. Hoogvelt, *Metingen aan de voegovergangen bij de brug bij Zaltbommel en Grubbenvorst*, TNO report 2002-CI-R1054, 2002.
2. A.C.W.M. Vrouwenvelder, P.H. Waarts, Traffic loads on bridges, Structural Engineering International, IABSE, August 1993.

TABLES & FIGURES

Side view



Top view of the modular expansion joint

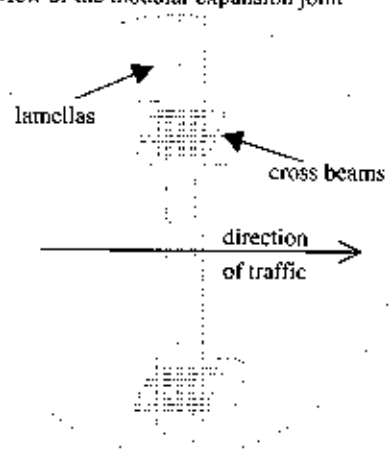


Figure 1 - Modular expansion joint in the bridge "Martinus Nijhoffbrug", The Netherlands

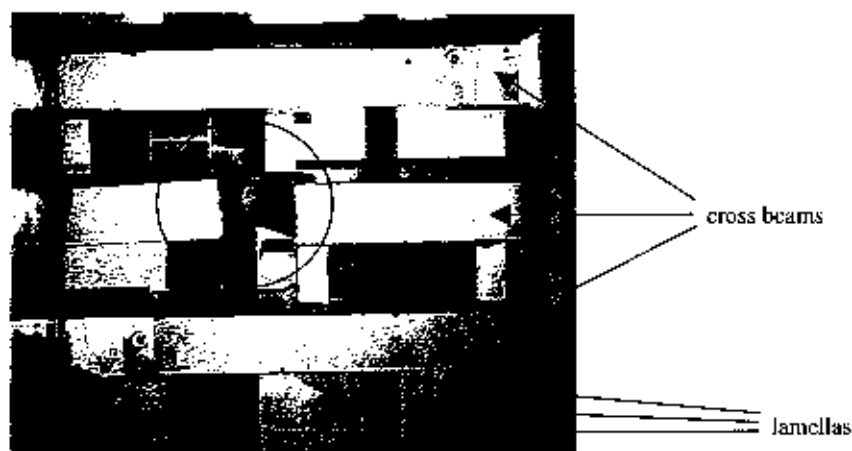


Figure 2 - Crack in the modular expansion joint of the "Martinus Nijhoffbrug", The Netherlands
(view from under the expansion joint)

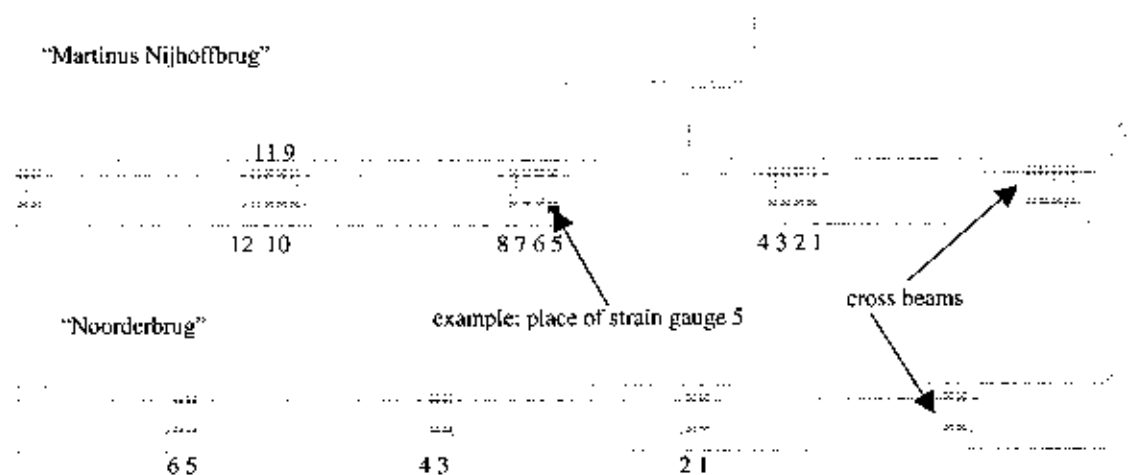


Figure 3 - Cross-section of the expansion joint construction



Figure 4 - Vehicle used for measurements

Table I – Vehicle parameters for the simulation of dynamic wheel loads from vertical accelerations

Element	Parameter	Unit of measure	Value
Chassis	Mass	kg	16700
	Pitch inertia	kgm^2	53718
	Roll inertia front	kgm^2	1040
	Roll inertia rear	kgm^2	1691

	Distance CG to front axle	m	3.46
	Distance CG to rear axle	m	2.13
Front axle	Mass	kg	650
	Pitch inertia	kgm^2	540
	Track width	m	2.02
Rear axle	Mass	kg	1125
	Pitch inertia	kgm^2	634
	Track width	m	1.85

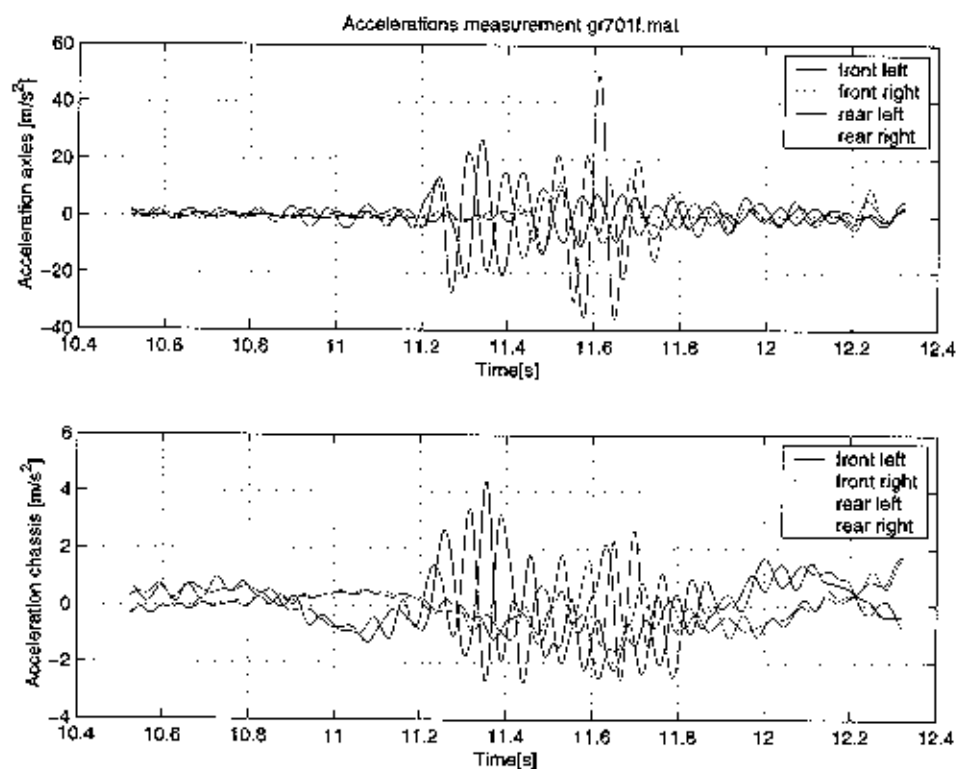


Figure 5 - Accelerations at the "Noorderbrug" at 70 km/h

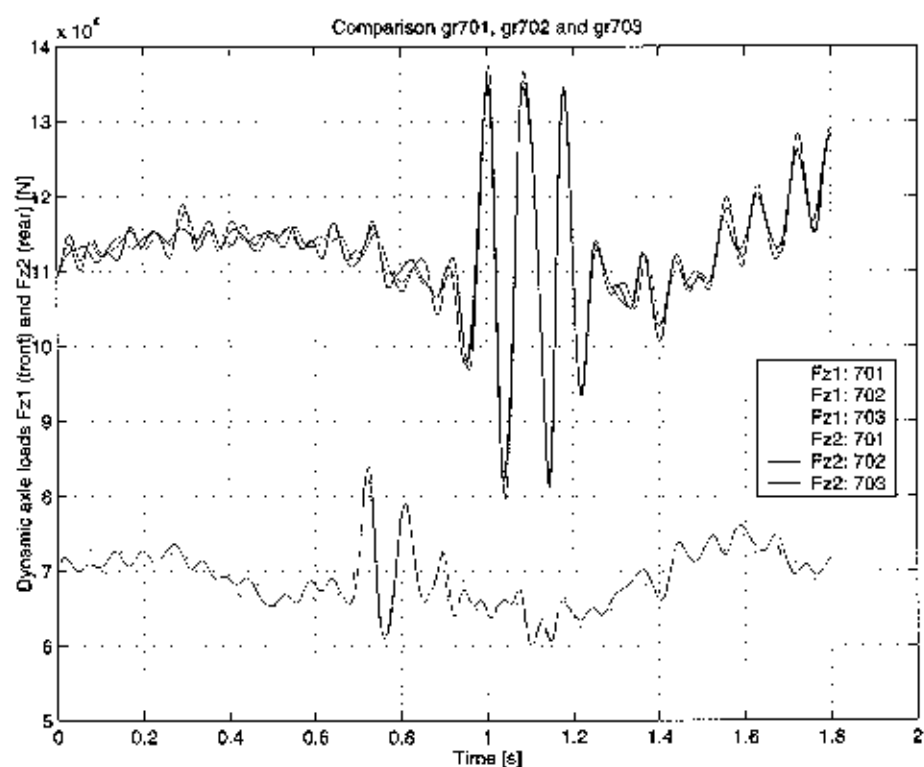


Table II - Dynamic amplification factors

velocity	dynamic ampl factor "Martinus Nijhoff"	dynamic ampl factor "Noorderbrug"
50 km/h	1.5	1.1
70 km/h	1.6	1.2
85 km/h	1.7	1.4

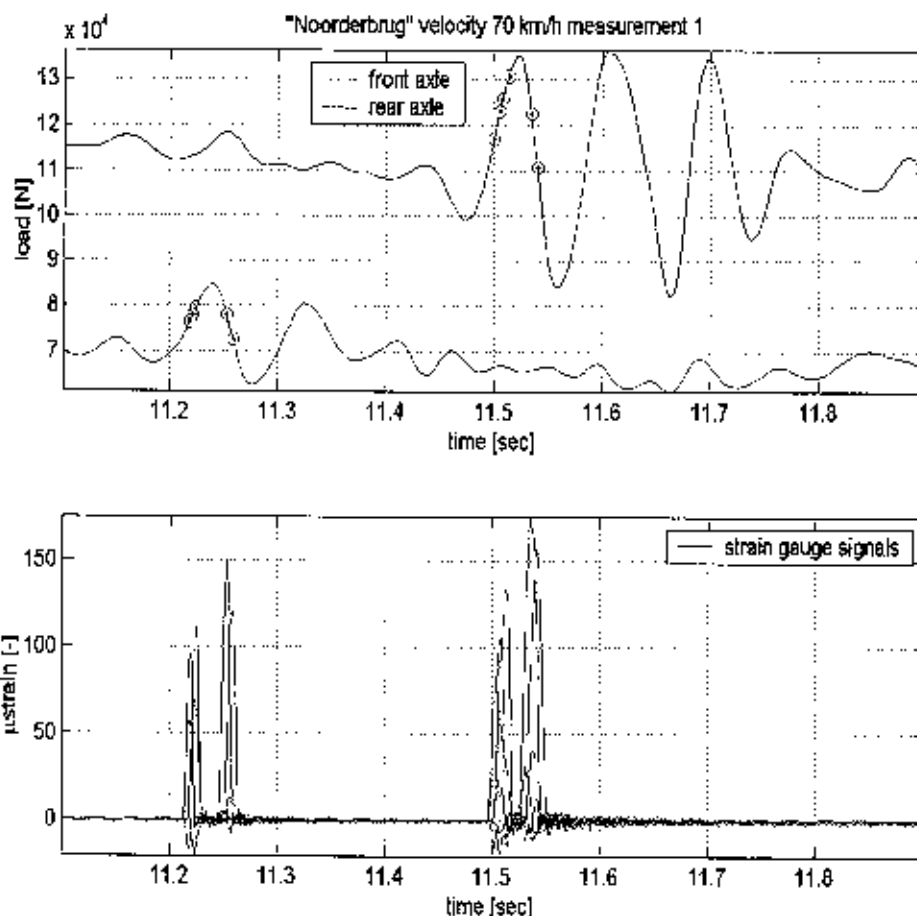


Figure 7 - Loads at maximum strains at a velocity of 70 km/h, "Noorderbrug"

Translation of text:
 snelheid - velocity
 belasting - load

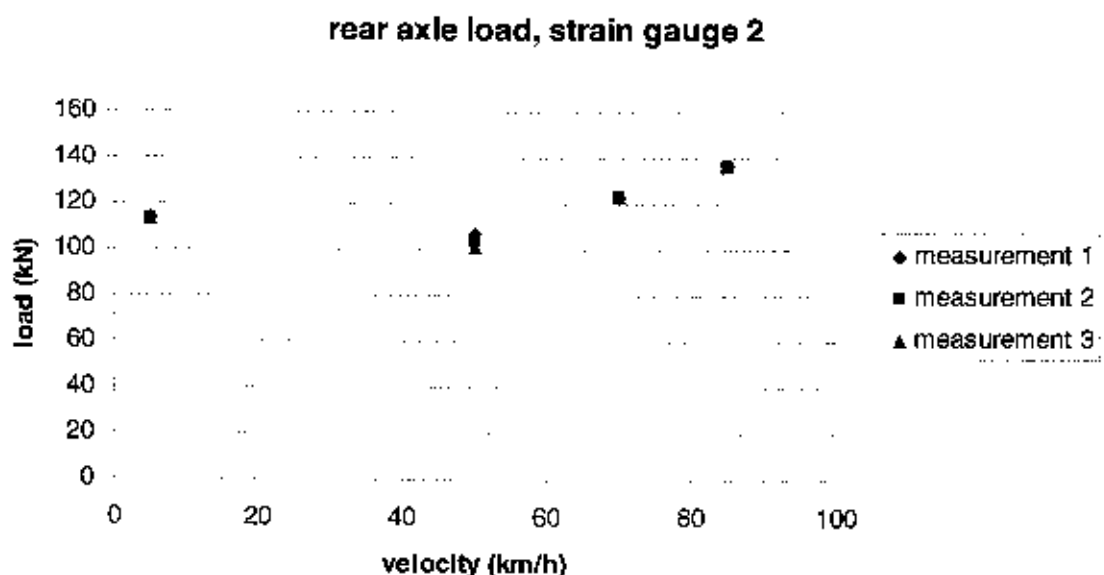
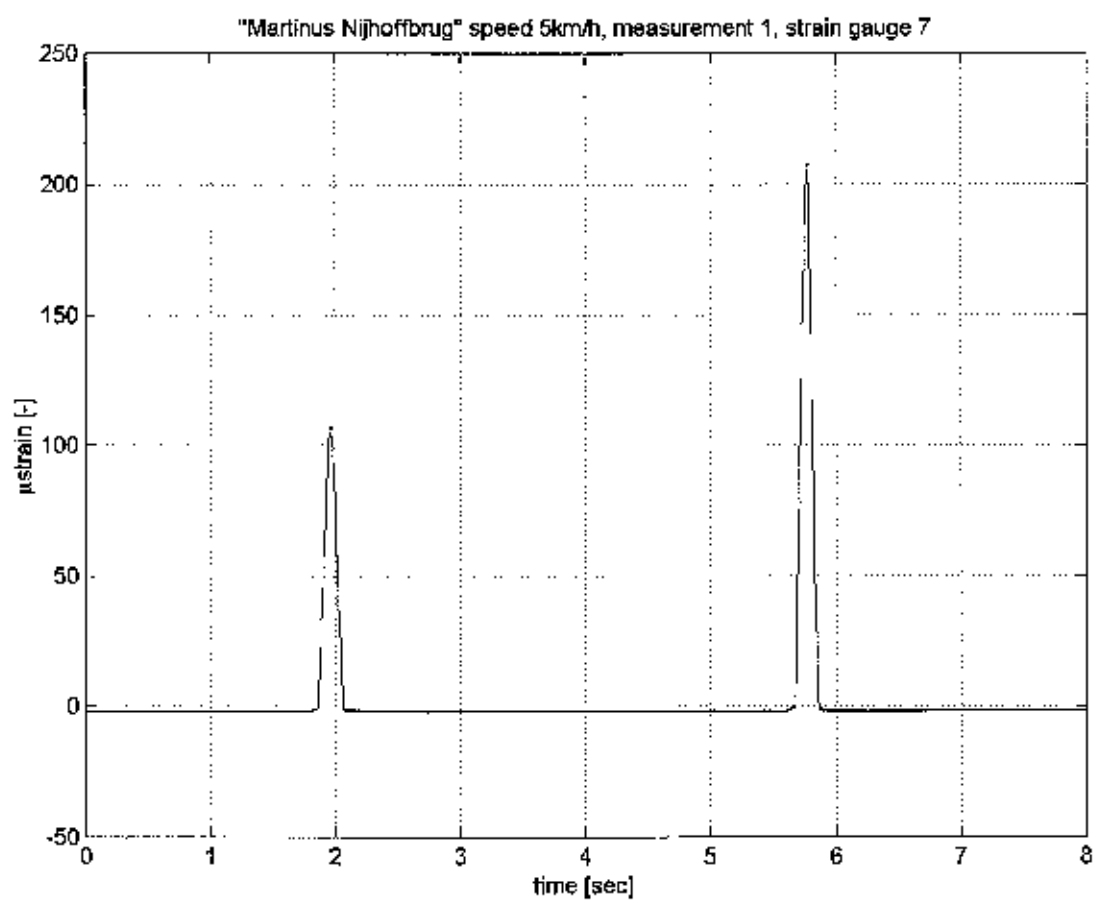


Figure 8 - Loads at maximum strains at strain gauge 2, "Noorderbrug"



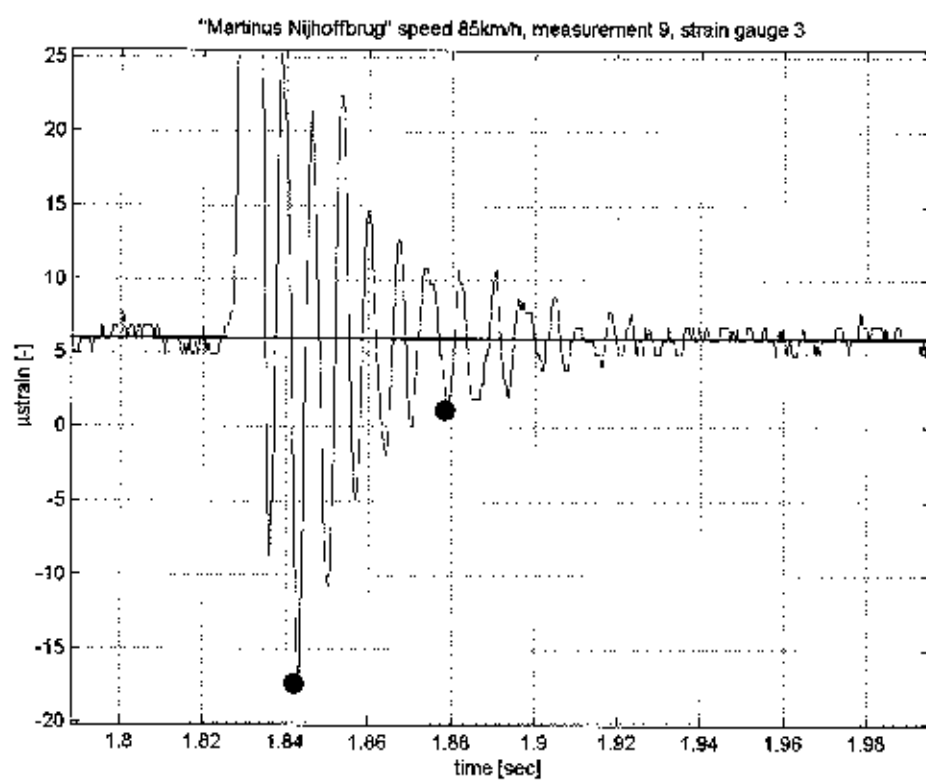
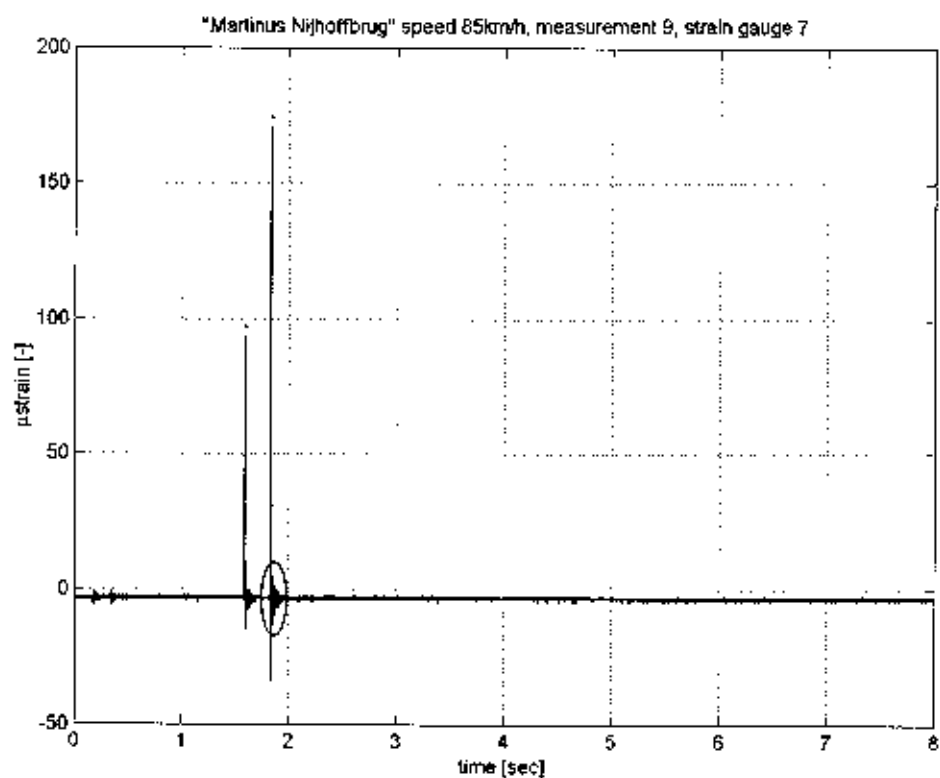


Figure 9 - Strain measured at cross beam 7, "Martinus Nijhoffbrug"

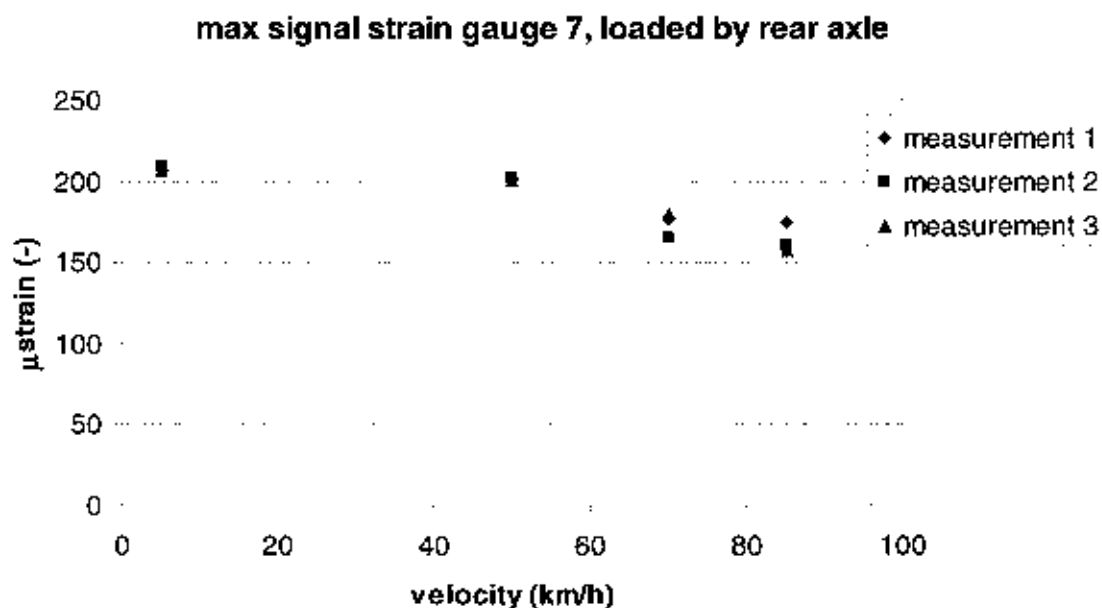


Figure 10 - Maximum strains at various velocities, cross beam 7, "Martinus Nijhoffbrug"

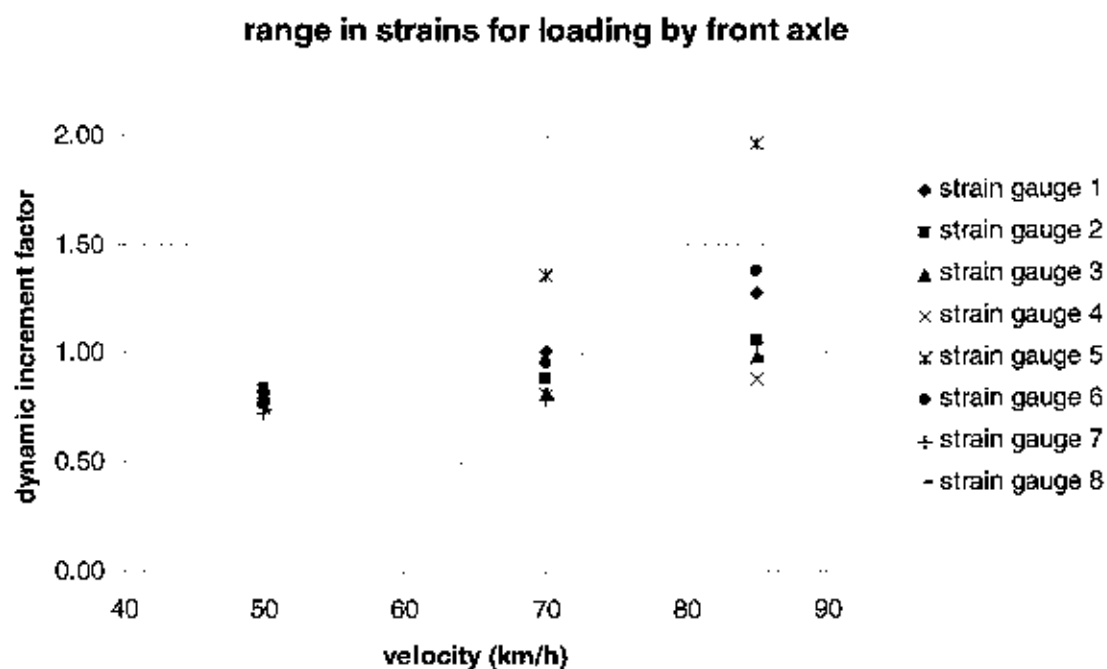


Figure 11 - Dynamic increment factors for range in strains at the "Martinus Nijhoffbrug"

Table III - Weighing average of dynamic increment factors, "Martinus Nijhoffbrug"

	50 km/h	70 km/h	85 km/h
Maximum, front axle	0.71	0.80	1.01
Range, front axle	0.77	0.94	1.38
Maximum, rear axle	0.68	0.64	0.56
Range, rear axle	0.73	0.76	0.78

range in strains for loading by front axle

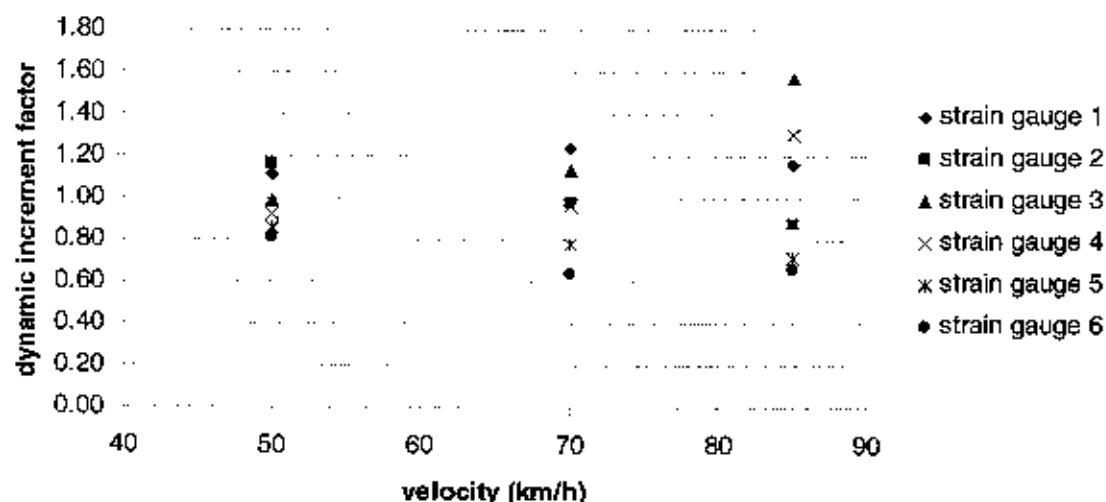


Figure 12 - Dynamic increment factors for range in strains at the "Noorderbrug"

Table IV - Weighing average of dynamic increment factors, "Noorderbrug"

	50 km/h	70 km/h	85 km/h
Maximum, front axle	0.93	0.92	1.02
Range, front axle	1.00	1.00	1.12
Maximum, rear axle	1.00	0.94	0.91
Range, rear axle	1.08	1.04	1.03

Table V - Equivalent axle loads and fatigue life factors for variation in damping and dynamic amplification factors

Damping	Dynamic amplification factors	Equivalent cycles	Equivalent damage	Fatigue life factor
Damping 10%	DAF = 1.8	2186	1.9	0.53
	DAF = 1.4	1145	1.0	1.0
	DAF = 1.0	631	0.6	1.7
Damping 2.5%	DAF = 1.8	5812	5.1	0.2
	DAF = 1.4	3005	2.6	0.4
	DAF = 1.0	948	0.8	1.3

RESEARCH ON THE PERFORMANCE OF THE HGVs IN THE MAJOR GREEK ROAD NETWORK USING WIM TECHNOLOGY

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ABSTRACT

The work presented within the framework of this paper concerns the first major research on heavy goods vehicles (HGVs) operating on the national highway network of Greece. Data was collected at seven sites along the two main road axes of the country. A total of about 3.000.000 records at all sites were collected concerning all vehicles classes. Data was collected using permanent Weigh-In-Motion (WIM) technology. In addition to that, comparison measurements were taken from high-speed and low-speed WIM systems. The results will be taken into account when considering enforcement for the overloaded HGVs along the national highway network. In the framework of this research emphasis was given in the development of a comprehensive database concerning the WIM measurements and more specific the dynamic and static characteristics of the HGVs. This database is unique for the Greek highway network and it is a necessary step in order to provide useful information to engineers and support all future actions in the area of pavement design and management.

1. INTRODUCTION

During the last decades a shift in the movement of goods towards the use of Heavy Good Vehicles (HGVs) has been observed. In Greece the total amount of HGVs transported goods for an average working day is about 500.000tns corresponding to about 70.000 trips [1]. According to the results of the National Origin – Destination survey, there is an annual 3% increase in the HGVs traffic during the period 1980-1995. This is accounted to the transport related costs, the requirement of the transport companies, the growing density and quality of the highway network and the trends in the transport industry. This increase of HGVs traffic has raised questions about the potential negative impacts in the Greek highway network, and therefore, revealed the need for a systematic recording of the HGVs characteristics.

A research project was carried out during 1992 concerning the collection and analysis of HGVs data along parts of the national highway network [2]. In 1998 the Ministry of Development, General Secretariat for Research and Technology assigned a research project [3] concerning the determination and assessment of the dynamic characteristics of HGVs and their impact to the national highway network, to the Department of Transportation and Hydraulic Engineering, Faculty of Rural and Surveying Engineering, Aristotle University of Thessaloniki, to the Central Laboratory of Public Works (responsible authority for pavement quality) and to the Egnatia Road S.A. (responsible authority for the construction, operation and maintenance of the Egnatia Road).

The project deals with the recording of the static and dynamic characteristics of HGVs using the Weigh-In-Motion (WIM) technology, with reference to the two main road axes in the country (Patras-Athens-Thessaloniki-Evzonoï -PATHE and Egnatia Road). PATHE connects the southern with the northern part of the country and has a length of 1.050 kilometers. Egnatia Road connects the western with the eastern part of the country and has a length of 680 kilometers (some parts of this highway are under construction). Both highways play a significant role in the freight transport sector of the country and therefore they are important to the traffic of HGVs. The project also deals with the analysis and evaluation of the collected data in order to create a database, in accordance to the guidelines imposed by the European Action COST 323.

Research was made on the characteristics of the overloaded HGVs and on their impact to the highway network. Within the project monitoring of the pavement performance due to the loading from HGVs, forecasts about the traffic loads in the examined road axes were also made. Finally the development of a database for pavement design and management and the support of the decision making process concerning the budget allocation for road construction, operation and maintenance was made together with the design of guidelines for the evaluation of the impact of HGVs on road axes and for the computation of the pavement construction elements.

2. DATA COLLECTION METHODOLOGY

The main criterion for the selection of sites for the survey was that each one must be characterized by significant HGVs traffic. An attempt has been made so that the number and position of these sites will secure the geographical coverage of the examined highways. These sites are presented in Figure 1. It was not possible to have simultaneous counts to all seven sites as it was originally scheduled. Since the number of the available loggers was smaller than the number of the available sensors and loops, the loggers were removed from one site to another according to a predefined program of counts. The counting period started in August 2000 and ended in March 2001. It is divided into four discrete periods as follows: summer period (August 2000), autumn period (September-November 2000), winter period (December 2000 - February 2001), spring period (March-April 2001).

Two WIM systems were installed (one for each of the two highways). Underground cabling was used to connect the sensors with the logger. In four out of seven sites PEEK TRAFFIC Ltd. sensors were used while in the rest two sites the Electronique Controle Mesure (E.C.M.) sensors were used. The layout of the installation consisted of two piezoelectric weight sensors having a length of 3,5 meters, another piezoelectric on scale sensor having a length of 0,5 to 0,8 meters and finally an inductive loop in order to create the necessary magnetic field. A total number of 3.053.116 records (referred to all vehicle classes) were finally collected.

Repeated visits to the sites were necessary in order to transfer data from the logger to a portable personal computer and also to replace the batteries of the system. The frequency of these visits depended on the traffic volumes and on the batteries used. Measurements of the evenness were also made in each site in order to assess the quality of the pavement. These measurements were made for a distance of one kilometer before and after the exact position of the WIM systems. Apart from the field data collection process, an extended survey was conducted in order to collect data on the construction elements of the pavement (e.g., year of construction, type and year of last maintenance etc.), on the available traffic volumes, on the classification of the HGVs, on the maximum allowed axle and gross weight etc.

Traffic volume data was based on the results of the National Transportation Survey which was carried out in 1993 by the Ministry of Environment, Physical Planning and Public Works [4]. Data on HGVs became available from the Ministry of Transport and Communications, division of freight transport.

3. PRESENTATION OF RESULTS

Due to the fact that two WIM systems were used within the framework of the project, there was incompatibility concerning the data obtained from the loggers. Therefore there was an effort in order to perform a common analysis process. The first WIM system (ADR-Peck Traffic) uses the Standard FHWA classification system but with one change (class 14 now refer to auto-calibration vehicles; passenger cars having wheelbase between 2,40 and 2,60 m.). Class 15 is now the default class for vehicles with more than 8 axles. The second WIM system (Hestia Station-ECM) uses a more detailed classification system created by the manufacturer. Data from this second WIM system was converted, by using the distances between axles, in

order to be compatible with the ADR-Peck Traffic WIM system. For the purposes of the project the FHWA classes 1, 2 and 14 (subclass of class 2) referred to passenger cars while the rest classes referred to HGVs.

A significant number of false records were identified before the statistical analysis. Filters checking basic parameters of the WIM data were applied to false records. These filters check the status code that the logger gives to each record, the vehicle speed, the wheelbase, the distances between axles and the axles weights. Due to the large amount of data, powerful tools were used in order to design the databases, to allow for computations and to produce statistical results. Data analysis process includes all the analytical information from every site, on every period of counts and on every direction of vehicles.

The identification of the overloaded vehicles was made taking into account the values of the maximum allowed weights for the international transport as defined by the Ministry of Transport and Communications in Greece. The transformation of the axle loads to Equivalent Single Axle Loads (ESALs) of 8.155 tn (18,000 lbr) was made using the respective coefficients of the American Association of State Highway Officials. (A.A.S.H.O.) Two coefficients were defined for the purposes of the analysis in order to express the "aggressiveness" of loads to the pavement. The first coefficient (Vehicle Class Equivalence Coefficient – V.C.E.C.) is defined as the ratio of the total number of equivalent axles per vehicle class over the total number of vehicles of this class (Dept. of Transportation and Hydraulic Engineering et.al. 2001).

The second coefficient (Area Equivalence Coefficient – A.E.C.) is defined as the ratio of the total number of equivalent axles over the total number of the heavy goods vehicles at a certain position or area. Geographic information systems technologies was used in order to produce thematic maps with ESALs and overloaded vehicles.

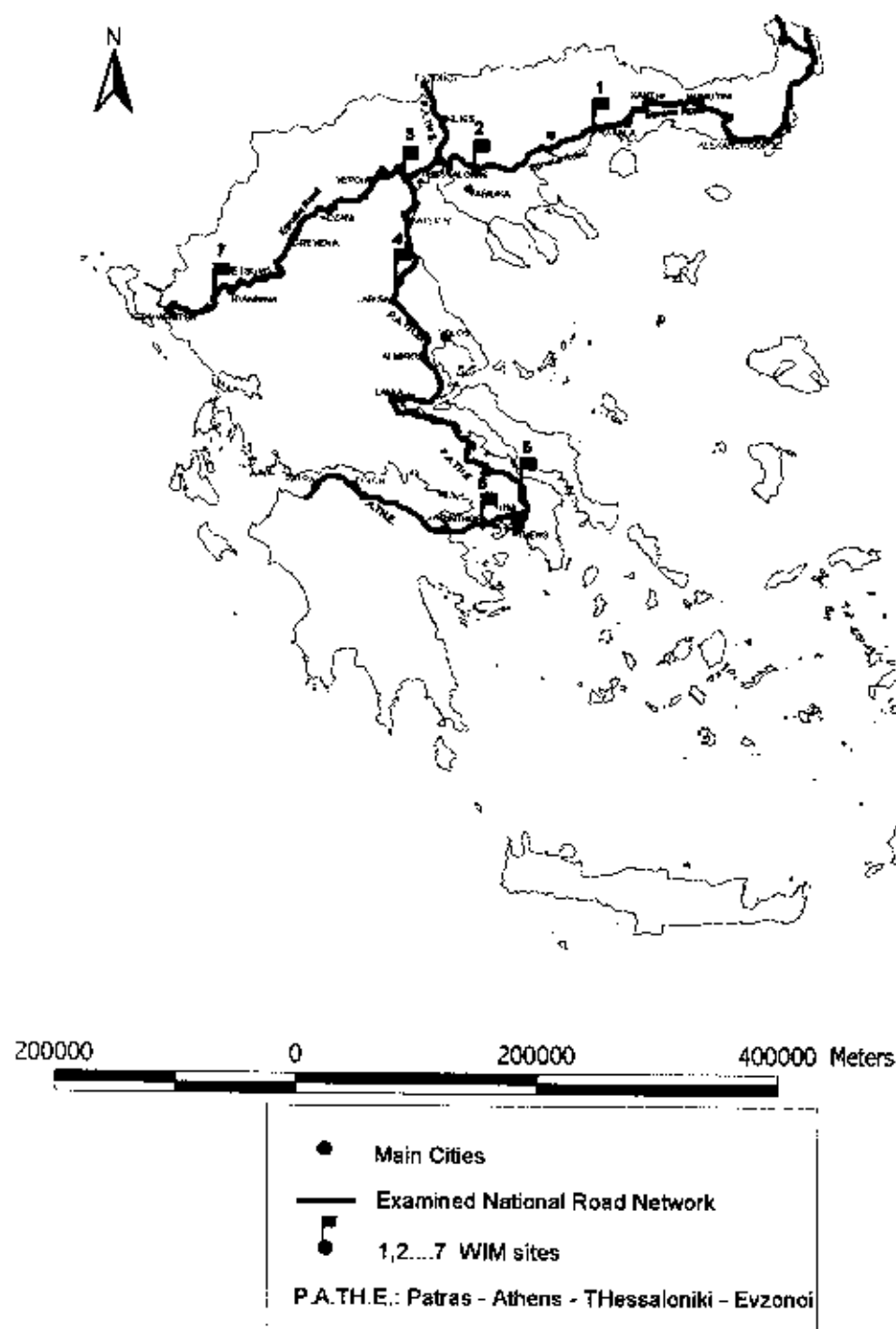
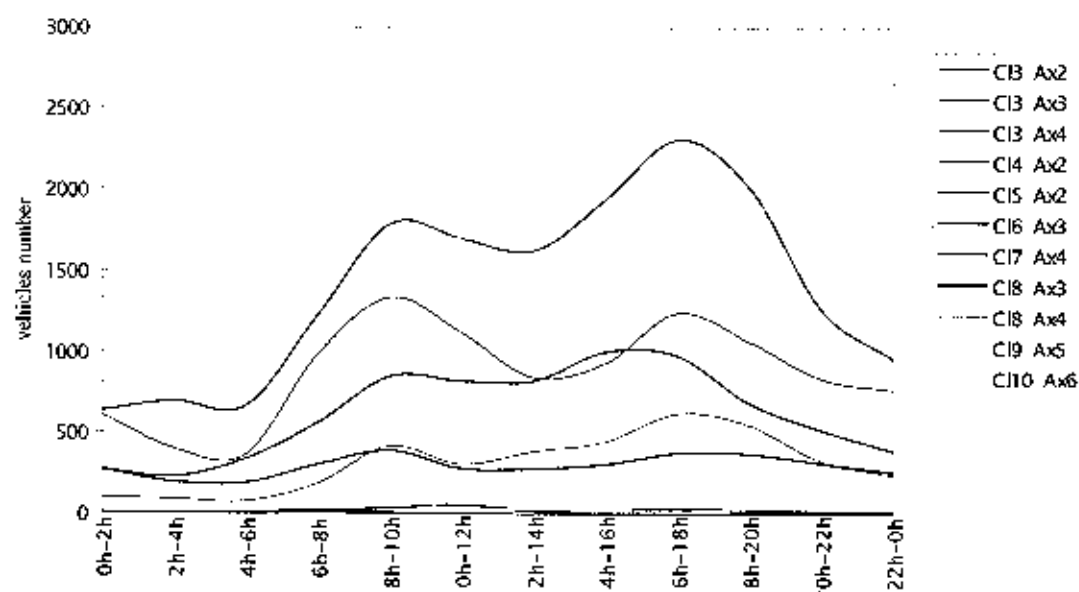


Figure 1 - Position of the WIM sites in the national road network

In the following figures and tables, detailed data concerning HGVs in a specific array (site 3) are presented. The results refer to the direction: Athens to Thessaloniki (PATHE). Results obtained during the period 13th September 2000 to 9th November 2000 (total number of days where measurements took place is equal to 58).

Figure 2 presents the variation of the cumulative number of the HGV for a period of 24 hours in a specific area (site 3). Figure 3 presents the distribution of the cumulative number of HGV per class in a specific array and for the specific period of 58 days. The majority of HGV belong to class 9 with 5 axles and to class 5 with 2 axles. Figure 4 presents the distribution of the cumulative number of HGV of class 9 with 5 axles. There is a high value concerning HGV with Total Gross of 38-42 tns something which is expected due to the fact that drivers usually utilize the maximum permitted capacity of their vehicles.

The number of overloaded HGV of this specific class is decline as the Total Gross increases, something which again is highly expected.



Variation of the cumulative number of HGV

Figure 2 -

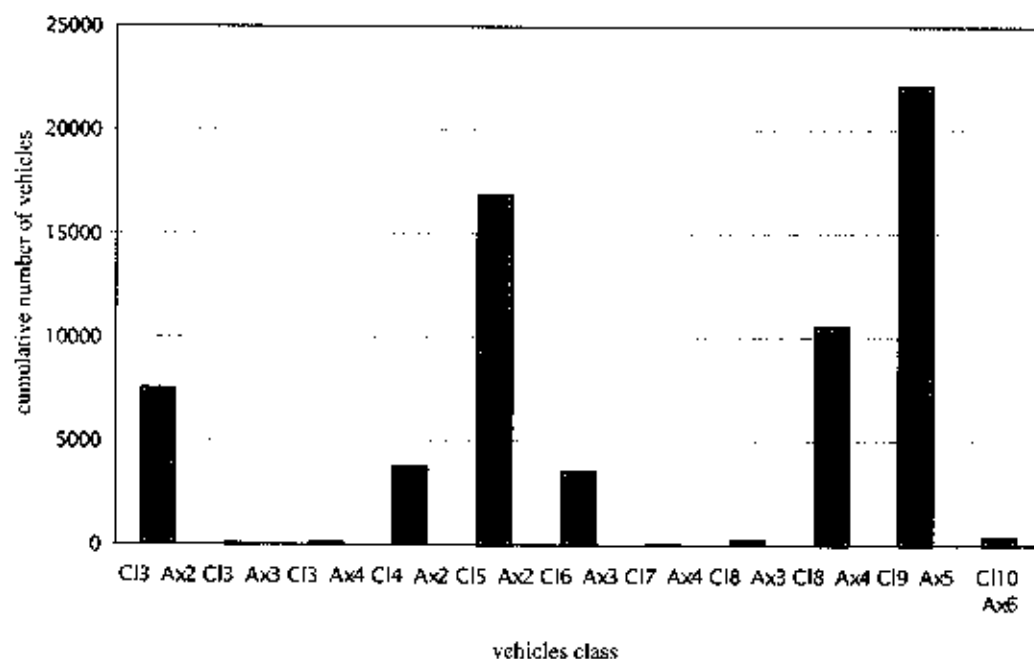


Figure 3 - Distribution of the cumulative number of HGV per class

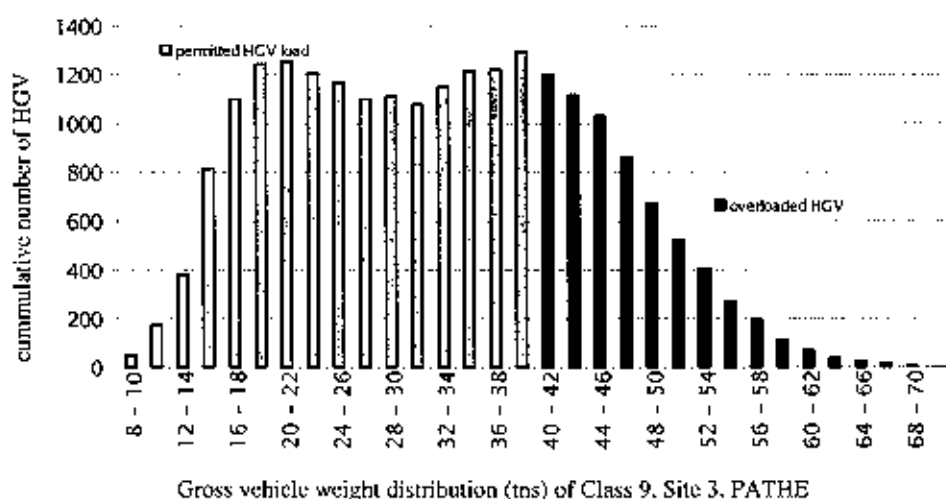


Figure 4 - Total Gross distribution of the cumulative number of HGV (class 9 with 5 axes)

Figure 5 presents the distribution of the cumulative number of ESALs per Total Gross category and per vehicle class. High values refer to 16-18 tns, although no direct comparisons can be made due to different HGV characteristics and limits imposed by the current legislation.

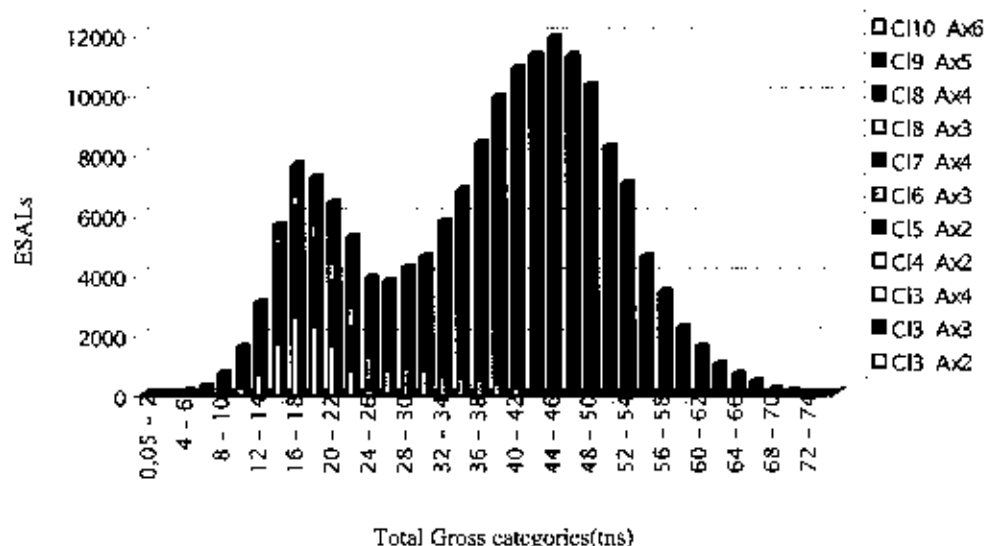


Figure 5 - Distribution of cumulative number of ESALs per total Gross category and per vehicle class

Characteristics of the HGV as recorded by the WIM system are presented in tables 1,2 and 3. More specifically, results concerning minimum, average and maximum values of Total Gross per HGV, values of wheelbase, length of vehicle, speed and finally average and maximum values of axle systems weight are included in these tables. Tables 4 and 5 include data on the distribution of overloaded HGV per class and VCBC and AEC.

In figures 6 and 7 the percentage of overloaded HGVs in each site and the average number of ESALs in each site are presented. Overloaded vehicles in figure 6 refer both to axle weight and gross vehicle weight. The percentages presented in the specific figure are the summation of the above two cases of the overloaded vehicles.

Table 1 - Characteristics of HGV as recorded by the WIM system (per HGV class)

Class	Axles	Vehicles Number	Percent (%)	Wheelbase (m)			Total Length (m)			Speed (km/h)		
				Minimum Value	Maximum Value	Average Value	Minimum Value	Maximum Value	Average Value	Minimum Value	Maximum Value	Average Value
	2	7499	11,5	3,1	3,9	3,5	3,3	10,2	5,6	36	148	90
3	3	42	0,1	5,6	7,9	7,2	6,9	9,8	8,8	47	124	76
	4	124	0,2	7,9	9,5	8,8	8,8	22,0	11,1	66	116	89
4	2	3775	5,8	6,1	7,6	6,3	6,5	18,1	12,2	45	148	98
5	2	16852	25,8	4,0	6,1	4,9	4,3	16,7	8,7	21	162	90
6	3	3562	5,5	2,0	17,6	6,0	5,3	18,6	9,3	30	149	89
7	4	69	0,1	5,0	8,8	6,2	6,2	11,5	8,6	50	106	80
8	3	278	0,4	6,8	12,3	10,3	9,3	16,5	12,9	48	149	90
	4	10539	16,1	5,1	22,4	12,1	5,8	21,9	15,1	34	164	88
9	5	22129	33,9	8,0	25,0	13,2	8,7	21,9	16,4	21	170	87
10	6	434	0,7	8,9	18,3	14,6	12,7	21,8	17,4	54	106	85
Totals:		65303	100,0									

Table 2 - Average and Maximum values of HGV axle systems weight as recorded by the WIM system

Class	Axles	Total Number of system axles			Average Weight (Kg)			Maximum Weight (Kg)		
		Simple	Double	Triple	Simple	Double	Triple	Simple	Double	Triple
	2	14998	-	-	1797	-	-	4000	-	-
3	3	126	-	-	2074	-	-	5841	-	-
	4	248	124	-	2971	4227	-	5910	13906	-
4	2	7550	-	-	7392	-	-	16366	-	-
5	2	33704	-	-	5096	-	-	16379	-	-
6	3	3599	3536	5	6578	13939	16499	14527	31655	23370
7	4	82	10	58	6822	23780	23157	11293	30578	40357
8	3	834	-	-	5736	-	-	15927	-	-
	4	28309	6919	3	6371	12188	21111	16378	32389	26789
9	5	52011	8872	13630	7175	13209	17220	16376	32212	46800
10	6	734	713	148	6181	12283	16016	15060	30640	37416
Totals:		142195	20174	13844						

Table 3 - Minimum, average and maximum values of Total Gross per HGV class

Class	Axles	G.W. Average (kg)	G.W. Minimum (kg)	G.W. Maximum (kg)
	2	3594	1307	6945
3	3	6221	2184	12114
	4	10170	1775	23171
4	2	14784	5423	27099
5	2	10191	2197	25724
6	3	20507	6162	43605
7	4	31019	9710	48968
	3	17209	7708	38159
8	4	25121	7338	55981
9	5	32765	8441	71939
10	6	36094	12174	74734

Table 4 - Distribution of overloaded HGV per class

Class	Axles	Total Number of vehicles	Over loaded Vehs.	(%)	Overloaded Vehs. (Gross weight)	(%)	Overloaded Vehs. (axle weight)	(%)
	2	7499	0	0,0	0	0,0	0	0,0
3	3	42	0	0,0	0	0,0	0	0,0
	4	124	0	0,0	0	0,0	0	0,0
4	2	3775	860	22,8	756	20,0	104	2,8
5	2	16852	1242	7,4	1025	6,1	217	1,3
6	3	3562	964	27,1	873	24,5	91	2,6
7	4	69	45	65,2	41	59,4	4	5,8
	3	278	18	6,5	13	4,7	5	1,8
8	4	10539	3147	29,9	2960	28,1	187	1,8
9	5	22129	8356	37,8	6551	29,6	1805	8,2
10	6	434	180	41,5	167	38,5	13	3,0
Totals:		65303	14812	22,7	12386	19,0	2426	3,7

Table 5 - Distribution of ESALs per HGV class

Class	Axles	Vehicles Number	ESALs	V.C.E.C.
	2	7499	75	0,01
3	3	42	2	0,06
	4	124	12	0,10
4	2	3775	10233	2,71
5	2	18852	18382	1,09
6	3	3562	7779	2,18
7	4	69	268	3,88
8	3	278	372	1,34
	4	10539	40096	3,80
9	5	22129	92147	4,16
10	6	434	1343	3,09
Totals:		65303	170707	
A.E.C.			2,61	

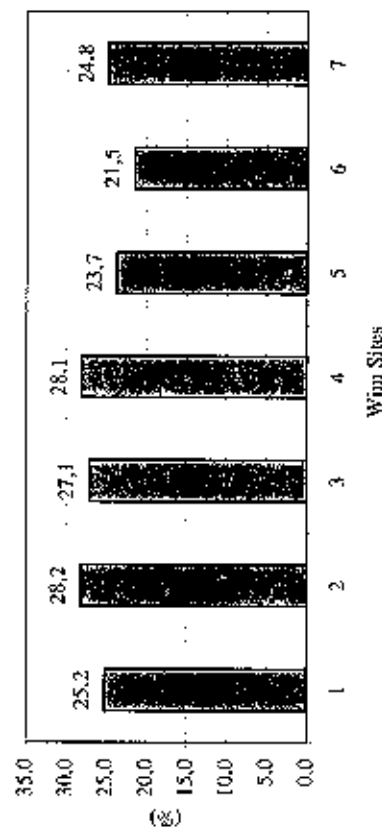


Figure 6 - Percentage (%) of overloaded HGVs in each site

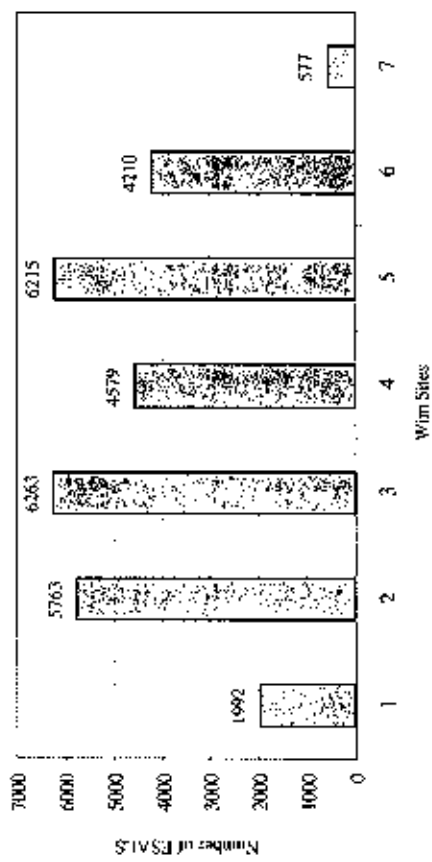


Figure 7 - Average daily number of ESALs in each site

4. CONCLUSIONS

The percentage of the WIM records finally used in the analysis was 65% (the rest concerns the false records). Concerning the traffic composition in the national network, high percentages appear in class 9 with 5 axles in class 8 with 4 axles in class 5 with 2 axles and in class 3 with 2 axles. The rest of the classes appear to have significantly smaller contribution to the traffic composition. The percentage of overloaded vehicles (considering only HGVs) varies between 21,5% and 28,2%. Within the framework of the project only some initial values for V.C.E.C. for specific vehicle classes can be accepted due to the fact that are based on large number of counts. The volume of data collected is considered to be insufficient for a reliable pavement evaluation. In order to do such an evaluation, repeated traffic counts are needed in the seven sites until deteriorations will be identified in the pavement. In this case there will be the opportunity for the development of prediction models in the area of pavement maintenance in the country. Comparison of Greek results to other studies concerning the loading of different class vehicles is not possible due to the small period of measurements. Taking into account the results of the project and the geographical distribution of the sites selected, a number of other sites for the continuation of this work was identified. Concerning the time periods of the measurements, a minimum period of one week every three months (seasonable measurements) is proposed for all permanent stations. Therefore, all range of peak values (taking again into account the experience gained within the project) will be included in a future work.

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ABNORMAL LOAD SUPER ROUTES – A STRATEGIC INVESTMENT FOR SOUTH AFRICA'S ECONOMY

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ABSTRACT

Because of a lack of inland waterways and the limited capacity of the railway network in southern Africa, the transportation of heavy industrial equipment to and from manufacturing, processing, power generating and other industries, in many cases, has to take place by road. When these large indivisible payloads, together with the combination of vehicles used to transport them, exceed 150 tons, 8 metres in width or 4,8 metres in height they are defined as Superloads and are of paramount interest to highway and traffic officials responsible for preserving the road network, infrastructure and furniture and minimising traffic congestion. The routes that they are allowed to travel are limited by bridge capacity and geometrics, involving mass, height, width and length restrictions. These critical routes are known as Super Routes. Failing to establish and maintain such Super Routes will place unfortunate limits on the South African industry by limiting opportunities for importing, exporting and moving superloads such as large machinery and transformers within the country. The national Abnormal Loads Technical Committee recognised a need to identify and monitor a minimum number of strategic routes that need to be preserved for the movement of Superloads. Short-sighted planning has in the past caused a reduction in the capacity of existing routes by the construction of overpasses with restrictive clearances on particular routes. CSIR Transportek was appointed by the Gauteng Department of Transport to develop a computerised tool for graphically displaying all relevant data related to these Super Routes. The system will be a useful tool for indicating to planning authorities the effect that a particular project such as a new bridge or a cable across a road could have on reducing or improving the capacity of a given route. The primary consideration is to arrest the increasing constraints being placed on existing routes by the encroachment of restrictive features. Route improvement based on favourable cost/benefit ratios will become a reality. The effectiveness of given routes may, for instance, be improved by raising a power line cable or a single bridge.

INTRODUCTION

In many parts of the world, there has been an increased demand for moving larger and heavier machinery and industrial components during the past few decades. On the oceans, many of the 20 000 ton ships of yesteryear have been superseded by supertankers, which may exceed half a million tons. On land, large loads are required to be moved from factories or seaports to various inland destinations. Because of the absence of inland waterways and the limited capacity of the railways in southern Africa, these loads are transported on the road network. These large and heavy loads generally comprise equipment for electric power generation and chemical plants. Demands for such components are economically justified since the potential savings to these industries are vast assuming a useful life of about 25 years. When these large indivisible payloads, together with the combination of vehicles used to transport them, exceed 150 tons, 8 metres in width or 4.8 metres in height they are defined as Superloads and are of paramount interest to highway and traffic officials responsible for preserving the road network, infrastructure and furniture and minimising traffic congestion (Figure 1). The routes that they are allowed to travel are limited by bridge capacity and geometrics, involving mass, height, width and length restrictions. These critical routes are known as Super Routes. During the past few decades, numerous routes in South Africa have been checked, cleared and documented in order to ensure the safe movement of these loads. Since 1994, the number of major road authorities in South Africa has more than doubled. This includes the establishment of five new provinces and the National Roads Agency, the awarding of four 30-year concession contracts (to date) to manage and maintain sections of the national road network, and the re-definition of six metropolitan areas. The national Abnormal Loads Technical Committee recognised a need to identify and monitor a minimum number of strategic routes that need to

be preserved for the movement of superloads. This paper describes the development of an electronic Super Route Map, which can be used to graphically display relevant data related to these Super Routes. The system will be a useful tool for indicating to planning authorities the effect that a particular project such as a new bridge or a cable across a road could have on reducing or improving the capacity and effectiveness of a given Super Route. It will also be used by transportation consultants for preliminary planning of the movement of superloads. The primary consideration is to arrest the increasing constraints being placed on existing routes by the encroachment of restrictive features that reduce the effectiveness of these strategic routes.

NATURE AND PURPOSE OF SUPERLOADS AND SUPER ROUTES

The primary restriction offered by overhead bridges, overhead cables, pipes and road furniture to the movement of large units is height clearance and the flexural and shear strength of supporting structures such as bridges and culverts. **Superloads** may therefore be defined as loads in excess of approximately 150 tons and 4.8 metres in total height. While the width and length may present practical problems, these problems can usually be overcome by using steerable dollies or multiple axle steerable trailers.

The items associated with superloads are normally large specialised vehicles such as mobile cranes and piling machinery, transformers, pressure vessels for the chemical and petroleum industries, precast building units, industrial, mining and earthmoving equipment. In addition, there is also the occasional need for the transportation of yachts and other unusual items such as trees and classic aircraft.

It is self-evident that general limits have to be set for normal daily movement of vehicular traffic on public roads. In South Africa and generally in the southern region of Africa the legal height limitation is 4 300 mm and the maximum allowable gross vehicle mass is a generous 56 tons for vehicles with a maximum overall length of 22 m. Modern bridges normally have a minimum vertical clearance of 5 500 mm. It is generally found that vehicles up to the superloads as defined, encounter few restrictions. When understrength bridges (as a result of deterioration or damage, or dating back to an era when a 20 ton truck was thought to be monstrous and awesome) are encountered, then this restriction on use is usually clearly indicated at the approach to the bridge and well-known and alternative routes can timeously be selected.

It stands to reason that superloads need special roads or routes where the visual physical restraints are not present. These routes are termed **Super Routes**.

GENERAL SITUATION IN SOUTH AFRICA

It needs to be noted that in South Africa there are specific constraints that may not exist in many other countries. Firstly, the use of waterborne transportation is virtually non-existent. For all practical purposes there are no navigable rivers or canals and the only waterborne transportation is coastal, from one sea port to another. This is of limited use for the movement of superloads as the heavy industrial areas are largely situated inland in and around the Gauteng Province near Johannesburg. Certain ports are constrained by suitable crainage (such as Richards Bay harbour) and others may have restricted access to the inland roads (such as Durban harbour).

The other mode of transport that might have been used for superloads, namely rail, has other severe limitations. Firstly the rail network is poorly developed compared to Europe and the United Kingdom with limited access in terms of origin/destination considerations. The other aspect, which is worse, might be ascribed in a devilish way to South Africa's colonial past!

The legacy from British colonial times is a railway system with a three foot six inch gauge which is less than 1 100 mm. There were even some lines installed with a gauge of less than 800 mm! The apparent cost savings that were calculated during the nineteenth century appeared to be based on sound economics at the time but it has rendered the present railway system impractical for the purpose of transporting superloads, due to dimensional and load bearing restrictions.

PROCESSING EXEMPTION PERMIT APPLICATIONS

Any vehicle or combination of vehicles that is in contravention of the legal road traffic restrictions requires an exemption permit in order to operate legally on any public road. All such vehicles are classified as "abnormal vehicles", which operate under previously determined restrictions (and routes in the case of superloads) and need to be in possession of valid exemption permits. These permits ensure that regulations governing dimensions, mass, wheel loads, traction power, signs, escorts and so on, are complied with¹. Permits are normally issued by the Provincial Road Authorities and, if necessary, input is obtained from local and metropolitan authorities.

South Africa has a well-developed and sophisticated system in place for adjudicating and processing applications for such abnormal vehicle exemption permits. Routes and operating conditions can usually be selected even for smaller superloads.

However, there are instances where superloads present major challenges. There are situations in South Africa where transformers have been installed that would be virtually impossible to remove or replace due to the fact that the access routes utilised at the time of their installation have been compromised by subsequent alterations or additions such as overhead bridges.

THE NATURE OF THE PROBLEM

There is a natural conflict of interest inherent in this situation. On the one hand there is a need to transport superloads. SASOL, the oil-from-coal enterprise, the mines and refineries could never have been established on an economic basis if it had not been possible to move large industrial units, pressure vessels and equipment on the public roads. These units have to be imported, exported, moved within the country or moved from coastal ports to land-locked countries to the north of South Africa. Lacking suitable super routes would render these operations practically impossible or exorbitantly expensive.

On the other hand, building and maintaining super routes costs money. Additional funds need to be invested in order to create and maintain the capacity to transport such superloads.

The crux of the matter is that this must be seen as an investment in the infrastructure of a country in order to cope with the need to develop and maintain the heavy industrial sector.

Due to the need to apply limited funds as judiciously as possible, and often through sheer ignorance, existing super routes may be reduced in capacity or rendered useless for the purpose by the construction of a single overhead footbridge or other obstruction.

ROUTE CLEARANCE

Whenever the need arises to find or assess a given route to transport one or more superloads, then a process known as **Route Clearance** needs to be carried out. This is an expensive and time-consuming process that entails a careful inspection, measurement and charting of all obstacles that may obstruct a superload. Because every reserve of structural strength may be used, all load bearing structures are inspected and their load bearing characteristics assessed, taking into consideration their current condition, including signs of damage, crack formation, corrosion and foundation subsidence. Construction drawings often have to be consulted in the case of older bridges that were designed and constructed in accordance with older, less demanding loading codes. In the case of doubt, special supports (normally temporary supports) have to be designed and installed. This, of necessity, is highly specialised and expensive work. Alternatively, the structure may be bypassed, or in exceptional cases, replaced.

The disturbing fact is that the results of these assessments may not be commonly available. Such a route clearance is normally commissioned and funded by a private concern and carried out by a specialist firm who may naturally regard this as intellectual property that may be provided at a price and need not be freely available. This information also has a limited shelf life, as subsequent deterioration, alterations or additions may change the characteristics of the route.

There is therefore a need to establish a mechanism for accessing this information (summaries of the assessments) and to store such information in a central database where it may be available to all practitioners and where it can be updated on a sustainable basis.

This is perfectly feasible and would only need access to the restrictions imposed by the route and not the process or technical details. This information could be made accessible to other users and roads authorities to the common benefit of all concerned. The maintenance and updating of the information would then be the responsibility of the roads authorities.

PROPOSED SOLUTION

The proposed solution to these problems has been the establishment of an electronic **Super Route Map** that will house relevant information regarding the network of super routes (Figure 2).

The first step was to establish a national working committee comprising participants from both the public and private sectors in order to identify the minimum super route network of roads necessary for the purpose. It was necessary to consider the position of the ports for import and export purposes as well as major border posts to neighbouring countries that may wish to import large items either through South African harbours or directly from South Africa. By considering the positioning of heavy industries and industrial users, a minimal network of internal routes was established to form the **National Super Route Network**.

The super route map was thus based on this basic national super route network. This map, its potential and its implications will be brought to the attention of all roads authorities from the South African National Roads Agency to the provincial, metropolitan and local authorities. The initial approach was made through the national Abnormal Loads Technical Committee in March 2002 and will be followed by a presentation at the South African Transport Convention in July 2002.

THE SUPER ROUTE MAP

Data regarding the current height, width and load limitations on each segment of the super route network were obtained from provincial road authorities and a consulting firm that has been extensively involved in route clearances for the past 25 years. In order to obtain the maximum benefit from the data and present it visually, the data was captured into ArcView GIS. A layer containing height, width and mass limits was prepared. Power stations, oil refineries and other large industrial plants are also indicated, as they are the most common destination points. The super routes were classified in three groups namely, major cleared routes; cleared routes and routes cleared for height only.

The next step was to obtain more detailed information regarding structures on the super route network. The South African National Roads Agency, as well as most provinces and metropolitan authorities utilise bridge management systems for optimising their bridge maintenance and rehabilitation programmes. Relevant data (and electronic photos) are being imported from these systems for viewing in the super route map (Figure 3). This includes minimum vertical clearances, width between kerbs, load capacities, design codes and other general inventory information such as date of construction and history of strengthening and rehabilitation..

The Super Route System comprises the following components: a GIS map, the Super Routes Desktop system and a Super Routes Website. The Super Routes Desktop System is targeted at the various Road Authorities that need to query data. The system was developed in Visual Basic 6® with Microsoft Access® as the desktop database. The mapping component used is MapObjects LT®. As the user moves the mouse over a route, details such as route number and limits appear. Where photos are available, a thumbnail appears and by double-clicking on the photo, a full photo can be viewed. The GIS file used in the program is the same base file created in ArcView®.

The website is also being developed to ensure that the relevant roleplayers and stakeholders will have access to the latest information (Figure 4). It will allow the users of the Super Routes System to update their systems on a regular basis. The website will also enable users that do not have the Desktop System to query selected data.

CONCLUSIONS

The road network in South Africa is essential for the movement of people and goods on a daily basis. Certain routes in the country have been upgraded during the past few decades in terms of vertical and horizontal clearances as well as load carrying capacity in order to accommodate large indivisible loads, primarily from ports to inland industrial areas. Because of the strategic importance of moving these large loads in terms of the economic growth of South (and southern) Africa, it is of great importance that these super routes are maintained or in some cases upgraded in order to allow the movement of similar loads in the future.

The Super Route Electronic Map has been developed for the purpose of sharing relevant information with stakeholders in South Africa's road infrastructure, so that when road rehabilitation, upgrading or new construction is being planned, cognisance will be taken of the dimensional and load capacities of existing super routes, with a view to preserving these capacities.

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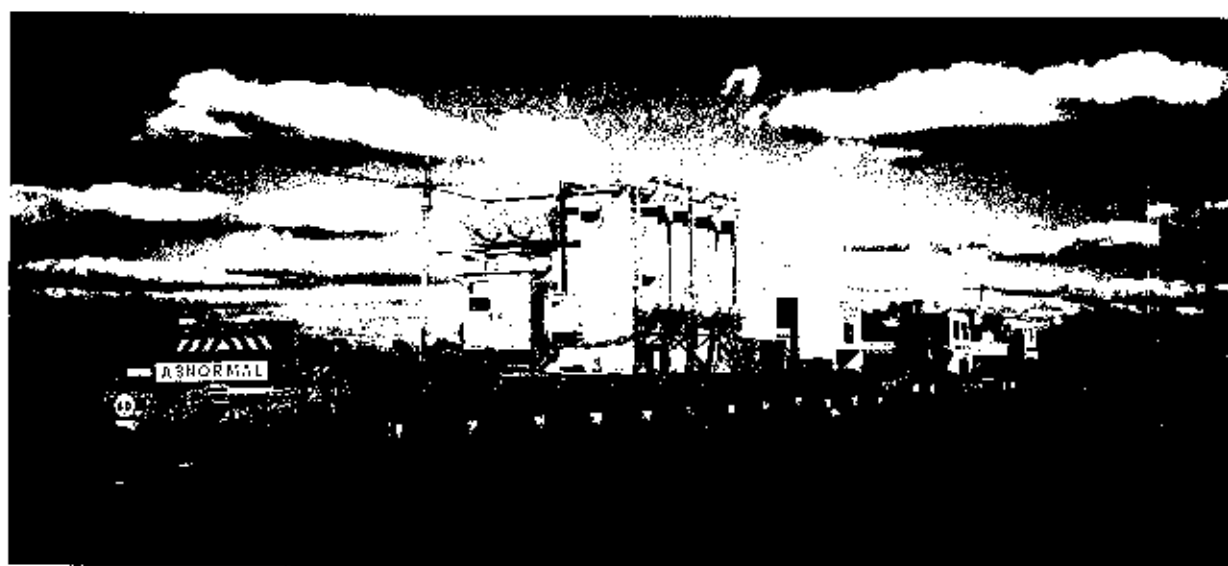


Figure 1 Typical medium-sized superload motorised by two haulers

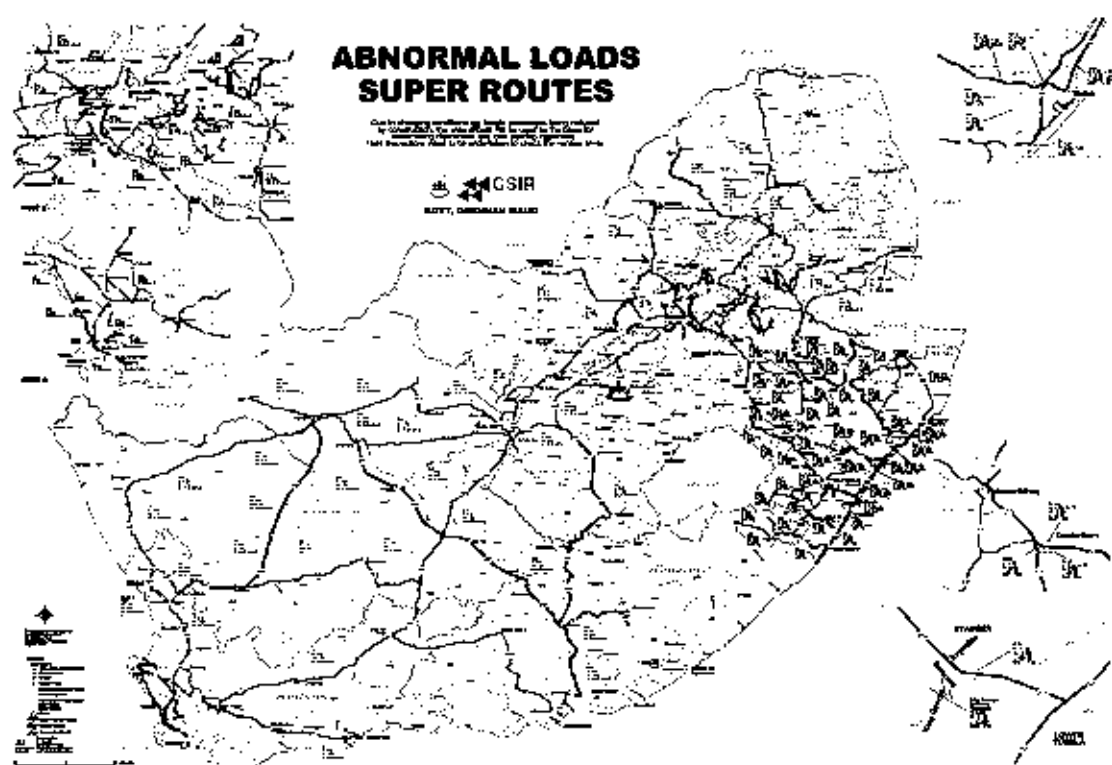


Figure 2 Super Route map

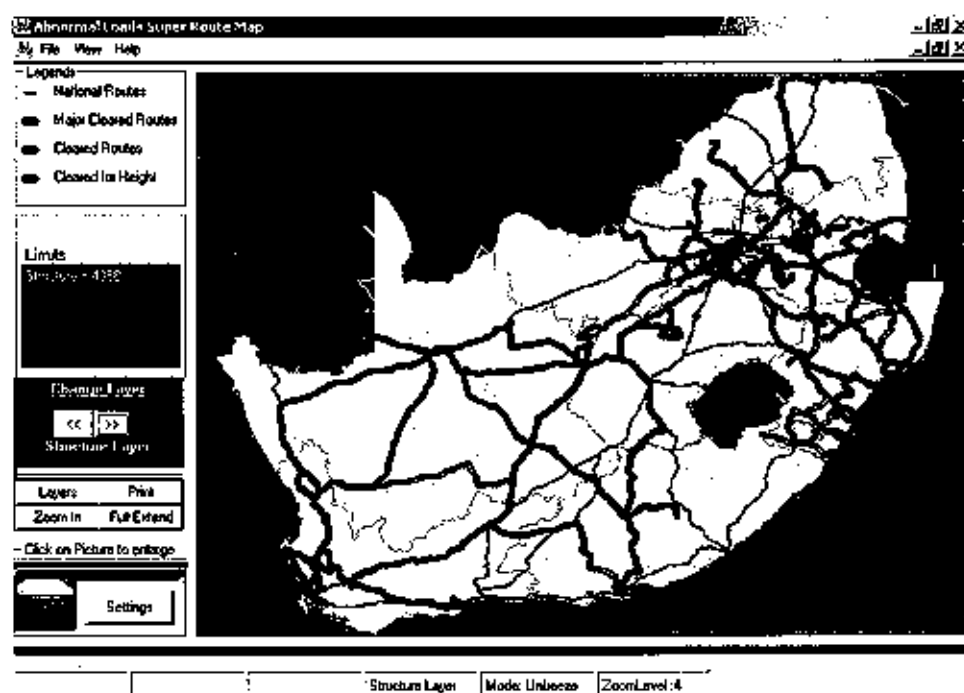


Figure 3 Super Route map showing structure information

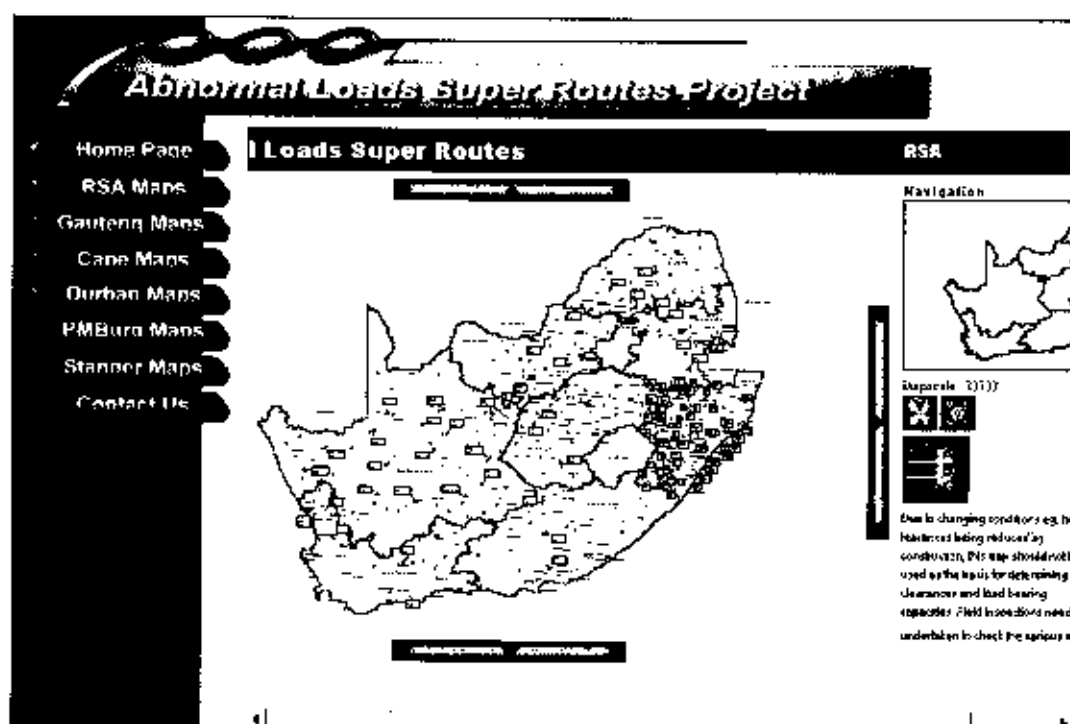


Figure 4 Website home page of Super Route map

IMPROVED PERFORMANCE OF EUROPEAN LONG HAULAGE TRANSPORT (EXTRA)

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ABSTRACT

The TFK project EXTRA with participants from Sweden, Finland, Denmark, Switzerland and the Netherlands started in March 2001 and will be finished in June 2002. By use of authentic freight information collected during a three-month period from hauliers operating on the European continent the advantage of an extended modular heavy vehicle system will be demonstrated. The reduced number of road trains which is needed for the same transport volume will be calculated showing the potential in savings of vehicle kilometres, fuel consumption, CO₂ emissions and cost etc. Other identifiable positive effects such as decreased traffic density and road wear as well as improved traffic safety are also expected.

In 1997 Sweden and Finland, by exception from the EU Council Directive 96/53/EC, was permitted to retain longer lengths and 60 tonnes maximum weight for vehicles in national traffic. To avoid unfair competition the exception required that the maximum lengths should be extended to contain either a truck or trailer or a semi-trailer combination with an attached trailer unit according the EU Directive. This way foreign road trains can be extended with an available extra trailer for additional capacity when entering Sweden and Finland. Theoretically the loading capacity of a semi-trailer combination may be increased by about 60 % consisting of 51 m³ and 16 tonnes. The extra loading capacity for a truck and trailer combination is about 40 % consisting of 37 m³ and 11 tonnes. The system is based on modules consisting of the CEN standardised 7.82-metre long unit load carrier and the 13.6-metre long semi-trailer being the longest single vehicle allowed in EU and largely used. The units are well adapted for rail transport in combined transport either as single vehicle units or as separate load carrier units. The system is built on existing vehicles and load carriers available in large quantities on the European continent.

1 INTRODUCTION

The EXTRA project, initiated by Swedish and Finnish industry, has been carried out to support the improvement of the road transport system in Europe. The economic, environmental and safety benefits of extended heavy vehicle combinations has been studied and indicated by use of authentic transport information. The following organisations and their representatives have contributed to the accomplishment of the project financially and/or by the provision of expertise and data services. Financial support has also been received from the Swedish Agency for Innovation Systems (VINNOVA).

Börje Jönssons Åkeri AB
Confederation of Swedish Enterprise
Dansk Transport og Logistik, DTL
Finnish Forest Industries Federation/Confederation
of Finnish Industry
FOCWA
IKEA AB
IRU, International Road Transport Union
NEA Transport Research and Training
Reining Transport bv

Rockwool Group
SCANIA AB
Swedish Forest Industries Federation
Swedish National Road Administration
Swedish Road Haulage Association
TFK - Transport Research Institute
VBG Produkter AB
Volvo Truck Corporation
Vos Logistics

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1.1 Background

The maximum weight and dimensions that are allowed for certain vehicles in national and international traffic were prescribed in the EU Council Directive 96/53/EC of May 25th 1996. By exception Sweden and Finland was permitted to retain the longer lengths and the maximum weight of 60 tonnes for vehicles operating in national traffic. This exception was given on the condition that the lawful maximum length for road trains consisting of modules applicable in international traffic was extended to 25.25 metre. Foreign hauliers this way are enabled to extend their road trains with an available extra trailer for additional capacity at the border to Sweden or Finland.

Articulated vehicles, usually called semi-trailers, are dominating in international traffic within the EU. Their regular loading capacity, of about 85 m³ and 26 tonnes is provided by a 13.6 metre long semi-trailer. Also road trains, consisting of a truck and trailer, are used. The most common combination in Sweden and in other northern countries is truck and trailer combinations on the contrary. They have trailers with different lengths and axle configurations in combination with a load carrying motor vehicle. In recent years a relatively short trailer for this combination, built like a cart with a centre "axle" in the form of a bogie, has become customary on the continent (stiff draw bar trailers). The loading capacity of the truck and trailer combination is dependent on the type of trailer used and the length of the draw bar between the cargo units. In general the maximum loading capacity for a road train is 40 tonnes in weight. The maximum loading capacity in volume is approximately 96 m³.

The modular system practised in Sweden and Finland since 1997 is based on the CEN standardised 7.82-metre long unit load carrier and the 13.6-metre long semi-trailer being the longest single vehicle allowed in EU. The maximum length of this combination of "modules" is 25.25 metre. All cargo units in the system are well adapted for rail transport in combined transport either as single vehicles or as separate load carrier units. It is built on existing vehicles and load carriers available in large quantities on the European continent. Discussions about a possible adoption of this system for the entire Europe has been held in the EU Commission.

1.2 Objective

The objective is to demonstrate and by example confirm the possible benefits from an efficient and environmentally adapted cargo and vehicle modular system for heavy vehicle combinations on the European continent.

2 THE METHOD

Information about fully loaded vehicles (FTL) performing direct shipments on selected continental freight corridors during a fixed period of time in regular heavy vehicle combinations should be translated into cargo loads on 25.25 metre vehicle modules combinations. The less numerous vehicles needed for the same transportation would thus show the saving potential in vehicle driving distance, fuel consumption, CO₂-emissions and costs. Other positive effects expected was for instance less traffic congestion and road wear as well as improved traffic safety.

2.1 Preparation

A more detailed project planning should be carried out including appointment of data providing sources and selection of adequate transport routes (corridors) on the continent for the data acquisition.

Appointment of data providing sources and selection of adequate transport routes would be based on compiled information from a number of plausible international road transport routes. Information such as origin and destination, number of trips, freight volume (ton), transport distance (both short and long distances) and type of commodities would be listed for each route.

2.2 Specification and limitations

Requested data elements from goods flows on selected transport routes should be specified and data processing formats developed. Depending on the quality of the available freight bill information (CMRs) some data would be

transformed into standard values for further use. The data transformation should be made with factors decided by the project management group. The package volume, for instance, should be based on the specific density for the current goods if available. As the extended vehicle combination is composed of existing modules, the variation due to package dimensions would be only marginal. By use of internal or national registering numbers the type of vehicle used could be determined.

The following factors and vehicle configurations should be analysed:

Factors to be analysed with quantitative methods:

Energy (kWh, kg, litre)

Environment (CO₂-emissions)

Transport cost

Factors to be analysed mainly with qualitative methods:

Road space (road classification)

Road wear (number of equivalent 10-ton axles)

Road traffic safety (number of vehicles and drivers on the road)

Vehicle configurations to be included in the analysis:

Load lengths:

13.6 m

7.82 m + 13.6 m

13.6 m + 7.82 m

7.82 m + 7.82 m

Vehicle weight: max 60 ton

Vehicle length: max 25.25 m

2.3 Data collection and processing

During the determined investigation period transport information data should be gathered from the selected data provider's freight registers. Based on the calculated factors decided by the project management group the data processing should then be performed.

For data simulation also official national statistics of goods flows on other similar routes in different European countries on the continent should be collected and prepared.

2.4 Simulation and scenarios

The vehicle systems presently used and the alternative extended vehicle combinations should be compared for the same transport operation in data simulation showing how many less shipments would have been needed by use of the alternative modular vehicle combination. In scenarios based on official statistics of goods flows shipped in long haul transports on the continent the full advantage would be focused by extrapolation. Data simulations by use of regional and national data would thus show the gains in road space and the environmental advantages of road trains with extended loading areas in the European road transport system.

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TABLES & FIGURES

Table 1- Overview project work plan

PREPARATION	March 2001 - April 2001
SPECIFICATIONS	April 2001 - June 2001
DATA COLLECTION/PROCESSING	June 2001 - December 2001
SIMULATION	December 2001 - April 2002
ANALYSIS	February 2002 - April 2002
REPORT	December 2001 - May 2002
NOFOMA/7th Int. Symp.	June 13 th - 20 th

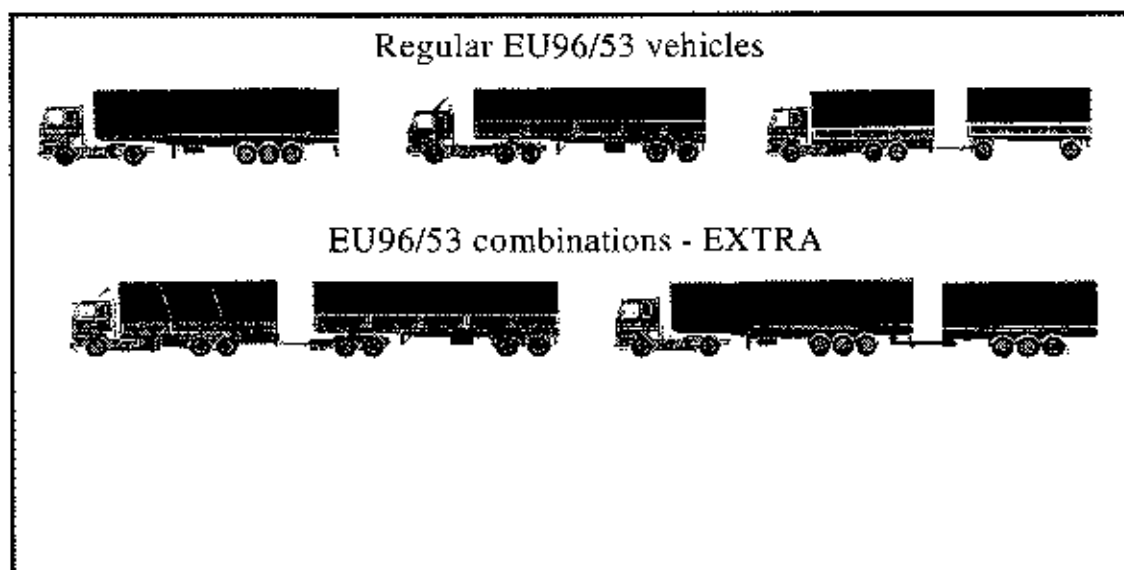


Figure 1 - The same amount of cargo loaded on EXTRA combinations

MICHELIN VIEWPOINT ON WIDE BASE SINGLES AND OTHER FUTURE TRUCK TYRE TYPES.

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ABSTRACT

The White Paper on a common transport policy was published by the European Transport Commissioner in September 2001. For goods transportation, the target in the European Union is to reduce the growth rate of goods road haulage to 38% by 2010 in comparison with its 1998 level. The main concerns leading to this decision are road traffic congestion, road safety and road haulage environmental effects.

As a responsible tyre manufacturer committed to sustainable development, Michelin strives to contribute to an acceleration of road transportation progress regarding these concerns.

The wide base single is part of this commitment. With lower rolling resistance, leading to reduced fuel consumption and subsequent environmental impacts, it has a comparatively lower weight, resulting in a greater payload and a reduced amount of material during manufacture and dismantling after use. The X One tyre range was developed with these criteria in mind. It is already on the European market for urban buses and available in North America for long haul and regional transportation. Other projects are in progress in order to extend this range to other uses.

Other notable and envisaged trends for future truck tyre types are :

- *A continuation of traditional tyre performance improvements, such as reliability, wear life, rolling resistance, grip, noise, dynamic active safety....*
- *The introduction of new tyre sizes in order to increase the payload volume, through the design of smaller diameter tyres.*
- *The introduction of electronically instrumented tyre-wheel assemblies offering new services.*
- *In general, a more integrated tyre design covering the entire life cycle of the tyre, targeting the reduction of societal impacts, including their effects on the road surface.*

GENERAL CONTEXT AND INTRODUCTION

The White Paper on a common transport policy was published by the European Commission on September 12th, 2001. Unless major new measures are taken by 2010 in the European Union, heavy goods vehicle traffic on the road will increase by almost 50% in comparison with 1998 levels. The approach endorsed by the White Paper consists of a number of measures to reduce the growth rate of road haulage to 38%, through more effective use of other means of transport. This level is the equivalent of a return to the 1998 market share for road transportation. Even in these conditions, important and urgent efforts must be taken by the road transport industry, including truck and tyre manufacturers, road authorities etc, to progress towards sustainability and obtain acceptable or improved levels of road congestion, pollution emissions, noise, road safety and transport efficiency by 2010.

Wide base single tyres are part of the answer to this challenge.

WIDE BASE SINGLE TYRES

Tyre manufacturers introduced wide base single tyres about thirty years ago as a replacement for dual tyre fitments on towed vehicles.

Their presence became more widespread in Europe about twenty years ago, with the adoption of five axle, 40 tonne articulated vehicles.

The advantages of these tyres for operators are significant; their lower assembly weight allows payload increases, and their decreased rolling resistance reduces fuel consumption.

They also allow an increase of the semi-trailer track, leading to improved safety achieved through better rollover stability.

Recent advances in tyre technology now offer a further step forward : the introduction of a new wide base single tyre range for the drive axle of motor vehicles in Europe as well as for the drive and towed axles in North America. This new range is called X One for Michelin products.

The advances required in tyre technology deal primarily with casing endurance, both in the crown and in the bead regions. Michelin developed new and patented solutions for the X One range, for instance "Infinitcoil"TM for the crown region.

The industrial production and commercialisation of the X One range started with two different configurations. North American long haulage 6x4 tractors and semi-trailer articulated vehicles in November 2000 and European Urban Buses in May 2001.

The most important customer benefits obtained with the Michelin 445/50R22.5 X One XDA and XTA on 80000 lbs 6x4 tractors and tandem axle semi-trailers are increased payload and reduced fuel consumption. For these vehicles, the two tractor drive axles and the two semi-trailer axles may be fitted with X One tyres replacing the 275/80R22.5 dual assemblies. See figure 1 for an example.



Figure 1 - Michelin 445/50R22.5 X One tyres on North American tractor and semi-trailer.

When fitted on the tandem axles of both the tractor and the semi-trailer and compared to Michelin's best twinned solutions, the payload increase is around 850 lbs (385 kg) and the fuel consumption reduction is about -4%. Fuel costs account for about 20% of the haulier's operating costs and these results lead to a very significant improvement in this respect.

Six X One tyre sizes are commercially available on the North American market:

445/50R22.5 X One XDA and XDA-HT for 6x4 tractor drive axles. Replacement of 275/80R22.5 tyres.

445/50R22.5 X One XTA and XTE for tandem axles of semi-trailers. Replacement of 275/80R22.5.

455/55R22.5 XDA-HT and XTE respectively for 6x4 tractor drive axles and for tandem axles of semi-trailers.

Replacement of 11R22.5 and 275/80R24.5 tyres.

Additional information about the interest of X One tyres for the North American transportation industry can be found in ref [4]. This paper details the improvements offered in weight, fuel economy, handling, pass by noise and comfort, while offering at the least equivalent tread wear and acceptable rapid air loss control in comparison with an equivalent dual tyre set.

The 495/45R22.5 tyre size was designed by Michelin for the drive axle of motor vehicles in Europe and is now also proposed by other tyre manufacturers.

Its societal impact, when compared to twinned 315/70R22.5, is described in detail in the COST 334 report, chapter 7, "Examination and consequences of using tyre types". A cost-benefit analysis was carried out, taking into account road surface maintenance costs, vehicle operating costs and environmental costs (mainly pollution emissions and manufacturing of material and recycling). It shows that the use of this tyre on the drive axle of European heavy goods vehicles and wide base single tyres of the same diameter on the trailer or semi-trailer would not significantly modify the road surface maintenance costs but would reduce the vehicle operating costs and environmental costs. Globally, the annual cost reduction would be about 600 million Euros at EU level.

This new tyre size is not yet commercially available for 4x2 trucks and tractors.

Indeed, a concern is what effect would the single tyre concept have on vehicle stability in the occurrence of a rapid air loss during road use. For a current dual tyre set if one of the tyres suddenly fails, usually one good tyre remains on the wheel-end to help support the vertical and traction load of that wheel-end. For similar reasons, there is no safety issue for vehicle stability for 6x4 tractors and semi-trailer North American configurations (ref 4).

Obviously this is different for the 4x2 trucks and tractor combinations.

Common studies are in progress with truck manufacturers to find practical solutions and reach the safety level of dual tyres. They will be developed within a relatively short period of time.

This safety issue does not exist in urban use, which is operated at very low speeds and transverse accelerations. It is the reason why this dimension is currently on the market for European urban buses, under the name 495/45R22.5 X One XDU.

The most significant customer benefit obtained from the use of this tyre in urban buses is an increased aisle width at the rear end of the bus. See figure 2 for an example.

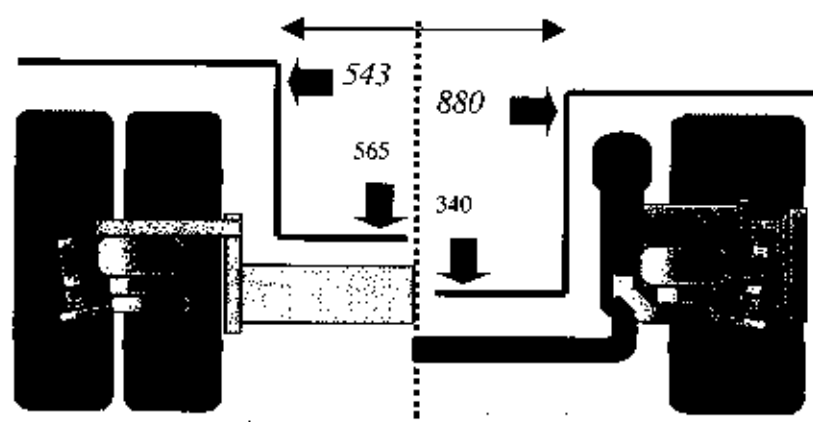


Figure 2 - Comparison of Irisbus Agora and Cavis aisle width.

The 495/45R22.5 X One XDU tyre size has a maximum load capacity of over 13000 kg per axle for urban use. This load capacity is necessary for France, which authorises such a load on its road network. A new tyre size, the 455/45R22.5 X One XDU will be available on the market very shortly, with a maximum load capacity of 12190 kg per axle for urban use. This load capacity corresponds to the requirements of all the other EU countries. This new size will still allow bus manufacturers to increase aisle width by several centimetres.

Additional information about the X One relation to the road surface in terms of response and wear can be found in refs [1], [2] and [3]. They show that this new generation of wide base single tyres is much less aggressive to the road surface than its predecessors.

Michelin's viewpoint regarding the new wide base single tyres is that they represent a major breakthrough for urban buses and heavy goods transportation (in the near future, for 4x2 trucks and tractor combinations).

In addition, COST 334 results make it clear that, considering vehicle operating costs and environmental costs as well as road surface maintenance costs, they reduce the overall societal costs related to heavy commercial transportation.

Thus there is no obvious reason to restrict their use in the European Union, as was the case with the former generations of wide base single tyres.

In particular, the load restriction applied to the tandem drive axles fitted with single tyres (18 tonnes maximum load instead of 19 tonnes with dual tyres) in directive 92/53 should be deleted.

OTHER FUTURE TRUCK TYRE TYPES

Naturally, the future truck tyre types will continue traditional improvements in tyre performance, such as reliability, wear life, rolling resistance, grip, noise, dynamic active safety, ... in order to provide lower operating costs and impacts on the environment as well as safety improvements.

As an illustration, during the last twenty years, the tread life and casing endurance of long haul Michelin tyres has approximately doubled, while their rolling resistance has been reduced by about 30 percent.

Additionally, new tyre sizes will be introduced in order to increase payload volume through the design of a lower outer diameter.

For an example, see figure 3. From this diagram, based on existing commercial tyre sizes, it can be seen that switching from an outer tyre diameter of approximately 1100 mm (315/80R22.5) to about 940 mm (295/60R22.5), the payload volume is increased by 10 m³, without any change in vehicle dimensions.

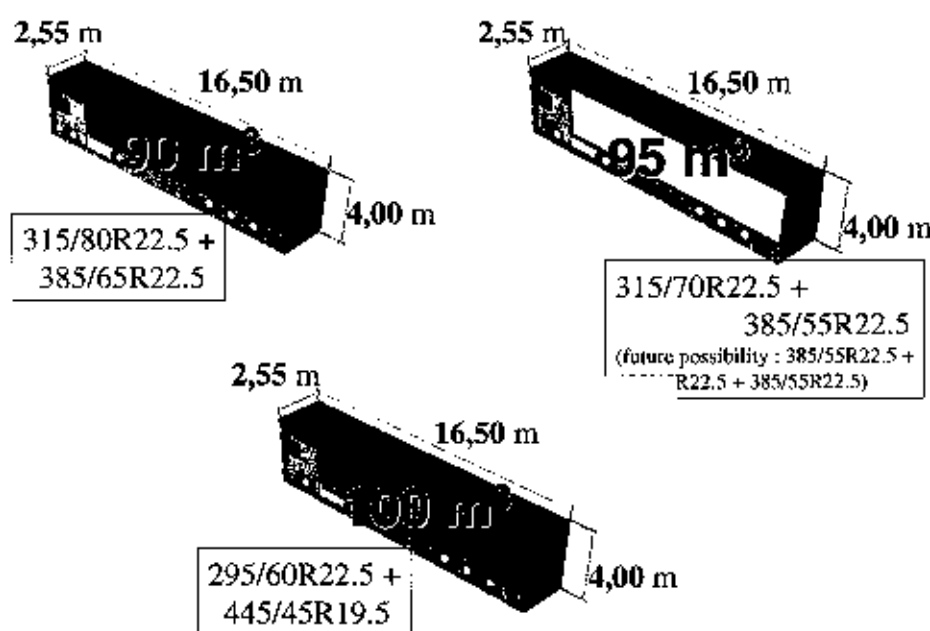


Figure 3 – Evolution of payload volume according to tyre size.

The trend for European transportation is to move progressively from bulky transportation (coal, steel, wood ...) to more sophisticated goods transportation with lower density. This evolution requires vehicles with greater volumes and this trend is set to continue. Smaller diameter tyres will be offered, in close co-operation with vehicle designers in order to maintain or simultaneously improve braking efficiency.

Another important trend for future tyre types is the introduction of "electronic" or "smart" tyre-assemblies.

These new products will include electronic devices and provide new services.

The first product of this kind to be available in the heavy vehicle business is IVTM (Integrated Vehicle Tyre Monitoring), jointly developed by WABCO and Michelin. This device provides an on-board tyre identification and pressure measurement. It includes an original algorithm, developed by Michelin, able to detect slow punctures and help to prevent major tyre failures. This advance will improve road safety, as well as hauliers operating costs especially in just-in-time transportation. See figure 4 for a schematic description of the system.

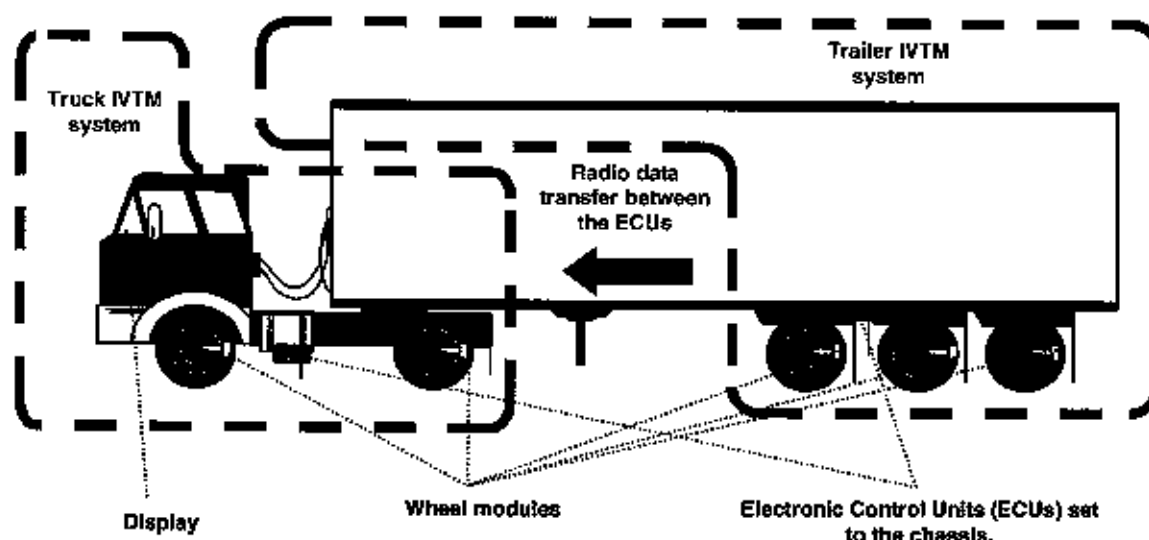


Figure 4 – IVTM general description.

Many new functions are being studied by the tyre and electronics industry. They will lead to new services designed to improve safety and vehicle operating costs, and to reduce traffic congestion and negative impacts on the environment.

In general, the future tyre types will offer a more integrated design covering the entire life cycle of the tyre and targeting the reduction of societal impacts, including their effects on the road surface. For example, with this aim in mind, new tyres with wider treads are progressively launched on the market. Rolling resistance reduction will also remain a very important goal.

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FUTURE EUROPEAN HEAVY GOODS VEHICLES

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ABSTRACT

The White Paper on a common transport policy was published by the European Transport Commissioner in September 2001.

For goods transportation, unless major new measures are taken by 2010 in the European Union, heavy goods vehicle traffic on the road will increase by nearly 50% in comparison with its 1998 level. The approach on which the White Paper consists of a number of measures to reduce the growth rate of road haulage to 38%, due to a better use of other means of transport. This level corresponds to a return to the 1998 market shares levels of the different transport modes.

Even in these conditions, important and urgent efforts must be taken by the road transport industry (including truck and tyre manufacturers, road authorities, etc) to progress towards sustainability and obtain by 2010 acceptable or improved levels of road congestion, polluting emissions, noise, road safety and transport efficiency. This is a difficult challenge. However it has to be achieved and, taking into account the present and envisaged infrastructure geometry of 2010, several new vehicle configurations can be defined in a win-win approach. They are intended to reduce the number of vehicles and the congestion they contribute to, the fuel consumption and the corresponding polluting emissions, the pavement damage and maintenance costs and to improve the vehicle stability and safety as well as to improve road transport efficiency. These vehicle configurations do not meet present European Regulations but are viewed to be possible responses to the challenge mentioned above. They are described and discussed, with a precise as possible evaluation of their main characteristics, advantages and drawbacks.

The acceptability of these new future trucks by the existing infrastructure will be checked with respect to wear of pavements and bridges, as well as to traffic safety conditions and fair competition pricing.

GENERAL CONTEXT AND INTRODUCTION.

The White Paper on a common transport policy was published by the European Commission on September 12th, 2001. For goods transportation, unless major new measures are taken by 2010 in the European Union, heavy goods vehicle traffic on the road will increase by nearly 50% over its 1998 level. The approach the White Paper is based consists of a number of measures to reduce the growth rate of road haulage to 38%, due to a better use of other means of transport. This level corresponds to a return to 1998 levels of market share for road transportation.

Even in these conditions, important and urgent efforts are to be made by the road transport industry (including truck and tyre manufacturers, road authorities, etc) to progress towards sustainability and obtain by 2010 acceptable or improved levels of road congestion, polluting emissions, noise, road safety and transport efficiency.

In this respect, a reflection group was initiated between Michelin, Renault V.I. and the Laboratoire Central des Ponts et Chaussées (LCPC).

It has been started on a purely French basis, for convenience reasons and a good existing relationship, but has the potential to be extended to a European Group.

The following parts will cover :

- Possible ways identified for improvement and their potential effects.
- Action plans.
- Conclusions.

POSSIBLE WAYS OF IMPROVEMENT AND THEIR POTENTIAL EFFECTS.

From the beginning of the reflection, two ways appeared as possibilities to improve the situation as regards road congestion, polluting emissions, noise, road safety and transport efficiency. The first four requirements are the

societal ones, expressed by the European Commission. The fifth and last one represents the possibility to achieve a win-win evolution for the Society and the transport industry and therefore guarantee its success.

These two ways of improvement are "heavy vehicle weights and dimensions" and "interactivity between the vehicles, tyres and infrastructures".

Heavy vehicle weights and dimensions.

The target, in this respect, is to find the best possible solution of heavy vehicles' weights and dimensions allowing the preceding criteria to be optimised: minimise road congestion, polluting emissions and noise as well as maximise road safety and transport efficiency.

The most likely direction is to increase the volume and weight available for the cargo. Indeed, it is the only way to reduce the vehicles number and to develop the use of the most efficient engines.

Due to infrastructure reasons it is not possible to increase either the vehicle height (bridges' height) or width (roads' width). Thus, the only possible direction is to increase the vehicle length, together with weight and axle number.

Present European Regulations (Dir 96/53) allow, for the heavier vehicles with five axles or more, the following features :

- Articulated vehicles : Maximum length : 16.5 metres. Maximum weight : 40 tonnes.
- Road Trains : Maximum length 18.75 metres. Maximum weight : 40 tonnes.

The resulting most common European heavy vehicles are :

- Articulated vehicles : 4x2 tractor and tridem axle semi trailer.
- Road trains : Three axles 6x2 truck and two single axle trailer.

Some former studies have been carried out, which give clues for the possible future optimised solutions.

In particular, in COST 334, a 6 axle configuration with a maximum weight of 48 tonnes was envisioned and compared to the present 5 axle 40 tonnes configuration.

The axle positions and chassis layout would be set so that the maximal axle weight would be homogeneous and equal to 8 tonnes. The tyres would be the same on all axles and would consist of low rolling resistance extra-wide-base single tyres. This solution would optimise the fuel consumption and reduce the road aggressivity, especially on the motor vehicle.

The vehicle length was not explicitly analysed but it was assessed that the load rate (ratio of actual payload versus maximum payload) was maintained, which implies a proportionality between maximum payload and cargo length.

The comparison was made regarding pavement damage, environmental impact (polluting emissions, noise, recycling) and vehicle operating costs.

The main conclusions were as follows :

1. The use of such vehicles would reduce the number of heavy vehicles by about 20% for the same traffic (in tonnes.km) when compared to present vehicle configuration.
2. The pavement distress would also be reduced by about 20% for the same amount of traffic. The pavement maintenance cost would consequently be reduced.
3. The fuel consumption would be decreased by about 14% for the same traffic. This benefit would both reduce the polluting emissions (including CO₂) and reduce transportation costs.
4. Safety would be improved by the use of extra-wide-base single tyres, due to better road handling. It could also benefit from the use of two driven axles on the motor vehicle. However such a feature would reduce payload capacity and increase fuel consumption and vehicle cost.
5. Haulage productivity would be improved due to the increased payload handled by each driver.
6. Tyre and wheel recycling would be improved because of the decrease of tyres and wheels material weight used for the same traffic.
7. Driver and payload comfort would be improved by the use of the extra-wide-base single tyres. This benefit could result in driving being less tiring (inducing improved safety) and possible savings in packaging of the payload.

The case of present Nordic 7 axle - 60 tonnes - 25.25 m vehicle was also envisioned by COST 334. It showed a very low road-aggressivity level per transported tonne of payload (about -30%). Even if the case study was not

carried out as precisely as the previous, this vehicle configuration would lead to very important progress in vehicle number reduction (about -35%), haulage productivity and probably fuel consumption reduction.

These examples show that there should be room for a win-win optimisation between the societal improvements required (congestion, pollution, safety) and the haulage industry.

In order to ensure the acceptability of such possible future truck designs, together by the infrastructures, by the traffic and by the national and European authorities, the following matters should be assessed:

- The maximum load effects on all types of bridges should not be increased, because it would be extremely expensive to perform reinforcement of all the existing bridges; local and distributed loads should be maintained within reasonable limits, i.e. those taken into account to calibrate the Eurocode EN1991-1-3 on 'Traffic Loads on Road Bridge' (Flint and Jacob, 1996). The issue of fatigue lifetime will have to be carefully investigated for steel and composite bridges, which are sensitive to repeated loading (Jacob, 1998).
- Longer and heavier trucks will have to be much safer than the current ones, in order to reduce the risk of accident, either induced by the truck itself or by other vehicles. In order not to disturb the traffic flow, some operation regulations may be necessary to be taken (minimum spacing, special routes or time for operation, etc.).
- It will be important to ensure a fair competition between transport modes, and between different types of road vehicles. The pay-load or gross weight of a truck should be one of the parameters of the tax or toll fees. This will require automatic weighing systems, such as WIM (Weigh-in-motion) equipment's, to be installed and operated (Jacob, 1999; WAVE, 2001; COST323, 2002).

Interactivity between vehicles, tyres and infrastructure.

The introduction of "smart" technologies inside vehicles, tyres and infrastructure presents a significant new potential for innovation in the field of road transportation.

Indeed, new types of information can be measured, acquired and transmitted simultaneously to all the actors.

For instance, we can anticipate the generation of many types of information by :

- The vehicle : Identification (type, engine, maximum load per tyre position...), geographic position, speed, acceleration, driver behaviour, travel optimisation type (time, economy...), status (OK, failure, breakdown, accident...), available payload capacity, forecasted route, actual load per tyre position, distance from the preceding vehicle...
- The tyres : Identification (type, age, load index, speed index...), inflation pressure, temperature, revolution count, wear, efforts in the contact area, strains, pavement grip....
- The infrastructure : Identification (motorway, trunk road, bridge,..., speed limit, weight limit...), status (dry, wet, icy, snowy..., unevenness, potholes, grooves...), temperature and other meteorological conditions (fog...), wheel, axle and/or vehicle weight, warnings (works, accidents...), distance from the preceding vehicle, presence of pedestrians on pedestrian crossings...

Some of these possible information sources can be combined in a relevant way in order to reach a specified goal. For example, fit trucks with tyres able to measure the local grip on the surface they are being driven on, detect slippery conditions (ice...) and transmit the information to the truck. The truck will then be able to include the relevant parameters in its electronic stability program (ESP) and correct its speed and applied torques. It will also be able to locate the concerned area with its GPS sensor and, with its GSM or other communication means, transmit the information to the infrastructure management systems. These systems will then be able to activate road surface salting, in the case of icy conditions and inform the other vehicles entering the concerned zone. The other vehicles will then integrate the information in their GPS/ESP systems to avoid any damage, follow its evolution with their tyres, give back information to the infrastructure and so on.

An alternative could be to equip identified problematic sections of the road network with appropriate sensors instead of equipping the vehicles' tyres.

This choice should be made by balancing the costs and benefits of the two possibilities.

Many other vehicle-tyres-infrastructure interactions can be imagined, such as virtual high speed truck trains (safety, congestion, pollution), inside-tunnel safety systems, on-road dynamic force limitation (road surface damage limitation)...

These examples show there is room in this field for dramatic improvements. However "smart" technology is rapidly evolving and becoming obsolete nowadays. Also this kind of evolution can be very expensive for the different actors. It is thus necessary to stand back and reflect carefully with the help of the different actors before defining a relevant and realistic target.

ACTION PLANS.

This action was decided and launched in 2001. The meetings held in 2001 were used to deepen and organise the ideas and concepts and to define an action plan starting in 2002. It is now described for the two discussed ways of actions.

Heavy vehicle weights and dimensions.

The provisional action plan is shown in figure 1. The project duration should be 3 years, including the building of a demonstration vehicle and the validation of its technical and economical performance.

The first part, with an 18 months duration, will be a feasibility study.

In a first step (about 9 months), the existing situations will be thoroughly examined regarding the different aspects

- Vehicles : Existing configurations linked with local regulations, in the EU, North America, Australia.
- New concepts recently proposed by the truck manufacturers, with their described benefits.
- Impacts of the possible vehicle configurations on the users (payload, costs, drivers' numbers and skills, specific driving licences, possible re-use of existing vehicles, logistic platforms, modularity, vehicles resale...).
- Impacts on the "external" aspects :
 - Traffic (insertion in traffic, manoeuvres, itineraries, overtaking by passenger cars, visibility including during raining and winter conditions...).
 - Infrastructure development (roads, bridges, crash barriers, escape lanes, roundabouts, service areas, tunnels ...).
 - Safety (accident seriousness, collisions, emergency manoeuvres, road handling, rollover, braking distances, sweeping radius, vehicle stability and braking in curves...).
 - Environment (emissions, noise, tyres and pavement wear particles, recycling ...)
- Impacts on the regulations, including vehicle lights...
- Impacts on the truck manufacturers (engine power, brakes, stability, ground clearance (logistic platforms, ferry-boats, intermodal shifts...))

In a second step, the most promising new possibilities for the EU will be listed and investigated. Their feasibility and technical and economical performances will be investigated.

The second part will also have a 18 months duration.

It will consist in designing, building and testing a full scale vehicle demonstrator.

All the technical and economical performances of the retained vehicle configuration will be checked and, based on the results, proposals will be made.

Interactivity between vehicles, tyres and infrastructures.

The provisional action plan is shown in figure 2.

It covers a 18 months period including the feasibility study and a preliminary 6-month period dedicated to the target definition.

The target definition will be achieved through the following process :

- Extensive examination of all the possible ways of interaction between tyres and vehicles, between vehicles and infrastructures, and finally between tyres and infrastructures will be carried out.
- From all these ideas, possible scenarios will be built. They will be designed with the goal of bringing improvements on road congestion, polluting emissions, noise, road safety and transport efficiency. They will also involve the three parties, vehicles, tyres and infrastructures.

Then the three or four best ones will be retained and a feasibility study will be carried out for each of them.

This feasibility study will deal both with the technical and the economical point of view.

This 18-month period will be followed by building the demonstrator and evaluating it but, in this particular field, we are not yet able to draw realistic outlines. Indeed the target definition must first be defined.

SUMMARY.

An action was started by Michelin, Renault VI and the LCPC in order to help reach the objectives of the White Paper on European transport policy.

This action will concern the whole transportation system, including the vehicle, tyre and infrastructure.

Two major parts will be dealt with, the vehicles weights and dimensions and the "interactivity".

Started with French partners, it is open to other European actors in order to become a European project.

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FIGURES

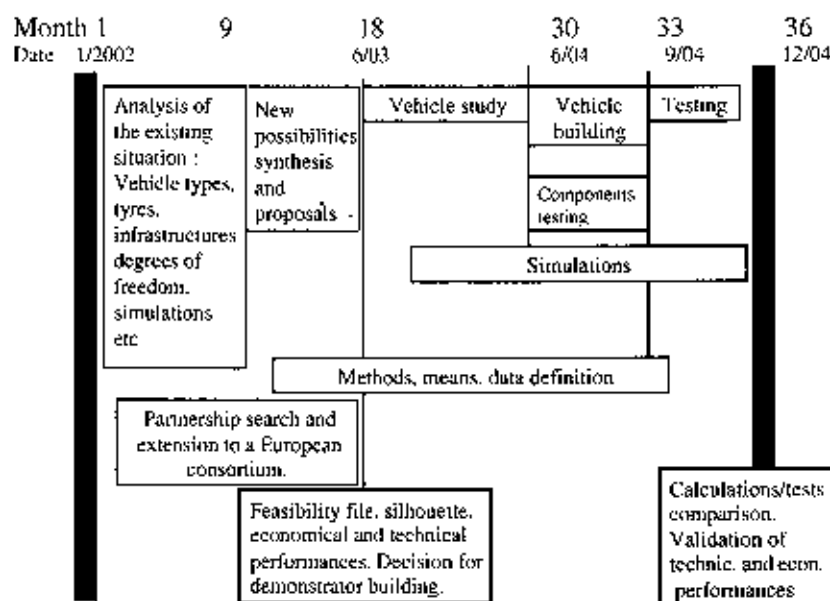


Figure 1 – Action plan : Heavy vehicle weights and dimensions.

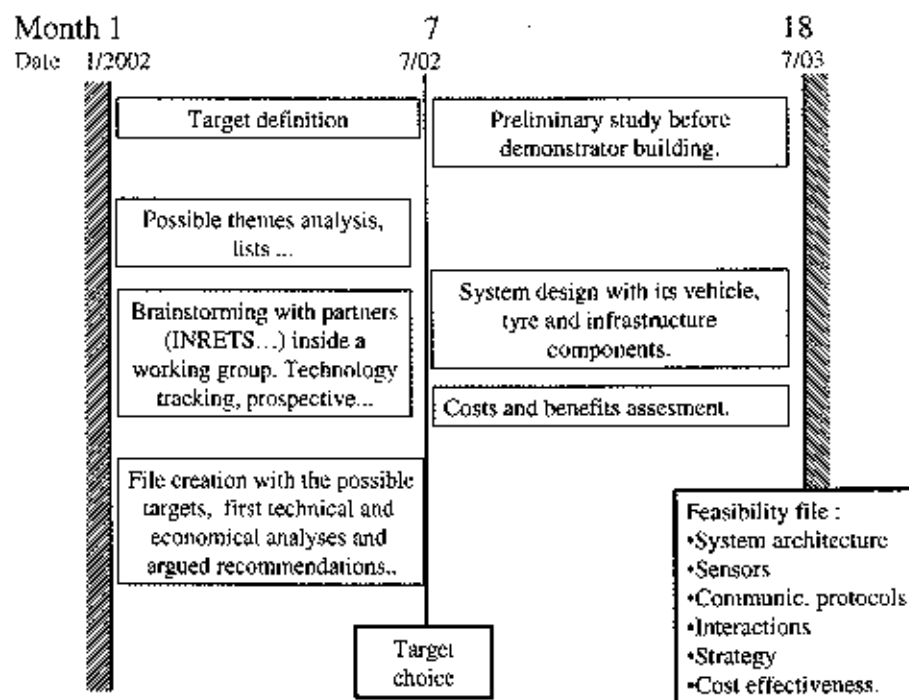


Figure 2 – Action plan : Interactivity between vehicles, tyres and infrastructure.

MANAGING ROAD TRAIN ACCESS IN A LARGE CITY: PERTH, WESTERN AUSTRALIA

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ABSTRACT

The western third of Australia is the State of Western Australia. It occupies a land area of 2.5 million square kilometres and has a population of 2 million of which 1.3 million live in the city of Perth. The geography of the State – outside Perth, lends itself to the use of road trains. The distances are large; the traffic density is low; the country is flat; and the freight task is quite small and dispersed. Terminating road train use on the outskirts of Perth is annoying and inefficient for the trucking industry and during the 1990s their access and use in Perth grew rapidly. But by the late 1990s the community was objecting and a new Government in early 2001 initiated a far ranging review to establish a consensus policy on the regulation of road trains. The review scope covered the entire State but the policy established for their regulation in Perth is the most relevant for the rest of the world.

The consensus approach used was a first for Western Australia and involved engaging a wide range of stakeholders in consensus conferences, followed by a small group distilling the findings into a workable and pragmatic policy. The outcome is a set of rules and regulations developed on a policy framework with the following eight components:

- ① *Integrating land use and transport planning*
- ② *Designating freight roads*
- ③ *Involving the community*
- ④ *Improving freight roads and intermodal facilities*
- ⑤ *Accrediting operators*
- ⑥ *Publicising rules and regulations*
- ⑦ *Enforcing the rules*
- ⑧ *Training and informing all road users*

This paper looks at the process used in developing and implementing the policy, and provides details of the policies and guidelines that are now used in Perth.

INTRODUCTION

Remote parts of Australia make extensive use of large trucks. Public roads in some areas have trucks hauling mineral concentrates with gross masses of nearly 200 tonnes and overall vehicle lengths of more than 50 metres. The economics to operators are significant and as a consequence there is considerable demand to extend the network of roads made available to the longest and heaviest trucks.

In Western Australia vehicles up to 36.5m long and weighing up to 100 tonnes are now permitted on some roads in Perth, a city of 1.3 million people. But the speed of their introduction and the process used in introducing them was not acceptable to many in the community. It contributed to the fall of a state government and led to a reappraisal of the approach taken to considering applications from operators to use larger vehicles. The outcome is a more rigorous and holistic approach with input from all concerned members of the community.

PERTH

Perth is a city of 1.3 million people on the western coast of Australia. Most people try and live close to the coast to take advantage of the Mediterranean climate – a cool wet winter and a hot dry summer. As a consequence Perth is very long – some 100 kilometres from the north to the south yet only 20 kilometres from the east to the west. It is also very spread out. Land values on the fringe are low and along with a well-developed road system there is a growing urban passenger rail network. Perth is quite flat with few serious waterways making for low cost road building.

LAND USE AND TRANSPORT PLANNING

Perth has been developed on the basis of a town-planning scheme – known as the Metropolitan Region Scheme (MRS), since the 1960s. This has resulted in industry and urban development in prescribed areas and the development of roads – in the main, to serve the city. It has however meant that there has been a need to build some major roads through areas that have been suburban housing estates for a long time and this has been very difficult. The MRS also placed some road reserves in wetlands, as at the time of the MRS creation these areas were considered wastelands. This is now not the case in many people's minds and it has become very difficult to build these roads – despite the land being reserved and never having been occupied.

SEA PORT

The Port of Fremantle is Perth's seaport. It was originally at one location – at the mouth of the Swan River, but in the past 50 years has expanded with the development of specialised berths to the south adjacent to new industrial areas and at locations better suited to the rail and road network. This has been very successful, with one exception. The original facility has been developed as Perth's sole container terminal. Container trade has grown rapidly (355,000 TEUs in 2000/01 – up 18% on 1999/2000) and nearly all of them get to and from the Port by road. Rail access is poor to this part of the Port and in addition most containers need to be moved by road as they are going to and from dispersed warehouses in Perth. The road outcome is the need to accommodate large numbers of trucks through highly desirable urban areas on inadequate roads with the local community strongly resisting the construction of suitable roads.

The truck movements in and out of the container port are significant. A survey carried out on 10 December 2001 on the access road to the Port provided the following statistics:

- 8490 vehicles;
- 2345 trucks comprising 260 'long vehicle' road trains, 146 B doubles, 1158 five and six axle articulated trucks and 792 mainly smaller trucks; and
- 93% of truck movements took place between 7am and 7pm.

About 4 million tonnes of freight were moved in and out of the port on this access road in 2000/01.

CHANGING TRUCK MASS AND DIMENSION LIMITS IN PERTH

Until the early 1980s trucks in Perth were no bigger than regulation limits – at the time 17m long and 38 tonnes. But the head of the State's road authority was authorised to permit larger vehicles wherever he felt it was in the interests of the community at large – on any public road. This authority stemmed from the fact that remote areas of WA had for 30 years been making extensive use of road trains.

In these areas traffic numbers were very low, the distances were very long and the country very flat and forgiving. But these trucks wanted to get closer to the bigger towns – and Perth. In the early 1980s permits were issued to allow some trucks in Perth to exceed the regulation mass limit – but not the length. This was trialled for some time. Truck and trailer combinations with a length of 17 metres and a gross mass of about 55 tonnes became quite common. In the mid 1980s this was followed by trials of two 17 metre long Canadian B-trains. These were designed to carry two 20-foot containers – without being overloaded, as was often the case with conventional

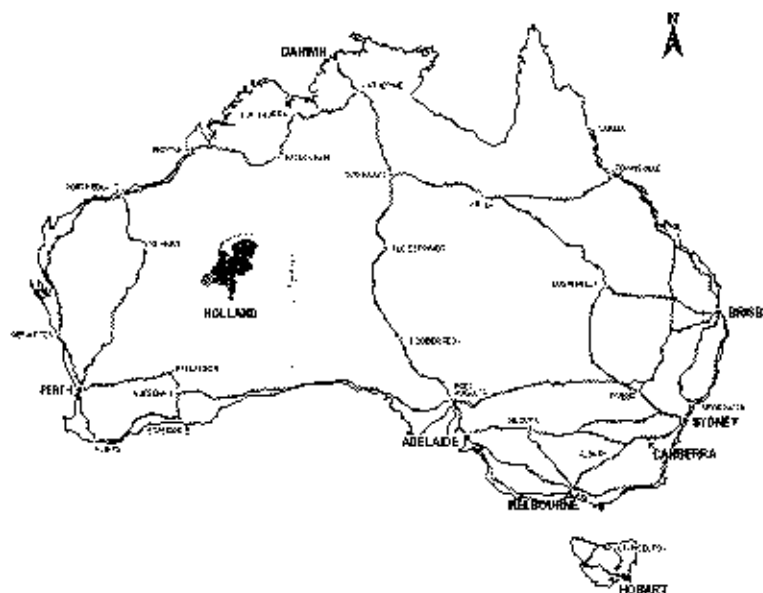


Figure 1: Perth, Australia and Its Major Roads

semi-trailer vehicles carrying two containers. The reform process was slow and cautious. But as was the practice then there was no involvement by the community or local government – something that in the present environment would not be tolerated.

THE 1990S

This cautious approach continued until the early 1990s. Then changes took place rapidly with the Government of the day keen to make economic progress. The State's economy grew rapidly and the Government's strong growth policy meant deregulation took place quickly. In the case of road trains it was very fast and largely without consultation. Rudimentary trials took place and provided the vehicles more or less kept in lanes routes were opened up to road trains – small ones initially but by the end of the 1990s vehicles up to 36.5 metres and over 100 tonnes were using parts of Perth's road network.

The process used in making the decision to allow these vehicles on Perth's roads left a lot to be desired. The deregulation environment brought with it quality assurance schemes which worked well in some cases but not others. During this period of rapid growth, deregulation and privatisation it was not possible to be sure that all checks and balances were taking place to ensure reforms were all in the interests of the community. Research, data collection and evaluation were minimal.

THE PROBLEM VEHICLES – 'LONG VEHICLES AND ROAD TRAINS'

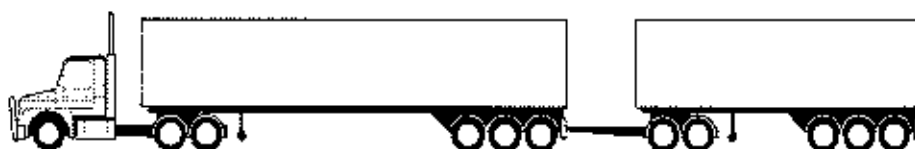


Figure 2 - 'Long vehicle' road train, Up to 27.5m long (GVM to 88t with container concession)



Figure 3 - B Double up to 27.5m long (GVM to 76.5t with container concession)

The vehicles shown above have access to 97% of the State's major road network including an extensive network through the Metropolitan area.

The following vehicles, which are longer than 27.5m, have limited access to the Metropolitan area.



Figure 4 - Double Road Train up to 36.5m long (GVM to 86t with concessions)



Figure 5 - Rigid truck plus two dog trailers up to 36.5m long (GVM to 102.5t with concessions)

The metropolitan network available to these vehicles is shown in Figure 6.

PERTH'S FREIGHT TASK

Perth does not have a large manufacturing sector but nevertheless has a considerable demand for road freight movement. Perth is an isolated and wealthy city, where much of the wealth is invested in housing. The houses are large and relatively low cost to build, as the climate is forgiving and the land good to build on. Most houses are masonry in construction and heavy! The wealth and large investment in housing result in the demand for a lot of freight movement – through personal consumption and housing.

Perth is also the focal point for all activities in Western Australia outside of the city. All regional centres are largely serviced from Perth and Perth is also the major destination for the agricultural products coming from the agricultural region – a region that produces considerably more grain, meat, wool, timber, wine and dairy products than is consumed in Western Australia.

And finally although Perth is not a large manufacturing city it does produce much of the hardware used by the mining and petroleum industries – the industries that are the key source of Perth's wealth.

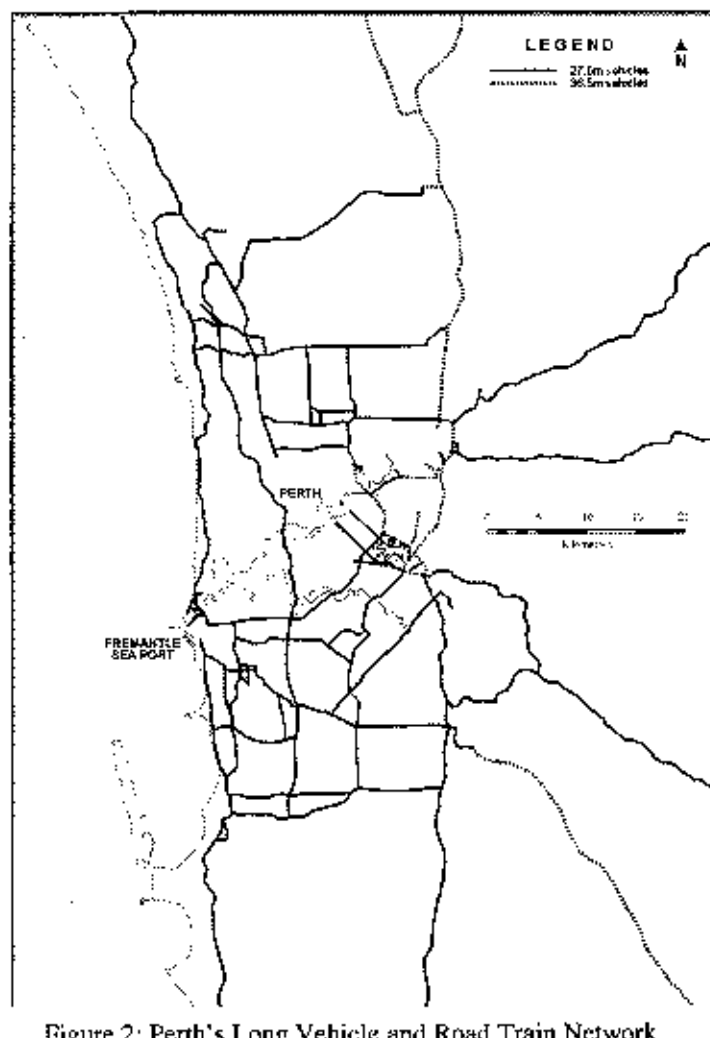


Figure 2: Perth's Long Vehicle and Road Train Network

WHAT IS THE PROBLEM?

The sum of all of these tasks is large and diverse. And as most of the road network is capable of taking the biggest trucks there is constant demand to extend the network available and in particular to gain greater access to the metropolitan area. Being able to use certain trucks on all but a very limited amount of the network is frustrating to the industry, but succumbing to their demands without adequate community input proved to be unwise.

There is no doubt that people do not like trucks on the road. They would prefer freight to be moved by rail. In Perth the trucks grew rapidly in size and number in the 1990s and the community resented this and the way in which it happened. The challenge was to go to the people and work with them on how to manage Perth's trucking needs. The safety record of the vehicles is unclear. Their speed compliance is relatively good as shown in Table 1.

A speed survey covering 200 sites and over 2.3 million vehicles in sites with speed limits from 60 km/h to 110 km/h showed that Long Vehicles and Road Trains were less likely to exceed the enforcement speed limit than any other vehicle class.

Only 2.9% of the 26,000 of these vehicle surveyed were recorded as exceeding the enforcement speed limit. This compares to 11.5% of the 2 million cars.

Table 1 - Vehicles Exceeding Enforcement Speed Limit in Western Australia - 2000

Vehicle Type	Metropolitan Area	Rural Area
Light Vehicle (mainly cars)	11.3%	11.2%
Rigid Truck	25.6%	13.3%
Semi-Trailer	5.5%	2.4%
Long Vehicles and Road Trains	4.7%	1.2%
Total	12.7%	11.1%

Their crash record is however less clear. The crash statistics for 2000 show a disproportionate number of casualty crashes involving heavy vehicles occur in the Metropolitan area. A casualty crash involves a fatality, a person being hospitalised or a person requiring medical attention. The annual number of casualty crashes involving trucks in the past 10 years is generally unchanged except for 'Other Trucks' in the Metropolitan area. The composition of 'Other Trucks' is not exactly known, but it does include the Long Vehicles and Road Trains. The crash statistics do not adequately discriminate between the types of heavy vehicles.

Table 2: Crash Rates for Heavy Vehicles in Western Australia - 2000

Truck Type	Number	Casualty Crashes		Casualty Crashes per million vehicle kilometre	
		Metro	Rural	Metro	Rural
Rigid Trucks	38511	212	44	0.6	0.1
Other Trucks	8694	88	54		

NATIONAL REFORM

The frenetic pace of change in the 1990s was taking place at the same time as national reform of the heavy vehicle industry was also taking place. In the early 1990s a national body – the National Road Transport Commission (NRTC) was established to put in place uniform national regulations in areas such as vehicle size and weight and charging. Western Australia was a participant, albeit reluctantly on many issues and slow to implement agreed reforms. It was the laggard when it came to national reform, arguing that the national reforms did not suit the special circumstances that applied in Western Australia.

STAKEHOLDER INVOLVEMENT

There were no formal links between the state road authority and stakeholders until 1999. Prior to then stakeholder involvement was through deputations to the Government and Commissioner of Main Roads, and through dialogue between Main Roads people, industry and local government. This meant decisions such as whether to allow road trains on a road were generally being made without any direct involvement by people on the route or others who might be directly impacted.

In 1999 a Heavy Vehicle Advisory Group (HVAG) was established with the following membership:

- Main Roads (Chair)
- Department of Transport
- Police
- Local Government
- Trucking Industry
- Chamber of Commerce and Industry

At this time the Western Australian Road Freight Council (WARFC) was also formed. Its predecessor had been disbanded in the mid 1990s. The membership of the WARFC was similar to the HVAG.

Essentially these groups had been put in place because Government and regulators alike knew that formal stakeholder involvement in the management of the road transport industry was needed. The community would not continue to tolerate decisions with major implications for them being made without their involvement.

Main Roads had started to heed these calls and developed and implemented guidelines for assessing the suitability of routes for road trains, commenced detailed community consultation in the case of a major route extension in Perth and the WARFC had Government endorsement to implement an operator accreditation scheme to lift the performance of the industry.

GUIDELINES FOR ASSESSING ROAD TRAIN ROUTES

Rudimentary guidelines were developed in the 1980s but it was not until the late 1990s that serious attempts were made to apply science to the question of whether a road was capable of safely accommodating multi-combination vehicles (road trains). Until then the little serious research that had been undertaken into what was needed in the way of road space for these vehicles had not been drawn together in a comprehensive manner.

Projects such as DIVINE had found out a great deal in regard to the dynamic loading characteristics of heavy vehicles but it was the dynamic stability work undertaken by people such as Sweatman and Prem for Austroads

and the NRTC in the 1990s that gave some indication of how much lane width was required by the various heavy vehicles. The lane width requirement report by Hans Prem for Austroads has proved to be very valuable in helping determine the suitability of roads for road trains.

The Guidelines have grown to a 30-page document covering issues such as:

- Bridges
- Overhead Clearances
- Turning at Intersections
- Railway Level Crossings
- Pavement Widths
- Clearance Requirements in Townsite Areas
- Overtaking and Passing Requirements
- Requirements for Steep Ascending Grades
- Entry Lanes onto Main Roads and Highways
- Off-Road Parking
- Dust, Splash and Spray
- Noise
- Vibration

The Guidelines are making a difference and proving to be very valuable. But there is still much to be done to better manage issues such as dynamic stability, noise and vibration.

CHANGE OF GOVERNMENT

But these moves were too little and too late for the government of the day. They were swept from power in February 2001 with these and other issues dominating the campaign – a campaign with a central theme of listening and involving the community in decision making.

A left of centre government replaced a right of centre government and one of its key platforms was the reform of the heavy vehicle industry in Western Australia, particularly so in Perth. In opposition they had argued strongly that it was out of control. They had committed to two major public reviews:

- A Road Train Summit; and
- A Freight Network Review

and a doubling of the number of transport inspectors. The Road Train Summit has been completed and the key findings are being implemented. The Freight Network Review has commenced but is yet to report.

ROAD TRAIN SUMMIT

This took place in mid 2001. Four consensus conferences were held, each involving up to 100 invited people from all sectors of the community. The people were not only drawn from the traditional stakeholder groups (industry, local government, police, etc) but also from areas where road transport was causing most angst – road train routes, wetlands where roads were planned to be built and roads where strong resistance was being voiced about proposals to permit road trains.

"The Consensus Conference process involved reading papers representing each viewpoint before the Conference. At the Conference, participants listened to short presentations from the authors, added information from the floor and asked questions of the panels. Time was spent ensuring each person's point of view was thoroughly understood. The key issues to be resolved were determined. Options were devised for each of these issues and the extent of consensus for each option was ascertained – ie can live with it, can live with it with modifications, or can't live with it at all. Finally, the most important options were prioritised."

The priorities for action from the Road Train Summit were:

- A more rigorous process for community consultation that is inclusive, transparent and accountable
- An improved process for designating freight routes that provides for enhanced consultation, review and appeal
- A review of all existing heavy vehicle 'hotspots', identification of solutions and prioritisation for budgeting
- A requirement for operator accreditation as a condition for the granting of a permit, to ensure at least minimum standards across the transport industry
- A better resourced, more capable compliance and enforcement system which improves monitoring and provides suitable penalties and sanctions for breaches
- Improved integration of land use and planning policy and decision making, with better consideration of social economic and environmental impacts

- Educating all drivers on safety requirements when sharing the road with heavy haulage vehicles, and provision of better information including identification of freight routes, and publication of processes

The Road Train Summit showed that the community appreciated the importance of road freight transport but also required it to be very safe and considerate. They expected the regulators to ensure this was the case.

IMPLEMENTATION

As history shows it is often easier to come up with good ideas for change than to implement them. Great care is however being taken in Western Australia to make sure that is not the case when it comes to implementing the findings of the Road Train Summit.

- An Implementation Committee under the Chairmanship of the Minister for Planning and Infrastructure has been proposed;
- A new and expanded Heavy Vehicle Branch has been established within Main Roads; and
- A four-year strategy for the Heavy Vehicle Branch has been established.

IMPLEMENTATION COMMITTEE

This Committee is comprised of:

- The Minister (Chair)
- Commissioner of Main Roads
- Key Industry Representatives
- Transport Worker's Union
- Local Government
- Community
- Regulators

It has agreed Terms of Reference and the objective: *"To oversee the implementation of the recommendations of the Final Report of the Road Train Summit (2001)"*. The Final Report of Road Train Summit identified many key actions and each of these is being fully scoped to be managed as individual projects, with reporting and sign off by the Implementation Committee.

HEAVY VEHICLE BRANCH

A Heavy Vehicle Branch in Main Roads has been established, bringing together the previously separate areas of heavy vehicle management and the additional resources to implement the findings of the Road Train Summit.

The Branch is structured around five functions:

- Road Train Summit Implementation – a temporary role that will stay in place while the need exists – between 12 months and 2 years
- Policy – freight routes, regulations, vehicle standards, permit policy, Local Government, coordination, vehicle tracking systems, computer systems, Internet, national policy, data, analysis, strategy
- Operations – permits for long vehicles and road trains, permits for oversize loads, accrediting operators, accrediting pilot and escorts, advice to stakeholders, communications
- Administration – finance, records, correspondence, customer service
- Compliance – field operations, smart compliance, prosecutions

The Branch will eventually have over 60 people, about 20 more than was the case prior to the change of Government and the Road Train Summit.

HEAVY VEHICLE STRATEGY

The Strategy has six objectives and typically six strategies for each objective. The objectives are:

- The stakeholders will be well informed and have a real say in all heavy vehicle issues that impact upon their welfare and quality of life.
- Freight routes will be fit for purpose, identified through close dialogue with the stakeholders and clearly designated.
- People working in the transport chain will be accredited in the competencies and systems needed to make the industry safe and acceptable.
- The level of compliance with road transport law will be in keeping with the expectations of the stakeholders.
- Permits (and notices) to exceed regulation mass and dimension limits will be issued in a timely manner using electronic means where possible, and be supported by readily available policies, guidelines and procedures.
- The Heavy Vehicle Branch will participate in and promote the development and adoption of better heavy vehicle management practices at the national, state and local level.

CONCLUSION

There were probably many reasons why there was a change of Government in Western Australia in February 2001. But without doubt a key factor was the perception that the previous Government was not listening to the people. In the case of Perth truck sizes had been considerably increased and the network of roads on which they could operate had grown rapidly with little or no consultation. Road trains had been admitted, but not called road trains. They were described as 'long vehicles'. Sectors of the community were objecting strongly.

The previous Government had had for most of its term a very energetic transport minister who had the view that he was going to get things done. He did. He drove reform more strongly than any minister in the Government. He doubled the roads budget. He contracted out road construction and maintenance. He contracted out the operation of the bus component of Perth's public transport system. He shut the state shipping service. He was a man of action. He did what he thought was right and if that meant 'bulldozing' an issue through he would do it. But the community grew tired of the approach. The record would probably show that his decisions were generally wise. Transport efficiency improved. The economy grew. Road safety probably did not decline. But his approach was not acceptable. He was replaced by a more moderate minister towards the end but it was too late. The community had decided to change the Government.

The real lesson for people charged with developing and implementing public policy in the heavy vehicle area is that in a modern democracy the community must have a genuine say in issues that impact them. Changes such as adding a lot of big trucks through a community cannot be made without the genuine involvement of the community, and even then it must be done in accordance with world best practice. Due process must be followed in decision-making. To make major changes in the size and access rights of heavy vehicles without the support of the community would be politically 'brave'. It was tried in Perth. It worked for a while. It eventually contributed to the fall of a Government.

INCLUDING PERFORMANCE MEASURES IN DIMENSIONS AND MASS REGULATIONS

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ABSTRACT

Heavy vehicle dimensions and mass regulations exist primarily to maintain safety and to preserve the infrastructure. In terms of these aims, prescriptive regulations are a crude mechanism but they are straightforward to measure and thus compliance checking is relatively simple and cheap. Since the RTAC study, which led to the first of these symposia in 1987, there has been increasing interest by regulators and users in the potential use of performance-based standards either in lieu of or as an adjunct to dimensions and mass rules.

In New Zealand the performance-based approach was adopted very early on for vehicles operating under permit. Vehicles operating within the existing dimensions and mass limits are not required to meet any performance standards for stability. However, New Zealand's dimensions and mass regulations are currently under review with the new rule scheduled for implementation in July 2002. The new rule includes a requirement that all heavy vehicles have a static roll threshold (SRT) greater than 0.35g.

In order to keep compliance costs reasonable an analytical method for determining SRT has been developed and implemented as an internet-based calculator. The calculator has been designed so that the required user inputs are easily obtainable. Default values are used for the other vehicle parameters. For the most significant parameters the user has the option to replace the defaults with real data. Where a vehicle fails to meet the target SRT, the load reduction at current height and the load height reduction at current mass required to achieve the target are both calculated. Validation tests comparing the calculator results with those obtained from tilt table testing and from computer simulations have shown a remarkably good level of accuracy.

INTRODUCTION

Since the Road Transport Association of Canada (RTAC) study of the mid 1980s, which led to these symposia, there has been a growing interest by regulators and road users in the potential of using performance-based standards rather than prescriptive limits to control heavy vehicle dimensions and mass. Ultimately the aim of any mass and dimensions controls is to promote safety without unduly inhibiting the efficiency of road transport operators. Performance measures are more directly linked to safety outcomes than prescriptive limits.

Since the early 1990s New Zealand regulators have required stability performance assessments to be undertaken for vehicles with divisible loads requiring permits to exceed the prescriptive dimension and mass limits. Until now it was considered too costly to require stability assessments for vehicles complying with the prescriptive limits. However, the method for calculating Static Roll Threshold (SRT) described in this paper is straightforward and cost-effective enough to be able to be applied to the heavy vehicle fleet. It is part of the New Zealand Dimensions and Mass Rule 41001 (LTSA, 2001) scheduled to come into force in July 2002.

SAFETY AT REASONABLE COST

Road safety in New Zealand is the responsibility of the Land Transport Safety Authority (LTSA), which is a Crown entity charged with promoting land transport safety at reasonable cost.

The approach that is generally used to determine reasonable cost is broadly as follows:

- Surveys of road users (the public) are undertaken regularly to determine the amount that the average person is prepared to spend to save one life on the roads. Currently this figure is about NZ\$2.8M. Values for serious injury, minor injury and property damage only crashes are also obtained. These values reflect the social cost of road crashes rather than the actual economic costs.
- The benefit of any proposed road safety counter-measure is then determined by multiplying the expected reduction in crashes by the social cost values associated with those crashes.
- The cost of the counter-measure is estimated by including all costs, not just those incurred by LTSA. Loss of productivity by road transport operations, for example, is a cost that is included.
- The decision on whether the counter-measure will be implemented then depends on whether the benefits exceed the costs.

Numbers of performance measures relating to stability and control have been developed. In New Zealand we have used Static Roll Threshold (SRT), Dynamic Load Transfer Ratio (DLTR), Rearward Amplification (RA), High Speed Transient Off-tracking (HSTO), High Speed Steady Off-tracking (HSO), Yaw Damping Ratio (YDR) and Low Speed Off-tracking (LSO). Within these measures there are variations in both the names used and the test procedures so care needs to be taken when comparing results from different studies. Evaluation of the performance measures can be through experimental measurement or by computer simulation. For most of the measures experimental measurement is relatively costly. Accuracy is very good provided the test procedures are followed rigorously. Computer simulation is generally cheaper but requires that the characteristics of the key components of the vehicle are modelled accurately. In New Zealand, computer simulation by approved analysts has been accepted by the LTSA as an acceptable method for evaluating performance measures.

The LTSA have generally required a performance assessment against all of the measures listed above and expected the vehicles to perform as well or better than the alternatives. Thus although the safety benefits may be unknown, there is reasonable confidence that there will be a safety benefit rather than a cost. The operator expects some economic gains from operating this vehicle. He or she knows the cost of applying for approval (including having the performance assessment done) and hence would not proceed if the expected economic benefits do not exceed the costs. The question of safety at reasonable cost resolves itself.

However, requiring all vehicles that meet the prescriptive requirements to also meet a performance standard is quite different. The productivity benefits are likely to be negative as some existing vehicles will not achieve the performance standard without a reduction in freight capacity. The reduction in social cost through reduced crashes must therefore offset both the cost of compliance and the loss in productivity. From this it can be shown that it is not viable to require all heavy vehicles to be assessed for their stability performance using computer simulation or experimental measurement because the compliance costs would be greater than the potential safety benefits using the social cost model outlined above. However, if some easily implemented low-cost method of estimating the performance measures to reasonable accuracy can be developed it may be possible to meet the reasonable cost criterion.

A recent study (de Pont et al, 2000, Mueller et al, 1999) determined relationships between some performance measures and relative crash rates for rollover and loss-of-control crashes involving heavy vehicles in New Zealand. Figure 1 shows the relative crash rate against SRT as determined by this study. From this figure we see that vehicles with poor SRT (less than 0.3g) have a crash rate about four times the average. The study also found that the 15% of the vehicle fleet with an SRT below 0.35g was involved in 40% of the rollover and loss-of-control crashes. This indicates that improving the performance of the poorest performing vehicles in the fleet should have a significant impact on the overall crash rate. Similar relationships were found for DLTR and HSTO. Not unexpectedly SRT and DLTR were strongly correlated so that improving one should improve the other.

SRT, which is the lateral acceleration at which rollover occurs during steady speed cornering, is one of the most fundamental of stability-related performance measures. Rather fortuitously it is also one where under certain assumptions an analytical solution is possible thus lending itself to the development of a simple low-cost algorithm for its determination.

THE STATIC ROLL THRESHOLD CALCULATION ALGORITHM

Figure 2 shows a cross-section view of heavy vehicle subject to a lateral acceleration, α . For this 2D vehicle model rollover occurs when $F_2 = 0$. It can easily be shown that, assuming small angles, this implies that:

$$SRT = \alpha = \frac{T}{2H} - \Phi$$

where T is the track width

H is the centre of gravity height

Φ is the total roll angle due to compliance

The most important term in this expression is the $T/2H$, which is often referred to as the static stability factor. It is an upper bound for SRT and represents what the SRT would be if the vehicle was a rigid body with no compliances. In practice, however, the compliances from the tyres, the suspension and the body itself are not zero and the actual SRT could be as much as 50% below the value given by $T/2H$.

Winkler et al (1992) proposed a three level screening approach to ensuring that vehicles achieved a minimum SRT level of 0.35g. At the first screening level vehicles with $T/2H$ greater than 0.58 almost certainly have an SRT greater than 0.35g and are acceptable. Vehicles with $T/2H$ less than 0.58 but greater than 0.46 are accepted if the tyres and suspensions on each axle meet certain minimum stiffness requirement. These requirements are load dependent. Finally at the third level of screening testing is required. Note again that if $T/2H$ is not greater than 0.35 the vehicle cannot have an SRT greater than 0.35g.

For implementation in New Zealand we have used a mathematical solution based on the graphical approach developed by Chalasani (Winkler et al, 2000) to estimate the actual SRT. We assume that the tyres are linear springs acting through the mid-point of the tyre. The suspension is assumed to consist of linear springs at the spring hangers with a rotational spring at the roll centre to represent the auxiliary roll stiffness. The suspension springs are assumed to have lash. The vehicle body is assumed to be rigid. Because each vehicle unit is registered and certified independently the algorithm is only required to be applied to single vehicle units with, at most two suspension groups. (The issue of roll coupled combination vehicles will be discussed in the next section on implementation.) For each suspension group force and moment balance equations can be written with the rigid body requirement generating a further equation coupling the two ends of the vehicle.

As lateral acceleration is applied to the vehicle, the vehicle body rolls until the onset of lash or wheel lift-off at one of the suspension groups. The roll stiffness of the system then reduces. Further lateral acceleration causes more body roll until the next event occurs when the stiffness again changes and so on until rollover occurs. For each suspension group, there are three possible sequences of events:

1. wheel lift-off before the onset of suspension lash
2. onset of lash followed by wheel lift before maximum lash is reached
3. onset of lash, full suspension lash, followed by wheel lift-off.

The sequence of events for the two suspensions can interleave each other. As each event causes the system stiffness to change it is necessary to determine the correct sequence of events as well as solving for each point. At each event the corresponding lateral acceleration is calculated and the maximum value achieved is the SRT. This may not be the last event in the sequence. For many vehicles the point of no return occurs before the last wheel has lifted off.

The list of variables needed to solve these equations are, for each suspension group:

- tyre stiffness
- tyre track width
- unsprung mass
- unsprung mass centre of gravity height
- suspension spring stiffness
- suspension track width
- suspension auxiliary roll stiffness

- suspension roll centre height
- sprung mass
- sprung mass centre of gravity height

IMPLEMENTATION

The above algorithm has been coded using javascript so that it can be run as a web-based application via the internet. In keeping with the principle of safety at reasonable cost, a number of assumptions were made to make the software easier to use and to reduce the potential cost of compliance.

Smaller trucks (with a GVM less than 12 tonnes) were exempted from the requirements. These vehicles operate primarily within urban areas and do not have a high rollover crash rate. As the calculator software is then only required for larger vehicles it was assumed that all vehicles operated at the maximum allowable width (2.5m) and the wheel track was calculated back from this based on the tyre configuration.

Three tyre configurations (dual, single and wide single) are permitted and three tyre sizes (22.5, 19.5 and 17.5). Generic tyre stiffness values are assumed for each tyre type and size. No allowance is made for low profile tyres or for variations in stiffness for different brands of tyre. There are two reasons for this approach. The first is that it simplifies the user input requirements and the second is that the tyres will be changed numbers of times during the vehicle's life and there is no easy way of ensuring that tyre-specific characteristics will be maintained at all times.

Input values for tare mass and payload mass at each axle group are required. Default values are used for the axle masses and wheel masses. These depend on the axle type, i.e. steer, drive or trailer axle and tyre size. From these values the unsprung mass is estimated by summing the axle mass and the wheel masses. By subtracting the unsprung mass from the tare mass, the empty vehicle sprung mass can also be estimated. The laden sprung mass is calculated from the payload mass and the empty vehicle sprung mass.

The unsprung mass centre of gravity height for each axle is set at the typical tyre radius for the tyre size. The error in centre of gravity height from ignoring low profile effects has a minimal effect on the final SRT value. For the empty vehicle sprung mass the centre of gravity height is estimated by adding a fixed value, which depends on vehicle type to the axle centre of gravity. Currently the values used are 0.56m for prime movers and 1.25m for trailers which are typical values given by Fancher et al (1986). Although these values are somewhat arbitrary for most typical vehicles they work quite well because they are combined with the payload centre of gravity to give an overall sprung mass centre of gravity and for most vehicles the payload contribution dominates. For prime movers the empty sprung mass component is more significant but the variations from the 0.56m value are small. For trailers, the 1.25m value is typical for a van body vehicle and is probably high for say a flat deck vehicle. However, the vehicles with lower empty sprung mass centre of gravity heights also tend to have low empty sprung masses so the error in the overall sprung mass centre of gravity is small. To estimate the payload centre of gravity the user specifies one of three load types, uniform density, mixed freight or other. For uniform or mixed freight he or she is required to enter the load bed height and the maximum load height. Uniform density assumes the load is uniformly distributed and thus the payload centre of gravity is midway between the load bed and the maximum load height. Mixed freight assumes a load distribution where 70% of the load mass is in the lower half of the load space and 30% is in the top half. This is a typical distribution given by Fancher et al (1986). The centre of gravity is therefore at 40% of the distance between the load bed and the maximum load height. The "other" category is for any load types not covered by the previous two categories. In this case the user is required to determine the height of the payload centre of gravity from the ground and input the value directly. Unusual vehicle bodies that do not fit the assumptions for empty sprung mass centre of gravity height outlined above can be handled by treating them as part of the payload using the "other" category.

The payload centre of gravity calculations do not currently take into account the case of moving loads such as partially filled liquid tankers or hanging animal carcasses. These cases can be approximated by determining the lateral shift in centre of gravity and reducing the effective track width accordingly but this has not been implemented in the current version of the software. Many liquid tankers in New Zealand are subject to the dangerous goods regulations because of the potential impacts of a rollover. These regulations are currently under

review but it is being proposed that these vehicles will need to meet an SRT limit of 0.45g when full and be required to have the tank subdivided into multiple sections to limit the proportion of the load that is free to slosh at any time. These two requirements together should ensure that even in the worst loading case the SRT will be better than 0.35g. Other liquid tankers such as dairy generally also have a very good SRT ($> 0.45g$) when full and thus even in the worst case sloshing of a partial load should still meet the 0.35g limit. The hanging carcasses load is not addressed but this involves only a small number of vehicles which have not been identified as having a stability problem.

The final inputs needed to solve the equations are the suspension characteristics. For each axle group, the user can select one of three suspension options, generic steel, generic air, or user specified. For the generic suspensions, default parameter values, which depend on the axle type are used. The current default values, which are shown in Table 1, are derived from Fancher et al (1986) and represent suspensions at the more compliant end of the spectrum. Thus the resulting SRT values are conservative. If the user-defined option is selected the user is required to input values for suspension parameters, which have to be either measured or obtained from the manufacturer. Some suspension manufacturers will readily make this information available while others regard it as commercially sensitive and are very reluctant to divulge it.

Once all the input data have been obtained or inferred, the calculator software determine the SRT and compares it against the target value of 0.35g. Originally it was proposed that the target value for trailers should be 0.4g, which, as can be seen from Figure 1, would give a greater safety benefit. However, estimates indicated that too many vehicles would need to reduce their load capacity and hence the economic losses from reduced productivity would be greater than could be justified by the safety benefits. If the vehicle fails to meet the SRT target, the calculator program iterates to calculate the reduced payload mass at the same load height and the reduced load height at the same payload mass needed to achieve the 0.35g SRT.

As mentioned in the previous section the algorithm has been developed to solve for the single vehicle unit with up to two axle groups because vehicles in New Zealand are certified and registered as individual units not as parts of a combination. This then raise the question of how to handle tractor units and semi-trailers, which normally operate as a combination and are roll-coupled so that both units together contribute to the SRT. Winkler (1992) has addressed this issue in tilt table testing by developing the concept of a virtual tractor for testing semi-trailers. In the calculator the same philosophy is used although the details differ. Tractor units, which carry no payload, are exempted from the SRT calculation requirement because without a trailer they will always meet the target value. Semi-trailers have their SRT calculated on the basis of the loads, suspensions and axles on the rear axle group alone. The philosophy is that if the rear axle group can withstand a lateral acceleration of 0.35g without rolling over it should not apply an overturning moment at the king pin till the lateral acceleration exceeds 0.35g. This is not necessarily correct because the total compliance is made up of several components (tyres, suspension, fifth wheel) and the relative magnitude of the different components is not necessarily the same for the semi-trailer axles and the tractor drive axles. However, given that any tractor could be coupled to the trailer it is not possible to determine whether this will be the case or not. In practice the approach seems to work reasonably well.

A demonstration version of the software was made available on the internet for three months during the public consultation phase of the dimensions and mass rule and was well received. The code is structured so that there is general public access to the calculator itself up to the point where the results are presented on the screen. There is then a login facility for certifying engineers to proceed further where a certificate containing all the vehicle data together with the parameters used in the calculation and the results is printed. Feedback from the consultation process suggested some modifications to the input data options but a general satisfaction with the ease of use of the calculator and the credibility of the results. A final version of the software is now being developed and will be available on the internet before the Dimensions and Mass Rule comes into force in July 2002. As with the demonstration version the general calculator part of the program will be freely and publicly available to all. Compliance with the legal requirement for heavy vehicles to achieve a minimum SRT level will be primarily determined by approved certifying engineers using the software on the internet.

VALIDATION

As outlined earlier, performance assessments have been undertaken by computer simulation for permit vehicles in New Zealand for over ten years. Most of these were done using the Yaw-Roll software from UMTRI. Yaw-Roll has also been used for a number of other studies in New Zealand to investigate potential changes in vehicle regulations. Results from some sample vehicles from these earlier studies together with a tilt table test on a log transport trailer have been used to validate the SRT calculator algorithm.

The tilt table test was conducted on a 4-axle full trailer with steel suspension loaded to 20 tonnes gross vehicle mass as shown in Figure 3. After the tilt table test the vehicle SRT was calculated using Yaw Roll software to simulate steady speed travel through a slowly tightening curve. The SRT was also calculated using the calculator software described above with both generic steel suspension option and with the "user defined" option using a linearised version of the manufacturer-supplied suspension data that was used for the Yaw Roll analysis. The results are summarised in Table 2. Using the generic suspension option, the calculator gives a slightly lower estimate of SRT which is conservative as it means the vehicle's actual SRT is probably better than this value. With user-defined suspension parameters, the calculator gives an SRT value which is the same as that measured using the tilt table test to within the measurement accuracy. Yaw Roll gives a value which is slightly higher than the measured value. This is probably because of the differences in the test conditions. Yaw Roll considers a moving vehicle in a tightening curve while the tilt table has the vehicle stationary on a slope. The SRT values obtained by all the methods are acceptably close to each other.

From records of previous simulation studies 11 vehicles were selected with calculated SRT values ranging from 0.32g to 0.62g. For each of the vehicles the SRT was recalculated twice using the SRT calculator program. The first time generic suspensions were used and the second time the "user defined" suspension option was used and the same suspension parameters as used in the Yaw Roll analysis of the vehicle were input into the calculator. Figure 4 shows a comparison between the Yaw Roll calculated SRT and the SRT calculator results when generic suspensions are used. For most of the vehicles the correspondence is very good. However, for two of the vehicles, the calculator gives significantly lower SRT values than Yaw Roll. This is not unexpected as the generic suspensions are at the more compliant end of the performance spectrum and thus if the actual suspension is significantly more roll stiff the true SRT will be significantly higher than estimated by the calculator. Figure 5 shows the comparison of the Yaw Roll results with the calculator when correct suspension parameters are input by the user. The correspondence is markedly better and there are now no outliers.

CONCLUSIONS

New Zealand has used stability performance assessment for many years for policy development and for issuing permits for vehicles with divisible loads to operate outside the prescriptive mass and dimensions regulations. It is now going a step further in requiring vehicles within the prescriptive mass and dimensions regulations to meet a stability performance level. As far as is known this is a world first.

The key to be able to implement this measure cost-effectively is having a low cost method for accurately and reliably estimating SRT. This paper describes the development of such a method and its implementation via an web-based software program called the SRT calculator. The user inputs that are absolutely required are generally known or easily measured by the vehicle operator. Increased accuracy can be achieved through the input of suspension parameters but for most vehicles this is not necessary.

The software will be publicly available and the demonstration version has already been used by some transport operators for parametric studies relating load height to stability for their particular operation. The calculator will be the primary tool for certifying that vehicles comply with the SRT requirement, although the option to use a tilt table test or a computer simulation as an alternative will exist.

The validation that has been undertaken shows that the calculator is consistent and sufficiently accurate.

Implementation of the new dimensions and mass rule with the SRT requirement is scheduled for July, 2002.

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TABLES & FIGURES

Table 1- Suspension properties for generic suspensions.

Suspension Name and Model Number	Generic - steer axle	Generic steel	Generic air
Suspension spring stiffness (N/m)	185000	900000	350000
Suspension track width (m)	0.8	0.8	0.8
Total roll stiffness per axle (Nm/radian)	130000	520000	780000
Suspension lash (mm)	15	15	1000
Roll centre height from axle (m)	-0.02	0.2	0.2

Table 2 - Predicted SRT from Tilt table, Yaw Roll software and SRT calculator.

Tilt table test	Yaw Roll	SRT Calculator Generic steel suspension	SRT Calculator User defined suspension
0.418 ± 0.006	0.428	0.407	0.415

The tilt table results are based on seven tests and the range given is the 95% confidence interval for the mean.

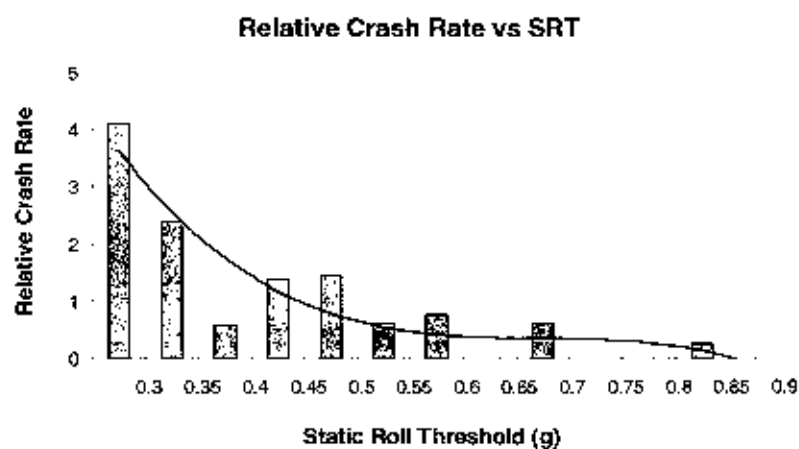


Figure 1 - SRT Relative Crash Involvement Rate.

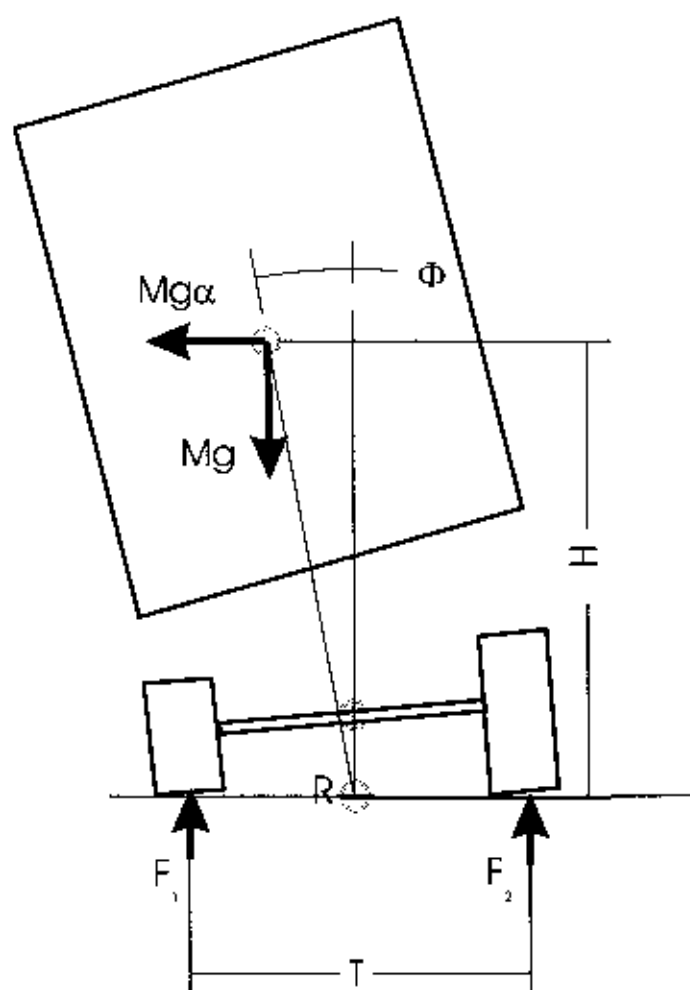


Figure 2 - Schematic showing heavy vehicle under lateral acceleration.



Figure 3 - Tilt table test on a 4-axle log trailer.

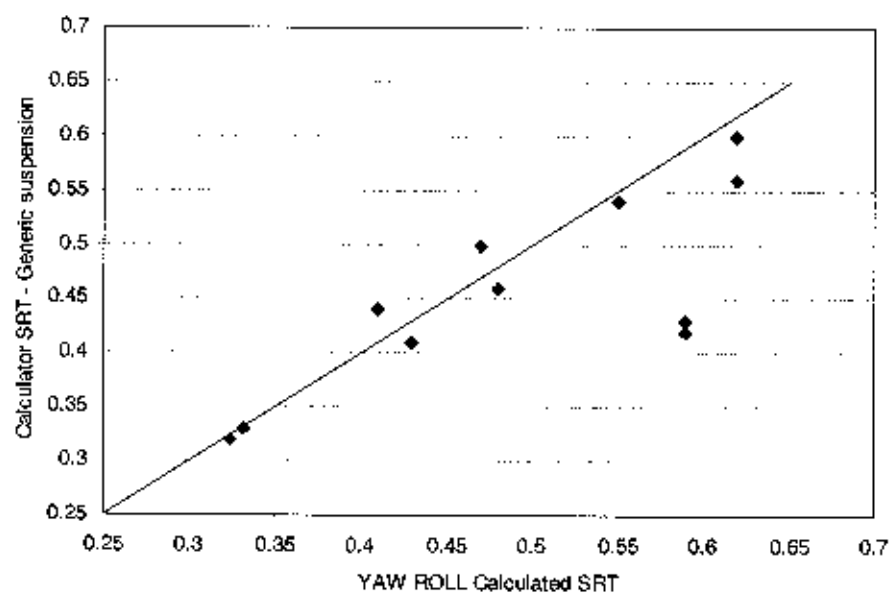


Figure 4 - Comparison of SRT Calculator using generic suspensions with Yaw Roll

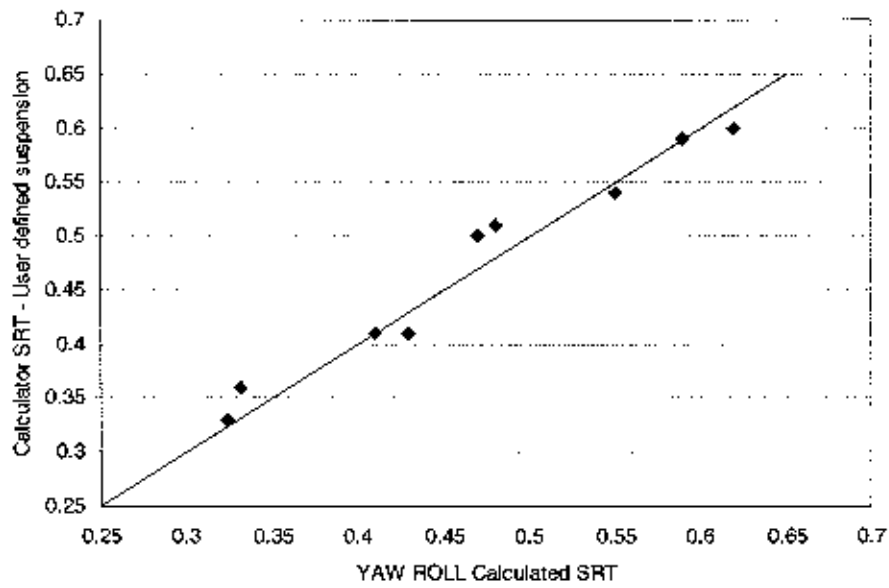


Figure 5 - Comparison of SRT Calculator using user defined suspension parameters with Yaw Roll

THE EFFECT OF MASS LIMIT CHANGES ON THIN-SURFACE PAVEMENT PERFORMANCE

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ABSTRACT

Pavement design and management including road user charging in New Zealand is largely based on the well known fourth power relationship between axle loads and pavement wear. However, the vast majority of the New Zealand highway network consists of thin surface unbound pavements, which are quite different from the pavements used in the AASHO road test where the fourth power law originated. Recent proposals to improve the efficiency of the road transport system have included options to raise the axle load limits as well as the GVM and GCM. Although the fourth power law predicts an effect, the true impact of these higher axle loads on the performance of the pavements is unknown.

To determine the impact on pavement wear, an accelerated pavement test was undertaken at the Canterbury Accelerated Pavement Testing Indoor Facility (CAPTIF) to compare the wear from a 10 tonne axle with that from an 8.2 tonne axle. The two loading units at CAPTIF were configured to be the same in all respects except load. They trafficked parallel paths on four different thin-surface pavements for approximately 1 million load cycles. The pavement was extensively monitored throughout the test.

From the data collected a new empirical model of the relationship between load and vertical surface deformation has been developed. This predicts a quite different relationship between load and wear to the fourth power law and has major implications for pavement management, particularly if the mass increase is implemented, and for the road user charging regime.

INTRODUCTION

Background

As in many countries, the New Zealand road transport freight industry is constantly searching for efficiency improvements. One of the obvious ways to achieve these is through increases in the allowable masses for heavy vehicles. These could result in economic benefits to the whole country provided the impact of the changes in mass limits are accurately known and considered in assigning the new limits and in determining appropriate road user charges. Of concern to the road-controlling authorities is the effect on increasing mass limits on the life of the pavements and how much more pavement rehabilitation and maintenance will be required.

In response to the industry's requests for larger and heavier vehicles Transit New Zealand (2001) has recently undertaken a study to assess the economic and safety impacts of increasing mass limits. This study investigated two scenarios:-

- Scenario A, where heavier vehicles subject to the same dimensional limits as are currently in place would be permitted to operate across the entire network and
- Scenario B, where longer and heavier vehicles would be permitted to operate only a selected set of key routes.

Within these two scenarios several gross combination mass limits were considered and associated with these increased axle mass limits. Transit's evaluation of these proposed changes in mass limits included research into the safety, geometric, economic, pavement and bridge impacts. In determining the pavement wear impact of these mass limits changes, existing theories for the relationship between vehicle loads and pavement wear were utilised.

The most widely used existing theory for determining the effect of mass limit increases on pavement life is the fourth power rule. This is used to determine the pavement loading as a number of Equivalent Standard Axles (ESAs). The formula for converting an actual axle load to ESA is:

$$ESA = \left[\frac{\text{Actual axle load}}{\text{Reference axle load}} \right]^4$$

This fourth power relationship between axle loads and pavement life has never been validated on New Zealand's thin surfaced unbound granular pavements. Power values between 1 and 8 have been suggested by different researchers throughout the world for different pavement structures and failure mechanisms (Cebon, 1999, Kinder and Lay, 1988, Pidwerbesky, 1996). Even the Austroads Pavement Design Guide, which is the basis of New Zealand design practice, uses a power of 4 for unbound basecourse performance and a power of 7.14 for subgrade performance.

The use of a fourth power relationship predicts that increasing the allowable loading for a single axle from 8.2 tonne (current limit) to 8.8 tonnes (proposed new maximum) will result in a 33% increase in pavement wear. On this basis a road controlling authority might expect a 33% increase in the length of pavement rehabilitation required per year. The actual situation is not as extreme as this because in the first place not all vehicles will change to the higher limits and secondly the higher axle load limits will result in higher payloads and consequently fewer trips for the same freight volume. Nevertheless, this change will represent a significant increase in annual expenditure on roads for a road controlling authority and requires budgeting for. The uncertainty in the validity of the fourth power rule poses difficulties when requesting increases in funding for the next financial year. Justifying an increase in the road user charges based on a fourth power rule that has not been validated in New Zealand is expected to be increasingly difficult, particularly as research results from accelerated pavement tests are suggesting different relationships.

This study investigates, via accelerated pavement testing on typical New Zealand pavement designs, the relative effect on pavement performance of an increase in axle load from 8.2 tonnes (the present load limit) to 10 tonnes. This increase is somewhat higher than the originally proposed change. There are two reasons for using this higher axle load. The first is that in response to the heavy vehicle limits study, the bus and coach industry has suggested that this level of increase should be considered. The second is that the larger mass difference between the two wheel paths was more likely to ensure that the difference in wear is sufficiently large to be able to draw clear conclusions. By interpolating the results of this research it would be possible to assess the effect on pavement life of loading increases directly (without necessarily using any power law). A more detailed report on this test (de Pont et al, 2001) has been published by Transfund New Zealand.

A key issue for this study is what constitutes pavement wear. The OECD DIVINE project (OECD, 1998) differentiates between functional and structural condition for pavements. Functional condition reflects the ability of the pavement to provide service to the road user and covers factors such as roughness, rutting and skid resistance. Structural condition relates to the ability of the pavement to support the loads applied to it and relates to cracking and other forms of distress. In some instances a loss of functional condition may also indicate a loss of structural condition. In New Zealand pavement maintenance and rehabilitation is driven primarily by measures of functional condition and thus, in this study, wear is taken to mean a reduction in functional condition. The primary measure of functional condition considered in this paper is permanent vertical surface deformation (VSD). VSD is the change in elevation of the pavement surface from a reference elevation measured at the start of testing.

The Canterbury Accelerated Pavement Testing Indoor Facility (CAPTIF)

CAPTIF is located in Christchurch. It consists of a 58 m long (on the centreline) circular track contained within a 1.5m deep x 4m wide concrete tank so that the moisture content of the pavement materials can be controlled and the boundary conditions are known. A centre platform carries the machinery and electronics needed to drive the system. Mounted on this platform is a sliding frame that can move horizontally by 1 m. This radial movement enables the wheel paths to be varied laterally and can be used to have the two "vehicles" operating in independent wheel paths. An elevation view is shown in Figure 1.

At the ends of this frame, two radial arms connect to the Simulated Loading and Vehicle Emulator (SLAVE) units shown in Figure 2. These arms are hinged in the vertical plane so that the SLAVEs can be removed from the track during pavement construction, profile measurement etc. and in the horizontal plane to allow vehicle bounce.

CAPTIF is unique among accelerated pavement test facilities in that it was specifically designed to generate realistic dynamic wheel forces. All other accelerated pavement testing facility designs we are aware of attempt to minimise dynamic loading. The SLAVE units at CAPTIF are designed to have sprung and unsprung mass values of similar magnitude to those on actual vehicles and use, as far as possible, standard heavy vehicle suspension components. The net result of this is that the SLAVEs apply dynamic wheel loads to the test pavement that are similar in character and magnitude to those applied by real vehicles. This was a significant factor in its selection for this project. A summary of the characteristics of the SLAVE units is given in Table 1. The configuration of each vehicle, with respect to suspensions, wheel loads, tyre types and tyre numbers, can be identical or different, for simultaneous testing of different load characteristics.

A more detailed description of the CAPTIF and its systems is given by (Pidwerbesky, 1995).

OBJECTIVE

The principal objective of the study was to determine the effect on pavement life and pavement performance of increasing the maximum allowable axle load for different pavement strengths (or aggregate depth) using data from an accelerated pavement test.

Based on this effect the aim was then to predict the increase in road expenditure resulting from an increase in the allowable axle loads and thus justify to the transport industry the increase in Road User Charges (RUCs) that would be required to offset this expenditure.

RUCs for heavy vehicles in New Zealand are based on mass and distance. The rates are based on the fourth power law so an increase in axle load will lead to a substantial increase in RUCs for vehicles with these more heavily loaded axles. The purpose of the second objective is to assess whether this level of increase is appropriate.

METHOD

Pavement design

The objective of the pavement design was to produce the relatively low levels of rutting observed in typical New Zealand Pavement while maintaining a balance between the life of the heavily loaded (10 tonne) outer wheel path and the lightly loaded (8.2 tonne) inner wheel path.

The track was divided into four sections with the same depth of a different basecourse material in each. The original experimental design called for only one basecourse material and different thicknesses but this was changed to accommodate the requirements of another project that needed to use data from the same test. The four pavement sections were not considered separately in the design as the material data available at the design stage was not sufficient to consider separate designs for each material. It was considered preferable to use pavements of the same depth in order to limit the number of variables in the project.

The pavement was designed in an iterative manner using the AUSTROADS Pavement design guide. The iterative designs assumed a 700 kPa tyre pressure with a 95.6 mm tyre radius on the inner wheel path and a 850 kPa tyre with a 97mm tyre radius on the outer wheel path. The basecourse layer was, from previous experience, modelled with a modulus of 400 MPa using AUSTROADS sub-layering and, as convention dictates, the thin asphaltic layer was not considered in the analysis. The subgrade was modelled with a 10th percentile design in-situ CBR of 10, based on test results at the top of the subgrade layer and used the standard 10 times CBR relationship to obtain the modulus.

The iterative analysis suggested the inner (8.2 tonne) wheel path would require a basecourse 250 mm deep to withstand the design 1,000,000 wheel passes and a depth of 290 mm for the outer wheel path assuming the

AUSTROADS subgrade strain criterion. Using the 4th Power law to convert the 10 tonne wheel to an equivalent number of standard axles rather than modelling directly suggested that the outer wheel path would have needed to be 270 mm deep.

The final design used a 275mm basecourse layer. Theoretically this was expected to fail at 2.9 million passes of the 8.2 tonnes axle and at 600,000 passes of the 10 tonne axle assuming the AUSTROADS subgrade strain criterion and directly modelling the tyre loads. Using the 4th power law suggested that the 10 tonne axle would cause failure at 1.2 million passes.

Pavement construction

The pavement was constructed in three primary segments: Segment A extended from station 00 to station 15; Segment B from 15 to 29; and Segment C from 29 to 43. An additional segment, Segment D, extended through the track access area from 43 to 00. There was a 4m transition zone allowed between segments. Within each segment on the track centreline there were 3 primary sites for intensive monitoring and 14 secondary sites for less intensive monitoring. A plan showing the layout of the different sections is shown in Figure 3.

Pavement construction at CAPTIF as far as possible utilises the same techniques as used for normal pavement construction in New Zealand. The main variations are that there is some scaling of the equipment because of the size of the track and some adaptations because of the curvature of the track. For example, a plate compactor is used for final compaction rather than a roller. The other big difference is that as each layer is placed it is intensively monitored to ensure that the pavement has a high degree of uniformity, particularly transversely. These measurements include density, moisture content, elevation, transverse profile, Loadman deflections, CBR, and penetrometer readings. In addition instrumentation to measure strains and stresses was placed within the layers during construction.

Zero measurements

Basic measurements of the surface profile and structural capacity of the pavement as constructed were undertaken. The SLAVE units were then loaded to 40kN each and 5,000 preliminary conditioning load cycles (10,000 ESA) were applied evenly across the full trafficable width of the pavement. Following these conditioning laps, a set of zero measurements to characterise the system at the start of the test were undertaken.

These measurements included:

- Loadman falling weight deflectometer measurements to monitor the structural capacity of the pavement layers during construction and CAPTIF deflectometer measurements on completion of construction.
- Transverse profiles at each of the 58 stations using the CAPTIF transverse profilometer. These are referenced back to the tank wall and give elevation readings at 25mm spacing across the track.
- Longitudinal profiles measured along five centrelines using the laser profilometer. The five centrelines consisted of one in the centre of each wheel path, one midway between the two wheel paths, one inside the inner wheel path and one outside the outer wheel path.
- The effect on pavement response to varying the transverse position of the SLAVE units was measured with ten sets of readings with the offset changing by 0.1m each time.
- The effect of speed variations was recorded by taking measurements at 20km/h and 45 km/h. For these measurements the vehicles were loaded to the test condition with 40kN on vehicle A and 50kN on vehicle B.
- The effect of variations in load was measured by setting the load on vehicle A to 21kN, 31kN, 40kN, 44kN and 50kN. The load on vehicle B was 50kN throughout these tests.
- Characterising the suspension response using the 80mm drop test at crawl speed as specified by the EC (Council of the European Communities, 1992) in their regulations for rating a suspension as "equivalent-to-air". For these measurements the vehicles were in the test load configuration with vehicle A at 40kN and vehicle B at 50kN.

The vehicles were weighed by standard New Zealand Police portable weigh scales at their operating configuration. The static weight for vehicle A was 4080 kg (40.02 kN) and the static weight for vehicle B was 5060 kg (49.64 kN).

Testing

For this project the SLAVE units were run in concentric offset wheel paths. Previous projects where the SLAVE units were run in offset wheel paths had used wide single tyres. However this project specified the use of standard dual tyre assemblies. As the dual tyre assembly is considerably wider than a wide single tyre this requirement significantly reduced the separation between the two vehicles, even with a narrow wander pattern. To overcome this limitation, a 300 mm long extension section was fitted to arm B. This meant that the radial distance between the centrelines of the vehicles was 1100 mm and the clear separation between the vehicles was 450 mm with the vehicles operating on a ± 50 mm normally distributed wander pattern.

The vehicles were run continuously at a mean speed of 45km/h with breaks for testing intervals at 20,000, 30,000, 50,000, 100,000, 150,000, 200,000, 250,000, 300,000, 400,000, ... 1,000,000 cycles. Starting at 20,000 cycles, every third test comprised a full set of tests that included: 58 transverse profiles; 5 longitudinal profiles; pavement strain and pressure readings at 45km/h and 15km/h; and dynamic wheel loads at 45km/h. At the other test intervals a reduced data set was taken. The reduced data set included: 58 transverse profiles; longitudinal profiles in the wheel paths only; and strains and pressures at 45km/h only. The Falling Weight Deflectometer was used to measure all stations in both wheel paths at 25,000, 200,000, 600,000, and 1,000,000 cycles.

After 200,000 laps the asphaltic concrete surfacing in Segment C began to show signs of shear failures in the outer (50kN) wheel path only. When the seal was removed it was noted that the top of the basecourse looked glazed and dirty with no significant bonding of the asphalt mix to the basecourse material. The damaged sections of seal were replaced but the problem recurred at 700,000 and 1,000,000 cycles in different places in the outer wheel path in section C.

Loading was halted after 1,000,000 cycles as specified in the research the brief. Although the pavement had not reached any of the predefined failure criteria, the rate of pavement deterioration (rutting) in both wheel paths had reached a stable state and failure would not occur for many more load cycles.

Post-mortem

The post-mortem analysis consisted of recording the condition of the pavement and excavating trenches across the width of the pavement to determine any changes in the properties of the different materials and to attempt to determine the amount of rutting in each layer. Three trenches were excavated in each section, at locations corresponding to the minimum, average and maximum locations of pavement rutting.

RESULTS AND ANALYSIS

Pavement Variability

In constructing a pavement at CAPTIF for a comparative study such as this the aim is to minimise the transverse variability in the pavement structure so that the two SLAVE units are, as much as possible, trafficking identical pavements. Longitudinal variations in the pavement structure are less of a concern but it is difficult to construct a pavement that is very uniform transversely and irregular longitudinally. To test a parameter such as layer thickness for uniformity between the two wheel paths we construct a new variable, which is the difference between the parameter's value on the inner wheel path and its value at a position on the same radial line in the outer wheel path. For a uniform pavement this new variable will have a mean equal to zero and a small standard deviation.

Table 2 and Table 3 show the statistics of this difference variable for the asphalt and basecourse layer thicknesses respectively. At the 95% confidence level, if the range of the average difference $\pm 2 \times$ standard error includes zero, then we cannot reject the null hypothesis that there is no difference between the inner and outer wheel paths. From Table 2 we can see that for the asphalt layer, only segment C shows no difference between the inner and outer wheel paths. For pavement segments A and B the inner wheel path asphalt layer is approximately 5 mm thicker

than the outer wheel path while, for segment D, the outer wheel path is approximately 2mm thicker. The average asphalt layer thickness for the whole pavement is about 27mm. Although the 5mm difference is a significant proportion of the overall layer thickness, thin surface pavement design assumes that the asphalt layer does not contribute to the structural capacity of the pavement and so this should not impact significantly on the performance of the two wheel paths.

From Table 3, which shows the analysis of the basecourse layer we can see that Segments B, C and D have an average, which is not significantly different from zero at the 95% confidence level. The average for Segment A is greater than zero but only just outside the 2 standard error confidence interval and as the average thickness of the basecourse layer is 276 mm the difference is less than 2%. The basecourse layer is the main structural element in a thin-surface pavement and to all intents and purposes the inner and outer wheel paths have the same thickness basecourse layer.

During construction Loadman falling weight deflectometer measurements of the modulus were taken at five transverse positions at twelve stations (three in each pavement segment) on the top of the subgrade layer. As the same subgrade was used for all four segments we consider the difference function on this layer for the whole track. This had a negative mean (average value -7 MPa with a standard error of 3.4MPa). The average value for both wheel paths was 68 MPa, so the inner wheel path average was 10% below the outer wheel path. For the basecourse layers readings were taken at nine stations (three in each of segments A, B and C. At the 95% confidence level there was no significant difference in the readings between the two wheel paths either by segment or for the pavement as a whole. Experience with the Loadman at CAPTIF has shown the repeatability of the results to be approximately 10 MPa. Thus, based on the Loadman measurements the inner and outer wheel paths appear to be effectively identical.

Vertical Surface Deformations and Rutting

At each measurement interval the transverse profile of the pavement was measured at each station. From these measurements the permanent vertical surface deformation (VSD) and the rut depth can be calculated. VSD has proved in past CAPTIF tests (de Pont et al., 1999) to be a fundamental measure of pavement wear that provides useful insights into the pavement performance and behaviour. Both rutting and surface roughness are related to VSD and so VSD reflects both these forms of pavement wear. Rutting is a direct result of VSD while roughness is the result of the variation in VSD. As VSD has been shown to be correlated to both dynamic wheel forces and the variability in pavement structure (de Pont et al., 1999), increasing VSD leads to increased roughness. With the transverse profiler at CAPTIF it is possible to measure VSD to a good degree of accuracy and reliability because the measurements are all referenced back to the edges of the concrete tank which are very stable.

Measurements of rutting and roughness do not have the same level of reliability (de Pont, 1997). Rutting is determined by calculating or measuring the depth of the rut from a straight edge laid across the wheel path. Thus the rut depth depends not only on the VSD in the centre of the wheel path but also that of the highest tangential points inside and outside the wheel path. Roughness is usually given in International Roughness Index (IRI) values, which are calculated from the longitudinal profile using the response of simulated quarter car. The dynamic characteristics of the quarter car are such that it responds to surface profile characteristics with wavelengths from 1m to 30m (Sayurs et al., 1986). To accurately sample the longer wavelength components in this range it is normally recommended that the section length for IRI calculation is greater than 100m. As the track at CAPTIF is only 58m long it does not meet this requirement.

Figure 4 and Figure 5 show the progression of VSD for each of the wheel paths in each pavement segment. Note that, as outlined in the test description, earlier parts of the pavement surface in the outer wheel path in segment C were repaired at 200,000 cycles and at 700,000 cycles. The VSD values for these repaired sections were therefore meaningless and so the average VSD for the outer wheel path of segment C is calculated from only those stations that had not been repaired. This left only four stations out of fourteen so it is expected that the variability will be greater for this dataset.

For all four pavement segments the VSD is greater on the outer wheel path, which was trafficked with the higher load than on the inner. The conventional approach to comparing the wear generated by two different axle loads is the power law method. This states that the amount of pavement wear caused by one pass of an axle is proportional

to some power of its axle load. The most widely used value for this power is four. Thus if a given level of wear is achieved by N_{inner} load cycles of a load P_{inner} or by N_{outer} load cycles of a load P_{outer} these are related as follows:

$$\frac{N_{outer}}{N_{inner}} = \left[\frac{P_{outer}}{P_{inner}} \right]^n$$

where n is the exponent of the power law

For every measured value of VSD and load cycles on the inner wheel path, the number of load cycles on the outer wheel path to generate the same VSD can be calculated by interpolation. Alternatively the number of load cycles on the inner wheel path needed to generate the measured VSDs in the outer wheel path can be calculated. In either case, as P_{inner} and P_{outer} are known, the value of the exponent, n , required to get the power law relationship to hold can be calculated for each VSD value. Whether the inner or outer wheel path VSD measurements are used as the reference makes relatively little difference to the results. For segment A, the average exponent value was 8.9 using the inner wheel path VSDs as the reference and 9.3 using the outer wheel path VSDs. For segment B, the corresponding values were 6.1 and 6.3, for segment C, 2.8 and 2.9 and for segment D, 3.8 and 3.9. Averaging across both sets of data gives exponents of 9, 6.2, 2.8 and 3.8.

Using these values in a model gives a reasonable fit to the measured data. If we use 8.2 tonnes as a standard axle load every load cycle on the inner wheel path corresponds to 1 "equivalent standard axle" or ESA. On the outer wheel path each load cycle would correspond to $(10.2/8.2)^n$ ESA where n is the exponent calculated above. Thus for segment A, where the exponent is 9, 1,000,000 load cycles corresponds to 5,966,000 ESA. Figure 6 shows the power law fits for each of the pavement segments. As can be seen this form of model provides a reasonable although far from perfect fit to the data. However, there are two major problems with this model. The first is that it does not explain why the rate of increase in VSD changes so much as the loading progresses and the second is that there is such a wide variation in the value of the exponent between the different pavement segments. These four segments are all variations of a basic thin surface pavement structure and yet the best-fit exponents for a power law model vary between less than 3 and 9.

Looking back at Figure 4 and Figure 5, we can see that VSD increases rapidly during the initial loading cycles and then slows to an approximately linear rate of increase. This suggests that the VSD consists of two components, an initial post construction compaction and then a wear related component. If we fit a straight line to the linear part of the VSD versus load cycles curve, then the intercept of this line with the vertical axis gives the post construction compaction component and the slope of the line gives the wear related component. We can then use the power law approach to compare both the intercept and the slope of these lines between the inner and outer wheel path for each segment.

Table 4 and Table 5 show the results of a least squares regression straight line fit to the linear portions of the VSD vs load cycles curves for each of the four segments in both wheel paths. As can be seen from the r^2 statistics these are very good fits.

A power law fit can be applied to the compaction and wear components independently. This power law approach implies relationships of the form:

$$\frac{\text{Intercept}_{outer}}{\text{Intercept}_{inner}} = \frac{\text{Compaction}_{outer}}{\text{Compaction}_{inner}} = \left(\frac{\text{Axle load}_{outer}}{\text{Axle load}_{inner}} \right)^n \text{ where } n \text{ is the exponent of the power law}$$

$$\frac{\text{Slope}_{outer}}{\text{Slope}_{inner}} = \frac{\text{Wear}_{outer}}{\text{Wear}_{inner}} = \left(\frac{\text{Axle load}_{outer}}{\text{Axle load}_{inner}} \right)^n \text{ where } n \text{ is the exponent of the power law}$$

Using logarithms the values of n can readily be calculated for compaction and wear for each of the pavement segments. The results for the exponent values are shown in Table 6. It is interesting to note how similar the exponents are for the two components of VSD. The exponents for the different segments, while still not identical, are much more alike than they were in the simple power law model, particularly if we discount segment C. Recall that a number of stations in the outer wheel path of segment C were repaired during the test and that the data from these stations were removed from the analysis leaving only four valid stations.

This model implies that the compaction is dependent only on the applied wheel load and not on the number of applications of this load (although a number of applications of the load are required to effect the compaction). The wear component is related to both the load and the number of load cycles. A logical extension to this model is that if after some large number of load cycles, the wheel load is increased the additional compaction associated with the higher wheel load would then take place as well as the higher rate of wear associated with a higher wheel load. The conventional power law approach does not predict an additional compaction with an increase in wheel load, only a higher wear rate.

Although this hypothesis was not tested explicitly because the test was completed before this analysis was done, there was a series of higher load cycles applied to the inner wheel path after the completion of this test as part of another project. VSD measurements were taken as part of that project and the results are shown in Figure 7. The project being reported here was finished at 1,000,000 load cycles. The subsequent project involved looking at the strain response of the pavement under various loading conditions, which included loads 40kN, 50kN and 60kN with varying tyre pressures and on both wide single and dual tyres. Thus it is difficult to make a direct assessment of the expected wear impact. However, the change in VSD after 1,000,000 load cycles does appear to reflect an additional compaction rather than just a change in wear rate.

Previous work at CAPTIF on the impact of dynamic loading on pavement wear (de Pont et al., 1999) also found a power law relationship for wear rate that had a relatively low exponent (between 1 and 2). This fits in well with exponent values found for this compaction and wear model and not with the higher powers normally associated with a power law model.

Note that the terms "compaction" and "wear" to describe the two components of VSD are not based on any knowledge of the underlying material behaviour. Previous measurements by (Patrick et al., 1998) measuring the density of in-service pavements found no significant increase in basecourse density during this post-construction compaction phase. Further investigation is needed to determine the mechanism generating the behaviour.

If this model is correct there are significant implications for road controlling authorities particularly if a change in axle load limit occurs. The wear rate will increase according to a power law with an exponent between 1.8 and 3. This means that if the axle load limit is raised by 7.3% as suggested the wear rate per axle for those axles at the higher loads will increase by between 13.5% and 23.5% depending on which exponent value is used. As road user charges are based on a fourth power law the increase in wear component of road user charges will be 32.5% for the higher loaded axles and thus will more than offset the additional wear. However, the compaction component also has an exponent of between 1 and 3.4. Increased axle loads will therefore also cause an additional one-off compaction of the pavement across the whole network. For a 7.3% axle load increase this will be between 7.3% and 27% of the compaction that occurred initially on the pavement after construction. For the four pavement segments tested here the magnitude of the initial compaction was equivalent to the wear induced by between 700,000 and 980,000 load cycles. Thus an increase in axle load limit of 7.3% would lead to an increase in VSD due to compaction which is equal in magnitude to the wear-related VSD associated with between 50,000 and 265,000 load cycles. For a typical New Zealand State Highway, the average heavy vehicle traffic is about 100 vehicle/day and it is assumed that the average heavy vehicle applies about 1 ESA. This implies about 36,500 standard axle loads per year. Thus increasing the axle load limit by 7.3% will result in an additional compaction that is equivalent to between 1.4 and 7.3 years of normal loading. This additional VSD would occur in a relatively short period, probably less than a year, of the change.

CONCLUSIONS

The aim of this study was to compare the pavement wear generated by a 10 tonne axle load with that of a standard 8.2 tonne axle with a view to predicting the cost implications of a change in the legal axle load limit in New Zealand. A pavement was tested at CAPTIF, which comprised four distinct segments, each of which was similar in design but utilised a different basecourse material. One of the SLAVE units at CAPTIF was configured to generate a 40kN wheel load (equivalent to an 8.2 tonne axle) and the other was configured for a 50kN wheel load (equivalent to just over 10 tonne axle load). The two SLAVEs were then used to apply 1,000,000 load cycles to parallel wheel paths on the pavement. During the testing measurements were taken to record the pavement wear, the pavement condition, the pavement response to the vehicle loading and the vehicle response to the pavement.

From these measurements a number of important findings can be deduced:

- VSD, which is a fundamental measure of pavement wear that results in both rutting and increased surface roughness, again proved to be useful for monitoring pavement wear at CAPTIF.
- Although a conventional power law relationship could be fitted to describe the differences in VSD between the two levels of loading for each of the four pavement segments, there was a large variation (between 2.8 and 9) in the exponent value required to give the best fit. As the pavement design of the four segments was substantially similar in character it does not seem reasonable that the exponents for a power law model should vary so much. It also makes it impossible to predict the appropriate exponent value in advance. Thus the power law approach does not appear to be an accurate or useful way of modelling the VSD wear of this type of pavement.
- Reviewing the progression of VSD with load cycles shows that the pavement underwent two distinct phases of VSD. There was an initial period of rapid change, which we will call compaction, followed by a period with a constant (linear) rate of change, which we will call wear. Least squares regression can be used to fit a straight line to the linear part of the VSD versus load cycles curve. The intercept of this line with the y-axis then gives the compaction component and the slope gives the wear. For each of the four pavement segments, a power law can be used to relate the compaction and wear between the normally and more heavily loaded wheel paths. Remarkably the best-fit exponent values for compaction and wear were quite similar for each pavement segment and did not vary too much between segments.
- This compaction-wear model implies that the compaction depends only on the magnitude of the applied load and not on the number of load cycles while the wear depends on both the load and the number of load applications. A corollary of this is that if the axle load level is increased at any stage further compaction will occur to reflect this higher load. As this model was developed well after the completion of the testing programme this hypothesis was not specifically tested for, but in the project that followed this one at CAPTIF, the same pavement was used and higher loads were applied to the more lightly loaded wheel path. The VSD did show an apparent increased compaction as would be expected.
- The exponent values for the compaction-wear model were between 1 and 3.4 for compaction and 1.8 and 3 for wear. (The values for pavement segment C were a little lower than this but repairs to the pavement surface during the test meant that very few data points could be used for this segment.) The implication of this is that if the axle load limit were increased the underlying wear rate would increase as indicated by a power of between 1.8 and 3. As the road user charges paid by these vehicles are based on a power of 4 the additional road user charges would more than offset the additional wear. In fact, there is some ground for considering a review of the road user charges schedule. But it must be noted the effect of the compaction phase will also be observed in new construction and rehabilitation so this effect cannot be ignored. The existing fourth power approach is based on a chord approach, that is, the amount of damage is considered only at the initial and terminal conditions. This research has shown that there maybe some merit in looking at a secant (tangential) rate of damage, but this approach would be difficult to incorporate into a charging model. However, an increase in axle load limit would also cause an immediate (over a year or two) additional compaction. Thus it would appear that the network had suddenly deteriorated substantially. This is a one-off effect but would need to be planned for by the road controlling authorities if pavement condition is to be maintained. If the road controlling authorities are not anticipating this additional compaction effect, the sudden apparent additional deterioration of the network will cause them great concern over the future maintenance demands.

From these findings a number of questions requiring further investigation arise:

- Further validation of the compaction-wear model is required. It is recommended that, in future CAPTIF trials where a wheel path has been trafficked with a constant load, after completion of the test some relatively small number (perhaps 300,000) load cycles with a higher load are applied. If the compaction-wear model is valid it will be possible to predict and test the amount of compaction that will occur. Work in progress at CAPTIF (2001) has attempted to address this issue but additional validation will be required.
- Although the compaction-wear model provides a good fit to the observed behaviour and is a useful predictor tool, the mechanisms underlying it are not understood. Further research is required to determine how the pavement materials are behaving and what is leading to the compaction and wear components of VSD. Note that the names, "compaction" and "wear" are speculative and not based on any fundamental consideration of the underlying material behaviour.
- More detailed analysis of the cost implications of the compaction-wear model are needed. The underlying wear rate appears to be related to load by a power lower than four. However, the compaction component is also related to load by a power law.
- The VSD associated with compaction is equivalent to the wear associated with a considerable number of load cycles. If this compaction can be induced without causing rutting or roughness, the performance of the pavement would be enhanced considerably. How this could be achieved requires further investigation.
- The performance of thin surfacings under the higher axle loads needs to be investigated further. The pavement surfacing exhibited distress and failures under the 50 kN axle load, but not the 40 kN axle load.

Overall the most significant finding of this study is the development of the compaction-wear model for VSD. This provides a much better fit to the observed behaviour than the conventional power law model and has very significant implications for pavement management practice in New Zealand and wherever thin surface unbound pavement structures are widely used. Further validation work should be undertaken as well as research to understand the material behaviour mechanisms that produce this behaviour.

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TABLES & FIGURES

Table 1. Characteristics of SLAVE units.

Test Wheels	Dual- or single-tyres; standard or wide-base; bias or radial ply; tube or tubeless; maximum overall tyre diameter of 1.06 m
Mass of Each Vehicle	21 kN to 60 kN, in 2.75 kN increments
Suspension	Air bag; multi-leaf steel spring; single or double parabolic
Power drive to wheel	Controlled variable hydraulic power to axle; bi-directional
Transverse movement of wheels	1.0 m centre-to-centre; programmable for any distribution of wheel paths
Speed	0-50 km/h, programmable, accurate to 1 km/h
Radius of Travel	9.2 m

Table 2. Difference (inner – outer wheel path) in asphalt layer thickness.

Segment (Station Nos)	Average (mm)	Minimum (mm)	Maximum (mm)	Standard Deviation (mm)	Range (mm)	Standard Error (mm)
A (0-15)	4.92	-1.38	14.25	4.13	15.63	1.03
B (16-29)	4.89	-3.00	10.38	3.74	13.38	1.00
C (30-43)	0.16	-4.75	6.13	3.45	10.88	0.92
D (44-57)	-1.79	-2.13	-1.25	0.47	0.88	0.27

Table 3. Difference (inner – outer wheel path) in basecourse layer thickness.

Segment (Station Nos)	Average (mm)	Minimum (mm)	Maximum (mm)	Standard Deviation (mm)	Range (mm)	Standard Error (mm)
A (0-15)	5.13	-10.13	25.25	9.81	35.38	2.45
B (16-29)	-3.94	-13.75	11.25	7.59	27.00	2.03
C (30-43)	1.93	-8.00	9.88	4.76	17.88	1.27
D (44-57)	0.21	-1.25	2.25	1.82	3.50	1.05

Table 4. Linear fit parameters for VSD vs load cycles on inner wheel path.

	Intercept – Compaction (mm)	Slope – Wear rate (mm/1000 load cycles)	Goodness of fit – r^2
Segment A	2.74	0.00321	0.960
Segment B	3.07	0.00327	0.993
Segment C	1.91	0.00245	0.965
Segment D	1.86	0.00233	0.904

Table 5. Linear fit parameters for VSD vs load cycles on outer wheel path.

	Intercept – Compaction (mm)	Slope – Wear rate (mm/1000 load cycles)	Goodness of fit – r^2
Segment A	5.37	0.00586	0.998
Segment B	4.68	0.00480	0.993
Segment C	2.24	0.00299	0.934
Segment D	2.30	0.00332	0.986

Table 6. Exponent values relating compaction and wear between the inner and outer wheel paths.

	Intercept - Compaction	Slope - Wear rate
Segment A	3.40	3.04
Segment B	2.13	1.94
Segment C	0.79	0.99
Segment D	1.06	1.77

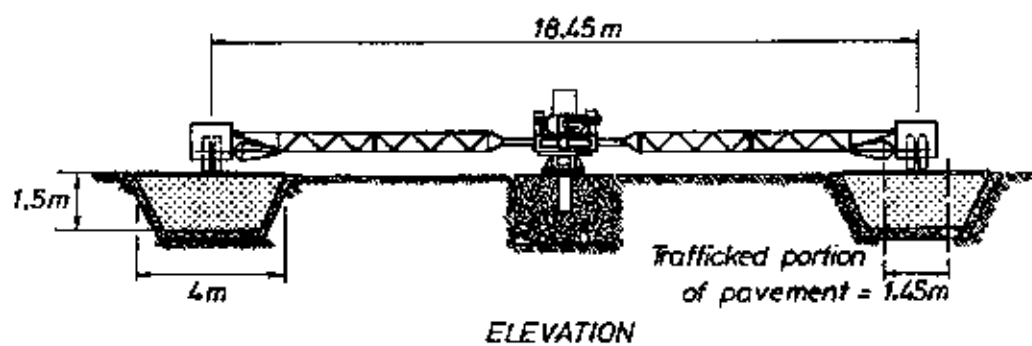


Figure 1 - Elevation view of CAPTIF.

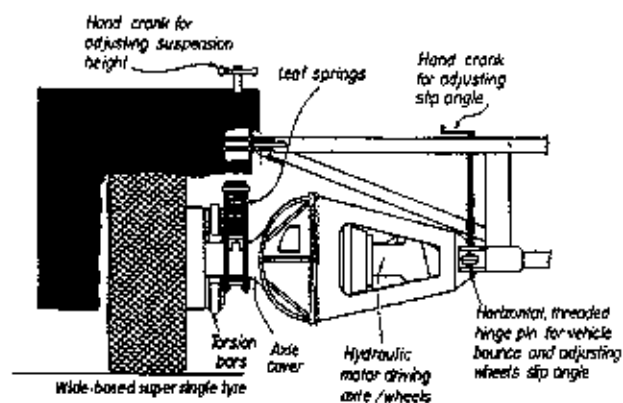


Figure 2 - The CAPTIF SLAVE unit.

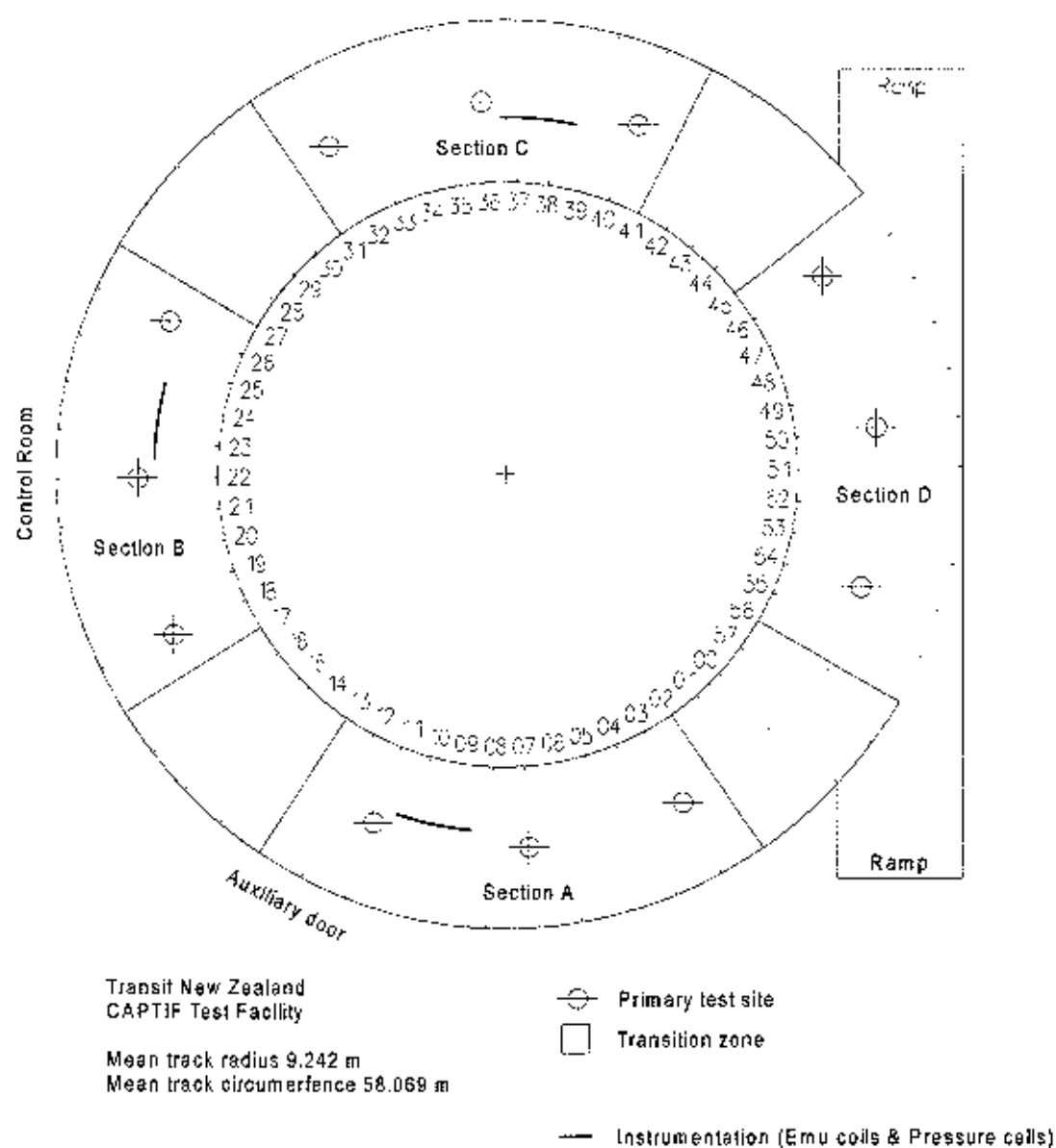


Figure 3. Plan showing the layout of the test sections.

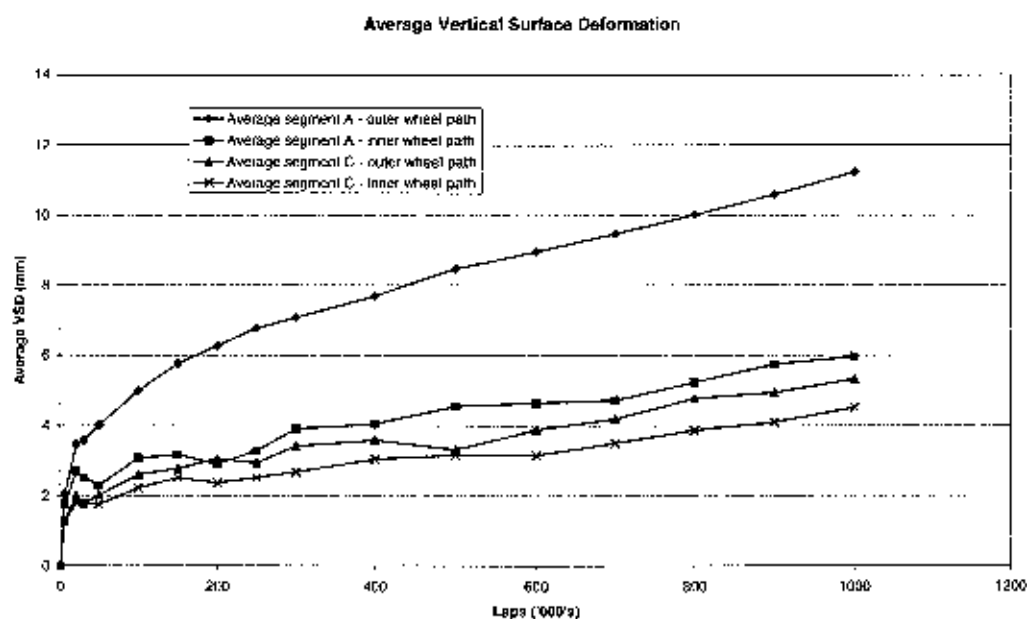


Figure 4. VSD for pavement segments A and C.

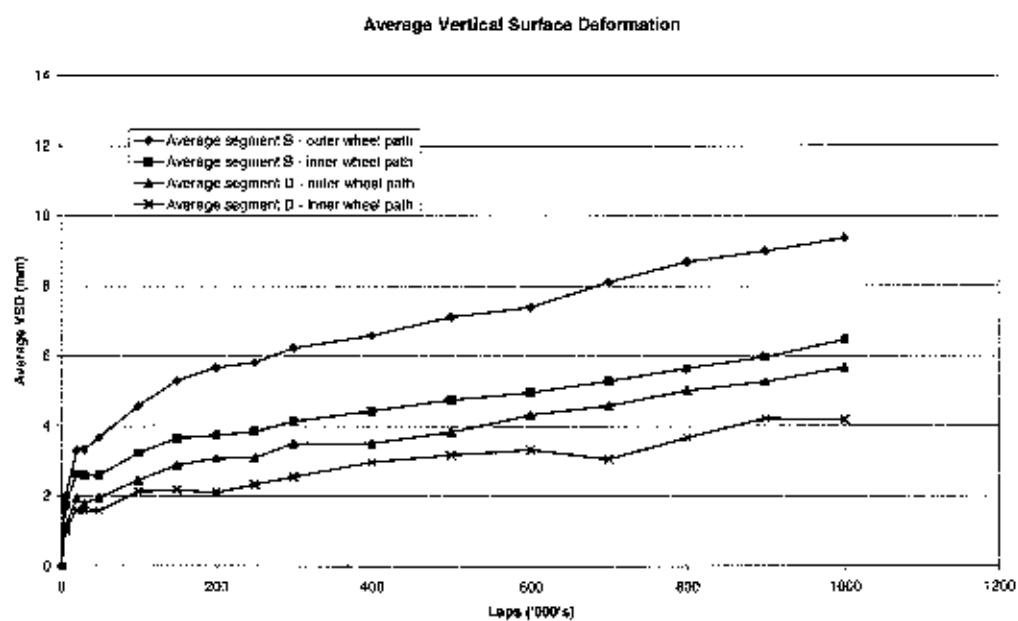


Figure 5. VSD for pavement segments B and D.

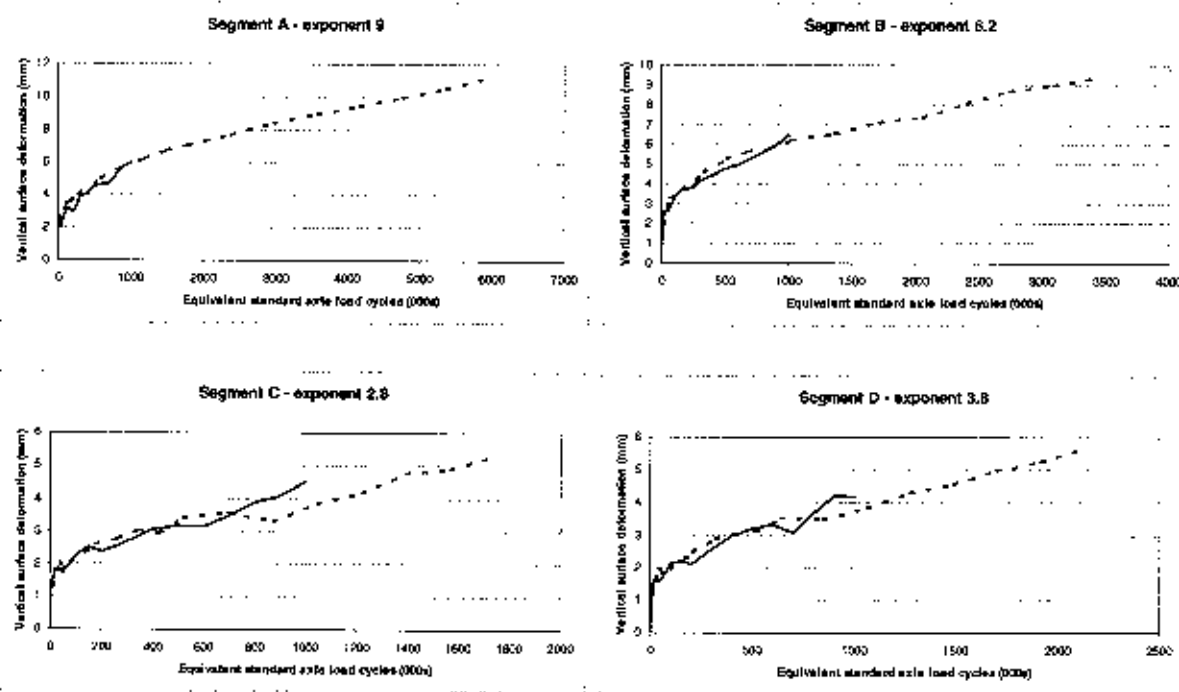


Figure 6. Power law fits to VSD for all four segments.

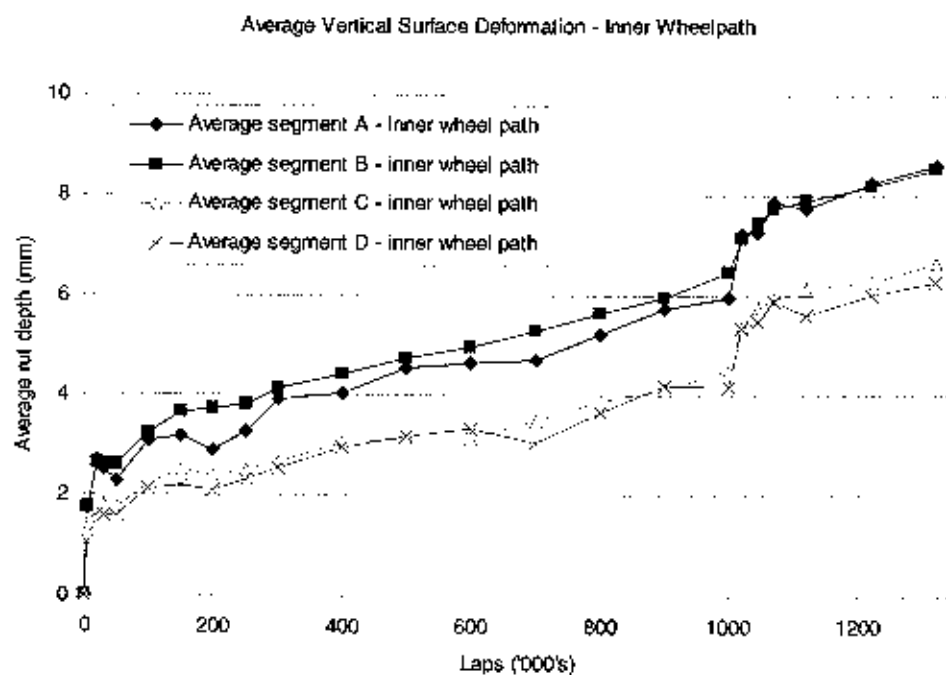


Figure 7. VSD vs load cycles for inner wheel path.

DYNAMIC STABILITY OF DOUBLE B-DOUBLE ROAD TRAINS

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ABSTRACT

In response to concerns over the lateral road space requirements and stability of a fleet of rigid plus double B-double (R2B2) road trains operating in remote Western Australia, investigations were conducted of various options for improving the on-road performance of the vehicles. The vehicles, having a gross mass of over 150t and an overall length of 51.4m, were observed in operation to exhibit excessive swaying of the rearmost trailers. This swaying was found to increase with vehicle speed, and has been raised as being of concern by other road users. An inspection of the vehicles revealed a rearward location of the tow coupling on the rigid truck, and elevated positions of the load bins, both of which contribute to increased rearward amplification, and hence to swaying and increased road space requirements. A change to a double B-double (2B2) configuration was proposed, replacing one of the load bins on the rigid truck and the first dolly with a turntable on the rear of the truck. This removes one articulation point and the long coupling rear overhang, with a slight reduction in payload capacity. In order to quantify the effect on swaying and road space requirements of the proposed change in configuration, comprehensive whole-of-vehicle computer-based models of the two vehicle configurations were created. In simulations of a standard lane change and pulse steer manoeuvres, the 2B2 exhibited superior dynamic stability. Additional simulations were conducted, revealing that reducing the load centre-of-gravity height, increasing the trailer suspension roll stiffness, and increasing tyre cornering stiffness all would lead to further improvements in dynamic stability and road space requirements.

1. INTRODUCTION

1.1 Road Transport in Western Australia

The State of Western Australia (WA) makes up the western third of the Australian land mass (see Fig. 1). The State is rich in mineral wealth with agriculture its second major industry. As there is a limited rail network, much of the mineral material and agriculture product is transported by road, usually over long distances, from areas remote from the export ports and to a limited number of high capacity, low unit cost processing plants/facilities often located at coastal centres.

Transport of livestock from inland areas to the coastal cities for processing or live export is also by road. General freight is also hauled over long distances, for example, from Perth, WA's capital city located in the South of the State, to destinations in the far North of the State.

To enable the mining and agriculture industry to maintain its international competitiveness, the transport component of the cost of export minerals and agriculture needs to be minimised. Most of the state is very flat and dry with low traffic volumes; circumstances that are ideal for the operation of large road trains which are often used.

1.2 Main Roads Western Australia Management of Road Trains

The government authority responsible for managing Western Australia's road asset is Main Roads Western Australia (MRWA). To enable road trains to access specific parts of the road network, MRWA issues individual permits to road transport operators and it also gazettes routes. MRWA works with local government when road trains need to travel on local government roads.

The two vehicles described in this paper – a road train comprising a rigid 5-axle truck towing a double B-double trailer set, shown in Fig. 2(a) and referred to herein as an R2B2¹, and another one comprising a 5-axle prime-mover towing a double B-double trailer set, shown in Fig. 3 and referred to as a 2B2 – operate under a permit. The permit allows up to 23.5 t to be carried on tri-axle groups as opposed to the 20 t normally allowed, as the operator has loading controls in place that ensure this higher axle-load limit is not exceeded.

The vehicles are used to transport nickel concentrate and they operate on local government roads for much of their journey.

1.3 Responsibility to Community

In issuing permits and managing the use of the road network by large road trains, MRWA must ensure the vehicles are safe, and that wear and tear on the infrastructure is acceptable. Further, MRWA must also ensure nuisance and inconvenience to other road users and the community is maintained at a level that is acceptable to all stakeholders.

People in the local community expressed grave concerns with the on-road behaviour of the rigid plus double B-double combinations (R2B2). They reported that at times the rear trailer swayed violently, often well outside the traffic lane. They advised that it was of great concern and some motorists would not share the road with these vehicles.

The operator was also well aware of the concerns of both the road users and the community. In an attempt to improve the vehicle's on-road safety performance the following series of changes was made and tested:

- Replacing dampers.
- Reverting to 20 t on tri-axle groups to achieve a lower centre-of-gravity (CG) height.
- Using dollies with steel spring suspensions in place of the original air spring suspensions.
- Using lower profile semi-oscillating turntables in lieu of the ball-race type turntables.
- Using low profile tyres.
- Restricting the operating speed to 70km/h. This proved to be a partial solution to the rear-trailer swaying problem but was not acceptable to the operator being able to keep to driving hours schedules and to other road users due to the higher speed differential between the trucks and other traffic.
- Different height settings on the air spring suspensions.
- Trialing the use of a 4-axle (twin-steer tandem drive) instead of a 5-axle (twin-steer tri-axle drive) hauling unit. This meant that the towing hitch rear overhang was reduced. An improvement in performance was evident from this step, which suggested overhang of the towing hitch on the hauling unit was a significant causal factor for poor dynamic performance. The axle-group loads of the road train with the 4-axle rigid hauling unit was raised to 23.5 t on tri-axle groups, and this showed no degradation in dynamic performance. This led MRWA to believe that for this configuration of road train coupling rear overhang was an important factor having a significant influence on dynamic performance. Therefore, conversion to a prime mover hauling unit, which would completely eliminate coupling rear overhang on the hauling unit (and one point of articulation), was believed to be a potential solution.

Another major factor found to affect dynamic performance is road unevenness (Prem, Ramsay and Fletcher, 2000). Some sections of the road used by these trucks had been widened by about one metre. This widening, which was done at the edge of the existing pavement, sometimes had greater unevenness than the pavement in the centre of the road. The path of the wheels near the edge of the seal on these widened uneven sections of the pavement translated to dynamic disturbances in the road trains' lateral motion.

The concerns of the community and other road users led to MRWA commissioning RTDynamics to investigate the performance of two road trains. MRWA proposed that the vehicle configuration in use be changed, from a rigid truck towing a double B-double (R2B2) to a prime mover and double B-double (2B2), to improve on-road performance.

¹ The nomenclature system for describing heavy vehicle configurations is fully described in Ramsay, Prem and Peters (2000).

The aims of the investigation performed by RTDynamics were to verify the benefits of the proposed configuration change, and through a parametric study quantify the influence on performance of key mechanical properties. The study was designed to both use and to confirm the main observations described above².

1.4 Performance-Based Standards (PBS) Approach

Austrroads³ and the National Road Transport Commission (NRTC) have developed a range of performance-based standards (PBS) for evaluating the safety and infrastructure related performance of heavy vehicles. The performance standards have been tested against a wide range of vehicles from the Australian heavy vehicle fleet (Prem et al, 2002). MRWA decided to adopt a PBS approach to the assessment of the performance of the road trains⁴ considered. This approach enables the dynamic performance of heavy vehicle combinations to be quantified and the performance of different vehicle combinations to be objectively compared.

2. OBSERVATIONS

2.1 On-Road

As part of the investigation carried out for MRWA by RTDynamics, the vehicles were observed in operation on the Geraldton to Mt Magnet Road. The road was found to be in generally adequate condition, with roadwork in progress to address some previously identified deficiencies. Several laden vehicles were followed and observed as they headed towards Geraldton. Although some swaying was evident, the lane width appeared to be sufficient to ensure that the vehicle did not encroach into oncoming traffic lanes or drop off the road shoulder (see Fig. 2).

2.2 Roadside Inspection

The rigid plus double B-double road train, or R2B2, shown in Fig. 3(a) was inspected near Geraldton. It was noted that the tow coupling on the rigid truck prime mover was located some distance rearward of the drive group centre, as can be seen in Fig. 3(b). Combined with an additional point of articulation (the front dolly), this would contribute to increased rearward amplification, and swaying of the trailers (Ervin et al, 1986; Austrroads, 2000).

Additionally, it was noted that the load bins (kibbles) were located high on the trailer chassis rails (see Fig. 3(c)). A raised CG of the trailers leads to a lower static rollover threshold and decreased dynamic stability (Ervin et al, 1986; Austrroads, 2000).

It was proposed by MRWA to change to a double B-double configuration (2B2) by removing the front tri-axle dolly and coupling the front trailer directly onto the prime mover, as shown in Fig. 4. One less kibble would be able to be carried by the new configuration; however, while it was known there would be an improvement in the dynamic stability it was not known how much would result from this change.

3. COMPUTER MODELS AND SIMULATIONS

Computer-based models were created of the R2B2 configuration and of the proposed revised configuration (2B2), and a range of simulations were performed based on a selection of the safety-related performance measures developed by Austrroads and the NRTC (Prem et al, 2002). These were used to quantify the effects of the proposed changes on dynamic stability and road space requirements, and also to give indications of the effects on performance of varying several key design parameters.

The models were created using the ADAMS multi-body dynamics simulation software (Mechanical Dynamics Inc., 2002), together with the truck modelling toolbox developed by RTDynamics (RTDynamics, 2002a; 2002b). Full three-dimensional, dynamic models of the vehicles were created, including controllers for speed, traction control and steering.

² Further observations of the behaviour of these road trains showed that on long downgrades a swaying of the rear trailers developed. This behaviour may be due to units of the combination being in longitudinal compression, which would produce a destabilising yaw moment on each unit in the combination. When the prime mover reverted to pulling the trailers uphill the swaying abated. This behaviour was not investigated as part of the work carried out by RTDynamics described in this paper.

³ Austrroads is the association of Australian and New Zealand road transport and traffic authorities.

Detailed non-linear air suspension models were used on the drive, dolly and trailer axle groups, and a non-linear multi-leaf steel spring suspension model based on Fancher et al. (1980) was used for the steer axle. Validated tyres models were based on those developed by Gim (1988).

Fig. 5(a) shows the computer model of the R2B2 that is depicted in Figs 3(a) to 3(c). Using the translucency features within ADAMS, some of the detail of the hauling unit is revealed in Fig. 5(b), showing the location of the drawbar hitch, drive group suspension, drive train, engine and gearbox, fuel tanks, steer axle suspension and steering arrangement.

The two road train models were simulated in the SAE lane change manoeuvre (Society of Automotive Engineers, 1993), in which a single lane change of width 1.46 m is executed at a speed of 88 km/h over a distance of 61 m. A second simulation was performed, in which a steer angle pulse is applied at the road wheel over a 0.1 s period. Test speed for the pulse steer manoeuvre is 100 km/h.

The responses of the computer models were compared to each other and to those of a reference vehicle model, representing a full-size, commonly used, triple road train with a low CG hauling similar mining product to the two vehicles studied. The triple road train is referred to in this paper as TRIPLE, as shown in Fig. 6.

The dynamic responses of all three models were compared using three of the performance measures developed by Austroads/NRTC (Prem et al, 2002).

4. PERFORMANCE MEASURES

The following three performance measures from those proposed by Austroads/NRTC (Prem et al, 2002) were selected to characterise the dynamic stability of the road trains considered in this paper: rearward amplification, high-speed transient offtracking and yaw damping coefficient. These are briefly described in the following sections; a full description is given in Prem et al (2001) and Prem et al (2002).

4.1 Rearward Amplification

Rearward Amplification (RA) is a measure of the tendency of the trailing unit(s) of a multi-articulated vehicle to amplify any lateral acceleration experienced at the hauling unit. The performance requirement for RA set by Austroads and the NRTC, which assumes the SAE lane change is representative of a typical evasive manoeuvre, is defined in terms of the rollover stability of the critical, rearmost roll-coupled unit, as follows:

$$RA = 5.7SRT_{rcu} \quad (1)$$

where:

RA = rearward amplification measured in accord with recommended practice
SAE J2179 or ISO 14791 (-)

SRT_{rcu} = static rollover threshold of the rearmost roll-coupled unit (g)

For a vehicle with a static rollover threshold of 0.35g, Eqn (1) sets the performance level for RA at 2.0, a performance level that is based on research carried out in the USA (Fancher et al, 1989; Winkler et al, 1992)⁴. Further, according to Eqn (1), larger values of RA are deemed to be acceptable only if accompanied by a commensurate increase in the rollover stability of the rearmost roll-coupled unit(s). Both SAE J2179 and ISO 14791 are accepted methods of testing, being well established and proven procedures that are fully documented (Society of Automotive Engineers, 1993; International Organisation for Standardisation, 2000). Further details on development of the revised performance level for RA can be found in Prem et al (2002).

In practical terms, a threshold value of 2.0 for RA means that the lateral (sideways) acceleration at the CG of the rearmost unit in the combination should not exceed twice the lateral acceleration at the centre of the steer axle of the hauling unit. In the SAE lane change manoeuvre the steer axle lateral acceleration has a peak value of 0.15g. Therefore, for the example cited, an RA that is less than 2.0 would be considered to be acceptable.

⁴ During development of the Austroads/NRTC performance standards a set performance level of 2.0 for RA was also proposed (see Prem et al, 2001). However, the form described by Eqn (1) is preferred because it directly links the performance requirement to rollover stability, discussed fully in Prem et al (2002).

4.2 High Speed Transient Offtracking

High Speed Transient Offtracking (HSTO) measures how far the rear of the combination vehicle tracks outside the path taken by the hauling unit during the SAF lane change manoeuvre. The performance standard for HSTO requires that the centre of the rear of the last trailer of the combination vehicle remain within 800 mm of the path taken by the centre of the steer axle (Prem et al, 2001; Prem et al, 2002).

4.3 Yaw Damping Coefficient

Yaw Damping Coefficient (YDC) quantifies how quickly oscillations of the last trailer take to reduce in amplitude, i.e. settle, after the application of a short duration steer input at the hauling unit. Vehicles that take a long time to settle increase the driver's workload and represent a higher safety risk to other road users. Under the Austroads/NRTC performance standards (Prem et al, 2002) the yaw-damping coefficient is required not to be less than 0.15 for yaw damping to be considered acceptable.

5. RESULTS

The results for the three vehicles are presented in summary form in Table 1. This shows that relative to the specified performance levels considered, the performance of the 2B2 was superior to the R2B2.

The greatest change in percentage terms was seen in the high-speed transient offtracking, which decreased from 1.28m to 0.59m, a reduction of 54%. The results for the 2B2 all met the Austroads/NRTC performance standards, with the exception of YDC, falling short by 7.3% of the performance level specified.

Table 1 also shows that both the previous configuration (R2B2) and the TRIPLE meet the RA performance standards but they do not meet the HSTO requirement. However, the performance of the TRIPLE, in terms of RA and HSTO, lies between the R2B2 and the 2B2.

6. PARAMETRIC STUDY

As a consequence of the 2B2 configuration meeting two of the three performance standards considered in this study, and almost meeting the third, a series of further simulations were conducted to determine if additional performance improvements could be achieved by making changes to some of the vehicle's mechanical properties. These changes may need to be considered if the vehicle were to operate under a concessional loading scheme where the tri-axle group loads would be increased from 20.0 t to 23.5 t. The following four scenarios were examined:

1. Reduce CG height by 100mm, such as by lowering the position of the kibbles (see Fig. 3(c)).
2. Increase the roll stiffness of the trailer and dolly suspensions by 20%.
3. Increase the tyre cornering stiffness by 20%; as could be achieved by using low profile tyres, or as could occur with normal tyre wear⁵.
4. All of the above three combined.

Table 2 presents the results of making these changes on the three performance measures considered. The greatest improvement in RA and HSTO, which would be reflected in road space requirements, was found to occur in changing to tyres that have greater cornering stiffness. It is possible to realise this by moving to low profile tyres, or, mindful of all the possible safety considerations and the associated implications, by using tyres with less tread on the rear trailers and saving the tyres with more tread for the front trailers. Increased cornering stiffness also gave the greatest increase in YDC, which would meet the performance standard, and cause sway oscillations of the trailer to both decrease more rapidly and execute fewer oscillations.

The influence of lowering trailer CG by 100mm (a reduction in CG height of about 6%) and increasing suspension roll stiffness is less than that of increasing tyre cornering-stiffness, but is still significant, emphasising the need to both design for as low a CG as practical while also using suspensions with the highest practical roll stiffness.

Table 2 shows the improvement toward achieving the specified performance levels is substantial when all three changes are introduced simultaneously, and ranges from about 20% for RA, in excess of 45% for HSTO, and 51%

⁵ It is useful to note that with wear the cornering-stiffness of tyres can increase by up to about 20% (Fancher et al, 1986).

for YDC. It is also useful to note that for all three performance measures (RA, HSTO and YDC), the improvement when all three design changes are introduced simultaneously is greater than the improvement due to each change made independently, emphasising the interactive nature of the parameters considered and their influence on vehicle dynamics.

The changes identified, if practical and introduced concurrently, are predicted by the modelling to produce significant improvements in the safety-related on-road performance of the 2B2 configuration by meeting the three performance standards considered.

7. CONCLUSIONS

7.1 Summary of Findings from Computer-Based Modelling

The computer-based simulations have suggested that the double B-double (2B2) configuration would be expected to have a better dynamic performance than the rigid plus double B-double (R2B2) configuration. This improvement should be evident in less swaying of the trailers and in reduced road space requirement. Much of this improvement could be attributed to the elimination of the coupling rear overhang and one articulation point.

Further improvements in dynamic stability and road space requirements can be realised by lowering the load height and hence the CG, by increasing suspension roll stiffness, and by using tyres with greater cornering stiffness.

There is a range of other performance enhancing design changes that could be made to road trains, which could be explored and their effect quantified cost effectively through computer-based modelling as demonstrated in this paper.

7.2 Feedback on Operation following Conversion of Hauling Units

The initial findings of this investigation were published in June of 2001 (Ramsay and Prem, 2001). All hauling units have since been converted from rigid trucks to prime movers, and MRWA has now been receiving positive feedback on the operation of the revised configuration. This is in line with the improved dynamic performance predicted by the computer modelling.

Given this link between observed on-road performance and simulated behaviour, MRWA will continue using the PBS approach and whole-of-vehicle computer-based modelling for assessing the performance of large heavy vehicles in the future. The study and subsequent changes to the hauling units showed that the PBS approach is an effective way to assess heavy vehicle performance without the need to instrument and test actual vehicles.

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TABLES & FIGURES

Table 1 - Computer Simulation Results

Performance Measure (units)	R2B2	2B2	2B2 of R2B2	TRIPLE	Performance Level
Rearward Amplification (-)	2.72 (2.74)	1.92 (2.74)	-29%	2.11 (2.45)	$\leq 5.7SRT_{rru}$ (Performance Level)
High Speed Transient Offtracking (m)	1.28	0.59	-54%	0.91	≤ 0.80
Yaw Damping Coefficient (-)	0.101	0.139	+38%	0.116	≥ 0.150

Table 2 - Results of Parametric Study for the 2B2

Performance Measure (units)	Baseline	Lower CG	Increase Susp'n Stiffness	Increase Roll Stiffness	Tyre Cornering Stiffness	All
Rearward Amplification (-)	1.92	1.89	1.86	1.63	1.51	
High Speed Transient Offtracking (m)	0.59	0.55	0.56	0.37	0.32	
Yaw Damping Coefficient (-)	0.139	0.144	0.143	0.192	0.210	

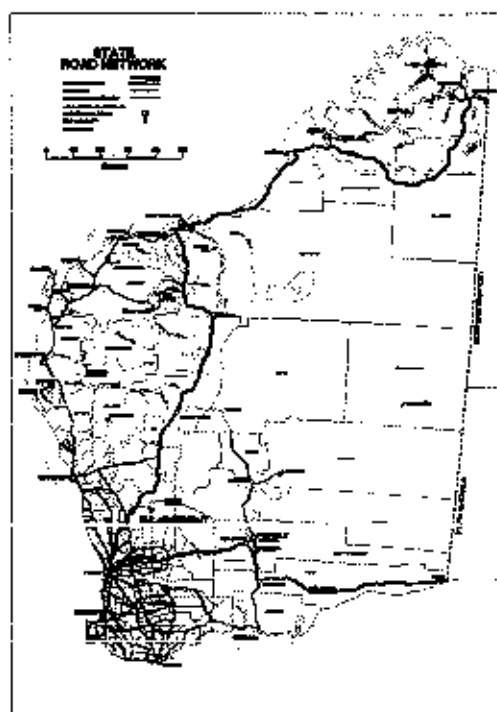


Fig. 1 The roads network in Western Australia (from <http://www.mainroads.wa.gov.au>).



Fig. 2 Typical area of concern over road space requirement of road trains.



Fig. 3(a) Rigid plus double B-double (R2B2) road train inspected near Geraldton.



Fig. 3(b) Side view of rigid truck (hauling unit) showing the coupling rear overhang dimension.



Fig. 3(c) Side view of trailer showing approximate location kibble CG.



Fig. 4 Proposed double B-double (2B2) road train configuration.



Fig. 5(a) RTDynamics computer-based model of the R2B2 shown in Figs 3(a) to 3(c).

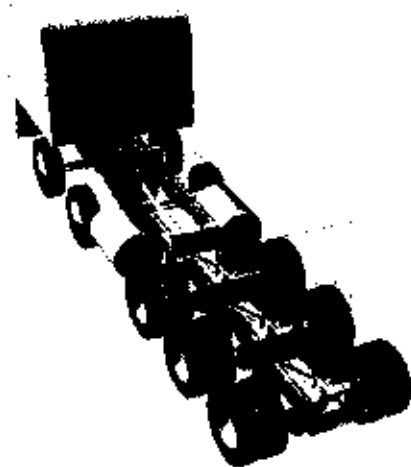


Fig. 5(b) Computer-based model of R2B2, hauling unit detail shown.



Fig. 6 Low CG, reference triple road train (TRIPL,E); 12S3-2S3-2S3.

COMPARISON OF THREE PROGRAMS FOR SIMULATING HEAVY-VEHICLE DYNAMICS

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ABSTRACT

Australia and New Zealand are developing a performance-based standards system of heavy vehicle regulation as an alternative to the current prescriptive systems, which is intended to encourage and foster innovation in road transport. Expected outcomes of the new regulatory regime are improved road safety, better use of the infrastructure, reduced environmental impacts and enhanced productivity. Computer-based modelling will play a central role in both the development and initial demonstration of innovative vehicles and concepts. Some Stakeholders have expressed the concern that performance predictions from computer-based models may not be reliable and may substantially differ with software package and with the Service Providers that use them. To address these concerns three computer-based modelling packages were compared: ADAMS, The University of Michigan Transport Research Institute (UMTRI) constant velocity Yaw/Roll model and AUTOSIM. Models of two heavy vehicles were created by two Consultants using the three packages. The same input datasets were provided to each Consultant and simulations were performed using the same test manoeuvres. Time histories of a wide range of variables from the simulations were compared as well as numeric values from a selection of performance measures. Simulations that provide a direct measure of vehicle responses to precisely defined steer inputs (open-loop control) showed excellent agreement and the results were generally more consistent than simulations requiring steer controllers and closed-loop path following. With due care, acceptable agreement between modelling packages can be achieved. Recommendations are made that reduce the variability between models in path following tasks to acceptable levels.

1. INTRODUCTION

1.1 Brief Review of Heavy Vehicle Dynamics Modelling

Realistic models and simulations of heavy vehicle dynamics have been developed over a period of almost 40 years (Segel, 1993). Early simulations were based on mathematical models whose equations were laboriously and painstakingly derived by hand and solved using purpose-written computer programs. The development of these simulation programs and verification of their outputs against experimental tests was undertaken at considerable expense in government funded research programs and through the involvement and support of equipment manufacturers and the transport industry.

In the late 1970s engineers began using newly available computer programs that automated the process of creating the mathematical models and solving the equations (Orlandea and Chace, 1977). These programs are generically referred to as general-purpose multi-body dynamics programs, allowing detailed models of literally any mechanical system to be created and simulated rapidly with relative ease (Schiehlen, 1986; Kortüm and Sharp, 1993; Richard and Gosselin, 1993; Sharp, 1994).

Multi-body simulation programs are now widely used throughout the automotive industry, and they are also used to solve dynamics problems that arise in areas such as biomechanics, machine dynamics, rail vehicle dynamics, robotics and in spacecraft dynamics and control.

In the field of vehicle dynamics, particularly in Australia, New Zealand, Canada and the USA, three computer-based programs have emerged as being the most popular and widely used for simulating heavy vehicle dynamics.

They are ADAMS[®] (Ryan, 1993; Mechanical Dynamics, 2002)¹, The University of Michigan Transportation Research Institute (UMTRI) "Constant Velocity Yaw/Roll Model" (Mallikarjunarao, 1981; Gillespie and MacAdam, 1982) and AUTOSIM[™] (Sayers, 1990; Sayers, 1993; Mechanical Simulation Corporation, 2002)².

1.2 Modelling Applications in Performance-Based Standards

In a major initiative in heavy vehicle regulation, Austroads³ and the National Road Transport Commission are developing performance-based standards as a national alternative to the current prescriptive system (National Road Transport Commission, 2002). Expected outcomes of the new regulatory regime are improved road safety, better use of the infrastructure, reduced environmental impacts and enhanced productivity.

Given that performance-based standards are intended to encourage and foster innovation in road transport, computer-based modelling will play a central role in both the initial demonstration and the development of innovative vehicles and concepts. However, some Stakeholders have expressed the concern that performance predictions from computer-based models may not be reliable and may differ significantly with software package and with the Service Providers that use them.

This paper attempts to address these concerns by comparing three computer-based modelling packages – ADAMS, UMTRI's constant velocity Yaw/Roll model and AUTOSIM – in a range of manoeuvres.

2. METHODOLOGY

2.1 Verification of Models

The literature confirms that models created with each of the modelling packages considered in this paper have been previously validated against experimental tests (Orlandea and Chace, 1977; Gillespie and MacAdam, 1982; Lam, 1986; El-Gindy and Wong, 1987; Sayers, 1990; Sayers and Riley, 1996; Latta, 1999; Prem et al, 1999; Sweatman and McFarlane, 2000; Schade and Hamill, 2000; Elliot et al. 2001).

A review of a selection of verification studies was carried out as part of a separate study and is reported in Prem et al (2001b). The review generally confirmed that an acceptable level of agreement can be achieved between experimental data and responses predicted with any one of the three modelling packages considered in this paper. However, while this has been demonstrated, it is important to note that the level of agreement achieved between test data and simulations very strongly depends on the quality of the test data and on factors that are related to the models. These factors include the level of detail included in the model, accuracy of the component models, accuracy and integrity of the input data (dimensions and mechanical properties), and the manoeuvres performed.

It should be further noted that obtaining accurate test data that can be used for validating computer-based models is a non-trivial and exacting task. It necessarily covers issues such as the use of correct instrumentation and its proper installation and operation, pre-processing and recording of signals from a wide range of different transducers, the conduct of controlled tests under less than ideal in-the-field conditions – which at times may involve performing near-limit manoeuvres – and recording all inputs and disturbances to the vehicle (steering, roadway and environmental) that may influence the measured responses.

2.2 Testing the Modelling Systems

Given that the modelling packages had been validated against experimental tests, the aim was to devise a series of simulations, or test problems, in a heavy vehicle dynamics context, that would test and explore the boundaries of the modelling packages as well as the Consultants (Service Providers) using them.

Roads and Transport Dynamics (RTDynamics) performed the modelling in ADAMS, Transport Engineering Research New Zealand (TERNZ) used AUTOSIM and the UMTRI Yaw/Roll model.

The following sequence of activities outlines the process, discussed in further detail in the subsequent sections of this paper:

¹ ADAMS is an acronym for Automatic Dynamic Analysis of Mechanical Systems. In addition to its widespread use in Australia and North America, ADAMS is also used widely in Europe and Asia.

² The AUTOSIM acronym is derived from "AUTOMatically generate SIMulation codes".

³ Austroads is the association of Australian and New Zealand road transport and traffic authorities.

- i) Two reference vehicles were developed and a comprehensive list was prepared of the parameters required to create the vehicles in the computer-based modelling package;
- ii) Standard manoeuvres were developed that would be simulated in each modelling package – these manoeuvres are based on the ones presented in Prem et al (2001a);
- iii) The reference vehicles were created in each of the modelling packages, the standard manoeuvres were simulated, and a range of outputs from the simulations was compared and the level of agreement between them was determined.

3. VEHICLES MODELLED, INPUT DATASETS AND MANOEUVRES

3.1 Reference Vehicles

Two reference datasets describing two separate heavy vehicles were created. The datasets describe a generic B-double and a generic truck/trailer combination. Sufficient information was sourced and prepared that would fully describe the vehicles and enable computer-based dynamic models to be created in each of the packages. The two vehicles chosen for this exercise were considered appropriate for the purposes of this work because they contain features that are present not just in a significant proportion of the Australasian heavy vehicle fleet, but also in heavy vehicles used throughout the world. Where possible, hardware elements were included in the models, such as non-linearity in suspensions components and tyres, to ensure a range of features within the modelling packages and capabilities of the Consultants could be tested and demonstrated.

It is important to note that the models were developed to test a certain range of base level and advanced computer modelling features and skills pertaining to heavy vehicle dynamics simulation. The models do not necessarily test all of the capabilities or features of the modelling packages that were used or the Consultants using them.

Perspective views of the two reference vehicles are given in Figs 1(a) and 1(b). Full descriptions of all the parameters used in the models are provided in Prem et al (2001b), which also contains the complete set of values for each parameter of each reference-vehicle⁴. The parameters used in the models are briefly described below.

3.2 Input Datasets and Steer Control

3.2.1 Geometry, Mass Properties, Suspensions and Tyres

The input datasets for the two reference vehicles are presented and fully described in Prem et al (2001b). These include:

- geometric layout and key dimensions;
- mass properties of bodies⁵ (centre-of-gravity CG location and mass moments of inertia);
- couplings (turntables, pin-couplings and drawbar);
- characteristics of the suspensions (springs, dampers, roll centre heights and roll steer coefficients); and
- tyres characteristics.

The force/deflection characteristics of the suspension springs used in the models, for example, are shown in Fig 2(a) and 2(b). These were derived from the data presented in Ervin and Guy (1986). The force/deflection values selected for the vehicle are simply the average of the compression and extension envelopes from the selected spring tables. As can be seen in Figs 2(a) and 2(b), with the exception of the steer axle, the springs are highly non-linear and there is a substantial increase in stiffness for deflections that are outside the range $\pm 30\text{mm}$.

The springs do not include inter-leaf coulomb friction (hysteresis). It was not included because no documentation could be found that described how it had been modelled in UMTRI's Yaw/Roll program. Further support for this decision came from a similar exercise conducted by Sayers and Riley (1996) in which various Yaw/Roll models were compared with models of the same vehicles created in AUTOSIM. They concluded that it would be difficult to get agreement with UMTRI's Yaw/Roll program (particularly with regard to transient responses) unless the damping due to hysteresis has been set to zero.

In the models identical tyres were used on the steer, drive and trailer axles. The tyre side force characteristics due to slip angle are based on data presented in Ervin and Guy (1986); the corresponding data for aligning moments were obtained from El-Gindy and Kenis (1998). These data are plotted in Figs 3(a) and 3(b) and describe tyre side

⁴ Prem et al (2001b) can be downloaded from the National Road Transport Commission website at <http://www.nrtc.gov.au>

⁵ For simplicity all parts used in the models were assumed to be rigid bodies.

force and aligning moment characteristics over a range of vertical loads and varying levels of slip angle in the absence of longitudinal slip.

Side force generation due to inclination (camber angle) has not been included in the models because there is no provision for this feature in UMTRI's Yaw/Roll program; though it is a feature within ADAMS and it can also be included in AUTOSIM. Further, there is also no provision in UMTRI's Yaw/Roll program to include relaxation length, and for this reason the tyre relaxation length parameter in the respect ADAMS and AUTOSIM models was set to zero. This means that the build-up of tyre side force and aligning moments is assumed to occur instantaneously, and the side force and aligning moments will always be at their maximum values for the prevailing slip and vertical load conditions.

3.2.2 Steer Control

Directional control of the models is achieved through the steer controller. Depending on the manoeuvre specified, steer control is intended to accomplish either a specific, precisely defined sequence of steer angle input (open-loop control), or it will guide the vehicle along a prescribed path in the same way that a real driver would steer the vehicle along a path (closed-loop control).

The characteristics of the steer controller for closed-loop control were not specified in the dataset. It was assumed that suitable controllers were either included as a feature of the specific software package (see, for example, Gillespie and MacAdam, 1982) or it was created from first principles, as was the case with the simulations in ADAMS.

3.3 Manoeuvres Simulated

Previous similar studies have used lane-change manoeuvres, steady (or quasi-steady) turns, and step steer inputs (Gillespie and MacAdam, 1982; Lam, 1986; El-Gindy and Wong, 1987; Sayers and Riley, 1996). These simulations were designed to test for specific performance attributes thereby revealing different aspects of the vehicle models and/or controllers.

The lane change, for example, is a very useful manoeuvre but it is not a reliable way of comparing vehicle models because identical vehicle models with different steer controllers may yield different responses. Hence, the controllers may have an influence on how well the responses of the vehicles match.

Steady turns, or quasi-steady turns, are also useful for comparing vehicle models but they also have limitations; typically, when the steady conditions have been reached all of the terms in the equations-of-motion that are related to the transient response have reduced to zero. These terms may be the only sources of differences between the response of the models which would not be revealed in the steady turn test⁶.

For the present study the following two types of simulations were performed:

- i) one to characterise vehicle-only responses under both transient and steady state conditions involving an open-loop control strategy requiring the precise application of a predetermined steer sequence and observation of vehicle response;
- ii) the other to provide information about the combined driver/vehicle responses under both high- and low-speed conditions using closed-loop steer control in two well-defined path following tasks.

In total, four simulations were devised that would test and compare a range of features in the models. Pulse steer and step steer inputs were used in the two simulations that employed open-loop steer control, and closed-loop control was used in path following tasks of a high-speed lane change and a low-speed 90° turn. These are described below under the two general headings of open- and closed-loop control.

⁶ When a dynamic system reaches a steady state condition many terms in the equations that govern its motion will have decayed to zero and thereby become numerically insignificant. By contrast, all of the terms in the equations of motion contribute to the transient response and matching the transient responses of two dynamic systems is considered a more challenging task than matching the steady state responses.

3.3.1 Open-Loop Control

a) Pulse-steer (transient response)

The pulse steer simulation is identical in concept to the test described in Prem et al (2001a) for estimating yaw damping. From an initial straight path, and a speed of 100km/h, the vehicle is subjected to a short duration input of steering; a very rapid application of steer is applied from the straight-ahead position over a short time-period and then quickly returned to the straight-ahead position over a time period of similar short duration. Once the steer has been applied the vehicle is then allowed to follow its own course without any further control action or external disturbances.

In response to the "pulse" of steering applied to the steered tyre, the towing unit (prime-mover or truck) will change its direction of travel (its heading) and the towed units (trailers) will follow. For a stable vehicle, and provided the subsequent motion is not extreme, the transition to the new heading is accompanied by vehicle oscillations that will progressively reduce in amplitude. For an unstable vehicle the oscillations may increase in amplitude culminating in rollover. This type of disturbance is commonly used to quantify the vibration characteristics of dynamic systems (Doebelin, 1980), including passenger cars and heavy vehicles (International Standards Organisation, 1988; International Standards Organisation, 2000). When applied to a heavy vehicle it will provoke a response from all of the key modes of oscillation: lateral/directional, yaw, roll and pitch.

The pulse steer input that was developed for this application is based on the haversine function described by Eqn 1, providing a smooth ramp-up and ramp-down transition.

$$\delta = \begin{cases} \delta_0 & t \leq t_0 \\ \frac{\delta_0 + \delta_1}{2} + \frac{\delta_1 - \delta_0}{2} \sin \left[\left(\frac{t - t_0}{t_1 - t_0} \right) \pi - \frac{\pi}{2} \right] & t_0 < t < t_1 \\ \delta_1 & t \geq t_1 \end{cases} \quad (1)$$

where:

- t = time (s)
- t_0 = commencement of steer application (s)
- t_1 = termination of steer application (s)
- δ = steer angle (deg)
- δ_0 = initial value of the steer angle (deg)
- δ_1 = final value of the steer angle (deg)

A steer pulse of 10° and 0.5 s duration was chosen, which is significantly larger in amplitude and longer in duration than the 3.2°, 0.1 s pulse specified for yaw damping estimation described in Prem et al (2001a).

The much larger steer input used in this study was designed to ensure the vehicle models were well into the non-linear operating regions, particularly the suspensions and tyres, in order to test the capabilities of the modelling packages at the boundaries and over the range of conditions that might realistically occur in performance-based standards related applications.

The pulse was also carefully designed to ensure that:

- i) the component models were not operated outside of the range of the real-world data that were used to create them, these include the measured tyre characteristics that are shown in Figs 6(a) and 6(b) and the suspension characteristics illustrated in Figs 4(a) and 4(b); and
- ii) modelling assumptions were not violated, such as the small angle approximations for pitch and suspension relative roll angle used in UMTRI's Yaw/Roll program.

b) Step-steer (steady-state response)

The step steer simulation is very similar in form to the pulse steer except that the steer angle, instead of returning to zero, is held at its maximum value for the duration of the simulation. Once the transients have settled after the initial application of steer, the vehicle enters a steady turn and follows a circular path on a constant radius.

The steer angle input of 10° that was used for the pulse steer, if sustained, is much too large for the step steer test and would cause the vehicles to rollover. A steer angle of 1° was found to be acceptable, making this manoeuvre consistent with the baseline manoeuvre developed by Sayers and Riley (1996).

3.3.2 Closed-Loop Control

Inertia forces and the prevailing vehicle dynamics are directly influenced by speed; at near-zero speed the inertia forces will be close to zero and the steer controller will be presented with a system that has different characteristics to one travelling at speed. For simulations involving closed-loop steer control it was therefore considered important to investigate the influence of speed on the closed-loop responses.

a) SAE lane change

The first of the closed-loop simulations is the SAE lane change (Society of Automotive Engineers, 1993). In the standard SAE lane change the vehicle is driven at 88km/h along a straight path approximately 100m in length. The vehicle is required to execute a lane change manoeuvre and the path of the centre of the steer axle not deviate by more than $\pm 150\text{mm}$ from a precisely prescribed path over a distance of 61m. The lateral displacement of the lane change manoeuvre is 1.46m. These dimensions were chosen to give a peak lateral acceleration at the steer axle of 0.15g.

b) Low-speed turn

In the low-speed turn the centre of the steer axle is required to follow a path comprising straight approaches to an 11.25 m radius 90° circular arc (Prem et al, 2002). This corresponds to the outside front wheel following a path of radius 12.5 m. A constant vehicle speed of 10 km/h is specified.

4. COMPARISON OF SIMULATION RESPONSES

The responses from the simulations described earlier are compared in this part of the paper. Only a selection of the key outputs and variables from each simulation are presented and compared. The key findings are discussed and where appropriate they have been compared with the results from other similar studies.

4.1 Open-Loop Control

4.1.1 Pulse-Steer (Transient Response)

a) B-double

The first variables considered, shown in Figs 4(a) and the first plot in Fig. 4(c), are tyre slip angles, the corresponding vertical forces (the lateral forces are not shown), and the tyre aligning moments, respectively.

These plots show that there is very close agreement of outputs from the three modelling systems with only small differences evident in the vicinity of the three largest peaks. The differences are particularly apparent in the tyre self-aligning moments, which are shown in Fig. 4(b). These differences are discussed in more detail shortly.

The initial peak in slip angle of about 9° , developed at the steer tyre and evident in the three plots, is due to the 10° pulse-steer input. This peak, and the subsequent slip angles developed by the other tyres in the ensuing motion, is well within the slip angle range (0 to 12°) of the measured tyre data shown in Fig. 3(a). Further, the vertical loads on the tyres do not exceed the range of the measured tyre data, and when considered together with the slip angle data it confirms the tyres are well into the non-linear operating region but not outside the range of the component data.

As discussed above, the main differences in the tyre forces and self-aligning moments occur at the peaks, with ADAMS typically predicting larger side forces (not shown) and aligning moments than either Yaw/Roll or AUTOSIM. This can be attributed to the way the tyre data are used in the component models within each modelling package. In Yaw/Roll and AUTOSIM linear interpolation/extrapolation is used (Gillespie and MacAdam, 1982; University of Michigan Transportation Research Institute, 1997), whereas the tyre models available within ADAMS use continuous and smooth analytical functions, which are based on tyre behaviour theory, to estimate the intermediate values (Bakker et al. 1987; Pacejka and Bakker, 1993). For slip angles (and

vertical loads) between the measured points the side force and aligning moment values from the ADAMS tyre models will generally be slightly greater than the corresponding values from either Yaw/Roll or AUTOSIM, as illustrated in Figs 5(a) and 5(b). With a greater number of input measured data points used in the component model the match between the models may be expected to improve.

The models created in Yaw/Roll and AUTOSIM assumed that pitch motion of the sprung masses are small. The pitch angles were checked (both for the B-double, and for the truck/trailer described later) and they were found to be less than 0.3° thereby confirming that the small angle assumption had not been violated.

The suspension forces are next compared in the second plot in Fig. 4(b). The suspension forces reach a maximum of about 55,000N confirming they are within the range and in the non-linear region of the component models that are based on the suspension data shown in Figs 2(a) and 2(b). Fig. 4(b) shows there is also close agreement between the outputs from the three modelling systems, except for the slightly larger amplitude 2Hz oscillation in the drive axle suspension, which appears in both the Yaw/Roll and the AUTOSIM outputs; it may be a by-product of the assumption of small pitch angles used in these two models but not used in the ADAMS model.

Fig. 4(c) shows there is little difference in the lateral accelerations and the roll angles, thus confirming there is close agreement between the three modelling systems. The small differences that do exist are likely due to the differences in the tyre models discussed above.

Due to the very abrupt nature of this manoeuvre, it can be seen in Fig. 4(c) that the prime mover momentarily reaches a peak lateral acceleration of 0.45g. In a steady turn if this value were sustained it would be sufficient to cause rollover. However, because the prime-mover is coupled in roll through the turntable to the unit behind it, and because the event is short in duration, a significant roll angle is prevented from developing. Similarly, the lateral acceleration of the second trailer reaches a peak that is slightly greater than 0.35g, and the roll coupling to the unit ahead of it also prevents a large roll angle developing.

b) Truck/Trailer

The truck/trailer is a dynamically more active system than the B-double, comprising two sprung masses that are not coupled in roll but connected through a pin coupling. Its motions are much more complex than that of the B-double providing a greater challenge for the modellers and testing additional features of the modelling packages.

The simulation outputs for the truck/trailer are compared in Figs 6(a) and 6(b). These include tyre slip angles, yaw rates, lateral acceleration and roll angles of the sprung masses.

The first observation that can be made is that the responses of the truck/trailer are not as smooth as those of the B-double and take much longer to reduce in amplitude; there being residual oscillatory motion even after 10 seconds of run time.

Fig. 6(a) shows the greatest differences in slip angle, exhibited by the trailer, is small, being no more than about 0.5° .

Yaw rates, lateral accelerations and roll angles are shown in Figs 6(a) and 6(b). These variables also indicate there is close agreement between the models. The differences that are apparent mainly occur around the peaks, and there is a small difference in the zero crossings at several locations.

Lateral acceleration, which is a key measure for vehicle stability, is compared in Fig. 6(b) showing that all three units are subjected to high levels in this manoeuvre; approximately 0.4g for the truck and about 0.45g for both the dolly and the trailer. Overall the agreement between the models is considered to be good.

4.1.2 Step-Steer (Steady-State Response)

a) B-double

While the pulse steer compares the transient responses of the models over a broad range of lateral accelerations and frequencies, the step steer simulation was designed to compare the performance of the models in a steady turn at one specific level of lateral acceleration corresponding to 1° of steer angle.

The results for the B-double step steer simulations are compared in Fig. 7, which show the lateral acceleration and the roll angles of the three sprung masses. There is very close agreement across all three models and both variables, and the maximum difference in the steady state values are 3.0%, 2.0% and 2.1%, respectively, for roll angle, yaw rate (not shown) and lateral acceleration.

In a similar exercise Sayers and Riley (1996) compared various Yaw/Roll models with models of the same vehicles created in AUTOSIM (aka TruckSim). Except for a small difference in speed (96.5km/h compared with 100km/h in this paper) the step steer test specified in this paper is identical to the one used by Sayers and Riley (1996).

The models created by Sayers and Riley (1996) were designed to match their Yaw/Roll models, and some of the modelling assumptions they used were adopted in this study⁷. For the same reasons given in Sayers and Riley (1996), hysteresis has been excluded from the suspension spring models in this study, though the non-linear load-deflection characteristics of the suspension springs was retained; Sayers and Riley (1996) further simplified their models and made the suspension springs linear.

Figs 9(a) and 9(b) taken from Sayers and Riley (1996) show the lateral acceleration and roll angle responses of the sprung masses of a 5-axle tractor and semi-trailer to a 1° step steer input. The lateral acceleration transients for the tractor sprung-mass shown in Fig. 9(a) are nearly identical, except for the sharp peak that is centred on 1 s, greater for one of the modelling programs (which one is not known) by about 15%. When the transients have reduced in amplitude and the steady state turn conditions have been established, the AUTOSIM model is seen to predict larger sprung mass lateral acceleration by roughly 4.5%, which is reflected in prediction of larger sprung mass roll angles by about the same amount, as shown in Fig. 9(b).

When compared to the results presented in Sayers and Riley (1996), there is closer agreement between the models in this study under fewer modelling assumptions and with a more complicated vehicle.

Nevertheless, Sayers and Riley (1996) make the point that some simplifications had been made that introduce small errors. They conclude that the Yaw/Roll equations contain a subtle error related to the coupling that accounts for about 2% of the difference from the equations derived by AUTOSIM. This may additionally account for some of the differences in the outputs from this study.

b) Truck/Trailer

The same variables used to compare the B-double step steer responses were used for the truck/trailer. The lateral accelerations and roll angles are shown in Fig. 8. The agreement in the outputs between the modelling packages for the truck/trailer is not as close as for the B-double. Even though the small amplitude oscillations matched almost exactly, there is a small constant offset between the program outputs; Yaw/Roll and AUTOSIM consistently showing lower values for the variables considered.

While the tyre models may account for part of this discrepancy, there are other differences between models such as the detail and treatment of the pin coupling in Yaw/Roll. In addition, the error in the hitch equations referred to in Sayers and Riley (1996), discussed above, may also account for some the observed difference.

The differences in the steady state values for roll angle are 2.8% for the truck and 8.5% for the trailer, and for yaw rate and lateral acceleration the differences are, respectively, 7.0%, and 7.5%. Given there is uncertainty of a few percent about the output from Yaw/Roll this match is still considered to be acceptable.

4.2 Closed-Loop Control

4.2.1 SAE Lane Change

a) B-double

The simulations presented in this part of the paper deal with the manoeuvres that require use of the steer controllers. The steer controller can be considered a mathematical equivalent of a driver trained to perform a specific task. Just as there are differences in steering performance between drivers, there will also be subtle

⁷ The assumption of small pitch angles that is used in the Yaw/Roll program was also used in the AUTOSIM models created by TERNZ but not in the ADAMS models created by RTDynamics.

differences between the performances of steer controllers created by different modellers. The task of the steer controller in the SAE lane change is to follow a prescribed path.

In the SAE lane change test procedure, the path of the vehicle, measured at the centre of the steer axle, is required to not deviate by more than $\pm 150\text{mm}$ from a precisely prescribed path over a distance of 61 m. The paths followed by the B-double models are shown in Fig. 10(a) together with the path specified in SAE J2179 (Society of Automotive Engineers, 1993). All three models are within the specified path tolerance, as revealed in second plot shown in Fig 10(a), showing the lateral error in position is within a range of $\pm 15\text{mm}$ for Yaw/Roll and ADAMS, and within $\pm 30\text{mm}$ for the model created in AUTOSIM. This level of precision in path following performance would be challenging for a driver to consistently achieve in practice.

The influence of the slightly different steer paths is compared in Figs 10(b) in the yaw rates and lateral accelerations for the three models. The yaw rates show the responses are similar; the most significant differences occurring on the peaks: of the order of 10 to 15% on the major peaks and much larger on some of the smaller peaks. Also evident is some high frequency in the Yaw/Roll and AUTOSIM responses due to activity of their respective steer controllers: seen more clearly in the lateral acceleration trace which is the second plot in Fig. 10(b).

b) Truck/Trailer

The two variables used to compare the B-double responses in the lane change for the truck/trailer are shown in Fig. 11. The first plot in Fig. 11 shows that all three models meet the path requirements. There is less error in the steer controllers for the ADAMS and Yaw/Roll models in this task than AUTOSIM.

It was also observed that the errors in lateral position are larger for the truck/trailer models than for the B-double models, suggesting the prescribed path task for this vehicle is more challenging for the steer controllers. Furthermore, it was also noted that the steer inputs for the truck/trailer were different to those for the B-double; the truck/trailer controller is required to be more active for the duration of the run, there are a larger number of path error reversals, and the nature of the error is more erratic indicating a dynamically more active vehicle. This is seen very clearly in Fig. 12, which compares the steer angle inputs from the steer controller to the ADAMS truck/trailer and B-double models in the same manoeuvre. The number and size of the steering reversals is much greater for the truck/trailer, indicating there is a corresponding increase in "workload" imposed on the steer controller.

A driver in a real-world lane change task matching this one may be exposed to a similar increase in workload. The contrasting examples of the B-double and the truck/trailer illustrated suggests that the number and size of the steering reversals is likely to be correlated with configuration and class of vehicle and may provide a useful vehicle-specific measure of driver workload or some other vehicle-performance related issue. Steering reversals have been used in driver behaviour studies as a measure of driver workload and performance, including driver fatigue (McLean and Hoffmann, 1975).

The increase in steering activity leads to a more dynamic response, as evidenced in the yaw rate plot in Fig. 11. The agreement in the outputs from the models is clearly affected by the now larger differences in the magnitude and sequencing of steer input between the controllers.

These results emphasise the central role of the steer controller in closed-loop prescribed path tasks, suggesting the path tolerance specification for the SAE lane change may be too large for computer-based modelling. This also suggests that the current path tolerance may also be too large for physical testing of heavy vehicles leading to significant variability between runs (discussed later in the paper).

4.2.2 Low-Speed Turn

The vehicle factors influencing responses in a low-speed turn are different to those in an evasive manoeuvre such as the lane change. In a low-speed turn the vehicle mechanical properties such as mass, suspension and tyre characteristics play only a very minor role because the (inertia) forces acting on the masses and on the suspensions due to motion are no longer large enough to be significant. In a low-speed turn wheelbases and the location of couplings play a significant role in determining offtracking performance.

For the above reasons, the task of the steer controller in a near zero-speed path following task is different to the lane change. As such, the steer controller – although still required to follow a path – is now dealing with a system possessing essentially no dynamics, and its characteristics would be required to be adjusted or adapted accordingly.

a) B-double

The paths of the steer, drive and trailer axles from the B-double low-speed turn simulations are compared in Fig. 13 and it to be of the order of a few percent.

b) Truck/Trailer

The same variables are compared for the truck/trailer in Fig. 14 showing a similar level of agreement between the models.

4.3 Performance Measures

4.3.1 Comparisons

Direct comparison of the traces from the simulations presented in the previous Sections of this paper provides a very detailed view of how well the responses match across a large number of variables in four separate and distinct simulations. However, in order to help the end-user to determine whether or not to consider the performance of a vehicle as acceptable, the outputs from the simulations are usually condensed into one or several numeric values corresponding to one or a number of different performance measures. Several performance measures from Prem et al (2001a) are normally calculated from the outputs for the same or similar manoeuvres described in this paper, and the B-double and truck/trailer models have been compared in this way in Tables 1 and 2 using the safety-related performance measures that are listed.

Multiple entries for rearward amplification and load transfer ratio appear in Tables 1 and 2. These entries correspond to calculations based on the outputs from the SAE lane change and pulse steer simulations. Rearward amplification and load transfer ratio can be calculated not just from the SAE lane change but also for any manoeuvre involving an abrupt steer input, though the results and threshold levels of acceptability may not be directly comparable with those from the SAE lane change.

The values in Tables 1 and 2 indicate percentage differences as low as 1.9% and generally well below about 7% for the majority of the measures from the simulations, except for the truck/trailer SAE lane change simulations, which shows a maximum difference between the models in excess of 30% on two of the measures. This is a direct result of the greater deviation from the desired path taken by one of the models and suggests that to achieve acceptable levels of agreement, equal to or better than say 10%, a path tolerance smaller than $\pm 150\text{mm}$ will be required.

For the B-double simulations a deviation of less than $\pm 30\text{mm}$ led to differences between models in the SAE lane change of 6.1%, 3.3% and 7.3%, respectively, for the high-speed transient offtracking, rearward amplification and load transfer ratio performance measures.

Using the above as a guide, it is recommended that for computer-based models proper execution of the SAE lane change requires that the front axle of the vehicle does not deviate by more than $\pm 30\text{mm}$ from the desired path defined by the test course.

This conclusion is equally applicable to physical testing and suggests the current tolerance on the path for proper execution of the test manoeuvre may lead to significant variability in test results. It is recognised that a path tolerance of $\pm 150\text{mm}$ may be difficult to achieve in practice, and a tolerance one-fifth this value will be more difficult to achieve consistently, if at all. A statistical approach may need to be considered.

4.3.2 Concluding Comments

The pulse and step steer simulations provide a direct measure of vehicle responses to precisely defined steer inputs in the absence of a steer controller. Tables 1 and 2 show that there was closer agreement between the performance measures for the pulse steer simulations than for the SAE lane change.

Given the above, the pulse and step steer tests could be considered as viable alternatives to the SAE lane change, having the benefit of avoiding introducing additional differences from the steer controllers that are necessary to perform the prescribed path lane change. The tests would provide very precise and repeatable steer inputs to the vehicle, and excite them over a range of frequencies, rather than a single frequency (0.4Hz) that may coincide with a lightly damped resonant response in a particular vehicle.

Test methods do exist for evaluating the road holding ability of passenger cars that are based on pulse, step, random and sinusoidal steer inputs (International Standards Organisation, 1988), and similar tests involving pseudo-random and pulse steer inputs recently have been proposed for heavy vehicles (International Standards Organisation, 2000).

5. SOURCES OF VARIATION AND SOME PRACTICAL CONSIDERATIONS

5.1 Sources of Variation

5.1.1 Input Datasets

One of the key inputs to any modelling work is the dataset describing the mechanical properties of each component. It is not a trivial task to collate, interpret and make use of the input data describing each and every part in the model. In some cases these data will not be available and will either need to be estimated or physically measured.

If the datasets are either incomplete or the input data are inaccurate, the predictions from the models will either be inaccurate or in the worst possible case the predictions will be meaningless. The level of accuracy is an important issue and some parameters will have a much greater influence on the accuracy of the models than others. The key parameters should be identified either from previous similar studies or through parametric studies, and every attempt made to obtain accurate input data for all of the important components.

5.1.2 Component Models

The component models are another source of variation that can lead to differences between the outputs from simulations. In the study of vehicle dynamics, the tyres and the suspensions are key components having a significant influence on vehicle responses. It is therefore essential to include all of the necessary properties in the component models.

Piecewise linear interpretations versus continuous functions for tyres, for example, as illustrated in this paper, can lead to differences in the estimation of tyre forces – though this can be addressed by simply using more data and better defining the component model characteristics. However, inappropriate simplifications in models, such as the use of linear characteristics for tyres and suspensions in manoeuvres where the components will be operating in the non-linear regions will lead to errors. Further, using the component models outside of the range of the measured data on which they are based will also lead to inaccurate outputs from simulations.

5.1.3 Vehicle Models

The number of components that are used to define the model, and the way the various component models are connected to each other, referred to as the topology, will define the overall vehicle model. The level of detail in the model and number of separate component models required will be dictated by the problem at hand; highly detailed models do not necessarily produce more accurate results, and a high level of detail may not be necessary. Further, if a high level of detail has been included it will require more input data and the computer simulation will take longer to run.

In the study described in this paper the topologies of the various models were not defined, each Consultant determined model detail by referring to the input dataset and the descriptions provided in the accompanying text.

5.1.4 Test Manoeuvres

If meaningful comparisons of the outputs from simulations are to be made, and vehicle performances compared, then the test manoeuvres that are simulated must be identical. Small differences in the specification of a test (size of steer input, test speed or path followed) can lead to significant differences in outputs from simulations. In the study described in this paper each Consultant used the same precisely defined test manoeuvres.

5.1.5 Modelling Packages

The modelling package (software) should not lead to any significant differences in simulation outputs nor should it introduce any errors. The software should be robust, free from coding errors and thoroughly tested, preferably in a wide range of applications or at least in the application in which it will be used.

As discussed in this paper, it was assumed that the three modelling packages used in this study would not need to be verified or validated against experimental tests because they have been verified in other studies. For the models and simulations carried out, excellent agreement in the outputs from the modelling packages was demonstrated, except in the high-speed prescribed path simulations where the path deviations led to poor agreement between models.

5.1.6 Post Processing

The end-user may require the outputs from the simulations to be condensed into one or several numerics corresponding to one or a number of different performance measures. In order to prevent errors from occurring when computing a particular performance measure, the same outputs from the simulations should be used and the same (or similar) calculations performed. This requires both a clear description of the outputs that will be used in the calculations and a clear description of either the actual calculation or the performance measure. If these considerations have not been addressed adequately there is chance of introducing errors that may range in significance from small to substantial, the latter occurring in the worst possible case.

5.2 Practical Considerations

While the modelling packages may be free from error and perform "as advertised", the predictions from the models created by a particular Service Provider will strongly depend on the Provider's ability to create a realistic and accurate model. The model will need to contain all the necessary features that will ensure accurate simulations and robust predictions, as well as conduct simulations in the standard agreed test manoeuvres to the specified tolerances.

The final test of whether simulation outputs are adequate can be determined when the predictions from the model are compared to actual measurements derived from physical testing. When computer-based modelling has been used to support a new design, final "sign-off" should only occur when the computer-based performance predictions have been confirmed in practice.

6. SUMMARY

Computer-based models of two heavy vehicles were created by two Consultants using three separate computer-based modelling packages; ADAMS, UMTRI's Yaw/Roll program and AUTOSIM. Comprehensive input datasets were developed for a non-descript B-double and a non-descript truck/trailer. The same datasets were supplied to each Consultant. Identical simulations were performed using the same test manoeuvres comprising a pulse steer, step steer, standard SAE lane change and a low-speed 90° turn.

Time histories of a wide range of variables from the simulations were compared as well as numeric values from a selection of performance measures. For the more stable of the two vehicle models, the B-double, the time histories from the pulse steer and step steer simulations were in close agreement between all three modelling packages. Agreement in the outputs from the simulations in all manoeuvres was generally better than 8% for the performance measures considered. These were marginally influenced by the characteristics of the steer controller in the SAE lane change though agreement was still generally better than 8%.

The truck/trailer model, representing a less stable and dynamically more active vehicle compared to the B-double, produced larger but acceptable amounts of variation in the pulse and step steer simulations and low-speed 90° turn. However, in the SAE lane change the differences between the models were much too large as a result of the greater deviations in the path followed. To achieve acceptable agreement in the SAE lane change between models, a deviation from the desired path not greater than $\pm 30\text{mm}$ is required and is recommended.

Simulations that provide a direct measure of only the vehicle responses to precisely defined steer inputs generally lead to consistent results than simulations that require steer controllers and closed loop path following. When there

is a choice, open loop manoeuvres should be selected in preference to closed loop manoeuvres that require the use of steer controllers.

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TABLES & FIGURES

Table 1 – Performance Summary for the B-double

Test ¹	Performance Measure	ADAMS	Yaw/Roll	AUTOSIM	Max % Diff
PS ²	Yaw Damping Coefficient ³	0.545	0.550	0.559	2.6
PS	Rearward Amplification (-) ⁴	0.873	0.890	0.887	1.9
PS	Load Transfer Ratio (-)	0.773	0.729	0.729	6.0
SAE	High-Speed Transient Offtracking (mm)	347	337	330	5.2
SAE	Rearward Amplification (-)	1.304	1.266	1.300	3.0
SAE	Load Transfer Ratio (-)	0.380	0.369	0.354	7.3
LST	Low-Speed Offtracking (mm)	5518	5658	5513	2.6

Table 2 – Performance Summary for the truck/trailer

Test ¹	Performance Measure	ADAMS	Yaw/Roll	AUTOSIM	Max % Diff
PS ²	Yaw Damping Coefficient ³	0.238	0.270	0.263	13.4
PS	Rearward Amplification (-) ⁴	1.269	1.163	1.230	9.1
PS	Load Transfer Ratio (-)	0.982	0.961	0.931	5.5
SAE	High-Speed Transient Offtracking (mm)	840	635	663	32.3
SAE	Rearward Amplification (-)	2.252	1.895	1.889	19.2
SAE	Load Transfer Ratio (-)	0.943	0.923	0.707	33.4
LST	Low-Speed Offtracking (mm)	2117	2174	2190	3.4

FOOTNOTES:

- 1) PS denotes Pulse Steer, SS denotes Step Steer, SAE denotes the SAE lane change, and LST denotes the Low-Speed 90° Turn.
- 2) The pulse steer used in this study is more aggressive than the one specified in Prem et al (2001a).
- 3) Calculation based on the amplitude of the second and fourth peaks of the yaw rate response.
- 4) Rearward amplification for the pulse steer is based on the ratio of sprung mass lateral accelerations.



Fig. 1(a) Computer-based model of the reference B-double (RTDynamics' model shown).

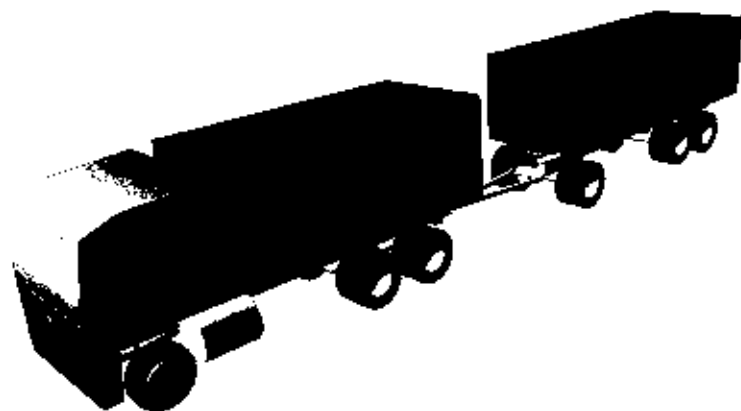


Fig. 1(b) Computer-based model of the reference truck/trailer (RTDynamics' model shown).

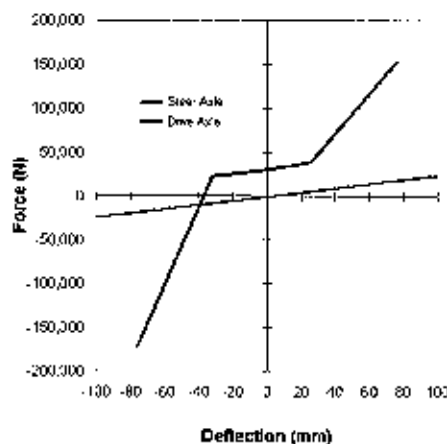


Fig. 2(a) Force/deflection relationship for the truck and prime-mover springs (adapted from Ervin and Guy, 1986).

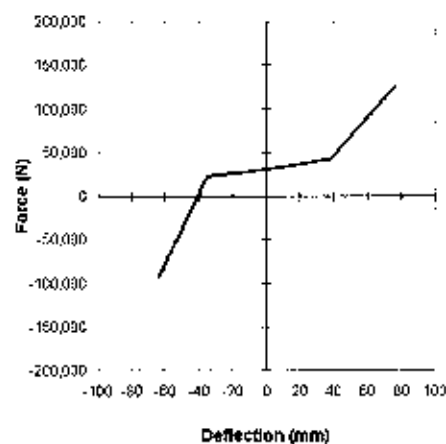


Fig. 2(b) Force/deflection relationship for the trailer springs (adapted from Ervin and Guy, 1986).

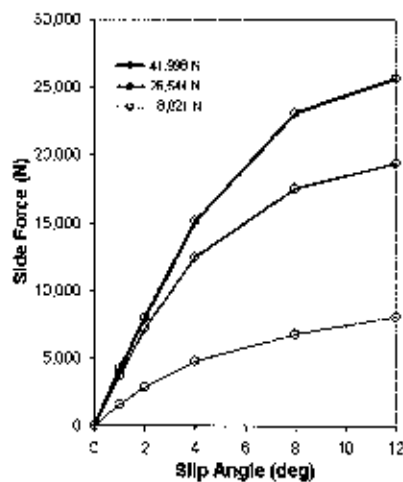


Fig. 3(a) Side force characteristics for a Michelin XZA 11R22.5 truck tyre (from Ervin and Guy, 1986).

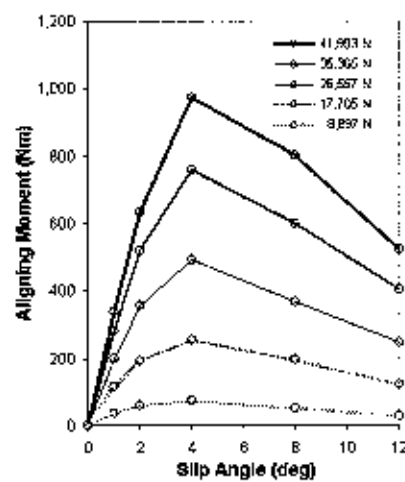


Fig. 3(b) Aligning moment characteristics for a Michelin XZA 11R22.5 truck tyre (from El-Gindy and Kenis, 1998).

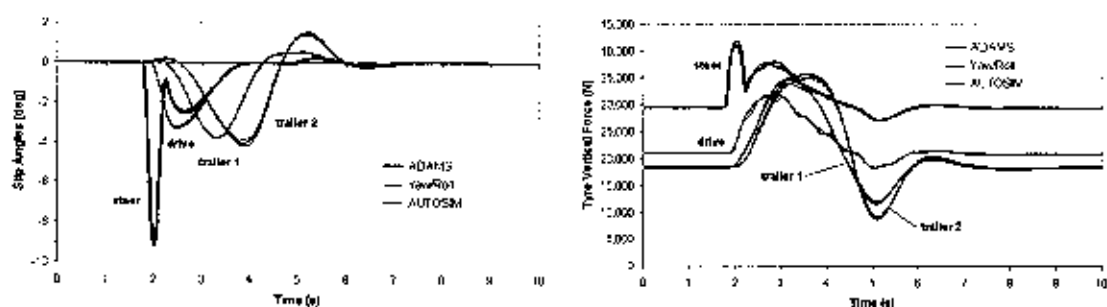


Fig. 4(a) Tyre slip angles and vertical forces from the B-double pulse steer simulations.

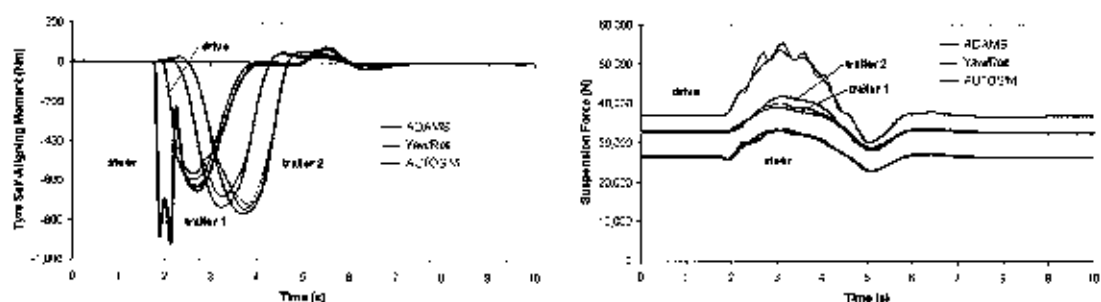


Fig. 4(b) Tyre self-aligning moment and suspension forces from the B-double pulse steer simulations.

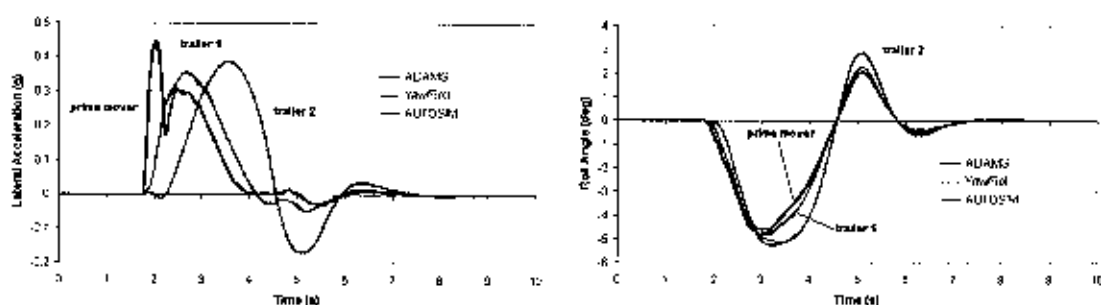


Fig. 4(c) Lateral acceleration and roll angle from the B-double pulse steer simulations.

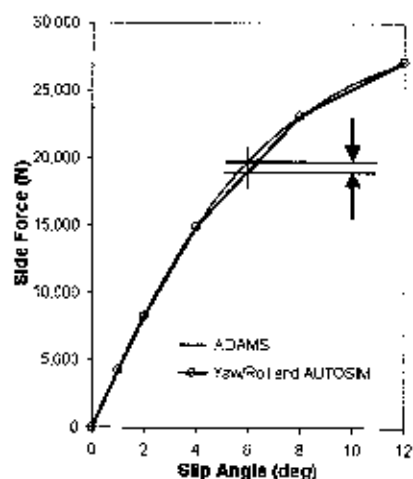


Fig. 5(a) Influence of tyre model on side

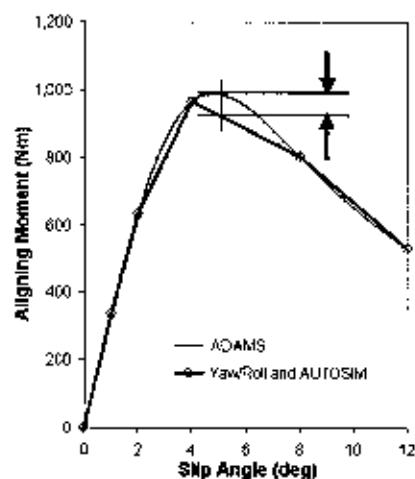


Fig. 5(b) Influence of tyre model on aligning

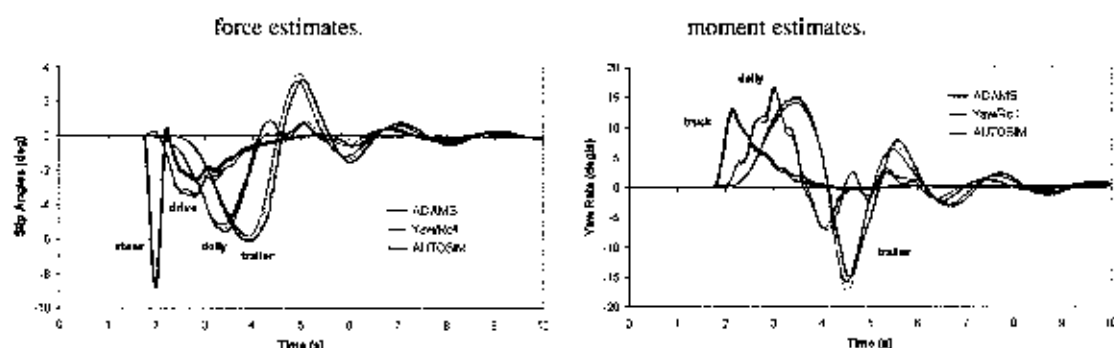


Fig. 6(a) Slip angles and yaw rate from the truck/trailer pulse steer simulations.

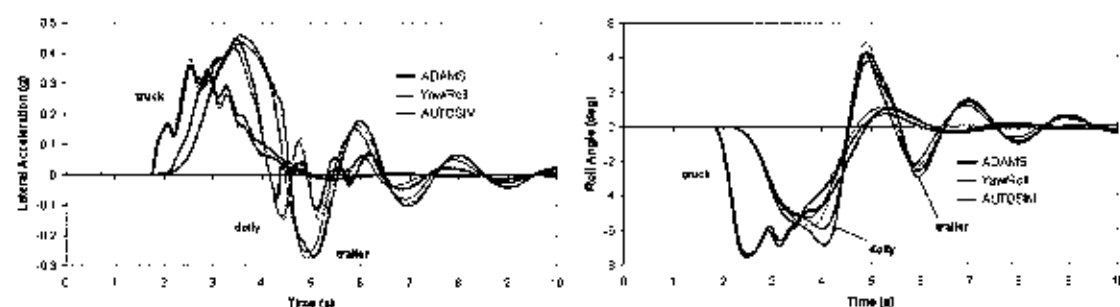


Fig. 6(b) Lateral acceleration and roll angle from the truck/trailer pulse steer simulations.

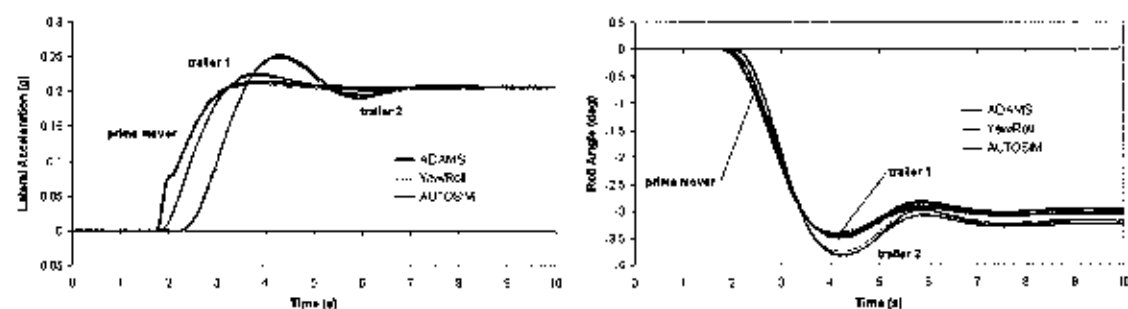


Fig. 7 Lateral acceleration and roll angle from the B-double step steer simulations.

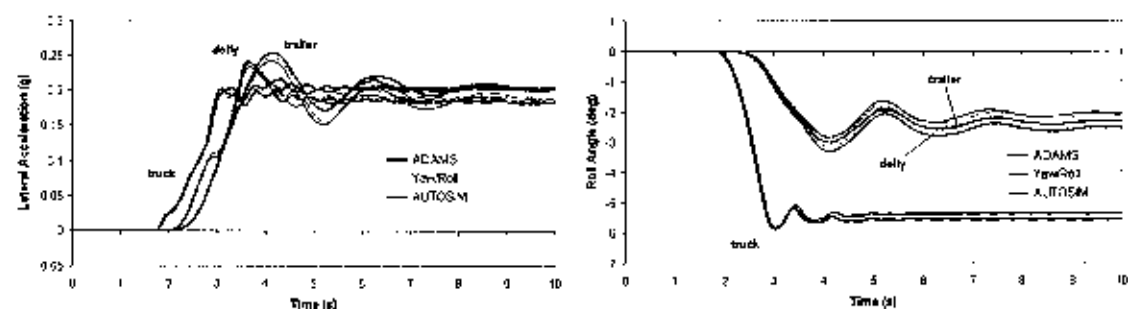


Fig. 8 Lateral acceleration and roll angle from the truck/trailer step steer simulations.

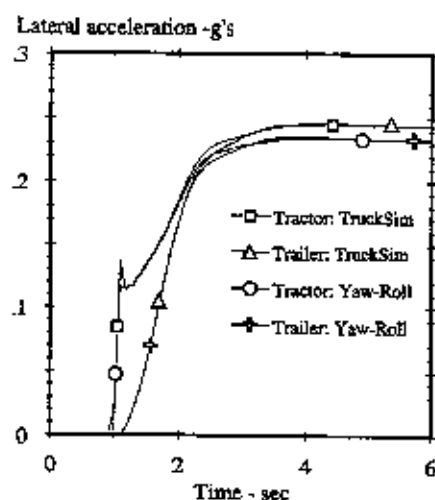


Fig. 9(a) Lateral acceleration response (from Sayers and Riley, 1996).

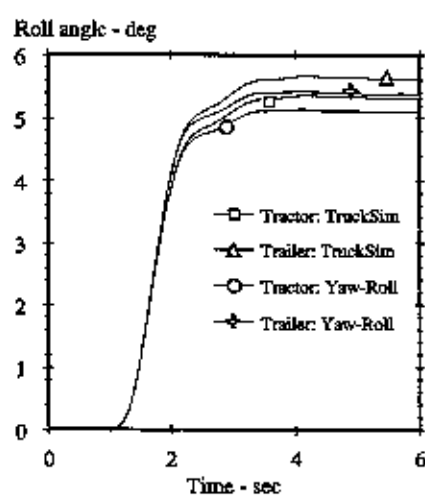


Fig. 9(b) Roll angle response (from Sayers and Riley, 1996).

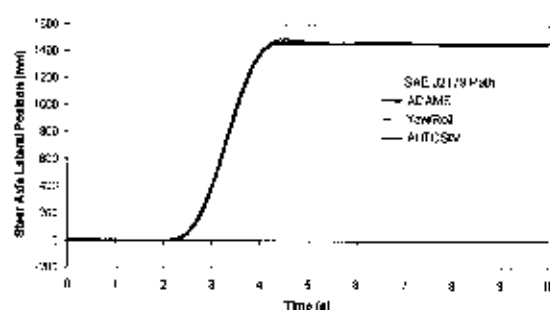


Fig. 10(a) Steer paths and steer axle lateral position error for the B-double in the SAE lane change.

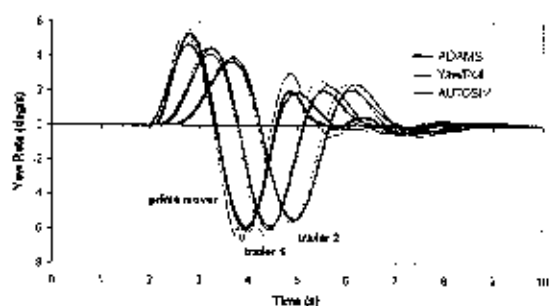
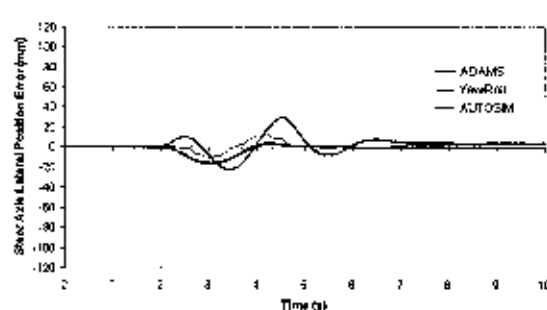


Fig. 10(b) Yaw rate and lateral acceleration for the B-double in the SAE lane change.

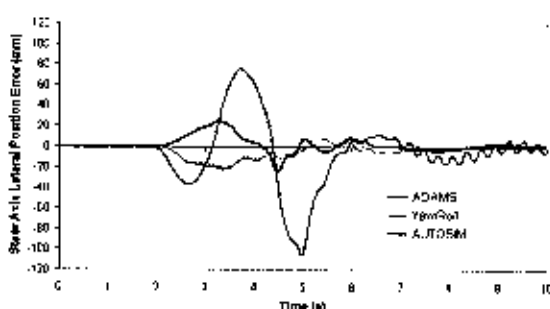
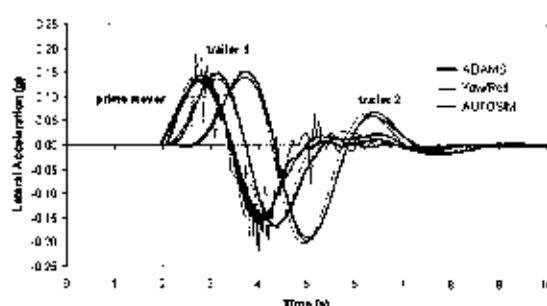
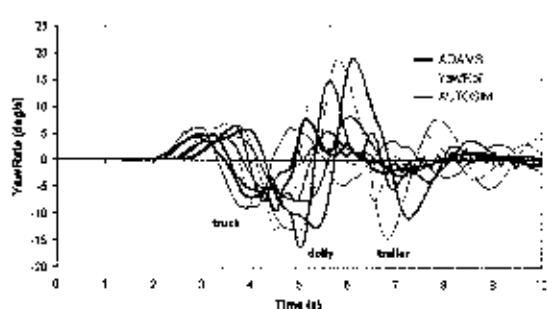


Fig. 11 Steer axle lateral position error and yaw rate for the truck/trailer in the SAE lane change.



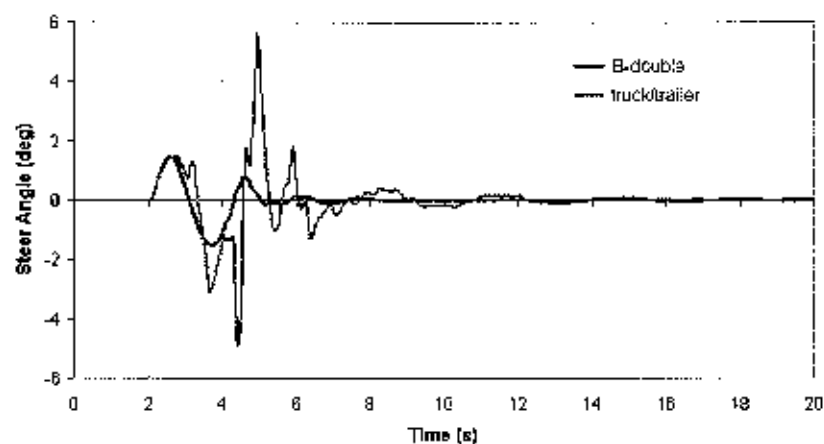


Fig. 12 B-double and the truck/trailer steer angle inputs in the SAB lane change simulations (RTDynamics' ADAMS models).

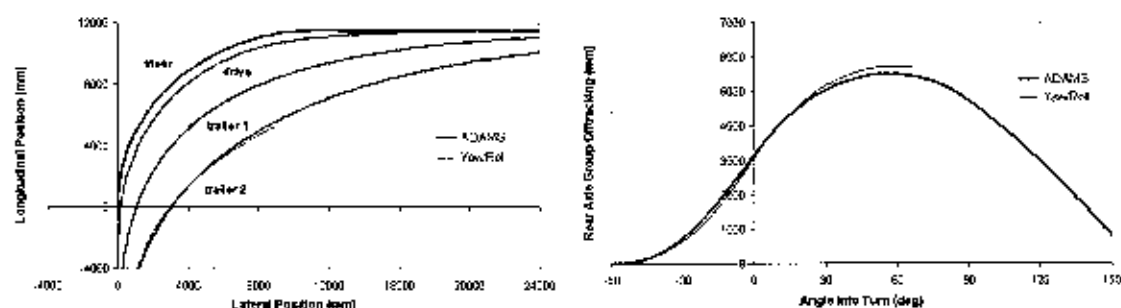


Fig. 13 Trajectories and offtracking for the B-double in the low-speed turn.

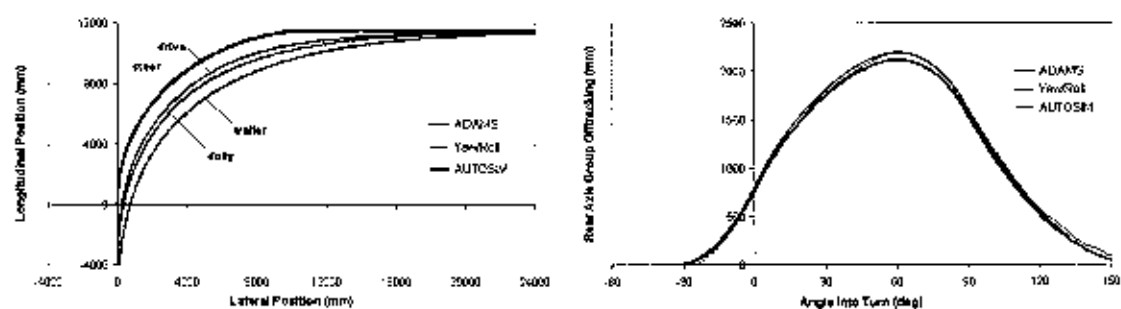


Fig. 14 Trajectories and offtracking for the truck/trailer in the low-speed turn.

PERFORMANCE EVALUATION OF THE TRACKAXLETM STEERABLE AXLE SYSTEM

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ABSTRACT

The use of steerable axles on heavy-vehicles is both a means of improving vehicle performance and a method of achieving higher productivity and improved access to the road network. The development in Australia and New Zealand of performance-based standards for the regulation of heavy vehicles provides one method of overcoming the prescriptive regulatory impediments to productivity and network access by directly addressing performance issues. Safety and infrastructure related performance measures were chosen from the set developed and proposed for use in Australia and applied to a new and novel steerable axle system known as "Trackaxle". The Trackaxle system, comprising a steerable axle group, allows increased trailer length by reducing low-speed offtracking. It utilises a rear, self-steering carriage rotating about its mid-point with respect to the trailer chassis. The front and rear of its three axles can oscillate with respect to the carriage in response to the angle developed between the body of the carriage and the trailer chassis. For on-road stability, the self-steer characteristic has two override devices: an automatic highway lock for straight ahead travel, and a spring loaded limiter system which controls the build-up of tyre side forces to limit the degree of rear-steer to a proportion of the articulation of the tractor at the fifth wheel. In the evaluation, full-vehicle computer-based models were created of a representative baseline prime-mover and semi-trailer combination, and another of a near identical vehicle featuring a longer semi-trailer equipped with the Trackaxle system. When compared to the baseline vehicle, the performance of Trackaxle was estimated to be better in several areas, including low-speed offtracking and horizontal tyre forces in a turn: the horizontal tyre forces were found to be low, suggesting a potential for reductions in pavement damage and wear. The evaluation of the prototype Trackaxle system indicates a range of potential benefits in the areas of safety, infrastructure and productivity, and illustrates a practical application of performance-based standards.

1. INTRODUCTION

The development in Australia and New Zealand of performance-based standards for heavy vehicle regulation provides one method of overcoming the current prescriptive regulatory impediments to innovation, productivity and network access by directly addressing safety and infrastructure related performance issues. By addressing performance issues rather than imposing constraints that are strongly linked to dimensions and mass limits, specific performance outcomes can be achieved.

One area of innovation with the potential to improve vehicle performance and productivity involves the use of steerable axles. While the focus of this paper is on steerable trailer axles, the use of steerable axles could also include drive axle groups.

The purpose of this paper is twofold. It presents both a technical description of the steerable trailer axle system known as "Trackaxle", and an evaluation of its performance using a performance-based standards approach. The evaluation draws on the most recent work and recommendations from a major initiative by Austroads¹ and the National Road Transport Commission (NRTC) in Australia in the area of performance based standards (National Road Transport Commission, 2002).

2. LOW-SPEED OFFTRACKING AND STEERABLE AXLES

2.1 Conventional Articulated Vehicles

It is known that for conventional articulated vehicles the amount of offtracking that occurs in a low-speed turn depends upon truck/prime-mover and trailer wheelbases, hitch locations and the lengths between articulation

¹ Austroads is the association of Australian and New Zealand road transport and traffic authorities.

points. The dependence varies with the squares of the lengths, and the longest trailer wheelbase is a critical parameter in determining the level of offtracking. Including additional articulation points to reduce the lengths of the longest wheelbases, and making trailer lengths equal in multi-unit combinations can achieve improvements in offtracking (Fancher et al, 1989).

2.2 Steerable trailer Axles

Another means of reducing the offtracking behaviour of articulated vehicles in low-speed turns involves the use of steerable trailer axles. Steerable trailer axles allow long-length trailers to be optimised to operate within the geometric design constraints of intersections and the road network. While increased length trailers with greater cubic capacity and improved road access have obvious and direct productivity outcomes, the less obvious benefits of steerable trailer axles are reported to be reduction in tyre scrub and tyre wear, as well as pavement wear. These in turn are claimed to lead to lower fuel consumption and reduced exhaust emissions (Gayat, 2000a).

3. "TRACKAXLE" AND ITS EVALUATION

3.1 Background

Gayat Pty Ltd has developed a steerable axle group that is designed to follow the swept path of the prime mover or other preceding vehicle wheel group. The system is referred to as "Trackaxle"², and derives its steering characteristics from a combination of force (linkage) steered and free steered sub-systems. For operation at highway speeds a mechanism locks the axles in the straight-ahead position.

A prototype semi-trailer featuring the Trackaxle system has been manufactured and field trials have been conducted. The prototype unit was compared to a similar conventional articulated vehicle in the field trials, and both a reduced low-speed offtracking and an acceptable tracking ability at highway speed have been demonstrated. To further test the performance of Trackaxle in everyday use and to assess the behaviour and responses of other road users with respect to areas of potential conflict, as described in Prem and Ramsay (2001b), in-traffic pilot trials are now underway.

As lead-up to the pilot trials, a detailed evaluation was carried out of the Trackaxle performance. The objective was to determine how the system would perform in a wide range of operating conditions and in standard manoeuvres. In particular, it was considered essential to assess both the safety-related performance as well as impacts on the infrastructure. This objective was met by using a combination of performance-based standards – drawing on the most recent work on performance-based standards by Austroads and the NRTC – and computer-based modelling and simulation.

It is important to note that the work described in this paper does not address any issues related to the mechanical strength of components, their durability and longevity, or their suitability to a particular application.

3.2 Methodology

Computer-based models were created of a representative baseline 19m long prime mover and semi-trailer combination, and another of a near identical vehicle featuring a longer semi-trailer equipped with the Trackaxle system. Safety and infrastructure related performance measures were chosen from the proposed final set of performance-based standards developed by Austroads and the NRTC and applied to the two vehicles. These standards are fully described in Prem et al (2001) and Prem et al (2002).

Simulations were performed with the two computer-based models in manoeuvres corresponding to each of the selected performance measures.

Outputs from the simulations were processed and performance was evaluated both in relative terms, by comparing the performance of Trackaxle to that of the baseline vehicle, and in absolute terms by comparing the performance of Trackaxle against the performance requirements of the selected standards.

In the analysis the performance of Trackaxle was considered acceptable if it met the proposed NRTC/Austroads standard. If Trackaxle met both the Austroads/NRTC standard and its performance exceeded that of the

² Gayat Pty Ltd has lodged a provisional specification for a patent (Gayat, 2000b).

benchmark vehicle then its performance was described as better. If its performance was less than the baseline vehicle then its performance was described as worse.

4. PERFORMANCE-BASED STANDARDS APPROACH

4.1 Considerations and Recent Developments

A performance based standards approach to the regulation of heavy vehicles involves specifying or identifying desired outcomes, setting objectives to achieve those desired outcomes, and establishing performance measures that promote those objectives (ARRB Transport Research Ltd., 2000).

A set of performance standards have been developed by Austroads and the NRTC that define performance requirements that should be met if a vehicle is to be considered both safe for operation on the road network and acceptable in terms of wear to the infrastructure. While it is desirable for vehicles that are acceptable in terms of safety and infrastructure performance to also be more productive than equivalent currently operating vehicles, improved productivity is not the fundamental purpose of performance-based standards. Therefore, when the safety and infrastructure related impacts have not been compromised acceptable performance is considered to have been achieved (National Road Transport Commission, 2002).

Under the current prescriptive regulations the vehicle featuring the Trackaxle system as described in this paper would exceed the 19m overall length limit for general access to the road network. On length alone it could be disqualified from receiving general access status. However, under a performance based standards regulatory regime, overall length – while important in some on-road situations – is not a prime determinant in all situations of whether a vehicle is acceptable or not, or whether access should be restricted to specific parts of the road network. The range of performance measures developed by Austroads and the NRTC are designed to test a vehicle's safety-related performance, to assess its ability to fit within the road network, and to determine its impact on the infrastructure.

4.2 Selection of Performance Measures

Of the recommended final set of twenty (20) performance measures from Prem et al (2002), fifteen (15) have been developed to a stage where they are considered to be useable and suitable for regulatory purposes. Of those fifteen (15), the following seven (7) were selected for this paper – these are described briefly in Section 7:

- i) Low-Speed Offtracking
- ii) Tail Swing
- iii) Static Rollover Threshold
- iv) Rearward Amplification
- v) Yaw Damping Coefficient
- vi) High Speed Transient Offtracking
- vii) Horizontal Tyre Forces

The above set of performance measures was chosen on the basis that they would show the greatest contrast between the Trackaxle system and the baseline vehicle considered. A number of performance measures were not considered in this paper. Analysis from a previous study estimated these would be identical or near identical for the two vehicles. The performance measures that were not considered included, for example, startability, gradeability, acceleration capability, frontal swing and steer tyre friction demand.

5. TECHNICAL DESCRIPTION OF TRACKAXLE

A general description of the Trackaxle system and details of its key technical features is presented in this section of the paper. Further details can be found in Gayat (2000a) and Gayat (2000b). In order to interpret the performance of the Trackaxle system it was necessary to first develop a basic understanding of how the system works.

5.1 General Description and Design Goals

Gayat (2000a) describes Trackaxle as:

"An improved trailer axle group, self-steering tracking system designed to closely follow the narrow swept path of a leading prime mover or other preceding vehicle wheel group with a limiting system to prevent excessive rear steer at highway speed."

Following are the six objectives the design needed to satisfy. These were set by the developer of Trackaxle:

- i) It must result in a major reduction in low-speed offtracking;
- ii) It must reduce tyre and pavement wear;
- iii) It must be stable on a highway;
- iv) It must have few moving parts;
- v) As much as possible, it must employ standard componentry; and
- vi) It must not diminish lateral stability.

Fig. 1 provides a rear view of the prototype vehicle partway through a turn. This shows clearly the most obvious feature of Trackaxle; a trailer axle group that steers relative to the trailer chassis. In field trials this feature has been demonstrated to reduce low-speed offtracking, as illustrated in the photo sequence shown in Fig. 2, showing the prototype vehicle negotiating an urban roundabout.

5.2 Technical Detail

5.2.1 Components

The key components and sub-assemblies of the Trackaxle system comprise:

- the subframe that supports the axle group;
- control (linkages) rods that limit or modify the steer of the subframe;
- mechanisms that steer the subframe via the front and rear axles of the axle group³;
- the mechanism that engages the tow-coupling turntable of the prime mover; and
- the steer control and steer limiter mechanism.

These components and sub-assemblies, which are described above, are identified in Fig 3.

5.2.2 Steer Response to Articulation Angle

As shown in Figs 3 and 4, a pair of linkages (rods) actuates the subframe (or bogie) steer mechanism allowing the subframe to rotate relative to the trailer chassis. The rods move in the fore-aft direction in response to the angle of articulation between the prime mover and the semi-trailer through the steer control and limiter mechanism (described later), which is shown in Fig. 6.

The basic relationship between subframe steer and articulation angle is shown in Fig. 5(a); this does not include the effect of the steer limiter.

In addition to steer rotation of the entire axle group through the subframe, the front and rear axles can be steered relative to the subframe by a pair of linkages (steer rods) through a connection to the centre axle. These steer rods are also shown in Fig. 4. The centre axle of the tri-axle group rotates with the subframe but is not steered relative to it. The basic relationship between subframe steer angle and the steer angle of the front and rear axles relative to the subframe is shown in Fig. 5(b).

As shown in Fig. 3, the subframe steer rods run the length of the semi-trailer between the centre of the trailer axle group and the tow-coupling turntable on the prime mover. The subframe steer rods are actuated by a mechanism controlling the angle of subframe on the trailer chassis. The mechanism has been designed to engage a standard turntable tow coupling (Gayat, 2000a).

5.2.3 Steer Control and Limiter

To cater for a wide range of on-road operating situations and conditions, the Trackaxle system also incorporates a number of design features that control and limit the degree of steering.

The steering actions transmitted along the subframe steer rods are generated by a slider/follower arrangement located at the tow coupling. These motions are modified under certain conditions by a steer limiter, comprising a pair of pre-loaded springs, with one located in each control rod. The slider follower arrangement and the steer limiter are identified in Fig. 6.

In order to keep the spring ends and the control rods a fixed length under certain conditions, the steer limiter springs are initially held in compression by a preload. When an axial compressive load exceeds the spring preload,

³ The links to the trailer body cause the front and rear axles to rotate in opposite directions, steering the subframe to follow the path of the prime-mover. This steering effect is opposite when the vehicle is reversing.

the length of the control rod decreases thereby modifying the simple steer relationship, shown in Fig. 5(a), between articulation angle and subframe angle. When active, the steer limiter permits a certain amount of self-steering, or free rotation to occur, in response to the yaw moment exerted on the subframe due to the build up of tyre side force. The force-displacement characteristics of the steer limiter are shown in Figs 7(a) and 7(b).

5.3.4 Highway Lock Pin

In addition to the above mechanisms for controlling trailer axle steer, a highway lock pin, as shown in Fig. 3, engages when the articulation angle is less than 2 degrees. When the highway lock pin is engaged the subframe and axle group are locked in the straight-ahead position, and the trailer behaves as if it were a conventional (long) trailer with a non-steered tri-axle group.

6. COMPUTER-BASED DYNAMIC MODELS AND SIMULATIONS

6.1 Vehicle Models

To evaluate Trackaxle the following two computer-based models were created using the ADAMS multi-body simulation software (Mechanical Dynamics, 2002) and the track modelling toolbox developed by RTDynamics (RTDynamics, 2002a; 2002b). The models include controllers for speed⁴ and steering as is required for many of the manoeuvres to be simulated:

- q A baseline design prime mover and semi-trailer using dimensions of the Austroads design prime mover and semi-trailer (Austroads, 1995), having a 5.4 m wheelbase prime mover and a 13.7 m overall length semi-trailer.
- q A prime mover and semi-trailer combination featuring the Trackaxle self-steering system as described in this paper, having the same 5.4 m wheelbase prime mover and a 15.81 m overall length semi-trailer.

The baseline vehicle has an overall length of 19.0 m, whereas the Trackaxle vehicle has an overall length of 21.08 m. Axle group loads for both vehicles were assumed to be identical and equal to the current legal limit for vehicles without road friendly suspensions. These loads are 6.0 t, 16.5 t and 20.0 t, on the steer axle and the drive and trailer axle groups, respectively, giving a gross mass of 42.5 t.

A generic, non-linear, full load-sharing air suspension model was used for both drive axle and trailer axle groups, the steer axle featured a non-linear multi-leaf steel spring suspension as proposed by Fancher et al (1980). 11R22.5 radial ply tyres were used on all wheels for both vehicles.

On the trailers a sprung mass centre-of-gravity (CG) height of 2.0 m was used, and it is an explicit assumption that this height was representative of worst case loading conditions.

The vehicle models described above are shown in Figs 8(a) and 8(b), respectively. Using the translucency features within ADAMS some of the detail incorporated in the Trackaxle system is revealed in Fig 8(b).

6.2 Simulations and Analysis

Both vehicle models were simulated using the test conditions and manoeuvres for the selected performance measures listed in Section 4.2 of this paper. Outputs from the simulations were post-processed and numeric values determined for each of the performance measures as defined in Prem et al (2002).

7. PERFORMANCE MEASURES AND RESULTS

This section of the paper briefly describes each of the performance measures considered, and presents the results of the simulations. The performance of the Trackaxle vehicle is compared with that of the baseline vehicle and the key results are presented in summary Tables 1 and 3. Table 2 is a summary of the performance standards used in this paper taken from Prem et al (2002).

⁴ Vehicle speed in the models is controlled through the application of tractive effort to the drive-axle tyres. Driveline characteristics in the model are generic and they are based on published engine torque-speed curves and gear reduction ratios.

7.1 Low-Speed Offtracking

For the simulation of low-speed offtracking the centre of the steer axle is required to follow a path comprising straight approaches to an 11.25 m radius 90° circular arc (Prem et al, 2002). This corresponds to the outside front wheel following a path of radius 12.5 m. A constant vehicle speed of 10 km/h is specified.

Trackaxle was found to have less low-speed offtracking than the baseline vehicle. For the baseline vehicle the maximum width of the swept path was 7.307 m, for Trackaxle it was 6.708 m. The swept path widths are less than the 7.4 m maximum for arterial road access, but neither is lower than the 5.0 m performance requirement to allow local roads access.

The low-speed offtracking simulations confirmed that for Trackaxle the path of the trailer axle group follows the swept path of the prime mover. This is consistent with the design goals of Trackaxle (Gayat, 2000a). The simulations also confirmed that for most of the turn less power was required by the Trackaxle prime mover to maintain the 10 km/h constant speed; approximately two-thirds of that required by the baseline vehicle towing the conventional trailer. This supports field observations and the claim of lower fuel consumption and exhaust emissions.

It is also worth noting that for the baseline (conventional) vehicle the location of the point of maximum offtracking through the turn occurs at a point that is near the centre of the rear axle group on the inside of the turn. For Trackaxle this point does not remain fixed at a single location relative to the vehicle, migrating forward from the rear axle during the initial stages of the turn and is well forward of the axle group at a point near the centre of the trailer body when offtracking reaches its maximum. These differences are illustrated in Fig. 9.

7.2 Tail Swing

For trailers with steerable axles, tail swing and low-speed steer behaviour will depend on the design goals set for the vehicle, how these goals are implemented in hardware, and how they are achieved in practice.

The Austroads/NRTC standards require tail swing to be evaluated on both approaches to the low-speed turn (entry and exit).

7.2.1 Turn Entry

On the entry to the turn, tail swing is greater for Trackaxle (0.325 m) than for the baseline vehicle (0.056 m). Fig. 10 illustrates and compares the difference in tracking between the baseline vehicle (upper illustration) and Trackaxle (lower illustration). It clearly shows how the baseline vehicle commences offtracking sooner, whereas Trackaxle initially holds a straighter path for a longer distance as it is steered relative to the trailer in response to articulation angle. This leads to a larger amount of tail swing, as recorded in the outputs of this performance measure.

Both vehicles are within the 0.35 m performance requirement specified by Austroads and the NRTC (Prem et al, 2002).

7.2.2 Turn Exit

Tail swing on the exit side of the turn is illustrated in Fig. 11, showing that the rear left corner of the Trackaxle trailer tracks approximately 0.275 m outside of the path of the front left corner of the prime mover when in the position shown. In practice, the driver would need to be aware of this overshoot and provide sufficient clearance to prevent the trailer from striking roadside objects that were successfully avoided by the prime mover.

For the baseline conventional vehicle (and other vehicles without steerable axles on trailers) tail swing on exit approach to the turn would not be an issue because the path of the rear left outside corner of the trailer approaches the path of the front left outside corner of the prime mover asymptotically.

Austroads and the NRTC (Prem et al, 2002) have specified the same performance level for tail swing on the exit, and both vehicles meet the 0.35 m performance requirement.

7.3 Static Rollover Threshold

To assess static rollover stability the vehicle is required to follow a circular path of constant radius (100 m). Test speed is slowly increased from 60 km/h until rollover occurs.

The first simulations under the prescribed conditions showed that the Trackaxle vehicle had a static rollover threshold of about 0.416 g. This was comparable to the value for the baseline vehicle, which was 0.417 g.

Gayat (2000a) claims that the increase in width of the tracking path of the trailer group when traversing curvatures and making sharp turns enhances rollover stability. This was attributed to the increase in "footprint" or effective track width when compared to a conventional tri-axle trailer group, as illustrated in Fig. 12.

To further explore this claim the simulations were repeated using a much smaller turn radius. This would increase the steer angle of the subframe and the corresponding width of the "footprint" with respect to the trailer.

On a turn radius of 20 m the static rollover threshold for the baseline and Trackaxle vehicles were estimated to be 0.410 g and 0.412 g, respectively. There was only a very small increase in rollover stability. The simulations showed the rollover stability of Trackaxle was expected to be similar to the baseline vehicle.

On close inspection of Fig. 12 it will be noted that while the effective track of the front axle on the left side of the Trackaxle system has increased, the effective track of the rear axle on the same side has decreased by a similar amount. The effective increase in track width, and corresponding increase in the roll stiffness on one axle, has been almost exactly offset by an opposite decrease in roll stiffness on another axle.

This result applies to a tri-axle group with identical roll stiffness on each axle and an idealised (perfect) load sharing airbag suspension. Increasing the roll stiffness on the front axle - either by increasing the auxiliary roll stiffness and/or the airbag spring rate - would better utilise the increase in effective track and this could lead to improvements in rollover stability. However, this would necessarily lead to greater front axle wheel loads. If implemented in Trackaxle it would need to be controlled and set at an appropriate level to ensure component loads did not exceed the manufacturer specified ratings. Hence, the potential for improving and enhancing rollover stability clearly exists, but the necessary design changes must take into account other aspects of the vehicle performance before being incorporated.

7.4 Rearward Amplification

Rearward Amplification (RA) is a measure of the tendency of the trailing unit(s) of an articulated vehicle to amplify any lateral acceleration experienced at the hauling unit. The performance requirement for RA set by Austroads and the NRTC, which assumes the SAE lane change is representative of a typical evasive manoeuvre, is defined in terms of the rollover stability of the critical, rearmost roll-coupled unit, as follows:

$$RA = 5.7SRT_{rcu} \quad (1)$$

where:

RA = rearward amplification measured in accord with recommended practice
SAE J2179 or ISO 14791 (-)

SRT_{rcu} = static rollover threshold of the rearmost roll-coupled unit (g)

For a vehicle with a static rollover threshold of 0.35g, Eqn (1) sets the performance level for RA at 2.0, a performance level that is based on research carried out in the USA (Fancher et al, 1989; Winkler et al, 1992)⁵. Further, according to Eqn (1), larger values of RA are deemed to be acceptable only if accompanied by a commensurate increase in the rollover stability of the rearmost roll-coupled unit(s). Both SAE J2179 and ISO 14791 are accepted methods of testing, being well established and proven procedures that are fully documented (Society of Automotive Engineers, 1993; International Organisation for Standardisation, 2000). Further details on development of the revised performance level for RA can be found in Prem et al (2002).

In practical terms, a threshold value of 2.0 for RA means that the lateral (sideways) acceleration at the CG of the rearmost unit in the combination should not exceed twice the lateral acceleration at the centre of the steer axle of the hauling unit. In the SAE lane change manoeuvre the steer axle lateral acceleration has a peak value of 0.15g. Therefore, for the example cited, an RA that is less than 2.0 would be considered to be acceptable.

⁵ During development of the Austroads/NRTC performance standards a set performance level of 2.0 for RA was also proposed (see Prem et al, 2001). However, the form described by Eqn (1) is preferred because it directly links the performance requirement to rollover stability, discussed fully in Prem et al (2002).

Both the baseline vehicle and Trackaxle were found to have acceptable rearward amplification and were well within the performance level value of 2.37 set by Eqn (1). Rearward amplification for Trackaxle, at 0.970, was lower than the value of 1.202 for the baseline vehicle. This is largely due to Trackaxle's longer trailer wheelbase, and there are no apparent adverse effects from Trackaxle.

7.5 Yaw Damping Coefficient

Yaw Damping Coefficient (YDC) quantifies how quickly oscillations of the last trailer take to reduce in amplitude, i.e. settle, after the application of a short duration steer input at the hauling unit. Vehicles that take a long time to settle increase the driver's workload and represent a higher safety risk to other road users. Under the Austroads/NRTC performance standards (Prem et al, 2002) YDC is required not to be less than 0.15 to be considered acceptable.

YDC for both vehicles met the performance requirement by a factor greater than 3; for the baseline vehicle it was 0.533, and for Trackaxle it was lower at 0.487. As shown in Table 1, the steer limiter was active for this manoeuvre, i.e. the highway lock was not engaged. While yaw damping is known to increase with trailer wheelbase (Prem et al. 2002), the increase in yaw damping with wheelbase is offset by the decrease in yaw damping due to Trackaxle. Consequently, with the highway lock engaged, yaw damping for Trackaxle would be expected to be greater than for the baseline vehicle.

7.6 High-Speed Transient Offtracking

High Speed Transient Offtracking (HSTO) measures how far the rear of the vehicle tracks outside the path taken by the hauling unit during the SAE lane change manoeuvre. The performance standard for HSTO requires that the centre of the rear of the trailer remain within 0.8 m of the path taken by the centre of the steer axle (Prem et al, 2001; Prem et al. 2002).

HSTO for both vehicles was found to be less than the specified 0.8 m performance requirement; the baseline vehicle having a lower value (0.201 m) than Trackaxle (0.365 m). With a longer trailer and the steer limiter active, there would be some steering of the trailer axle group leading to the increase in offtracking predicted by the simulations.

7.7 Horizontal Tyre Forces

This performance measure quantifies the influence of horizontal tyre forces on remaining pavement life. The same manoeuvre used for low-speed offtracking, described in Section 7.1, is also used for this measure. The performance measure can be applied to single and multi-axle groups as well as both driven and free-rolling wheels.

7.7.1 Lateral and Vertical Forces

Typical examples contrasting the key differences in tyre horizontal forces during the low-speed turn manoeuvre between the baseline and Trackaxle vehicles are shown in Fig. 13(a). These show the peak tyre side forces from Trackaxle to be more than 10 times lower than those for the baseline vehicle, and for the entire turn the forces are no greater than 1,450 N. This is a result of the Trackaxle system, achieving a balance between steer of the subframe in response to articulation angle, steer of the front and rear axles relative to the subframe, and characteristics of the steer limiter, which allows the subframe to self-steer along a path that reduces tyre side forces.

Prem and Potter (1999) found that on single axle and multi-axle groups the vertical tyre forces are modified by the horizontal side forces in a low-speed turn. The corresponding vertical forces for the two vehicles executing the low-speed turn are provided in Fig. 13(b), which shows that for Trackaxle the small change in tyre side force leads to a similarly small change in vertical force. For the baseline vehicle, however, the tyre side force imposes a roll moment on the axle resulting in a local increase in the vertical force on one side of the axle and a corresponding local decrease in vertical force on the other side, as shown in Fig. 13(b). In a low-speed turn on multi-axle group systems (two or more axles) there is a net increase in vertical load on tyres that are either at the front of the axle group and on the inside of the turn or at the rear of the axle group and on the outside of the turn, as discussed further in Prem and Potter (1999).

7.7.2 Tractive Forces

In a constant speed turn a reduction in tyre side forces would be reflected in a decrease in vehicle drag and a reduction in the tractive effort required to maintain a constant speed. The simulations showed that for most of the low-speed turn manoeuvre the tractive effort for Trackaxle was about 60% that of the baseline vehicle, and at times as low as 40%.

7.7.3 Pavement Damage (Wear)

The relative wear concept and the method described in Prem et al (2002), which is based on Prem and Potter (1999), was used to provide an indication of pavement wear due to horizontal forces, as required by this performance measure. The relative wear concept compares the estimated pavement damage (or wear) due to one vehicle (in this example Trackaxle) with that due to a representative reference vehicle; the Austroads design prime mover and semi-trailer.

Results for analysis of the horizontal forces, which are fully detailed in Prem and Ramsay (2001a), suggests that the Trackaxle tri-axle group, as described herein, will be between 160,000 and 190,000 times less damaging to the pavement than a conventional tri-axle group. It is important to note that the lower damage values for Trackaxle is due to the small side forces exerted on the pavement, as shown in Fig. 13(a), and on the use of a pavement surface layer damage model that is power law based involving horizontal tensile strain raised to the 5th power (Prem and Potter, 1999).

For the vertical forces the analysis suggests Trackaxle will be between 3.2 to 4.3 times less damaging than a conventional trailer with a non-steered tri-axle group.

The pavement damage values shown above for Trackaxle are presented relative to the baseline vehicle. The constants of proportionality for the damage relationships presented in Prem and Potter (1999) depend on pavement-specific properties. Because these constants are not known, they do not allow computation of absolute damage.

8. SUMMARY

A technical description of the steerable trailer axle system known as "Trackaxle" has been presented and an evaluation of its performance has been carried out using a performance-based standards approach and the performance standards developed by Austroads and the NRTC. Computer-based models were created of a representative baseline vehicle and another of a near identical vehicle with a longer semi-trailer and featuring the Trackaxle system. Safety and infrastructure related performance measures were chosen from the final set of measures developed by Austroads and the National Road Transport Commission (NRTC). These were applied to the two vehicles.

Performance of Trackaxle was evaluated both in relative terms, by comparing its performance to that of the baseline vehicle, and in absolute terms against the performance requirements of the selected Austroads/NRTC performance standards.

Trackaxle was able to meet all of the Austroads/NRTC performance standards that were considered. Its performance was found to be better than the baseline vehicle in respect of low-speed offtracking, rearward amplification, and pavement wear due to both horizontal and vertical tyre forces. Static rollover stability for both vehicles was similar. Compared to the baseline vehicle the performance of Trackaxle in respect of tail swing (on both the entry and exit approaches), yaw damping and high-speed transient offtracking was found to be worse.

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TABLES & FIGURES

Table 1 – Performance Summary

#	Performance Measure	Baseline (OAL=19.00 m)	Vehicle Trackaxle (OAL=21.08 m)	Steer Limiter
1	Low-Speed Offtracking	7.307 m	6.708 m	Active
2a	Tail Swing (turn entry)	0.056 m	0.325 m	Active
2b	Tail Swing (turn exit)	0.0 m	0.275 m	Active
3	Static Rollover Threshold	0.417 g (R = 100 m) 0.410 g (R = 20 m)	0.416 g (R = 100 m) 0.412 g (R = 20 m)	Active
4	Rearward Amplification	1.202	0.970	Active
5	Yaw Damping Coefficient	0.533	0.487	Active
6	High-Speed Transient Offtracking	0.201 m	0.365 m	Active
7	Horizontal Tyre Forces	Absolute value of pavement wear for baseline vehicle determined.	Pavement wear due to: a) <u>Horizontal forces</u> - 160,000 to 190,000 times smaller than baseline vehicle. b) <u>Vertical forces</u> - 3.2 and 4.3 times smaller than baseline vehicle.	Active

Table 2 – Summary of Performance Levels for Standards Considered (from Prem et al, 2002).

#	Performance Measure	Performance Level (Prem et al, 2002)
1	Low-Speed Offtracking (maximum swept path width)	5 m local roads; 7.4 m arterial roads; 10.1 m major freight routes; and 13.7 m road train areas.
2	Tail Swing (turn entry/exit)	0.35 m
3	Static Rollover Threshold	7 0.40 g, dangerous goods and buses 7 0.35 g, all other heavy vehicles
4	Rearward Amplification	5.7SR _{low} (refer to main text)
5	Yaw Damping Coefficient	7 0.15
6	High-Speed Transient Offtracking	0.8 m
7	Horizontal Tyre Forces	No greater than 1.8 times more damaging than the damage caused the reference vehicle (Austroads design primer mover and semi-trailer).

Table 3 – Summary of Performance Levels for Standards Considered (from Prem et al, 2002).

#	Performance Measure	Trackaxle of Baseline	Trackaxle of Austroads/NRTC
1	Low-speed offtracking	Better	Acceptable for arterials
2a	Tail swing (on entry side of turn)	<u>Worse</u>	Standard met
2b	Tail swing (on exit side of turn)	<u>Worse</u>	Standard met
3	Static rollover threshold	Similar	Standard met
4	Rearward amplification	Better	Standard met
5	Yaw damping	<u>Worse</u>	Standard met
6	High-speed transient offtracking	<u>Worse</u>	Standard met
7	Horizontal tyre forces	Better	Standard met

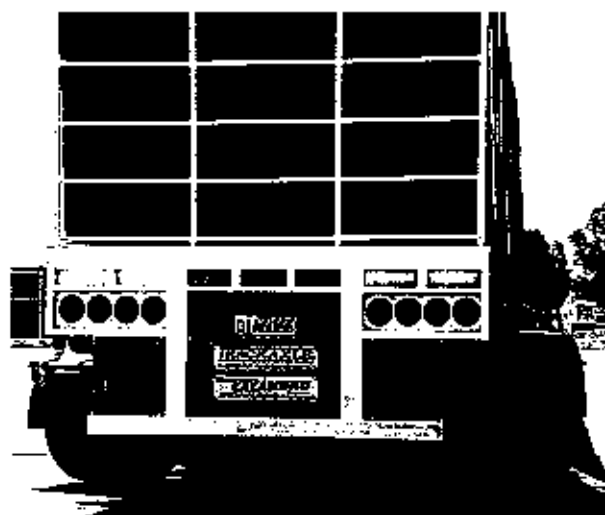


Fig. 1 Rear view of the Trackaxle system showing how the entire axle group is steered relative to the trailer chassis.

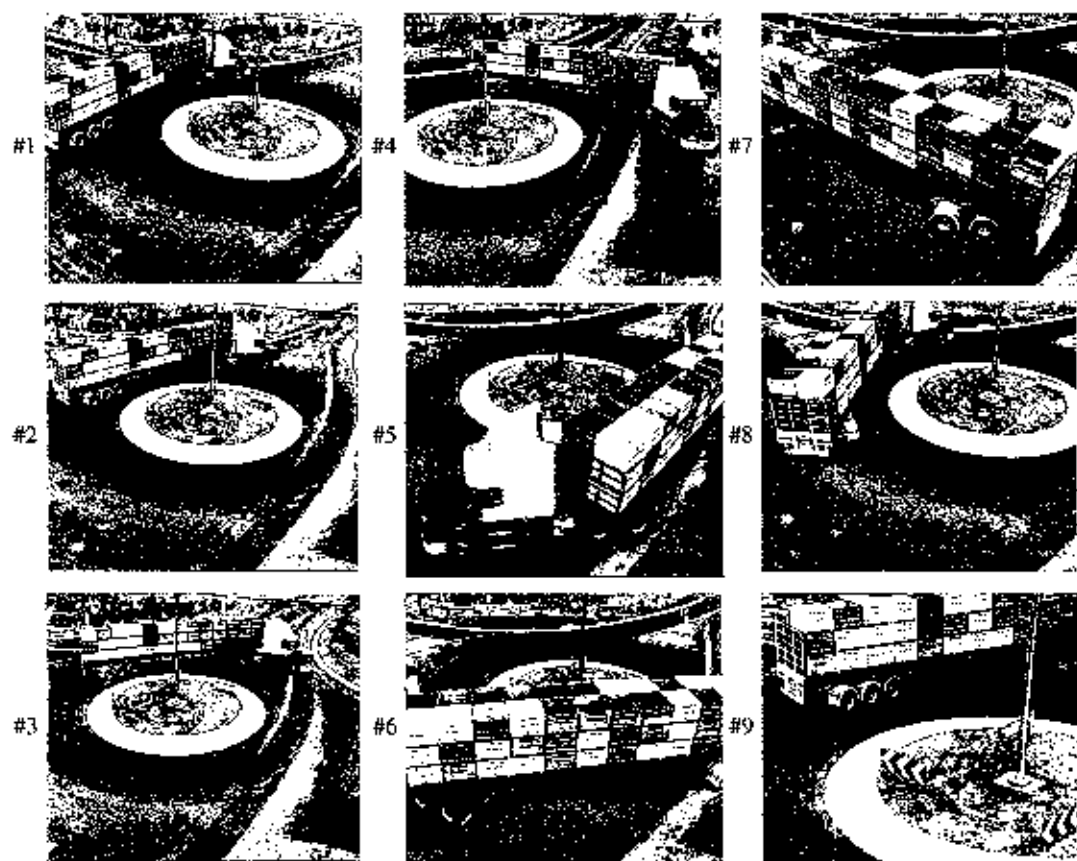


Fig. 2 Illustration of Trackaxle on an urban roundabout (commence left column, go top to bottom).

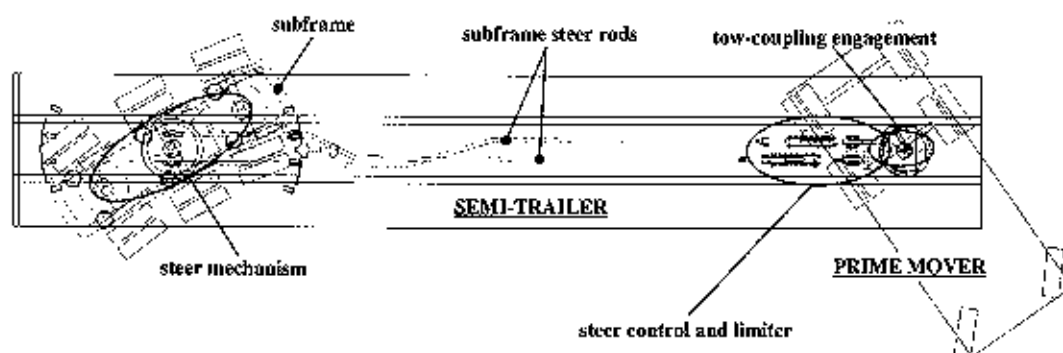


Fig. 3 Schematic showing the main parts of the Trackaxle system.

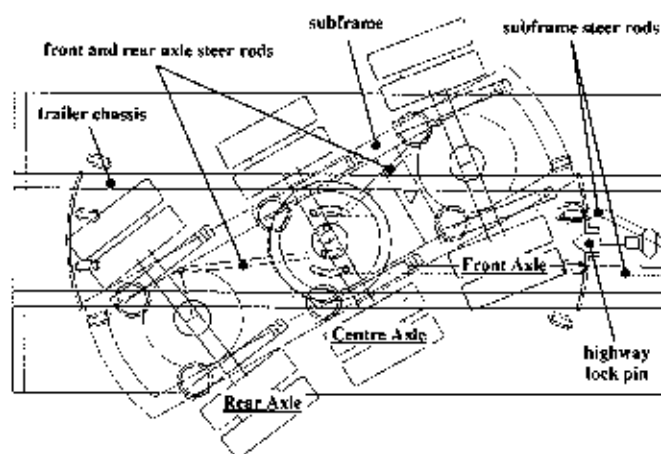


Fig. 4 Schematic showing steer mechanisms between subframe and chassis, and front and rear axles and centre axle (adapted from drawings supplied by Gayat Pty Ltd).

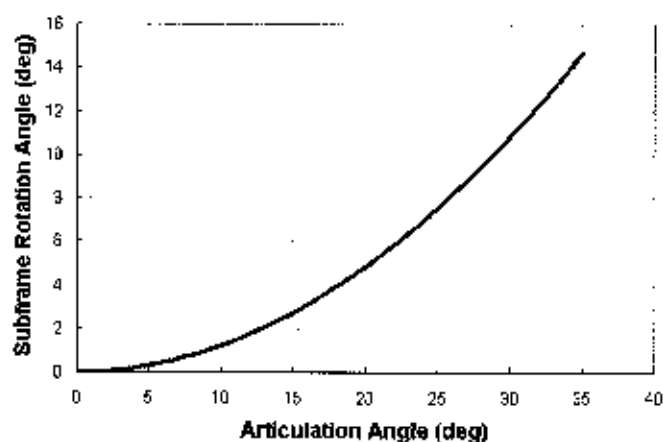


Fig. 5(a) Basic relationship between articulation angle and subframe rotation angle with the steer limiter not active.

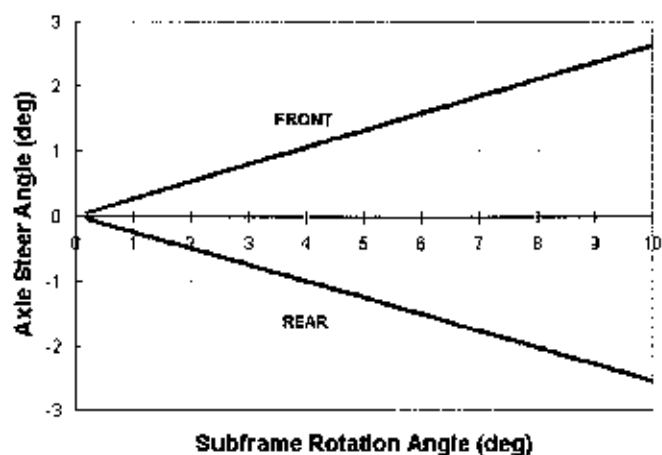


Fig. 5(b) Relationship between subframe rotation angle and the steer angle of the front and rear axles of an axle group.

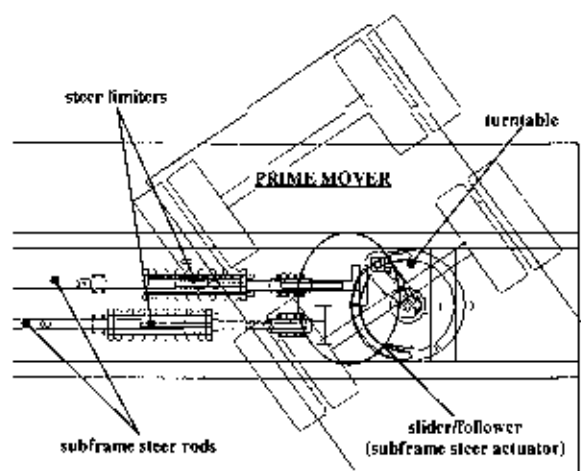


Fig. 6 Schematic showing subframe steer-actuating mechanism and steer limiter spring arrangement.

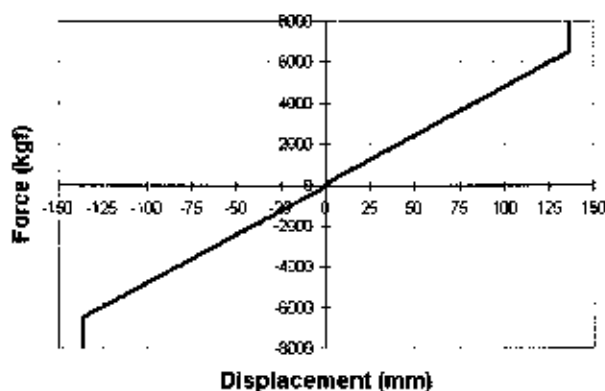


Fig. 7(a) Idealised characteristics of the steer limiter for the full range of spring displacements.

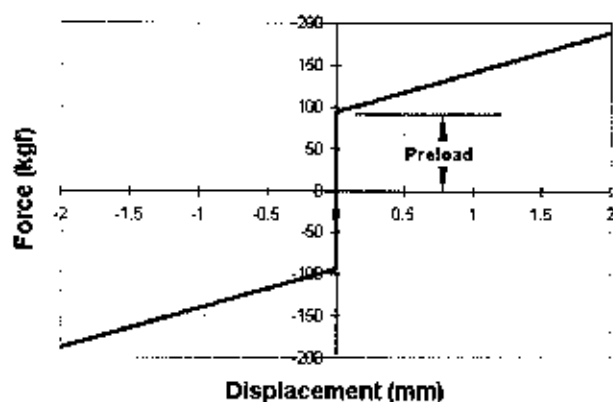


Fig. 7(b) Idealised characteristics of the steer limiter for small displacements showing the effect of spring preload.

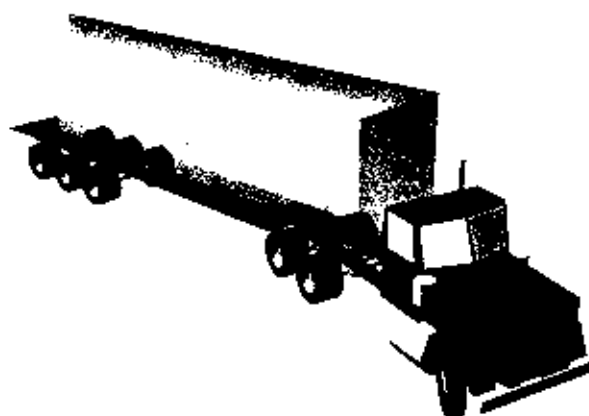


Fig. 8(a) RTDynamics model of the Austroads design prime mover and (13.7m) semi-trailer.



Fig. 8(b) RTDynamics model of the prime mover and the Trackaxle (15.8m) semi-trailer. For consistency the prime mover is the same for both models.

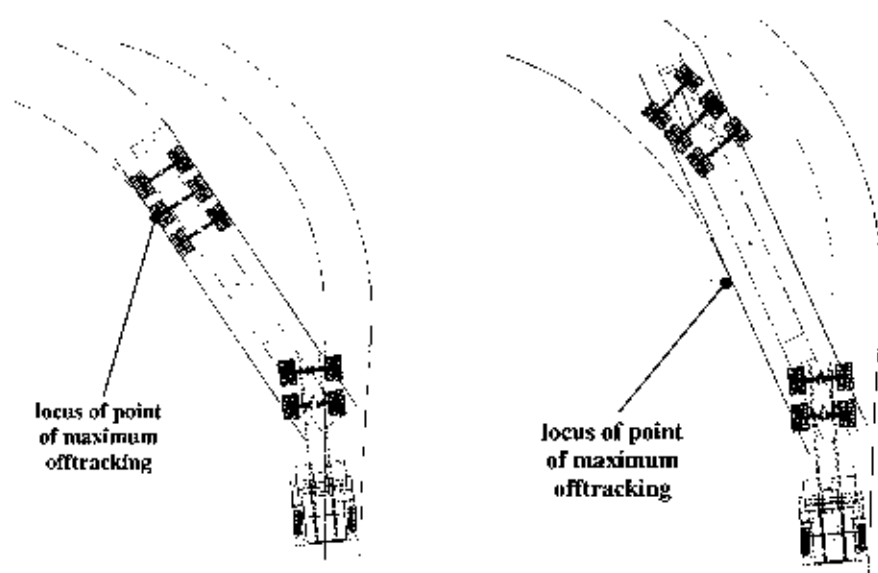


Fig. 9 Location of maximum offtracking for baseline vehicle (left) and Trackaxle (right). On the Trackaxle trailer the point of maximum offtracking is well inside the path of the rear axle group.

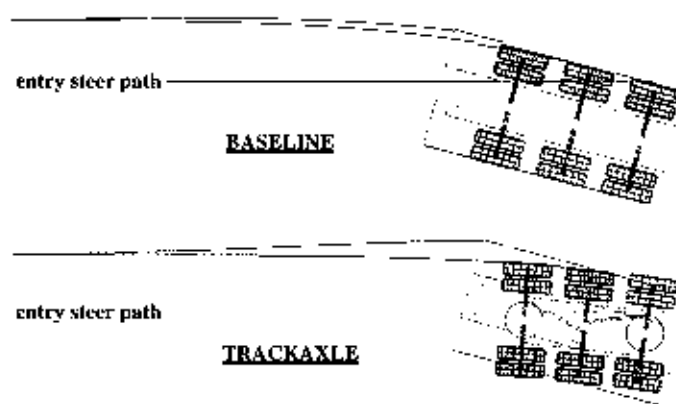


Fig. 10 Trackaxle tail swing compared to baseline during the initial stages of the low-speed turn. (Steer of the front and the rear axles on Trackaxle is evident.)

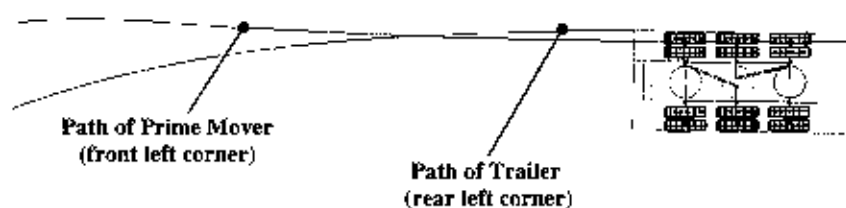


Fig. 11 Trackaxle tail swing (or overshoot) on the exit side of the turn.

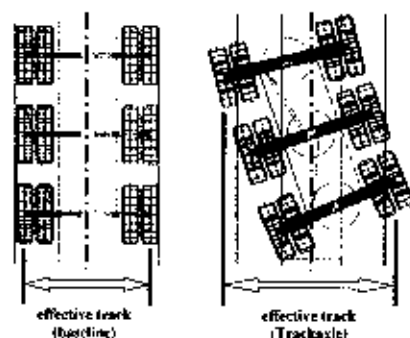


Fig. 12 Effective track for a conventional tri-axle group and Trackaxle, as defined by Gayat (2000a).

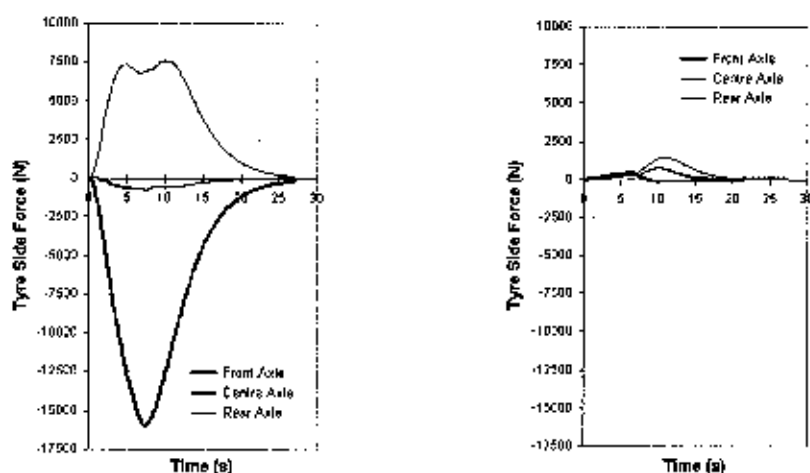


Fig. 13(a) Typical horizontal forces for tyres on the inside of the turn. These show larger forces are imposed on the pavement by the baseline vehicle (plot on the left) compared to Trackaxle (plot on the right).

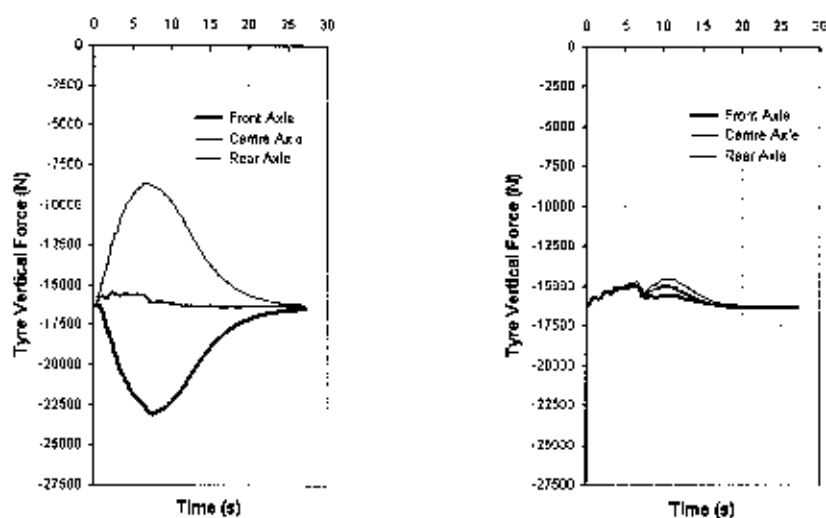


Fig. 13(b) The vertical forces corresponding to those in Fig. 13(a), showing that the horizontal forces modify the vertical forces.

DYNAMIC INTERACTION OF VEHICLES AND BRIDGES

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ABSTRACT

Many studies conducted worldwide over a period of more than a decade have confirmed that large dynamic effects can be induced in bridges by the combination of heavy vehicles and uneven road profiles. In Australia, approximately 75% of the bridges have spans between 5 and 15 m. Many of these short span bridges have shown dynamic effects due to vehicle-bridge interaction that far exceed the provisions of the current Australian Bridge Design Code, and in some cases dynamic increments in excess of 100% have been recorded, even for heavily loaded events. The introduction of the new Australian Bridge Design Standard will see the design load for bridges increase to allow for heavier and more innovative vehicles in the future, and it is important to investigate vehicle-bridge interaction associated with these future vehicle types.

This paper presents results from computer-based dynamic models that were developed to investigate the complex problem of dynamic interaction between heavy vehicles and bridges in the presence of uneven road profiles. The models were used to investigate the influence of the main parameters governing dynamic responses of bridges to the crossing of heavy vehicles, such as bridge natural frequency and damping, vehicle mass and suspension characteristics, road profiles on the bridge approaches and deck, and vehicle speed. Results from parametric studies were found to be generally consistent with field measurements and experience. The work described in this paper is part of an ongoing program of research in Australia that is aimed at delivering recommendations to AUSTROADS with respect to the dynamic load allowance both for the design of new bridges and the assessment of existing bridges.

1. INTRODUCTION

Many studies conducted worldwide over a period of more than a decade have confirmed that large dynamic effects can be induced in bridges by the combination of heavy vehicles and uneven road profiles. In Australia, approximately 75% of the bridges have spans between 5 and 15 m, and during dynamic testing using heavy vehicles, many of these short span bridges have shown dynamic effects due to vehicle-bridge interaction that far exceed the provisions of the current Australian Bridge Design Code (Austroads, 1996). In some cases dynamic increments in excess of 100% have been recorded, even for heavily loaded events.

In Australia there is continuing pressure for current bridges built throughout the last millennium to carry a diverse range of heavy vehicle configurations and types as illustrated in Fig. 1. The introduction of the new Australian Bridge Design Standard (Standards Australia, 2000) will see the design load for bridges increase to allow for heavier and more innovative vehicles in the future. Thus, it is important to understand vehicle-bridge interaction and the effects of heavy vehicles on bridges to quantify the dynamic loading to:

1. Allow the safe and efficient evaluation of the capacity of old bridges to carry present vehicles.
2. Provide design standards for new bridges that will be consistent with vehicles and bridges of the future.
3. Provide guidelines for the maintenance of road profiles to avoid unnecessarily high dynamic loadings on bridges from vehicles.

This paper looks at the issues associated with vehicle-bridge interaction in Australia, presents the development of a vehicle-bridge interaction model which was created to study the interaction between vehicles and bridges, and presents the results of a preliminary parametric study.

¹ Infratech Systems & Services, and RTDynamics (Roads and Transport Dynamics), are Business Units of Texcel Pty Ltd.

2. BACKGROUND

The dynamic response of bridges is the result of a complex interaction between the bridge, the vehicle/s that cross it and the road profile. Bridge design codes generally bundle this complex interaction into a single dynamic load allowance (DLA) or impact factor that is applied to the static effects of the vehicle/s. The Austroads² Bridge Design Code (Austroads 1996) follows this approach. It presents the DLA as a function of bridges natural frequency for frequencies between 0 and 7 Hz. These frequencies correspond to bridges with spans of 20 m or greater. In Australia, bridges with spans greater than 20 m represent less than 10% of existing bridges by number and less than 40% by area (National Road Transport Commission, 1996). Note that new bridges generally have longer spans.

Short span bridges were not included in the original international testing program that provided the basis of the DLA provisions in the Austroads Bridge Design Code. The considerable dynamic testing of bridges conducted in Australia and New Zealand over recent years has provided considerable data associated with both short and medium span bridges (Heywood, 2000). The medium span dynamic increment (DI) data is broadly consistent with the Austroads Bridge Design Code DLA provisions but the short span DI is often significantly larger than the extrapolation of the DLA to these higher frequencies. Fig. 2 summarises the test data and shows the relationship between bridge natural frequency, span and dynamic increment.

The analysis of the experimental data from these bridge tests highlighted the influence of gross laden mass, road profile, vehicle configuration/suspension, speed and bridge natural frequency on the dynamic increment. There was a strong trend supporting a reduction in the DI with increasing mass, which is consistent with the international literature. This would lead to possible increases in load carrying capacity for older bridges provided that road profiles are maintained.

The considerable variability of the DI data is believed to be associated with variations in the road profile, test truck suspension and speed. It is clear that these parameters are important as the dynamic response of a bridge changes from vehicle-to-vehicle and the dynamic response of similar bridges varies from bridge-to-bridge. The dynamic response at one speed may be considerably different to that at other speeds. The critical speed varies widely. It may be as low as 20 km/h. For short span bridges, the critical speed tends to be between 40 and 80 km/h.

The test data shows the variability of dynamic increment on bridges and the influence of some of the parameters. Because of this variability and the expense of testing, a computer-based model was developed to study the influence of various factors on dynamic increment. The model and typical results are outlined in the following sections of this paper.

3. MODEL DETAILS

The vehicle model is an extension of the 1/4-truck model developed by Prem et al (1998), consisting of a 1/4-truck sprung mass with three independent axles – in place of the more usual single axle – to form a tri-axle group. The three axles traverse a simply supported single span bridge, represented by an equivalent 1/4-bridge, in the presence of a road profile, as shown in Fig. 3. Either measured or artificially generated profiles can be included on the bridge and on its approaches. The model allows the user to study the interaction between the vehicles suspension, the road profile and the bridge, which deflects as the vehicle traverses the bridge.

3.1 Vehicle Model

3.1.1 General Description

The 1/4-truck tri-axle group vehicle model consists of a sprung mass, and three (3) unsprung masses representing the vehicle body and the three individual axles respectively. The model has four degrees-of-freedom, one degree-of-freedom in bounce for the sprung mass (vehicle body), and one for each of the three unsprung masses (axles). The model features non-linear spring and damper elements in the suspensions. The axles are connected to the road profile through a linear spring and viscous damper that represents the tyre.

² Austroads is the association of Australian and New Zealand road transport and traffic authorities.

3.1.2 Tyre-Road Contact

Contact between each tyre and the road profile is accounted for through the contact patch, with a fixed length moving average simulating the way the tyre envelopes small bumps and short, sharp, uneven features.

3.1.3 Air and Steel Suspensions

The suspension between the vehicle body and axle is modelled using two different types of non-linear spring, one that is based on the published models developed by Fancher et al (1980) representing a multi-leaf steel spring suspension Fig. 5(a), and one for an airbag suspension Fig. 5(b). The air spring model was developed from first principles assuming an adiabatic/isothermal gas compression/expansion thermodynamic process, which acts adiabatically for dynamic loads and isothermally for static loads. The isothermal feature allows the change in the air suspension stiffness with higher loads on the truck to be modelled, whereas the adiabatic feature accounts for the "stiffer" response from the air spring in the absence of heat transfer between the gas contained in the airbag and its surroundings when the excitation frequencies are higher (Prem et al. 1998).

3.1.4 Dampers

For both steel and air suspensions, a non-linear viscous damper has been added, similar to the one described in Prem et al (1998). The amount of damping in rebound (when the distance between the ends of the shock absorber is increasing) changes from low to high when a certain stroke velocity has been exceeded; in bump a single value of damping is used. This is consistent with the measured characteristics of shock absorbers reported by others, as discussed in Prem et al (1998).

3.1.5 Suspension Tests and Parameters

To tune and validate the parameters for the models, standard tests were applied. The responses from the model were analysed for natural frequencies and damping levels (bounce and wheel-hop modes) and compared with those published in literature (see, for example, OECD, 1997; Woodroffe, 1996). For the sprung mass bounce frequency the pull-up method was used (Anon, 1992), requiring the chassis of the vehicle (sprung mass) to be raised 80 mm above the axle and released from this position. The subsequent oscillations are analysed to determine the natural frequency and damping of the body bounce mode. To check the wheel-hop frequencies and damping levels the axles were displaced from their equilibrium positions and the subsequent free-vibration was analysed.

Analysis of the vehicle body oscillation from these tests for the steel and air suspension are listed in Table I. Also listed in this table are results of axle hop tests conducted on the model. The model results compare well with published (OECD, 1997; Woodroffe, 1996) and demonstrate the ability of the model to reproduce the key behaviour of air and steel suspensions. By changing suspension and damper characteristics it will be possible to model and evaluate the effect of suspension modifications, or, for example, air suspensions with failed dampers, vehicle behaviour and more importantly the effect of these suspension characteristics on bridges.

3.2 Bridge Model

In Australia short span simply supported bridges, such as those illustrated in Fig. 1 and Fig. 6 dominate the nations bridge inventory, as shown in Fig. 4. Thus the bridge model developed at this stage represents a simply supported single span, though it can be extended to model simply supported multiple span bridges. The model consists of a single degree-of-freedom linear model with the addition of transformations to account for the position of the vehicle on the span. The bridge is modelled as a mass on a linear spring with a linear viscous damper, as shown in Fig. 3. A method has been developed to calculate an equivalent mass, spring stiffness, and level of damping for the bridge – effectively creating a ¼-bridge analogue of the ¼-truck model – that is consistent with typical bridges based on measured natural frequency, percentage of critical damping and mid-span deflection.

3.3 Road Profiles

Measured road profiles or artificially generated profiles can be used in the model. The road profile is specified as an elevation versus chainage. The road profile before, on, and after the bridge can be specified, and the truck model passes over this profile. The section of road profile on the bridge accounts for any initial static deflections as well as deflecting under the dynamic loads imposed on the bridge by the vehicle. Fig. 7 shows a typical road profile recorded on the Camerons Creek Bridge.

Thus, the influence on vehicle bridge interaction of profiles on bridge approaches at the abutments and on the bridge decks can be studied.

3.4 Vehicle-Bridge Interaction Modelling

Both the bridge and the vehicle are modelled as interacting dynamic systems. The road profile causes dynamic wheel forces, which induces a dynamic bridge response which in-turn modifies the road profile. Thus there is an interaction between the vehicle and the bridge. Simulations can be performed with interaction between vehicle and bridge either enabled or deactivated.

To evaluate the effectiveness of the model and to demonstrate its capabilities, the model was used to simulate experiments that were conducted during the dynamic load testing of the Camerons Creek Bridge as part of the OECD DIVINE Project (OECD, 1997). During the testing of Camerons Creek bridge, road profiles were recorded, as well as dynamic wheel forces and bridge responses for a range of truck test speeds. Because the recorded results are based on a 6-axle prime mover and semi-trailer combination, two simulations were used to model the results. One model was used for the prime mover and one model for the trailer with the results superimposed to represent the passage of the truck over the bridge. Fig. 8 compares the bridge response from the model for a truck with steel suspension, and Fig. 9 for an air suspended truck. The results from the model compare well with the experimental results and show the ability of the model to simulate vehicle-bridge interaction. The model can also be used to simulate dynamic wheel forces and comparison of these wheel forces with the recorded wheel forces at Camerons Creek also shows that there is good agreement. Fig. 10 shows a comparison of results based on dynamic increment versus speed. The graph compares the results from the model using a tri-axle group with the experimental results for a 6-axle prime mover and semi-trailer. Again there is good agreement between experimental and modelled results.

4. PARAMETRIC STUDY

To begin the study into the influence of the various parameters which influence vehicle bridge interaction including: road profile, bridge type, vehicle suspension, configuration and mass a parametric study using the developed vehicle bridge interaction model was undertaken. The results from the study presented in this paper are based on the Camerons Creek bridge, use a variety of road profiles and vary the vehicle mass. Both air and steel suspensions were studied.

Fig. 11 shows the dynamic increment versus speed relationship for the steel suspended vehicles. For each profile: smooth, smooth with an initial bump, and measured profile, two truck masses were analysed (standard legal mass and 1.5 times the standard legal mass) with no change to the suspension parameters. In a similar analysis, the resultant relationships for the air suspension are shown in Fig. 12.

The results show the influence of road profile on the dynamic increment versus speed relationship for this bridge for both suspension types. The addition of a bump at the abutment considerably increases the dynamic increment compared to the smooth profile. The results for the actual measured profile (rough) are also significantly higher than those for the smooth profile as expected.

In each case, the increase in truck mass led to a reduction in the dynamic increment, however the reduction due to the increased mass on a smooth profile is very small.

Air suspensions produce either similar or reduced dynamic increments when compared to equivalent steel suspension systems. Due to the stiffness of the air suspension being linked to truck mass, the reduction in dynamic increment due to increased mass is less pronounced for air suspension than observed for steel suspension.

This study will be expanded during the remainder of the project to incorporate other bridges, road profiles of varying roughness (IRI), and a wider range of vehicle masses. This work will lead to a basis to make recommendations for the dynamic increment versus speed relationship for the new Australian Bridge Design Standard.

5. MODEL APPLICATIONS

The vehicle bridge interaction model developed for this project has a wide range of other applications. These include:

1. Studying vehicle-road interaction.
2. Assist with the design of "bridge friendly" suspension systems.
3. A basis for a *Bridge Roughness Index (BRI)*. A BRI could be developed, similar to IRI but designed to quantify the influence of road profile on bridge response. The BRI could then be monitored as part of routine network level roughness surveys to determine when unevenness levels on bridges and bridge approaches exceeded specified levels indicating that profile repair was required.

6. CONCLUSION

This paper has reviewed the development of a computer model, which includes the ability to study the effect of heavy vehicle suspension systems, vehicle mass, road profiles, and bridge behaviour on vehicle bridge interaction. Comparison of the model outputs with field test results from dynamic testing of bridges shows good agreement between the two results. The model is currently being used to conduct a parametric study which will allow recommendations to be made for appropriate dynamic increment values for the new Australian Bridge Design Standard. Dynamic increment is a measure of the increased effects of heavy vehicles on bridges due to dynamic effects over and above the static effects. To date the results of this study have shown that the road profile is a significant factor in vehicle bridge interaction and that dynamic increment values generally decrease with increasing vehicle mass. The model also has a number of other applications, which may be further developed in the future.

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TABLES & FIGURES

Table 1 Bounce and wheel-hop frequencies from the suspension tests.

Suspension Model	Body Frequency (Hz)	Bounce	Body Damping (% of Critical)	Bounce	Axle-Hop Frequency (Hz)
Air	1.55		17		10.6
Steel	3.2		7.1		11.7



Fig. 1 Triple road train crossing the Ward River, Central Queensland.

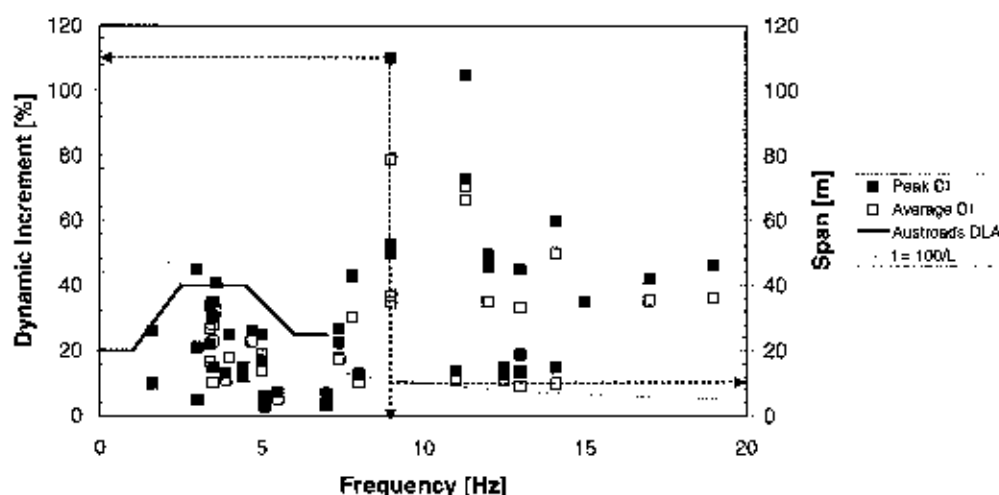


Fig. 2 Dynamic Increment Versus Frequency: All Materials - GLM 40-55 tonnes including Austroads DLA provision and relationship between frequency and span (Heywood, 2000).

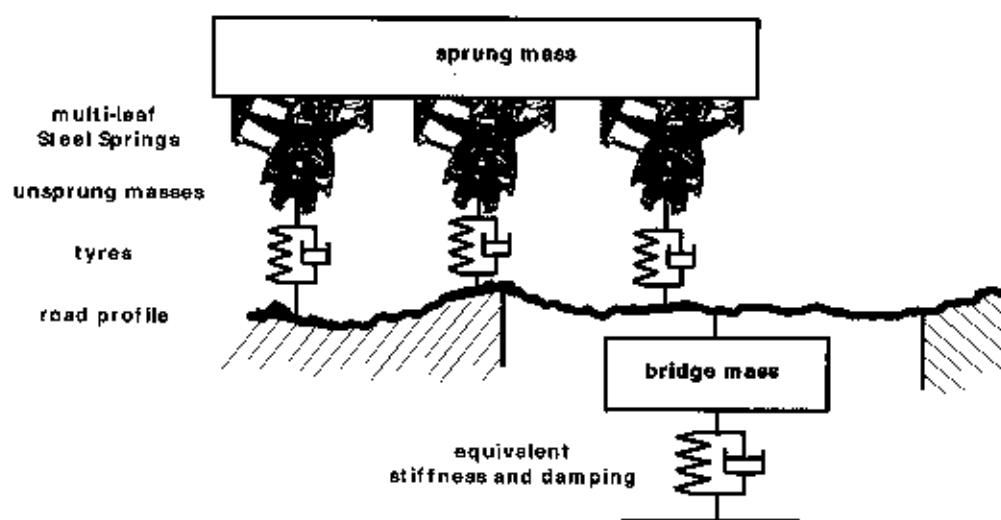


Fig. 3 Schematic of steel-spring suspension 3-axle 1/4-truck model and corresponding "1/4-bridge" model.

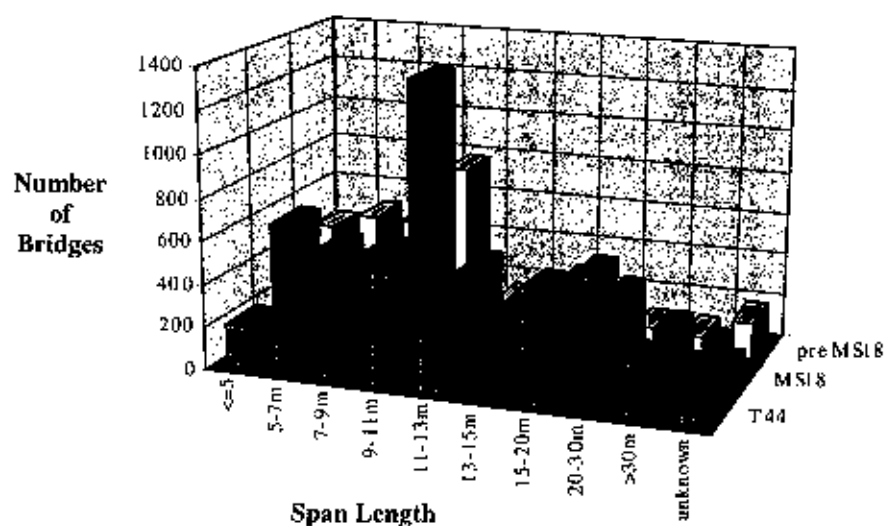
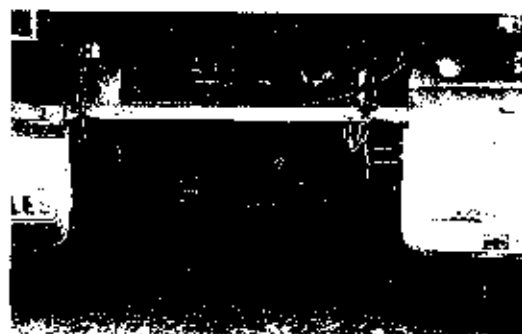


Fig. 4 Span lengths of bridges for the main design standards.
Source: National Road Transport Commission (1996).



(a) Multi-Leaf Steel suspension.



(b) Air suspension.

Fig. 5 Typical suspension systems.



Fig. 6 Ringarooma Bridge, Tasmania.

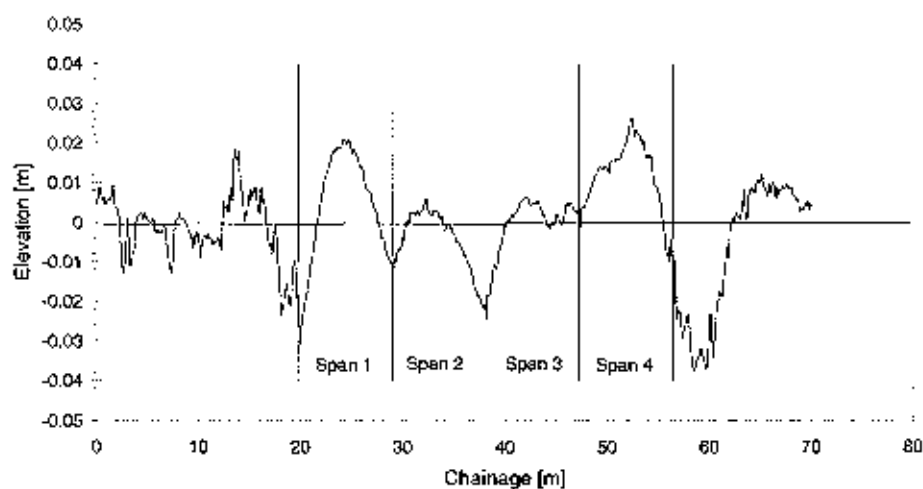


Fig. 7 Typical road profile (OECD 1997).

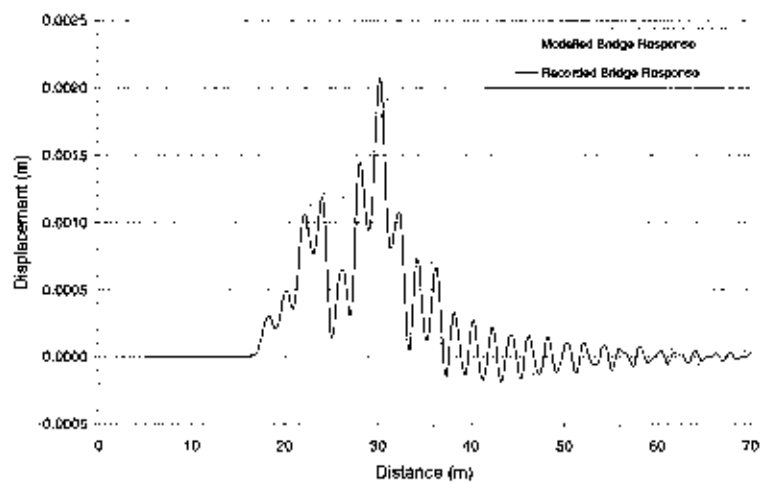


Fig. 8 Comparison of recorded field response and response from Vehicle Bridge Interaction Model (steel suspension).

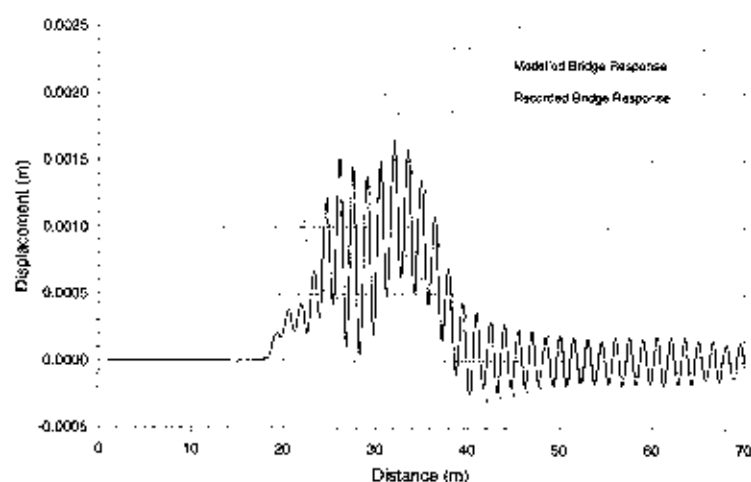


Fig. 9 Comparison of recorded field response and response from Vehicle Bridge Interaction Model (air suspension).

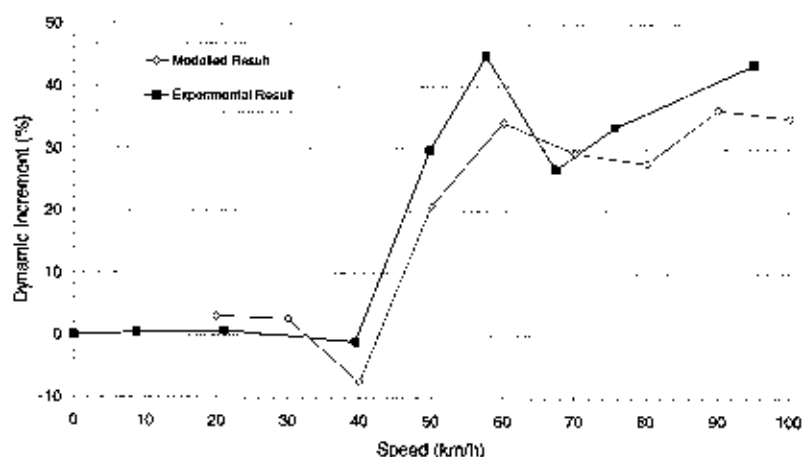


Fig. 10 Comparison of dynamic increment versus speed based on recorded test results and results from Vehicle-Bridge Interaction Model.

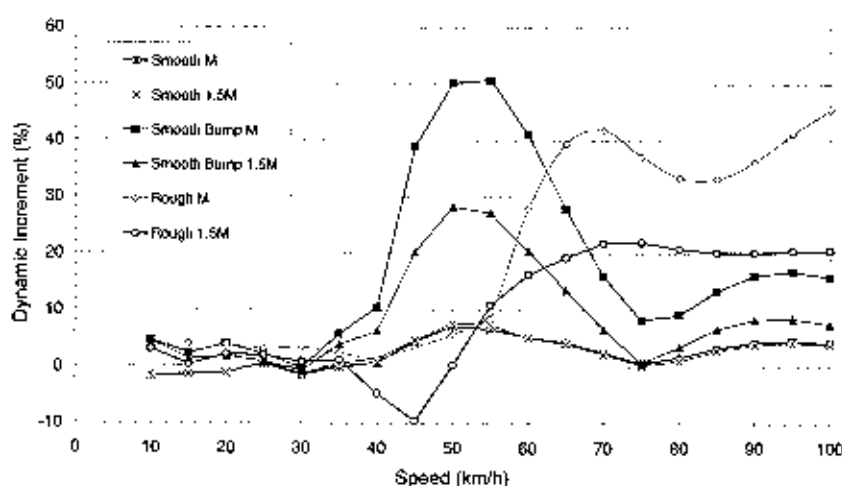


Fig. 11 Dynamic Increment Result for Steel Suspension.

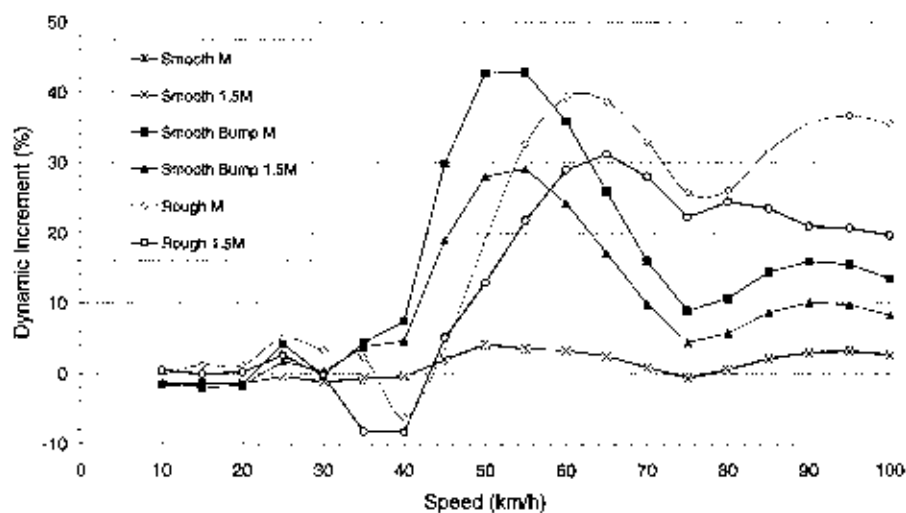


Fig. 12 Dynamic Increment Results for Air Suspension.

IMPACTS OF DIFFERENT JUNCTION TYPES ON HEAVY DUTY VEHICLES

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ABSTRACT

The paper includes the impacts which four different junction types cause on heavy duty vehicles. In order to indicate this clearly the light vehicles are surveyed, too. The impacts are estimated in quantitative units and in money terms. The junction alternatives are: an intersection, a roundabout, an interchange with one loop ramp and three rhombic ones and an interchange with four rhombic ramps. The type vehicles are a coach, a single unit truck, a truck + semi-trailer combination and a truck + trailer combination and two light vehicles; it is the passenger car and the van. The impacts are estimated by using a vehicle motion simulator based on dynamics. It is called VEMOSIM and developed in Finland.

The VEMOSIM outputs directly the data needed for quantifying the impacts, when vehicles move through the junctions. The impacts are: the fuel amount (and thus the variable vehicle operating costs), the emission amounts by components (NO_x , CO, HC, PM and CO_2), the time consumed, gear changes, etc.

Based on these output data the following cost items are calculated: the variable vehicle operating costs, the time costs and the emission costs and, additionally, the gear changes and information about the benefit distribution between the vehicle categories. The case study concerns a 4-leg junction of the main roads no. 5 and no. 14 in Finland (ADT ca 8500; 89 % of light vehicles and 11 % of HDVs).

The results and the conclusions are: the best alternative is the interchange with four rhombic ramps. However, its benefit/cost ratio is not high enough, because the construction costs are high and the traffic volume low. The roundabout is the worst. It causes extra costs compared with the present intersection, because of decelerations and accelerations, when the vehicles move through the roundabout. Among the different vehicle categories the greatest impact concerns the truck + trailer combination. Especially, the fuel consumption and emissions increase remarkably in the case of the roundabout. The VEMOSIM system is an effective tool for analyzing impacts of different junction types.

INTRODUCTION

The origin of the study is based on the traffic growth at a junction of two main roads (highways 5 and 14) in Eastern Finland. This is a normal junction with four legs, of which three are highways and one is a secondary road. The highway no 5 is locating in the direction South-North-South. The Eastern leg is highway no 14, and, especially, this has caused congestion during summer weekends. Because the present junction is of the type of the intersection, plans have been made in order to improve the traffic situation. The junction alternatives to be studied are: an intersection (current), a roundabout (abbreviated RA), an interchange with one loop ramp and three rhombic ones (IC L), and an interchange with all rhombic ramps (IC D).

The paper includes the impacts which four different junction types cause on different vehicle categories. These impacts are estimated in quantitative units and in money terms. The type vehicles are a passenger car, a van, a coach, a single unit truck, a truck + semi-trailer combination and a truck + trailer combination. The impacts are estimated by using a vehicle motion simulator based on dynamics. It is called VEMOSIM and developed in Finland.

STUDY METHOD

As a method we have used a vehicle motion simulator, VEMOSIM, based on dynamics.

The VEMOSIM simulates a motion of any vehicle. It is based on the technical characteristics of the vehicle (engine, powertrain and gear ratios, drive resistances etc.), the road vertical and horizontal geometry and the driving technique. In this case the time resolution for the simulation is 0.1 seconds. At the rate of this data updating frequency the VEMOSIM calculates the instantaneous power need and regulates the accelerator or brake pedal positions and gear locations according to these requirements, in other words simulates the real engine operating state ten times per a second.

The VEMOSIM outputs directly the data needed for quantifying the impacts, when vehicles move through the junctions. The impacts are: the fuel amount (and thus the variable vehicle operating costs), the emission amounts by components (NO_x , CO, HC, PM and CO_2), the time consumed, gear changes, etc.

Because all individual vehicles passing through the junctions cannot be simulated, a representative sample of different vehicles are selected for the vehicles categories. In this context they are called the type vehicles.

INPUT DATA

For the simulation concerning each type vehicle three different types of input data are needed: 1) the technical characteristics of the type vehicles, 2) the vertical and horizontal alignment of the different routes (continuously) and 3) the driving technique that is mainly composed of the goal speed pattern (meter by meter).

In order to calculate the total impacts (all vehicles passing the junctions), the traffic volumes by the vehicle categories and traffic routes ($4 \times 3 = 12$) must be known, see table 1. The cross section traffic volumes by road leg and vehicle categories are seen in table 2.

The goal speed pattern is in general an array of the instantaneous speed that the driver tries to maintain at any part of the route to be driven. The current speed limit values naturally determine the most of the patterns, but for example at turns (left and right) as well as at roundabouts those speed limit values (80, 60 and 50 km/h) cannot be maintained. In those cases the goal speed is 20...30 km/h unless the vehicle stops. But sometimes the vehicles are obliged to stop when swerving the priority traffic. Then the goal speed is instantaneously 0 km/h. In the goal speed patterns the share of stopping vehicles has been taken into account. For each route and junction type alternative two patterns have been determined, the one for the non-stopping vehicles and the other for the stopping vehicles, see table 3.

The traffic is composed of six vehicle categories, which are represented by the respective type vehicles. These are:

<i>Vehicle category</i>	<i>Abbreviation</i>	<i>Average mass</i> <i>kg</i>
Passenger car	P	1200
Van	V	2300
Bus & coach	C	15000
Single unit truck	T	20000
Truck + semi-trailer	TS	35000
Truck + trailer	TT	50000

The detailed vehicle technical data are not presented here.

IMPACTS STUDIED

The impacts caused by the different junction alternatives are surveyed both as quantitative and as economical. The results are presented as differences compared to the present junction type (intersection). An example of a drive simulation through a roundabout is shown in figure 1.

The most important quantity is the fuel consumption. On the basis of the fuel consumption the all variable vehicle operating costs are determined, too.

The variable operating costs are composed of the following items:

- fuel costs
- lubricant costs
- repair and maintenance costs
- tyre costs

In this study the lubricant, repair & maintenance and tyre costs are assumed to change in the same ratio as the fuel consumption (Wehner's principle). These costs are presented both at the market price (including indirect taxes) and at the production cost price (excluding indirect taxes).

The used time is surveyed, too. The time has also shadow unit prices for the different vehicle categories. In this respect there is no difference between the market price and production cost price.

The pollutant emissions to be surveyed are:

- nitrogen oxides (NO_x)
- carbon monoxide (CO)
- hydro carbons (HC)
- particulate matters (PM)
- carbon dioxide (CO_2)

These pollutants also have shadow unit prices, by which the emission amounts have been converted to monetary values.

RESULTS

Fuel amount

The changes in the fuel amount are shown in figures 2 - 3. Concerning the different vehicle types in this respect the truck + trailer plays a dominant role. Concerning the junction alternatives the roundabout increases remarkably fuel consumption while the both interchange types decrease it.

Nitrogen oxides

The changes in the NO_x amount are shown in figures 4 - 5. As regards the nitrogen oxides the same conclusions as the ones regarding fuel consumption can be made. However, the truck + trailer seems to have an emphasized role.

Time

The time changes are shown in figures 6 - 7. In this respect the passenger car is dominant because of its high traffic volume and, in addition, to the highway 5 (South - North - South) the highway 14 (East - South) has some significance.

Variable vehicle operating costs

The variable vehicle operating costs at cost production price are shown in figure 8 and the ones at market price in figure 9. According to the definition of the variable vehicle operating costs (Wehner's principle) the impacts on them are in the same relationship as the impacts on fuel consumption.

Total costs

The changes in the total costs are shown in figures 10 and 11. Concerning the different vehicle types in this respect both the passenger car and the truck + trailer play a dominant role. Concerning the junction alternatives the roundabout increase total costs of both the passenger car and the truck + trailer. The interchange with four rhombic ramps is the best solution and decreases the total costs for all vehicle categories.

CONCLUSIONS

The best junction type is an interchange in general, because it causes savings in the vehicle operating, time and emission costs for all vehicles.

In this case the interchange with the four rhombic ramps brings the most benefits to the traffic.

On the contrary, the roundabout is the worst because it only increases vehicle operating, time and emission costs for all vehicles. In general, the roundabout should be avoided as junctions of main roads.

The VEMOSIM system is an effective tool for analysing impacts of different junction types.

TABLES & FIGURES

Table 1- Traffic volumes ADT [veh/d] at the junction

	<i>DIRECTION</i>	<i>P</i>	<i>V</i>	<i>C</i>	<i>T</i>	<i>TS</i>	<i>TT</i>	<i>TOT</i>
1	SOUTH-WEST	131	15	3	4	0	4	157
2	SOUTH-NORTH	1060	118	24	32	40	168	1440
3	SOUTH-EAST	1710	190	41	38	6	43	2027
4	WEST-NORTH	126	14	3	4	0	4	151
5	WEST-EAST	124	14	3	4	0	4	149
6	WEST-SOUTH	131	15	3	4	0	4	157
7	NORTH-EAST	226	25	7	8	3	19	287
8	NORTH-SOUTH	1060	118	24	32	40	168	1440
9	NORTH-WEST	126	14	3	4	0	4	151
10	EAST-SOUTH	1710	190	41	38	6	43	2027
11	EAST-WEST	124	14	3	4	0	4	149
12	EAST-NORTH	226	25	7	8	3	19	287

Table 2- Cross section volumes ADT [veh/d] at the junction

	<i>LEG</i>	<i>P</i>	<i>V</i>	<i>C</i>	<i>T</i>	<i>TS</i>	<i>TT</i>	<i>TOT</i>
1	SOUTH	5802	645	134	146	91	428	7246
2	WEST	761	85	18	24	0	24	912
3	NORTH	2822	314	67	87	84	381	3755
4	EAST	4119	458	101	99	17	131	4925

Table 3- Proportion of stopping flows and average waiting time at the junction

	<i>ALTERNATIVE</i>	<i>1</i>		<i>2</i>		<i>3</i>		<i>4</i>	
		<i>STOP</i>	<i>TIME</i>	<i>STOP</i>	<i>TIME</i>	<i>STOP</i>	<i>TIME</i>	<i>STOP</i>	<i>TIME</i>
		%	s	%	s	%	s	%	s
1	SOUTH-WEST	22	6	8	4	45	6	35	6
2	SOUTH-NORTH	0	0	8	4	0	0	0	0
3	SOUTH-EAST	0	0	8	4	24	2	20	2
4	WEST-NORTH	100	2	8	4	65	10	30	10
5	WEST-EAST	100	2	8	4	30	0	0	0
6	WEST-SOUTH	100	2	8	4	30	4	0	0
7	NORTH-EAST	35	6	8	4	20	2	25	10
8	NORTH-SOUTH	0	0	8	4	0	0	0	0
9	NORTH-WEST	0	0	8	4	20	1	25	1
10	EAST-SOUTH	100	2	8	4	0	0	20	2
11	EAST-WEST	100	2	8	4	0	0	0	0
12	EAST-NORTH	100	2	8	4	0	0	0	0

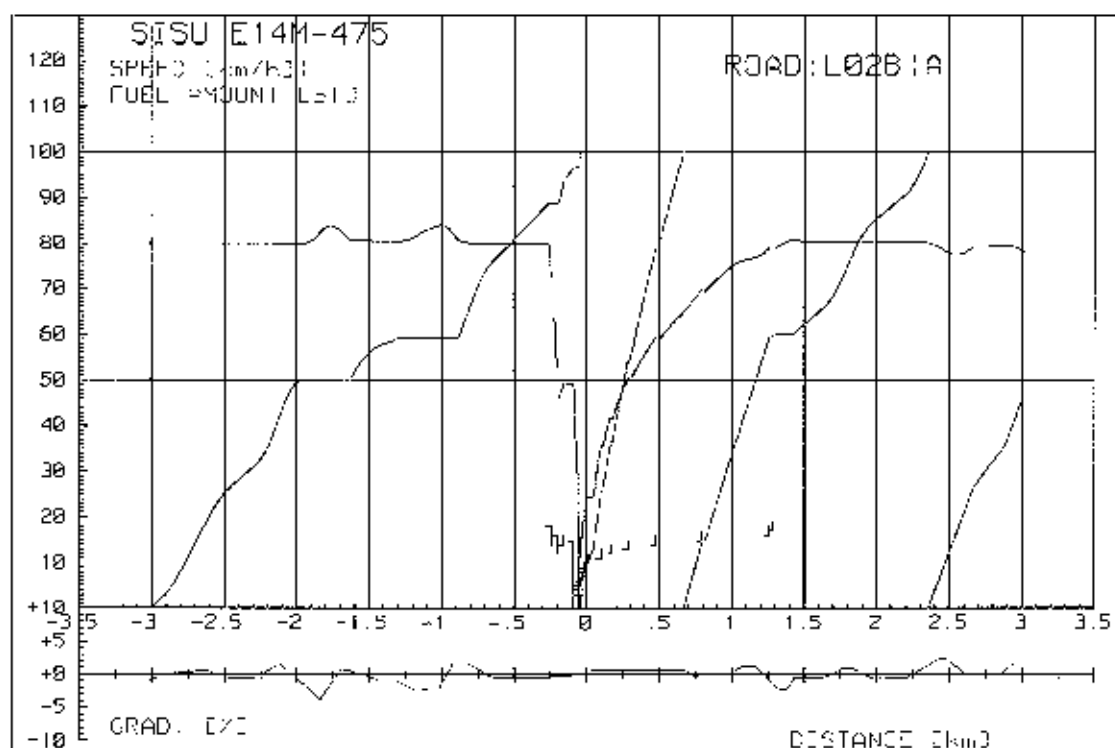


Figure 1 - An example of a drive simulation through a roundabout

Type vehicle: truck + trailer
 Route: South - North

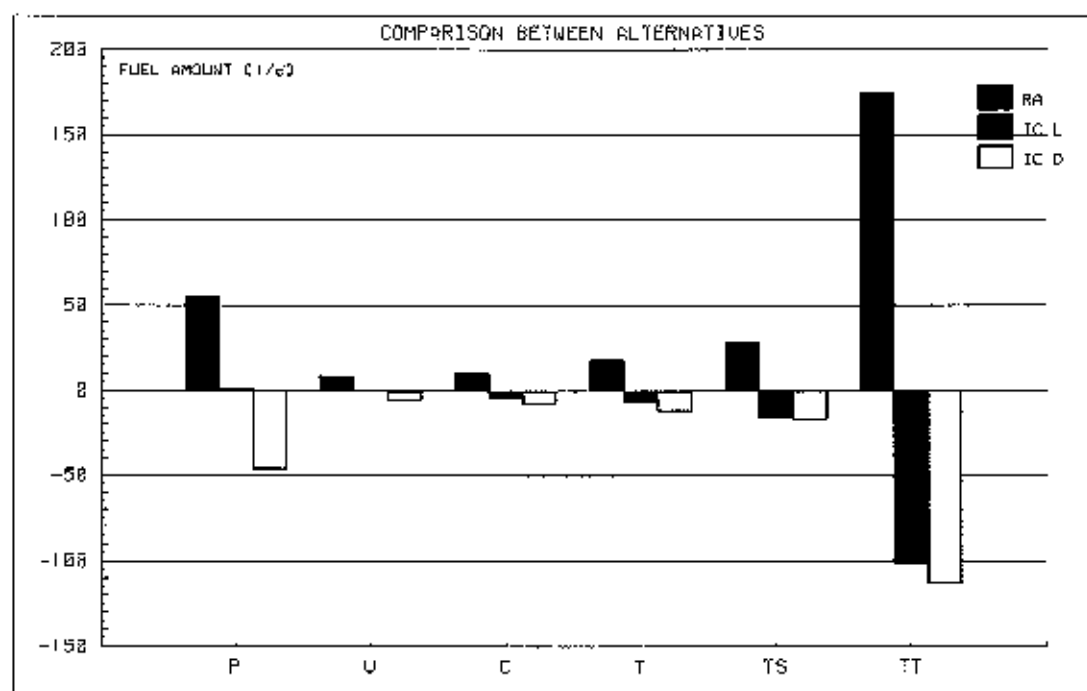


Figure 2 - Fuel amounts by vehicle categories at different junction types vs intersection

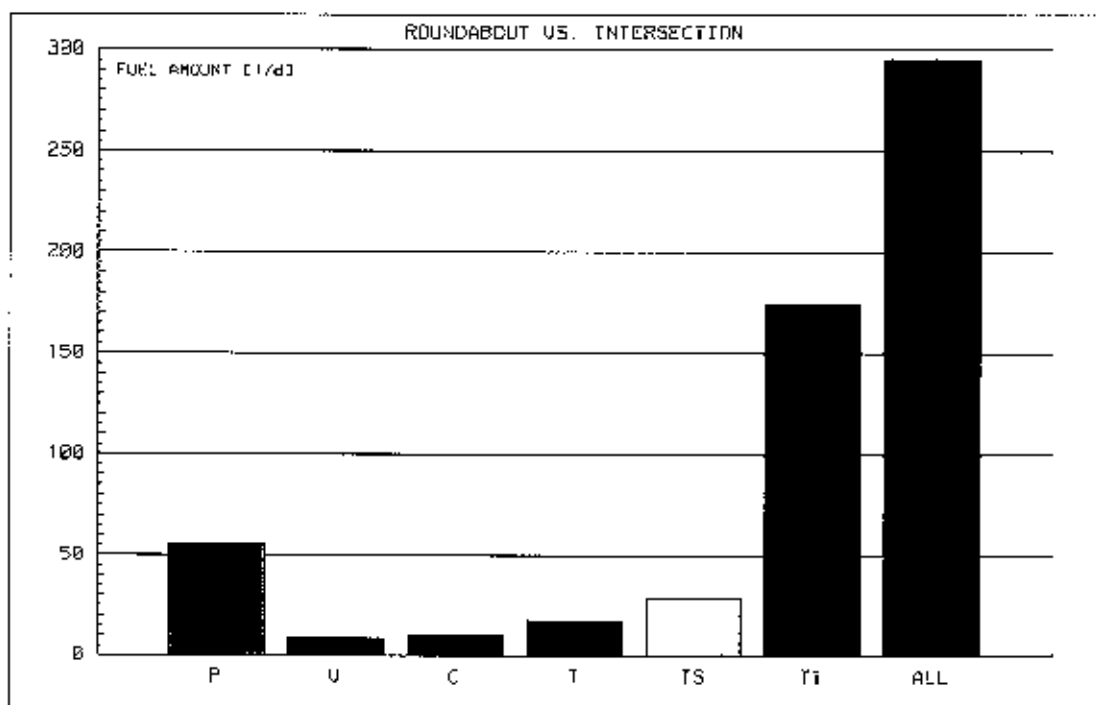


Figure 3 - Fuel amounts by vehicle categories at roundabout vs intersection

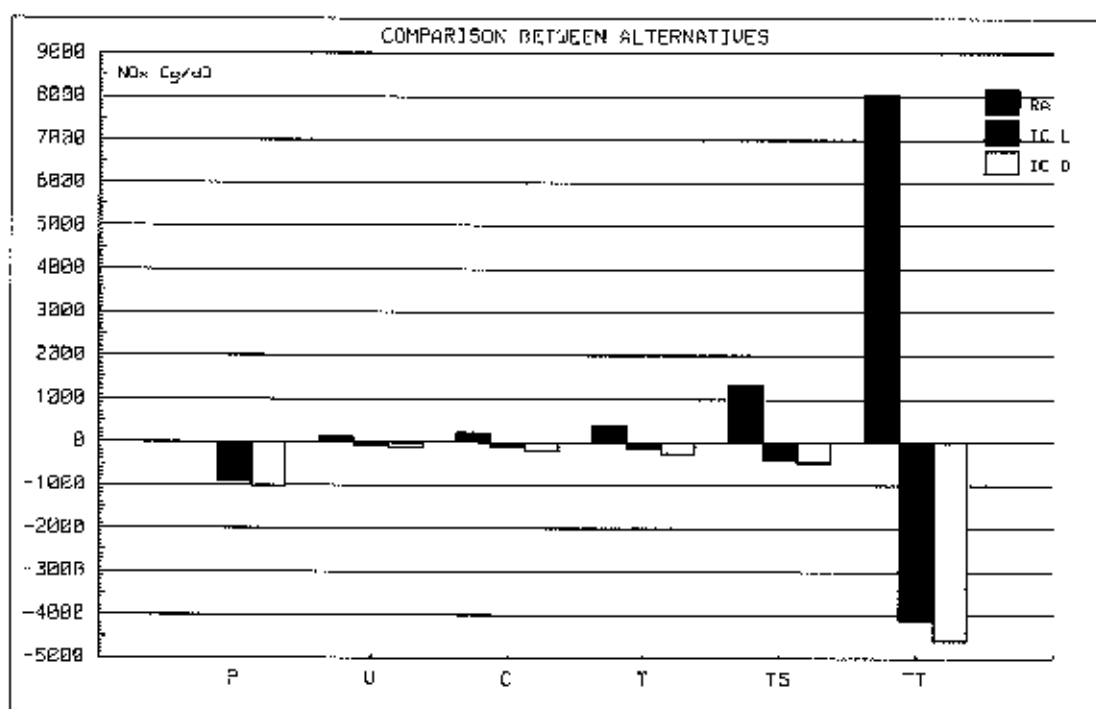


Figure 4 - NO_x amounts by vehicle categories at different junction types vs intersection

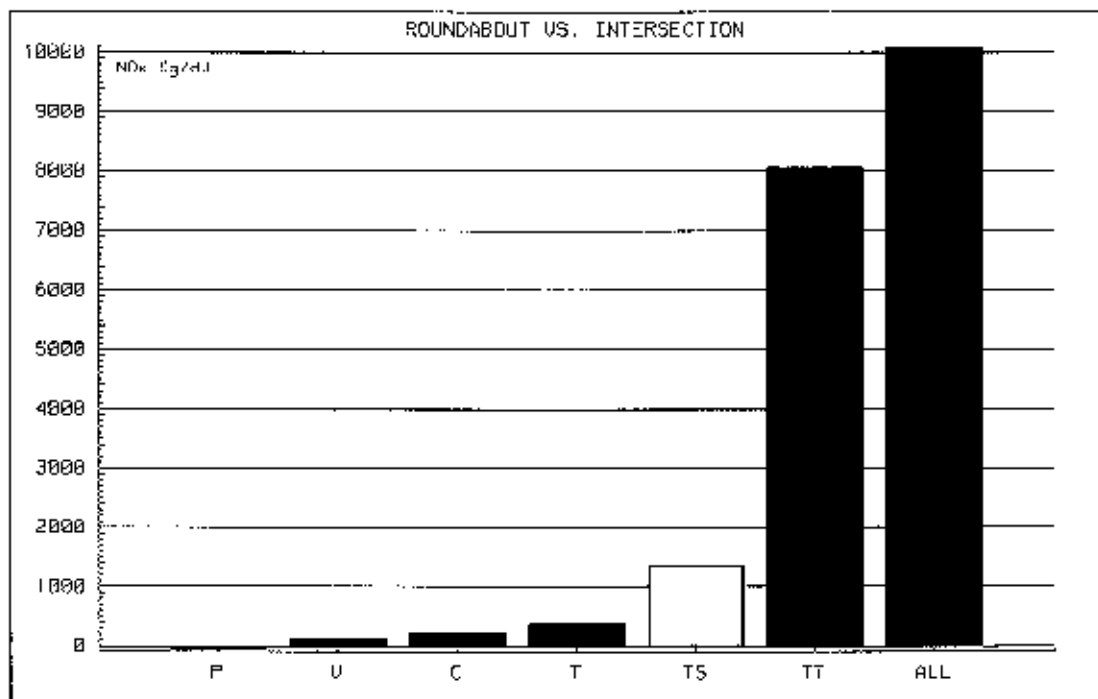


Figure 5 - NO_x amounts by vehicle categories at roundabout vs intersection

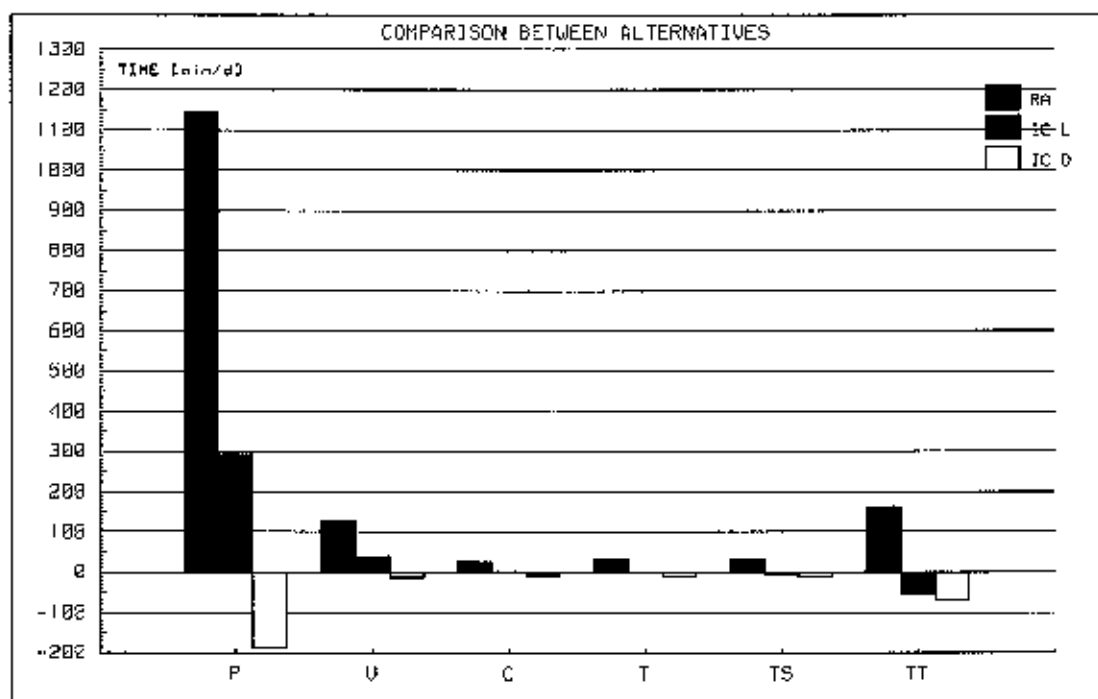


Figure 6 - Time consumed by vehicle categories at different junction types vs intersection

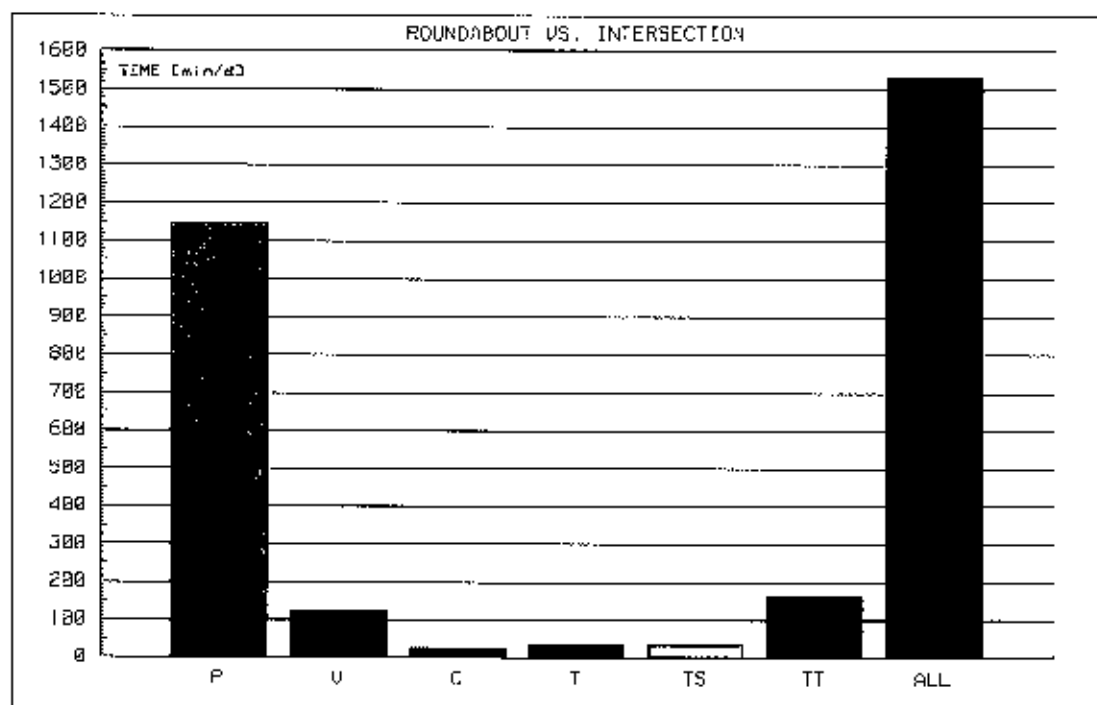


Figure 7 – Time consumed by vehicle categories and their sum at roundabout vs intersection

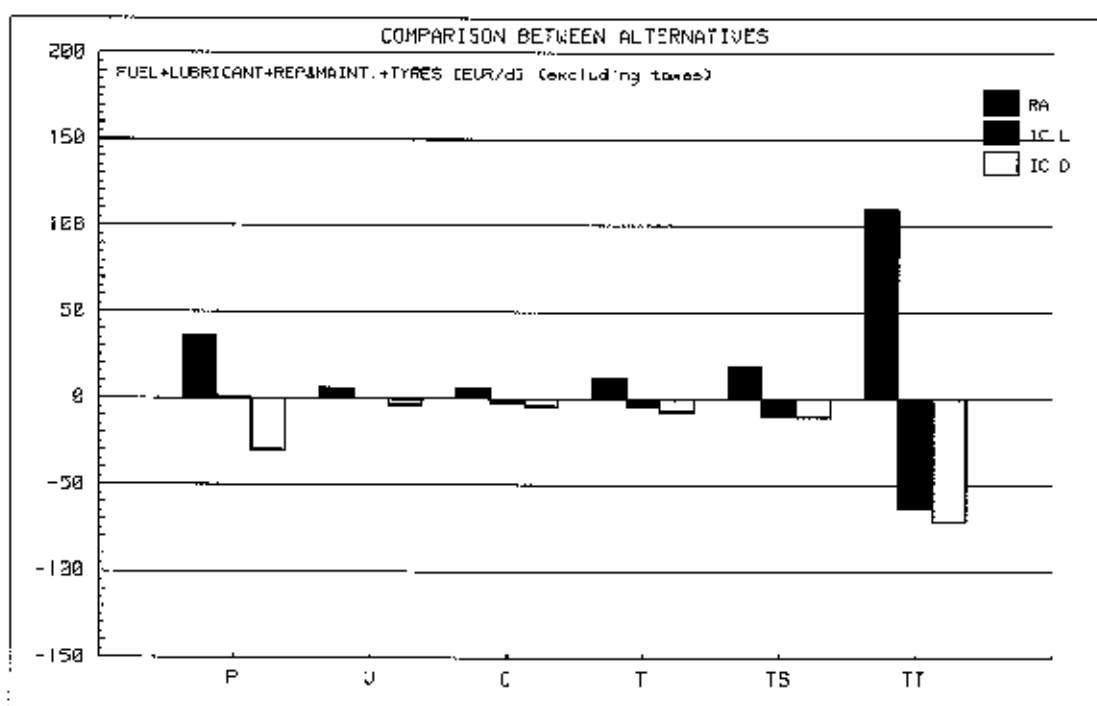


Figure 8 – Variable vehicle operating costs at production cost price by vehicle categories at different junction types vs intersection

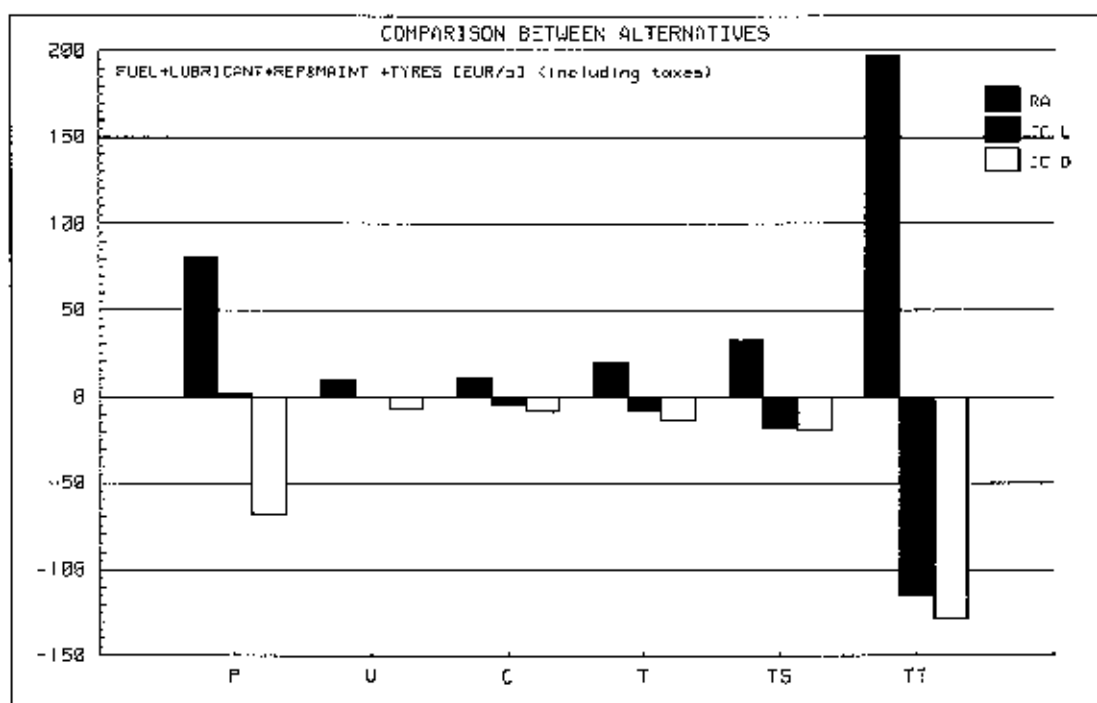


Figure 9 – Variable vehicle operating costs at market price by vehicle categories at different junction types vs intersection

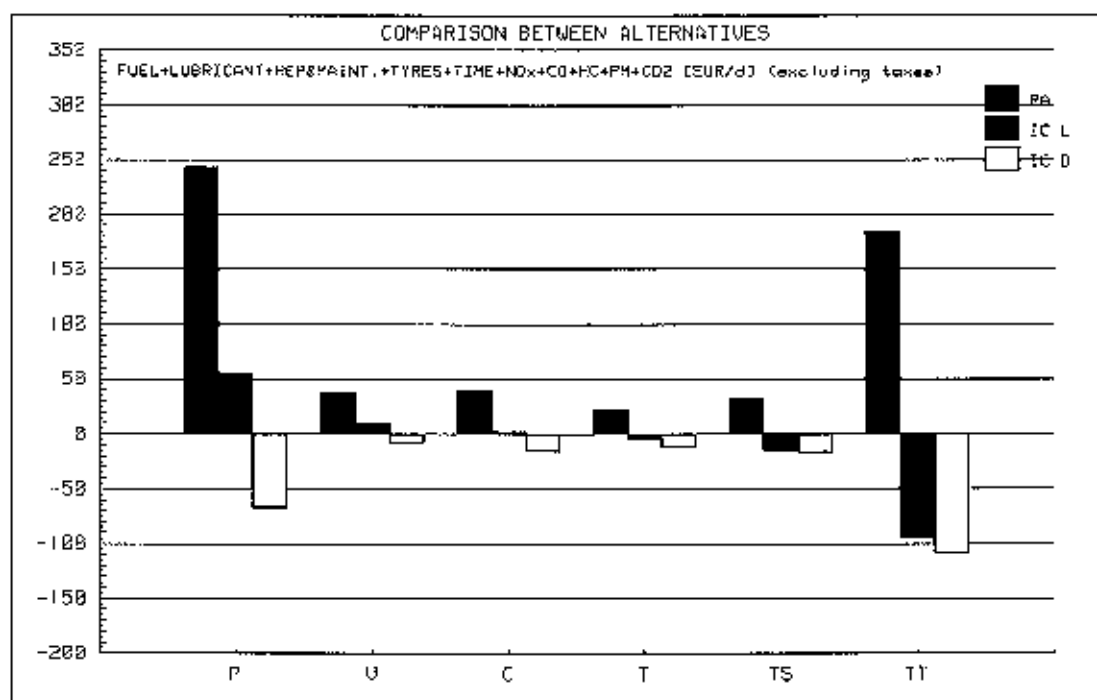


Figure 10 – Total costs at production cost price by vehicle categories at different junction types vs intersection

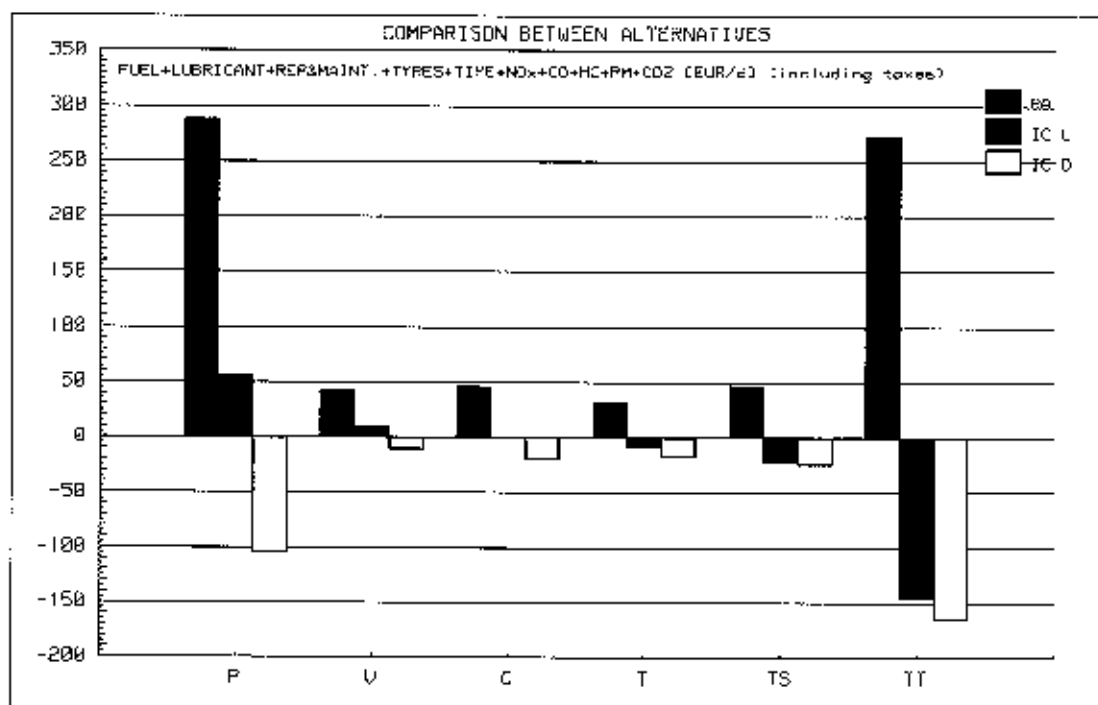


Figure 11 – Total costs at market price by vehicle categories at different junction types vs intersection

SASKATCHEWAN'S CENTRAL TIRE INFLATION SYSTEMS (CTIS) RESULTS FROM THE YEAR 2000 FIELD TRIAL

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ABSTRACT

This paper documents the results of the Central Tire Inflation System (CTIS) field experiments conducted in Saskatchewan in the summer of 2000 as a follow up to a field demonstration of CTIS technology conducted in the fall of 1999. A paper had been presented on the results from the 1999 field demonstration at the 6th International Symposium on Heavy Vehicle Weights and Dimensions held in Saskatoon, CANADA. The objective of the 2000 field experiment was to investigate the potential benefits of CTIS technology in more detail and specifically to determine to what extent the technology can offset incremental road damage when hauling in excess of regulated weights. The experiment used the same truck configurations as in the 1999 field demonstration trials. The research was sponsored by Saskatchewan Highways and Transportation, Manitoba Highways and Government Services, Agriculture and Agri-food Canada through the Canada Agri-Infrastructure Program (CAIP) and the Saskatchewan Wheat Pool. The authors wish to acknowledge and thank the Rural Municipality of Big Quill for allowing the use of their roads for the field experiment. The experiments were conducted in the vicinity of Wynyard, Saskatchewan and comprised of two separate test sites, one site to assess the benefit of CTIS for reducing road damage at equal gross vehicle and axle weights, and the other to assess whether the use of CTIS can offset any incremental damage caused by hauling the same tonnage at higher axle and gross vehicle weights than those specified by regulations.

1.0 INTRODUCTION

1.1 Background

In the fall of 1999, Saskatchewan Highways and Transportation conducted a field demonstration near Walpole, Saskatchewan to explore the difference in road damage that would occur from using trucks equipped with Central Tire Inflation Systems (CTIS)¹. The demonstration utilized fourteen 9-axle B-Trains, all loaded to the legal maximum primary axle weights allowed in Saskatchewan for a maximum gross vehicle weight of 70500 kg. Half of the trucks were operated at normal, highway (standard) tire pressures and the other half at optimised (reduced) tire pressures. The test trucks were run on two separate earth road test circuits joined by a common test road on which each group of 7 trucks passed each other in opposing lanes. Two modes of failure were investigated: washboarding and surface rutting. The results of the demonstration indicated that the use of optimised tire pressures reduced wash boarding by two-thirds. The rutting results were not conclusive; however, an analysis of the change in surface deflection, using data gathered before and after the demonstration, indicated that the strength of the surface crust on the standard-tire-pressure lane decreased significantly more than the opposing reduced-tire-pressure lane. This demonstration project is described in detail in a paper entitled "Field Demonstration Comparing Damage to Rural Saskatchewan Roads Caused by Optimised and Normal Highway Truck Tire Pressures" published in the proceedings from the 6th International Symposium on Heavy Vehicle Weights and Dimension. A review of related literature found no reference to the effect of variable tire pressures on damage to

¹ Central Tire Inflation Systems for commercial truck applications are often called Tire Pressure Control Systems.

low-volume roads constructed with a clay-capped or thin bitumen running surfaces, as are common in rural Saskatchewan.

1.2 Objectives

SHT offers an enhanced weight program to qualified trucking firms in the interest of improving truck efficiency and stimulating economic activity in Saskatchewan. In exchange for the granting of these weight privileges SHT requires enhancements in truck safety and operator qualification, payment for incremental pavement damage caused by the higher weights and a sharing of the haul cost savings from the granting of the weight privilege. SHT policy with respect to this program is that truck hauls utilizing CTIS technology do not cause incremental road damage and thus the company is not charged for any road damage. The basis for this policy position is research work done by the United States Forest Service, Forest Engineering Institute of Canada and undocumented research done in SHT Test Track facilities. The value of the benefits arising from the use of CTIS technology has been very controversial causing the results of the 1999 field demonstration project to be strongly debated. The stakeholder groups in Saskatchewan feel the extrapolation of results from research conducted in the United States and in the forestry industry were not applicable to Saskatchewan rural roads.

The significant reduction in the strength of the earth road surface crust caused by the trucks operating at normal tire pressures, as opposed to those operating at reduced tire pressure, prompted Saskatchewan Highways and Transportation (SHT) to conduct a second, more detailed, field experiment to further explore the potential benefits of CTIS in reducing road damage on rural roads. The purpose of the follow up research was two-fold:

- 1) to determine the magnitude of the benefit arising from the use of reduced tire pressures for equal axle loading conditions and
- 2) to determine whether the use of reduced tire pressures offsets the incremental road damage from using higher-than-regulated weights for an equal quantity of payloads hauled.

The work was conducted over a period of seven days in late July and early August of 2000. The details of this research will follow below.

1.3 Participants

The research was funded by SHT, the Canada-Saskatchewan Agri-Infrastructure Program (CAIP) with contributions in kind from Manitoba Highways and Government Services. The Saskatchewan Wheat Pool provided the trucks for the test. The Rural Municipality of Big Quill allowed the use of their roads to conduct the experiments. SOO Navigational Systems Inc. supplied and monitored the GPS-navigational systems used to track, in real-time, the movements of the vehicles over the test roads. Tire Pressure Control International Ltd. supplied the CTIS and provided technical assistance with the test including CTIS re-programming and gathering tire footprint measurements. Michelin (North America) approved the test tire pressures and evaluated the resultant tire heat build-up.

An advisory committee comprised of members from the Saskatchewan Association of Rural Communities, two Area Transportation Planning Committees, Saskatchewan Trucking Association, University of Regina, Saskatchewan Wheat Pool, the Forest Engineering Research Institute of Canada, Tire Pressure Control International Ltd., Manitoba Highways and Government Services, and the Saskatchewan Department of Highways & Transportation was established to provide advice and guidance for the 1999 CTIS Demonstration Project near Walpole, Saskatchewan. This committee was retained to monitor and advise on the progress of the research on the CTIS Field Experiments.

2.0 METHODOLOGY

The field study location was in the vicinity of Wynyard, Saskatchewan, Canada (see Figure 1). Two separate sets of earth roads were used to conduct the research: the Equal Axle Weight Experiment on Circuits 1 and 2 and the Equal Payload Experiment on Circuits 3 and 4 (see Figure 2). The two experiments were conducted simultaneously. The north-south roads common to Circuits 1 and 2, and Circuits 3 and 4 were the test sections. A

total of 17 test vehicles were used to conduct the experiments (ten 9-axle B-trains, four 8-axle B-trains and three 6-axle semi-trailers).

The test roads were low-volume earth roads constructed with the local clay-type soils of medium plasticity (Average Plasticity Index = 17.5, and Average Group Index = 8.3). The road surfaces were comprised of a compacted crust, 140–200 mm thick, overlain with a thin layer of gravel for wet weather traction. The pre-trial test road strengths were very similar, with the average surface deflection being 2.0 mm and 1.9 mm for the Equal Axle Weight Experiment test road and the Equal Payload Experiment test road, respectively. The principle difference between the test roads was running surface width—the width was approximately 8.0 m and 7.0 m for the Equal Axle Weight Experiment test road and the Equal Payload Experiment test road, respectively. The test roads were selected because they were representative of many rural roads in Saskatchewan, and because they were very similar in construction to the road used in the 1999 CTI demonstration project. Figure 3 illustrates a typical cross-section of the test roads.

Each test road included a thinly paved test section that was representative of the thin membrane surfaced (TMS) roads used as rural collector highways in Saskatchewan. Prior to paving the traction gravel was bladed away and the surface crust sprayed with an asphalt emulsion. A thin layer of bituminous cold mix (40 mm thick across the entire road surface) was placed in a 300 m-long section located at approximately the midpoint of each test road.

2.1 Equal Axle Weight Experiment

The purpose of this experiment was to determine the effect that reducing tire pressures will have on road damage, given identical gross vehicle and axle weight conditions.

Vehicle Description. The vehicles used in the Equal Axle Weight Experiment were five 9-axle B-Trains equipped with CTIS, air ride suspensions, 11R22.5 Michelin tires, and GPS-based navigational systems. Each vehicle was loaded with 5500 kg on the steer axle, 17000 kg on the tandem drive axles, and 24000 kg on each trailer trident axle group, for a Gross Combination Vehicle Weight (GCVW) of 70500 kg. The cold tire inflation pressures of the test vehicles cycling on Circuit 1 were reduced to 550 kPa (80 psi) in the steering tires, 414 kPa (60 psi) in the drive axle tires and 345 kPa (50 psi) in the trailer tires. All tires of the vehicles cycling on Circuit 2 were set to a normal, highway cold inflation pressure of 690 kPa (100 psi). The reduced tire pressure trucks cycled in the southbound lane of the test road and the standard tire pressure trucks cycled in the northbound lane. A description of the test vehicles is given in Figure 4.

Data Collection. The data collection on the test road consisted of taking samples for soil moisture and density, Atterberg Limits and the soil Group Index values as well as Dynamic Cone Penetrometer readings, surface rut measurements and surface deflections using a Benkelman beam. Data samples were taken before, during, and after trafficking. An analysis of the data collected indicated that the soil strength properties of the north and southbound lanes of the test road were the same. Relative changes in surface deflection were taken as a proxy for changes in strength of the road structure. Although deflections were gathered on the bituminous mix test sections, surface deflection within the bituminous mats confounded the results.

Dynamic Cone Penetrometer readings and Benkelman deflections were taken on all of the connecting roads. Benkelman beam readings were gathered on the test road two to three times every day of trafficking, at 400 m intervals on the earth sections and 100 m intervals on the bituminous mix sections. The readings were taken at approximately the same locations, however, variations in measurement location likely occurred and were accounted for by averaging all of the test road deflections together. Beam reading variability on the earth road was not very great; a limited sampling found a standard deviation of 0.05 mm for eight deflections taken on the same spot.

The test roads and connecting roads were videotaped during the experiment to record the test procedure and the road condition before, during and after trafficking.

Test Procedure. The five vehicles trafficked Circuits 1 and 2 on alternate days commencing with Circuit 2 (tire inflations set to standard tire pressures). The trucks cycled together and the lead truck maintained a pace of no more than 80 km/h. Every half-day (i.e., after approximately 70 passes) trafficking was halted while data was collected from the cycled lane. This alternating procedure was continued for the duration of the Equal Axle Weight Experiment.

Trafficking was continued on the test lanes until one of the two lanes failed. Once a test lane was considered failed, no more test trucks were cycled around the failed circuit. Trafficking was continued on the opposing lane until its average surface deflection was comparable to that of the failed lane (that is, it had approximately equivalent structural damage). Throughout the trafficking, video record was kept of the visual appearance of the test lanes.

2.2 Equal Payload Experiment

The purpose of this experiment was to determine whether a reduction in tire pressures would compensate for the incremental road damage caused by operating at higher axle weights. In this experiment, the two lanes of the test road were trafficked by five 9-axle B-trains, and a combination of four 8-axle B-trains and three 6-axle tractor/semi-trailers collectively carrying an equal payload to the five 9-axle B-trains. The resulting road damage in each lane was compared at given quantities of payload hauled.

Vehicle Description. Two groups of commercial trucks were used to conduct this experiment: five 9-axle B-trains (the high efficiency fleet), and a combination of four 8-axle B-trains and three 6-axle tractor/semi-trailers (the conventional fleet). All of the vehicles were equipped with CTIS, 11R22.5 Michelin tires, air ride suspensions and GPS-based navigational systems. Each high efficiency fleet truck was loaded to maximum legal primary axle weights—5500 kg on the steer axle, 17000 kg on the tandem drive axle group, and 24000 kg on the trailer tridem axle groups. The GCVW of the 9-axle B-trains was 70500 kg. The cold inflation pressures used by the high efficiency fleet were 550 kPa in the steering tires, 414 kPa in the drive tires, and 345 kPa in the trailer tires. Each of the conventional fleet trucks was loaded to maximum legal secondary axle weights—5500 kg on the steering axle, 14500 kg on tandem axle groups, and 20000 kg on tridem axle groups. The GCVW of the 8-axle B-trains and the 6-axle units was 54500 kg and 40000 kg, respectively. All of the 8 and 6-axle units utilized (standard) cold tire inflation pressures of 690 kPa. The high efficiency fleet cycled on the southbound lane of the test road whereas the conventional fleet cycled on the northbound lane. A description of the test vehicles used in the Equal Axle Weight Experiment is presented in Figure 5.

Data Collected. The data collected for the Equal Payload Experiment was identical to that of the Equal Axle Weight Experiment.

Test Procedure. The experiment was designed so that the combined payload hauled by the five high efficiency fleet trucks was approximately equal to that carried by the seven conventional fleet trucks. In a single cycle, the high efficiency fleet carried 236.5 t of payload whereas the conventional fleet carried 205.7 t of payload. Extra cycles were completed when necessary, by some or all of the conventional fleet, to keep the payload over the northbound lane of the test road equal to that over the southbound lane. The high efficiency fleet cycled around Circuit 3 while the conventional fleet cycled around Circuit 4. Cycling on Circuits 3 and 4 occurred simultaneously, with the fleet location coordinated so that only one fleet occupied the test road at a time. The trucks in each fleet cycled together and the lead truck maintained a pace of no more than 80 km/h. Trafficking was stopped twice daily to allow data to be collected from the test road.

Trafficking was continued on the test road until one of the two lanes failed. Once failed, no more test trucks were allowed to traffic that lane. Trafficking was continued on the opposing lane until its average surface deflection was comparable to that of the failed lane (that is, it had approximately equivalent structural damage). Throughout the trafficking, video record was kept of the visual appearance of the test lanes.

3.0 RESULTS

3.1 Equal Axle Weight Experiment (Circuits 1 and 2)

The northbound lane of the test road was declared failed after 200 passes (9350 t payload) by the trucks utilizing standard tire pressures. Cycling on the northbound lane had to stop because of the large shear areas that were threatening to damage or upset the test trucks. The southbound lane of the test road sustained 721 passes (33800 t payload) by the trucks utilizing reduced tire pressures before the experiment was stopped. The southbound lane never reached the level of damage observed in the northbound lane.

Comparison of Test Road Properties. A comparison of road properties was done prior to the commencement of testing to insure the test results were not biased by physical differences between the test lanes. Changes in road properties resulting from the application of the trucks onto the test road are documented below.

Comparison of Soil Properties. A comparison of the physical properties (e.g., moisture, density, plasticity (Atterberg Limits), and strength (Group Index)) of the subgrade soils in the north and southbound lanes prior to the experiment indicated no significant differences that might bias the test results.

Comparison of Benkelman Beam Deflections. The average Benkelman beam deflection values on the test road were monitored for a period of nine weeks (data collected every two weeks) prior to the commencement of the experiment. No significant differences in the average deflection values were observed between the north and southbound lanes of the test road (Figure 6).

At the start of the experiment, both lanes of the earth test section had an average deflection value of approximately 2.0 mm. After approximately 140 truck passes over each lane, the average deflection value observed on the northbound (standard tire pressure) lane was 4.7 mm whereas the average deflection value observed on the southbound lane was only 3.2 mm. After 200 truck passes, the average deflection of the northbound lane was 5.0 mm, the lane was declared impassable, and no further truck passes were made on it. After 200 passes, the average deflection on the southbound lane was 3.6 mm.

At the conclusion of the trafficking (i.e., 721 truck passes), the average deflection value of the southbound lane was 3.5 mm. This lane never reached an impassable condition. The average deflection observed on the southbound lane fluctuated from a low of 3.2 mm to a high of 4.4 mm during the test.

Comparison of Observed Road Damage. The northbound lane was declared impassable and cycling was stopped on this lane after 200 passes by the trucks utilizing standard tire pressures. The middle and northernmost sections of the northbound lane sustained the greatest amount of damage. It was estimated from the video records that 50% of the length of the northbound lane's earth section had sustained shear failure in the outer wheel path (Figure 7). The entire 300 m-length of bituminous mix completely failed in shear over, at least, half of its width (Figure 8). In many places, the shear failures resulted in the displacement of the outer wheel path crust by more than 30 cm. This raised concerns about damage to the test vehicles and led to the termination of the truck cycling on the lane.

After 200 passes by the trucks utilizing reduced tire pressures, the southbound lane sustained significantly less damage than the northbound lane. From the video record, it was estimated that only 7% of the earth section of the test road sustained shear failure in the outer wheel path. The video record also indicated that only 10% of the length of the bituminous section was observed to have failed in half the lane after the 200 truck passes. After 721 truck passes (33900 t payload), the video record indicated that approximately 10% of the length of the earth section and 40% of the length of the bituminous mix section of the test road had failed from the shearing of the outer wheel path.

3.2 Equal Payload Test (Circuits 3 and 4)

In this experiment, an effort was made to obtain the same payload on each lane of the test road over the duration of the testing. To accomplish this it was often necessary to stop cycling the trucks to allow for the repair of serious shear failures to prevent damage to the test vehicles. The requirement for subgrade repairs was far more extensive on the northbound lanes (cycled by the trucks utilizing standard tire pressures) than for the southbound lanes (cycled by the trucks utilizing reduced tire pressures). A total of 17970 t payload (380 truck passes), was hauled over the southbound lane and 13634 t payload (478 truck passes) was hauled over the northbound lane before the experiment was terminated.

Comparison of Test Road Properties. A comparison of the road properties was done prior to the commencement of the test to insure no bias existed in the test from differences in the physical properties of the road. The changes in the road properties resulting from the application of the trucks on to the test road are documented below.

Comparison of Soil Properties. A comparison of the moisture, density, plasticity (Atterberg Limits) and strength (Group Index) properties of the subgrade soils in the north and southbound lanes prior to the application of the trucks to the road indicated no significant differences to cause a skewing of the test results.

Comparison of Benkelman Beam Deflections. As was the case for the Equal Axle Weight test road the average Benkelman beam deflection values on the Equal Payload test road were similarly monitored for a period of nine weeks prior to the commencement of the experiment. No significant differences in the average deflection values were observed between the north and southbound lanes of the test road.

Prior to the start of the experiment, both lanes of the earth road section had an average deflection value of approximately 1.88 mm (Figure 9). After approximately 3400 t payload had been carried over each lane, the average deflection value observed on the northbound (conventional fleet) increased to 4.0 mm whereas the average deflection value on the southbound lane (high efficiency fleet) increased to only 3.2 mm. As the payloads continued to increase, the average deflection values ranged between from 3.73 mm to 4.61 mm on the northbound lane and ranged from a low of 3.70 mm and a high of 4.05mm on the southbound lane. Deflection measurements continued to be taken until payloads of 13634 t and 17970 t had been carried over the north and southbound lanes, respectively.

Comparison of Observed Road Damage. A review of the video record indicated that shear failures started to appear on the earth section of the northbound lane of the test road after the movement of approximately 3500 t payload. No shear failures were observed on the southbound lane at this payload. At a payload of 10300 t, shear failures were observed on 40% of the length of the northbound lane compared to 6% of the length of the southbound lane (Figure 10). After 10000 t payload had been moved, significant repair effort was necessary on the northbound lane in order to continue trafficking. After 13634 t payload, the poor condition of the northbound lane, and the extensive effort and delays required to prolong trafficking, resulted in the termination of the experiment. At the end of the experiment, 10% of the southbound lane length and 60% of the northbound lane length had sustained shear failures in the outer wheel path.

The video record indicated that failure of the bituminous mix section on the northbound lane of the test road began after the movement of approximately 200 t payload. Failure of the bituminous mix surface on the southbound lane progressed more slowly with first indications of damage appearing after approximately 500 t payload. After approximately 1000 t payload had been moved over the bituminous mix surface, shear failures were observed on 5% of the length on the southbound lane compared to approximately 50% of the length of the northbound lane. After approximately 13000 t payload had been moved, both lanes of the bituminous mix surface had completely failed.

4.0 DISCUSSION ON THE TEST RESULTS

4.1 Effect of Reduced Tire Pressure on Road Performance

The results from both experiments clearly indicate that damage to earth roads and thin bituminous surfaces can be reduced with the use of CTIS. For the 9-axle trucks tested at common axle weights, the effect of using optimised tire pressures was to reduce road damage by an order of magnitude (i.e., 1/10 the amount observed for the standard tire pressure trucks). It was also noted that, for equal payloads hauled, the high efficiency fleet (operating at enhanced weights and reduced tire pressures) did less road damage than the conventional fleet (operating at regulation weights and standard tire pressures). In both experiments, the use of reduced tire pressure significantly reduced road damage for an equal quantity of payload hauled.

4.2 Other Factors Observed to Affect Road Performance.

Road Width. The Equal Axle Weight test road was approximately 0.6 m wider than the Equal Payload test road. A comparison of surface deflections versus payload on the two southbound test lanes (lanes cycled by identically configured 9-axle trucks operating at reduced tire pressure) suggests that width did not have a large effect for these trucks (Figure 12). A similar conclusion can be drawn from a comparison of the road damage versus payload (Figure 13).

On the connecting roads used to complete the loading cycles the test vehicles typically drove down the centre part of the road because of the narrowness of the running surface (typically 7.4 meters in width). It is interesting to note that the video record from these roads indicates little or no damage done by either the reduced or standard tire pressure trucks. This observation would suggest that road performance is related to the distance between the outer wheel path track and the edge of the road surface. This distance was 0.5 - 0.9 m for the Equal Axle Weight experiment and 0.3 - 0.6 m for the Equal Payload experiment. (The range of distances is due to the drivers steering closer to centreline as the outer edge of the lanes failed and the road's 4% cross slope causing the trailers to track towards the road edge.) In the case of the connecting roads, the distance was in the order of 2.3 m. This suggests that, for the conditions tested, outer wheel path shearing can be reduced if an outer wheel path-to-edge of road offset distance of greater than 1.0 m exists.

Damage to the connecting roads also may have been reduced because they followed well-established, densified wheel paths in the centre of the road and, by straddling centreline, they remained level instead of leaning with the cross slope of the surface. Based on measurements made with the same test trucks during the 1999 CTI Demonstration, the outer wheel loads were increased by approximately 13% during these trials when driving on test lanes having a 4% cross slope.

Vehicle Configuration. Comparisons of the average surface deflections, and of the visual estimates of damage, for the two northbound test lanes indicate a significant difference in road performance. This suggests that, for an equivalent payload, the five 9-axle units operating at enhanced axle weights and standard tire pressures did more damage to an earth road than the conventional trucks operating at regulation axle weights and standard tire pressures (Figures 14 and 15). Assuming road width did not affect road performance, as discussed above, this observation supports the Highways & Transportation Department's ESAL damage model that predicts, for equal payloads hauled, trucks operating at higher axle weights will cause approximately 16% more road damage. The Department's ESAL model does not directly account for differences in tire inflation, however, and was not suitable for estimating damage potential of the test trucks utilising reduced tire pressures.

Accelerated Testing. The experiments were completed in an accelerated fashion on account of time and budget considerations. The effect of accelerated testing is uncertain however it is believed that failure is accelerated because of the insufficient time provided for pore water pressure to stabilize prior to the application of the next loading cycle. This effect is not considered to be critical in the comparison of the results from the experiments given that the experiments were all carried out in an accelerated environment however careful consideration needs to be given to the application of the results to actual in-service conditions to adjust for the acceleration effects that are built into the results.

The effect of accelerated testing also needs to be considered in the context of a road's exposure to seasonal strength variations and the effect this will have on the pavement components. This is particularly important in northern countries that are reliant on frost susceptible glacial lacustrine soil deposits in the construction of road subgrades.

5.0 CONCLUSIONS

The purpose of the CTI Road Experiments was two-fold:

- 1) to determine the magnitude of the benefit arising from the use of reduced tire pressures for equal gross vehicle and axle loading conditions and
- 2) to determine whether the use of reduced tire pressures offsets the incremental road damage caused by using higher-than-regulated weights for an equal quantity of payload hauled.

The results from both experiments clearly indicate that damage to the earth roads and thin bituminous surfaces tested was significantly reduced with the use of reduced tire pressures – as controlled by a CTIS. It was also noted that, for equivalent payloads carried, the high efficiency fleet (operating at enhanced weights and reduced tire pressures) did less road damage than the conventional fleet (operating at regulation weights and standard tire pressures). In both experiments, the use of reduced tire pressure significantly reduced road damage for an equal quantity of payload hauled. The rate of damage observed on the test roads did not appear to be influenced by the 0.6 m difference in test road widths, however, road damage was dramatically reduced when the offset from outer wheel path to edge of road exceeded 1.0 m. The results from the lanes trafficked by test vehicles using standard tire pressures appear to support the ESAL damage model for equal payload hauled.

The test road, subgrade soil types, 9-axle test vehicles, and test procedures were common to both the Equal Axle Weight Experiment and the 1999 CTI Field Demonstration. However, because of seasonal differences the CTI Field Demonstration test road was dryer and stronger, and the resultant differences in observed road performance were significantly less.

Opportunities for further research with the data collected from these trials include:

- an analysis of pre- and post-trial deflection data to determine if the results can be extrapolated to other seasons in a meaningful way;
- an analysis of deflection data taken in 2001 and 2002 to determine the rate of damage recovery in the earth road as it is subjected to climatic processes;
- analyses of rutting and gravel loss rates to estimate potential surface maintenance benefits; and,
- extrapolation of the results to other road types and travel speeds using pavement-road performance models calibrated with the data.

6.0 REFERENCES

"Field Demonstration Comparing Damage to Rural Saskatchewan Roads Caused by Optimised and Normal Highway Truck Tire Pressures" in Proceedings from the 6th International Symposium on Heavy Vehicle Weights and Dimension – Past, Present, Future. International Forum for Road Transport Technology, Regina, Saskatchewan. June 2000. pp 237-251.

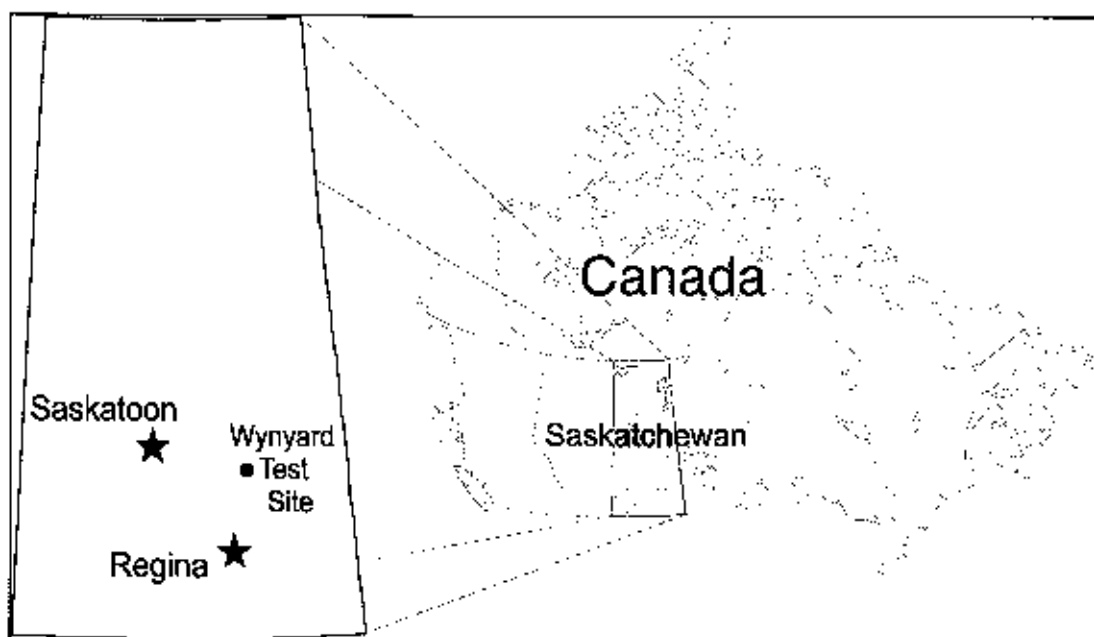


Figure 1. Location of CTI Road Experiment near Wynyard, Saskatchewan, Canada.

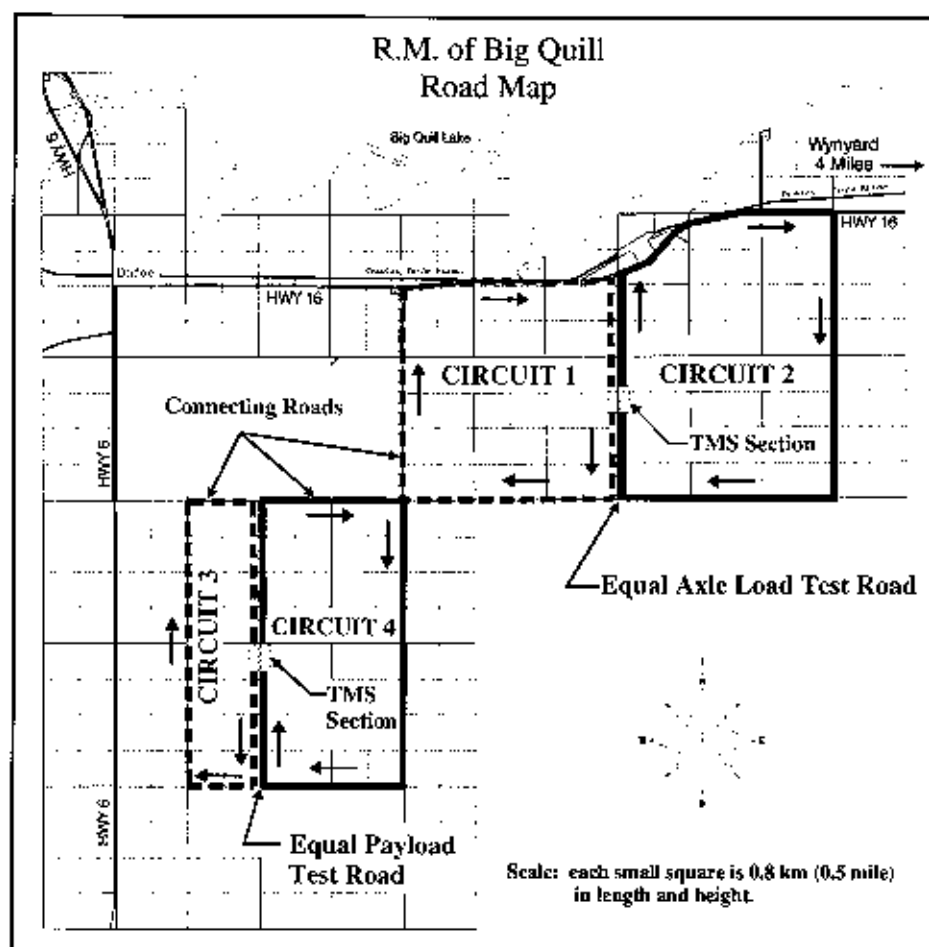


Figure 2. Equal Axle Weight Experiment (Circuits 1 and 2) and Equal Payload Experiment (Circuits 3 and 4).

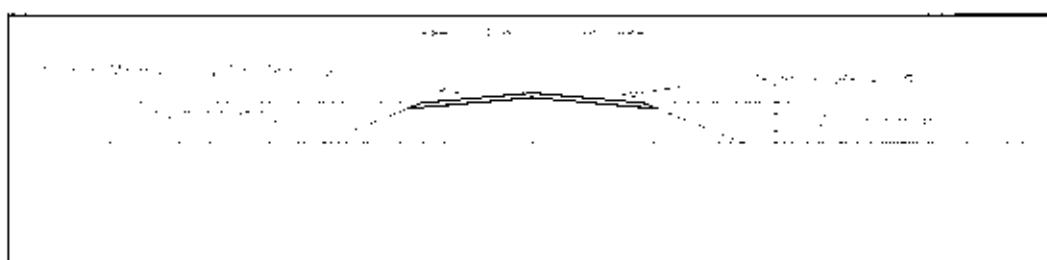


Figure 3. Typical test road cross-section.

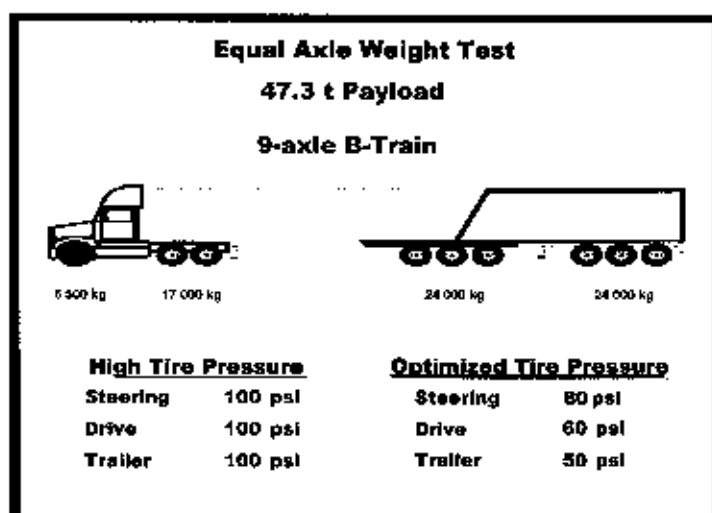


Figure 4. Test vehicle used for the Equal Axle Weight Experiment.

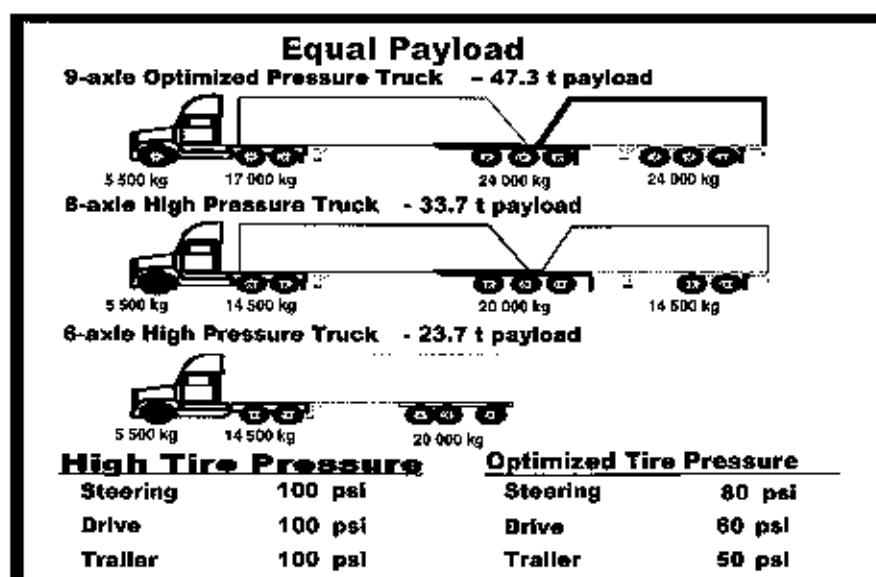


Figure 5. Test vehicles used for the Equal Payload Experiment.

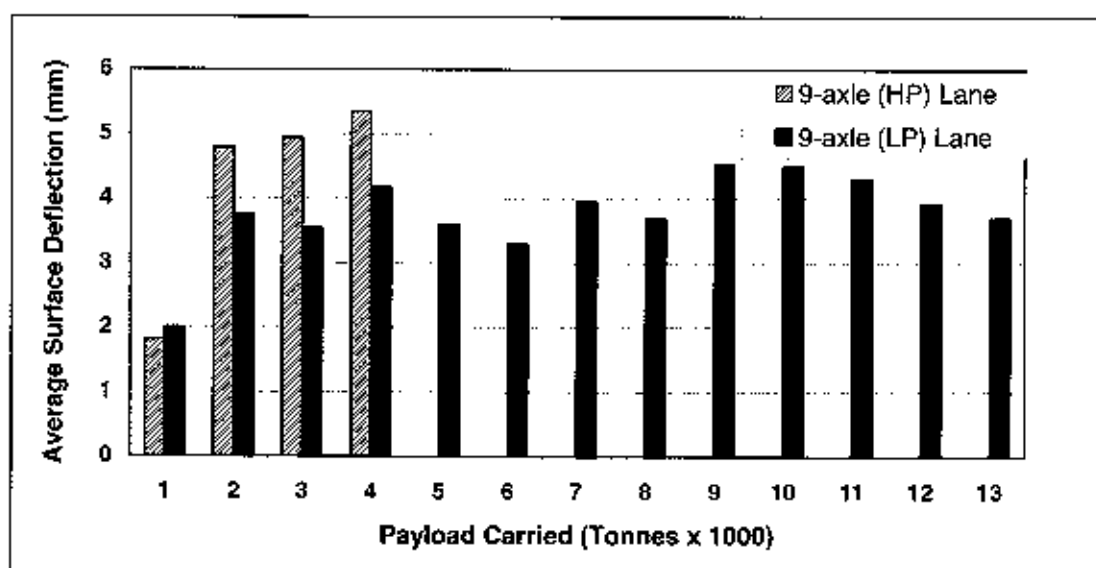


Figure 6. Comparison of the average surface deflections of the reduced tire pressure (LP) and standard tire pressure (HP) test lanes on the earth section of the Equal Axle Weight test road.

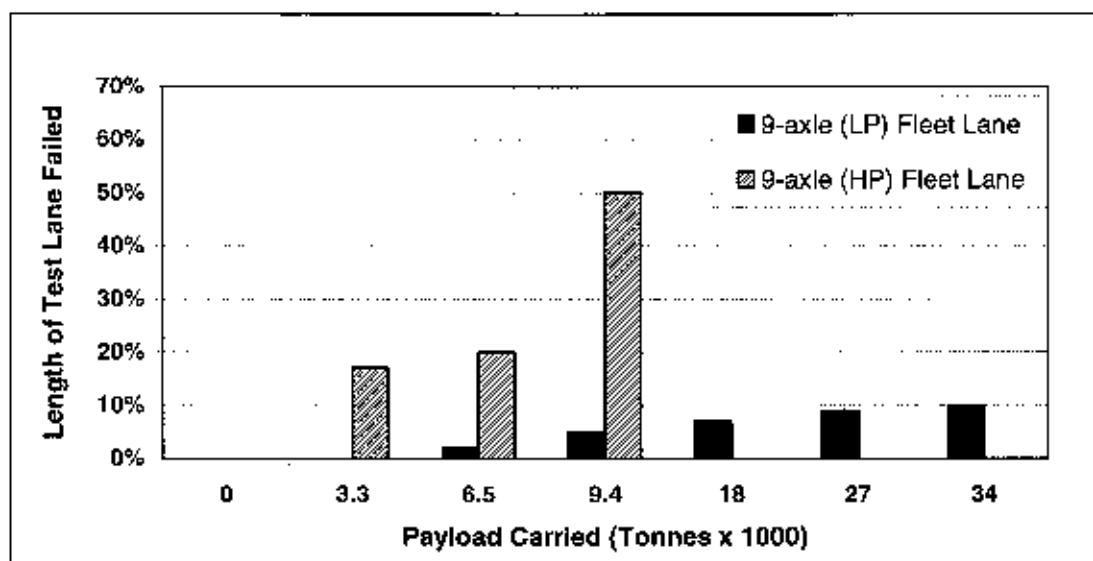


Figure 7. Comparison of the failed length of the reduced tire pressure (LP) and standard tire pressure (HP) test lanes on the earth section of the Equal Axle Weight test road.

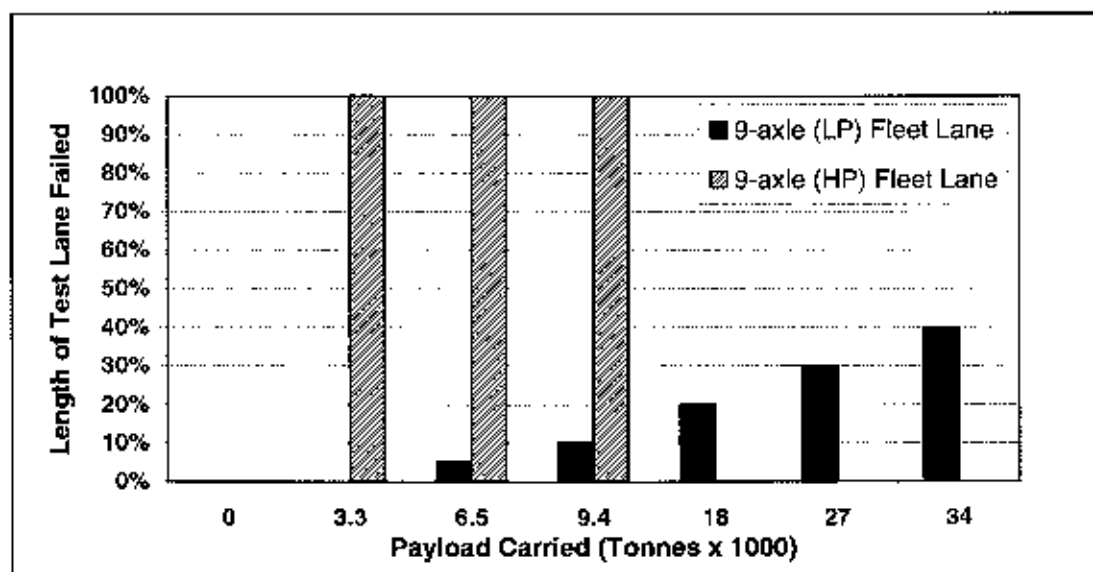


Figure 8. Comparison of the failed length of the reduced tire pressure (LP) and standard tire pressure (HP) test lanes on the bituminous mix section of the Equal Axle Weight test road.

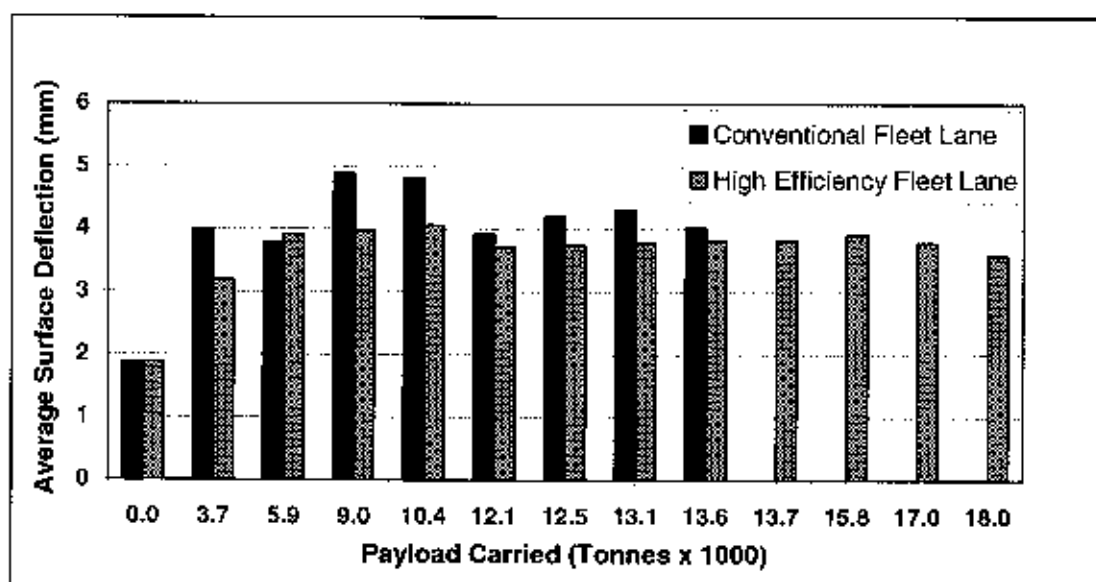


Figure 9. Comparison of the average surface deflections of the high efficiency fleet and conventional fleet test lanes on the earth section of the Equal Payload test road.

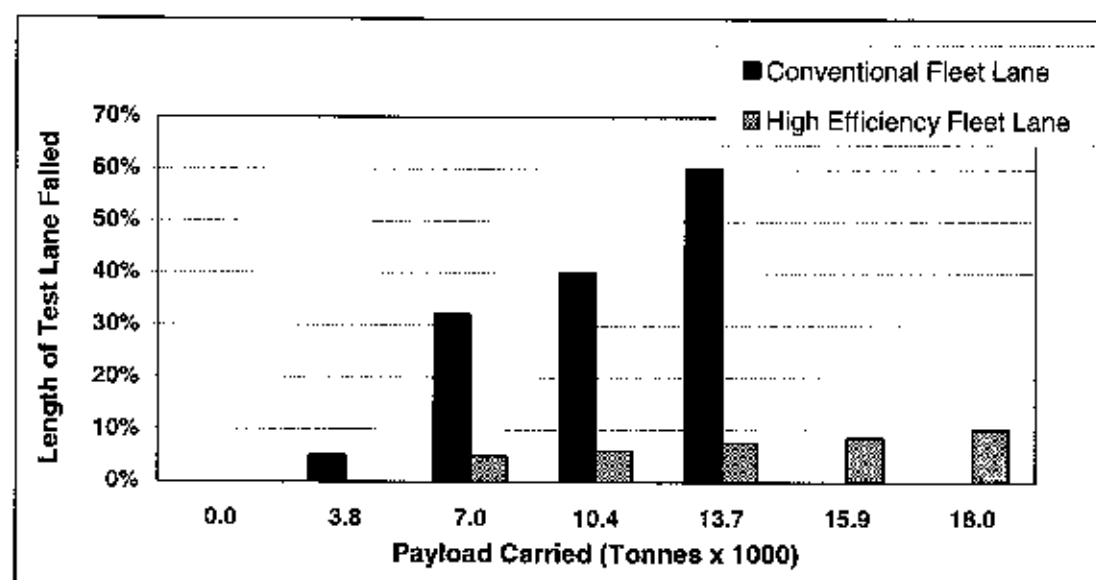


Figure 10. Comparison of the failed length of the high efficiency fleet and conventional fleet test lanes on the earth section of the Equal Payload test road.

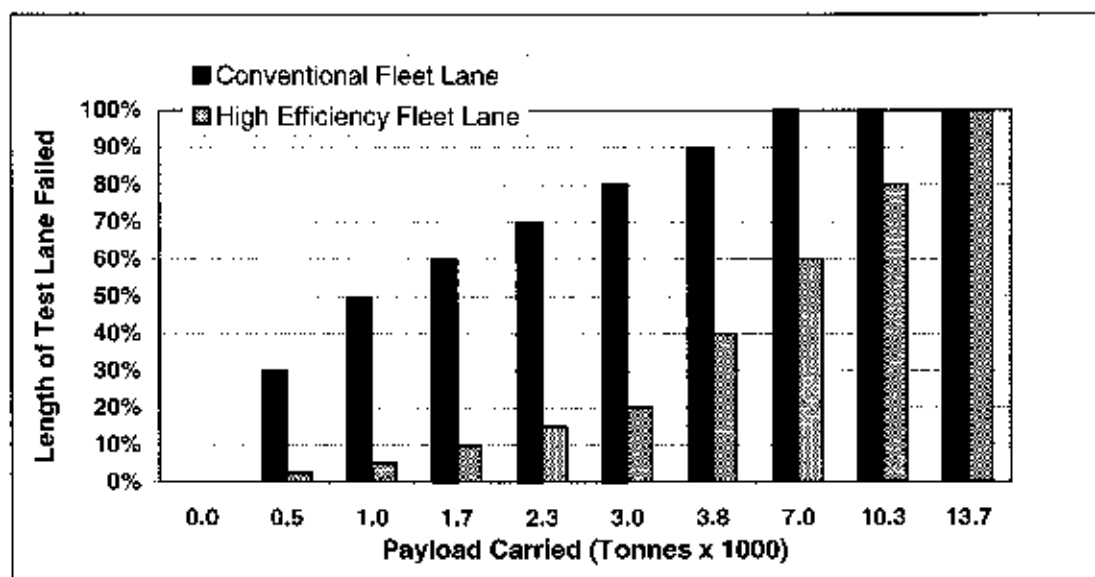


Figure 11. Comparison of the failed length of the high efficiency fleet and conventional fleet test lanes on the bituminous mix section of the Equal Payload test road.

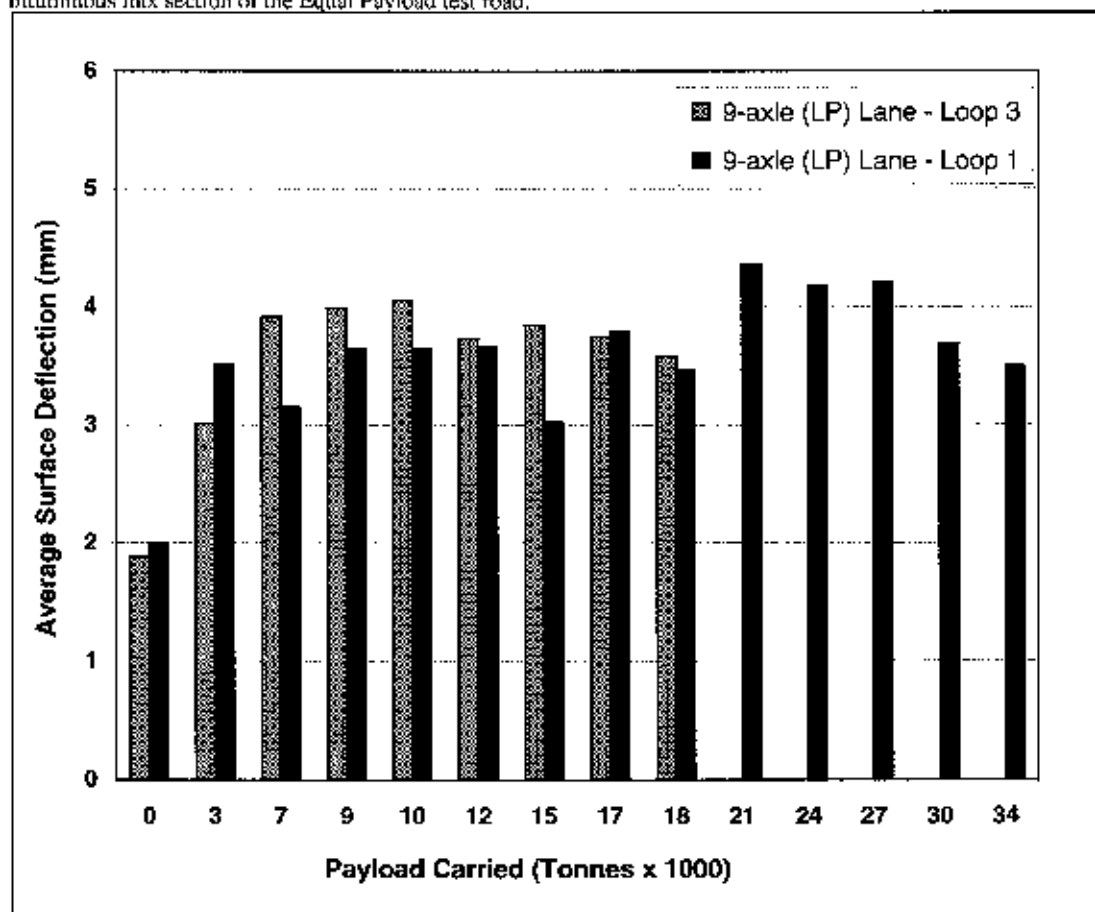


Figure 12. Comparison of the average surface deflections of the two earth road test lanes subjected to 9-axle B-train trucks loaded to primary highway (enhanced) axle weights and utilizing reduced tire pressures (LP).

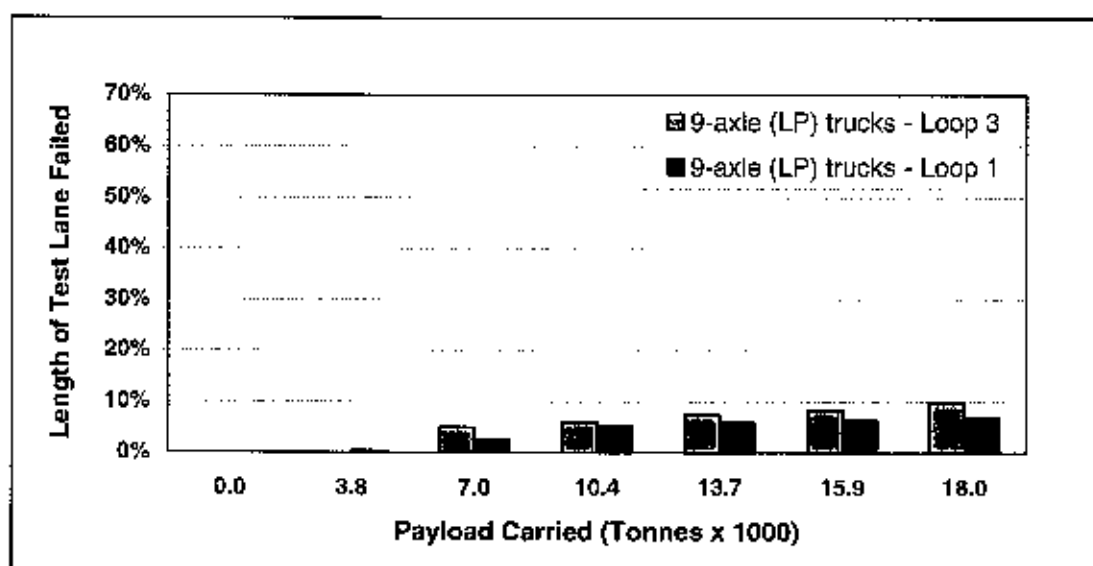


Figure 13. Comparison of the failed lengths of the two earth road test lanes subjected to 9-axis B-train trucks loaded to primary highway (enhanced) axle weights and utilizing reduced tire pressures (LP).

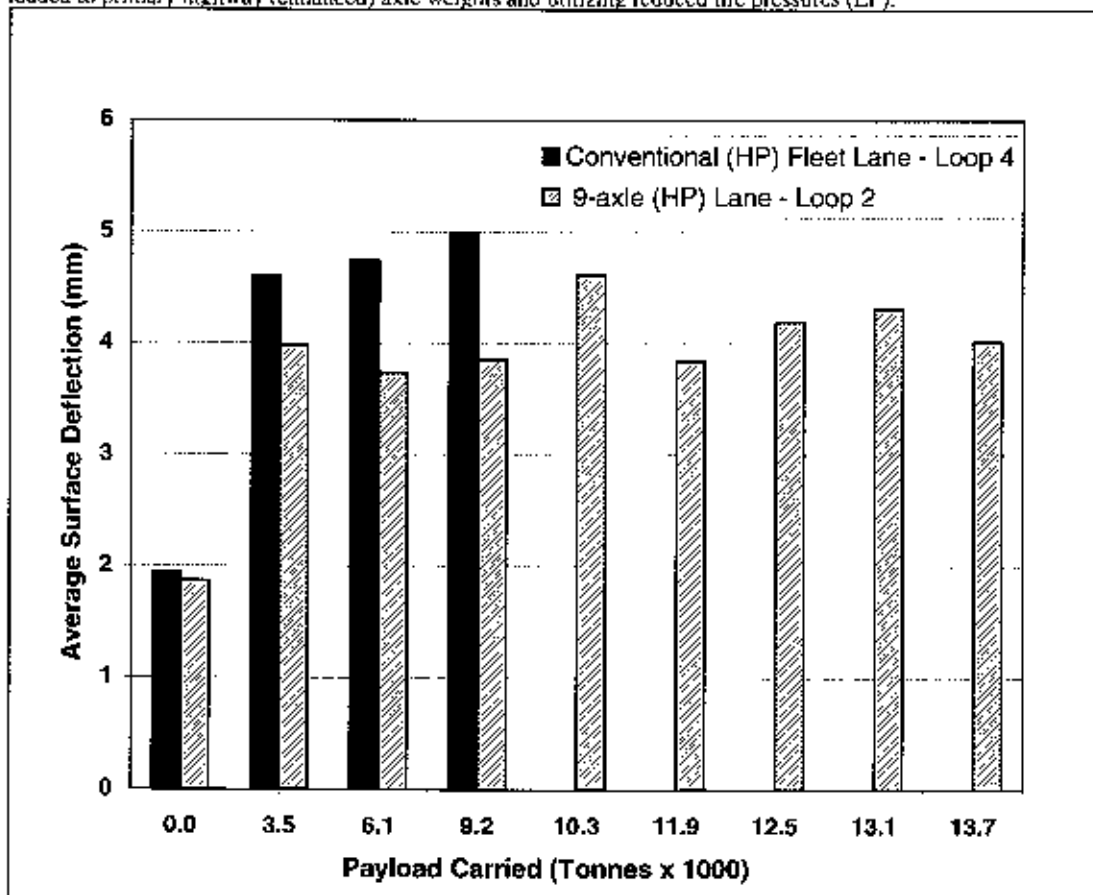


Figure 14. Comparison of the average surface deflections of the two earth road test lanes subjected to trucks utilizing standard tire pressures: Conventional 6 and 8-axis trucks loaded to Secondary Highway Axle weights and 9-axis B-train trucks loaded to Primary Highway Axle weights.

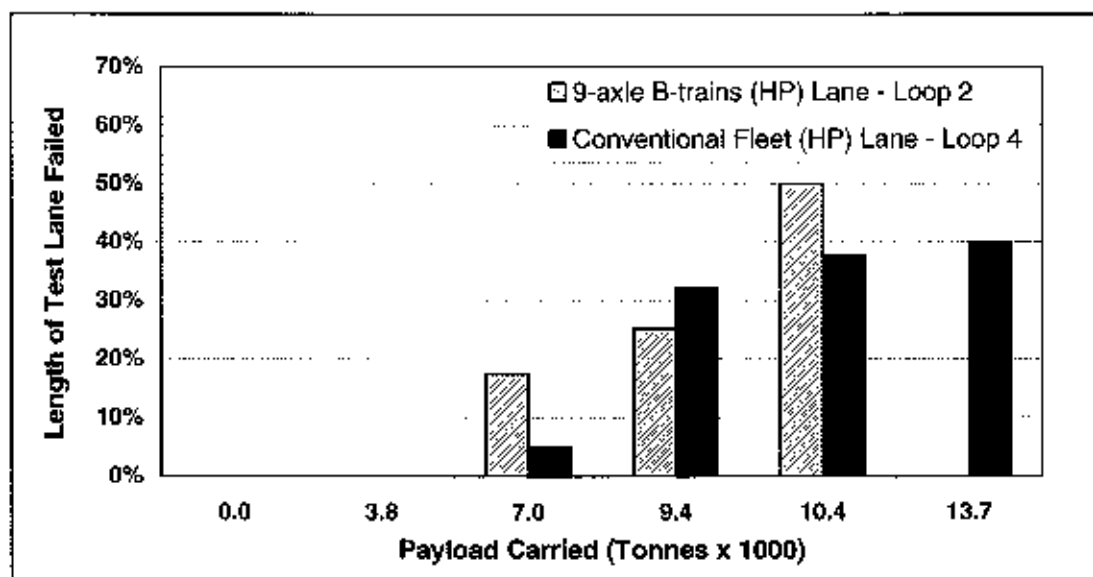


Figure 15. Comparison of the failed lengths of the two earth road test lanes subjected to trucks utilizing standard tire pressures: 9-axle B-train trucks loaded to Primary Highway Axle weights and conventional 6 and 8-axle trucks loaded to Secondary Highway Axle weights.

ROAD ROUGHNESS AND IT'S EFFECTS ON THE INFRASTRUCTURE

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ABSTRACT

The intention of this paper is to shed light on the interaction between road roughness and the resulting road damage caused by static and dynamic wheel forces. For this purpose several trucks have been mathematically modeled as multi-body systems in great detail. The models are based on actual data of the truck industry, taking concern of geometric details as well as non-linear behavior, such as Coulomb damping in leaf springs and characteristic curves of springs and dampers.

In order to describe the relation between road roughness and its effects on road damage in a mathematical manner the first chapter of this paper deals with the description of the road in terms of the power spectral density. Assuming that the road damage depends on the fourth power of the instantaneous wheel force, and the wheel force itself is distributed Gaussian, a relationship to the road roughness can be found. Based on this mathematical approach an easy to use formula has been deduced that allows to relate road roughness data to road damage.

In order to prove, whether and under which circumstances the above mentioned formula can give a realistic estimate of road loading the previously described complex vehicle models have been used to simulate test rides over a variety of different road surfaces. The results of these runs gave the necessary information to feed the formula, especially concerning the dynamic characteristics of the single axles and axle configurations. They showed that despite of the complex and clearly non-linear behavior of the vehicles the amount of dynamic loading on the road as well as the road damage can be estimated by a rather simple formula based on the theory of linear vibrations. This formula even takes into account that there might be distinct periodic phenomena contained in the road profiles.

DESCRIPTION OF ROAD ROUGHNESS

The power spectral density (PSD) proved to be a good means to describe the longitudinal profile of a road. It allows to find a mathematical relationship between road roughness and the vibration response of vehicles moving along that road. For our purposes the displacement PSD is used, i. e. the PSD of the vertical displacement of the road profile. It is the result of a Fourier Transformation and proportional to the square of the Fourier Transform. For those who are used to the spectral presentation the PSD gives a good impression about the roughness characteristic of a road, for instance whether rather the short or the long waves are dominant in the surface. Fig. 1 shows a simplified example of a road's PSD. On the abscissa we find the 'spatial angular frequency' (2π divided by the wavelength), so we find the big wavelengths (up to 100 m) to the left and the small ones (down to 0,2 m) to the right of the graph. On the ordinate we have got the PSD. It is proportional to the square of the corresponding amplitudes. According to the graph we can conclude: the road surface includes large wavelengths with high amplitudes and small wavelengths with low amplitudes. This is typical for roads. Indeed, plotted in a log-log scale, the most roads exhibit a displacement power spectrum that can be described by a straight line having a negative slope. This line again can be described by two parameters [1]:

- the PSD corresponding to a spatial angular frequency of $\Omega_0 = 1 \text{ m}^{-1}$ (i. e. a wavelength of 6,28 m) and
- the negative slope of the line

The first parameter is called the 'spectral roughness index' and shall be denoted 'RI'. It is proportional to the roughness. The higher the RI, the higher the straight line representing the PSD. $RI = 1 \text{ cm}^3$ is typical for a good and $RI = 10 \text{ cm}^3$ typical for a bad highway. The second parameter is called the 'waviness', w , and describes the frequency characteristic of the road. For instance, a low waviness ($w = 1,5$) stands for flat spectrum and so for relatively small amplitudes in the long wavelength range (left side of the graph) and relatively large amplitudes in

the short wavelength range (right side of the graph). Such a road in a whole seems very flat, but exhibits noticeable short-wave deflections. It causes vehicle vibrations at higher frequencies, for instance axle hop (10 Hz). A road with a high waviness ($w = 3$) on the other hand has got a steeply decreasing spectrum, which shows relatively large amplitudes in the long-wavelength and relatively small amplitudes in the short-wavelength range. Such a road in a whole seems uneven, but when looked at it in detail (for instance a short part of 5 by 5 meters) it is rather even. It causes vehicle vibrations at lower frequencies, for instance body bounce (1 – 2 Hz).

Having the two spectral indices, RI and w , we can easily generate road profiles by performing the Inverse Fourier Transformation. Fig. 2 (bottom) shows the profiles of a 400-meter-segment in the left and right wheel track of a truck (track width: 2 m). In order to generate the profiles the coherence function between left and right wheel track has to be considered. Fig. 2 (top) displays the corresponding two power spectral densities. From the graph you can easily identify an RI of $1 \text{ m}^6 = 1 \text{ cm}^3$ and a waviness of $w = 2$. Being able to generate roads with exactly defined spectral properties is helpful, when proving the relationship between road roughness and dynamic wheel loads by performing calculations in the time domain. This is being done later by using rather complex nonlinear truck models.

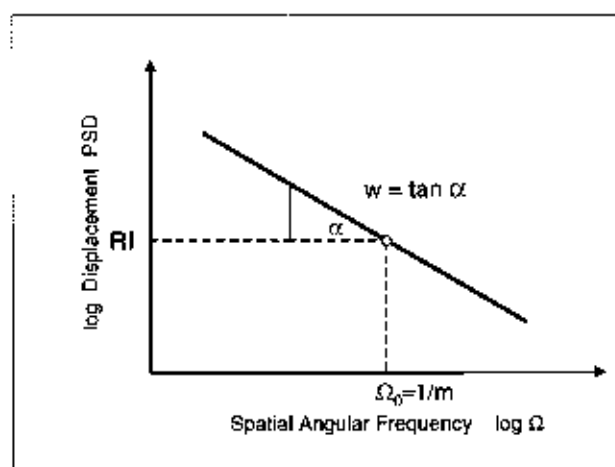


Figure 1 - Power Spektral Density of a Road: Definition of Spectral Roughness Index (RI) and Waviness (w)

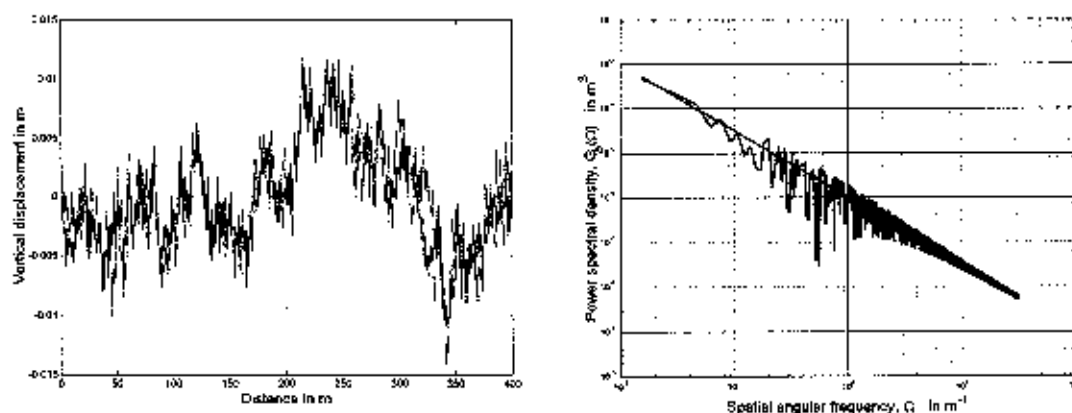


Figure 2 - LEFT: PSD for left and right wheel track ($RI = 1 \text{ cm}^3$, $w = 2$).

RIGHT: Resulting longitudinal profiles for left and right wheel track.

RELATIONSHIP BETWEEN ROAD ROUGHNESS AND DYNAMIC WHEEL LOADS

It can be shown, that a mathematical relation exists between road roughness and dynamic wheel load [2]. For a linear single axle system the dynamic load factor (DLC), which is defined as the ratio of the root mean square (RMS) dynamic force to the static force,

$$DLC = \frac{RMS\ dynamic\ tire\ force}{static\ tire\ force} \quad (1)$$

is a function of the roughness:

$$DLC^2 = v \cdot RI \cdot VDF \quad (2)$$

In the latter equation v is the speed of the vehicle (in m/s), RI is the spectral roughness index (in m^3) and VDF the Vehicle Dynamics Factor (in s/m^4). VDF represents the dynamic properties of the vehicle and is an index for the 'road friendliness' of the truck's suspension. Its rather complicated mathematical expression [3] reveals dependencies on

- the dynamic properties of the truck (such as damping, stiffness and masses),
- the speed of the truck
- the waviness of the road and
- the coherence between left and right wheel track

RELATIONSHIP BETWEEN ROAD ROUGHNESS AND ROAD DAMAGE

The repeated loading of pavements by passing trucks will eventually cause fatigue and cracking of bound materials in all types of pavements as well as permanent deformation (such as rutting) in flexible pavements. In the context of this paper road damage is looked upon as fatigue and cracking. In order to draw a relation between road roughness and road damage some kind of assumption about pavement wear has to be made. The so called 'fourth power law' obtained from the 'AASHO (American Association of State Highway Officials) Road Test', conducted in Illinois between August 1956 and November 1960, is such an assumption. One of the findings of this test was that pavement wear is exponentially related to the wheel loads (the contact forces between wheel and pavement), approximately to the fourth power [4]. Even though the fourth power law is unsatisfactory as a general approach of predicting long term pavement performance and road damage, for comparison purposes, i.e. comparing different axle configurations and suspension systems in terms of their influence on road damage, it is still a plausible rule of thumb and a commonly used approach. And the latter is the objective of this paper: to propose an indicator for 'road friendliness', which considers the dynamic properties of the truck suspension, and to rate different types of trucks.

Provided, that the dynamic wheel loads are distributed normally, which has been proved by experiments [5], it can be shown that the mean value of the fourth power of the wheel load can be calculated from the static wheel load,

F_{stat} , by the following equation [6]:

$$\overline{F^4} = F_{stat}^4 (1 + 6DLC^2 + 3DLC^4) \quad (3)$$

Taking into account that the DLC in most cases is beyond 0,3 (which denotes the instant when the wheel is beginning to loose contact to the ground) the fourth power term in the above equation can be neglected. By substituting Eq.(2) for DLC^2 Eq.(3) reduces to the following,

$$\overline{F^4} = F_{stat}^4 (1 + 6 \cdot v \cdot RI \cdot VDF) \quad (4)$$

In order to account for the influence of the wheel configuration (single or dual tires) and tire contact pressures the following expression for a so called 'road stress factor' was proposed [6]

$$\text{road stress factor} = (\eta_1 \eta_H F)^4 \quad (5)$$

with η_1 tire configuration: 1 for single and 0.9 for dual tires

η_H contact pressure: 1,1 / 1,0 / 0,9 for 0,9 / 0,7 / 0,5 N/mm²

Substituting Eq.(4) into Eq.(5) we get:

$$\text{road stress factor} = (\eta_1 \eta_H F_{stat})^4 (1 + 6 \cdot v \cdot RI \cdot VDF) \quad (6)$$

Because the road stress factor yields rather large values (proportional to F_{stat}^4), it is referred to a 10-ton axle for the further considerations. The result is the amount of pavement distress expressed in number of (quasi-static) passes of a 10-ton reference axle. This quantity is an extended but essentially similar formulation of the 'load equivalent factor' (LEF) derived from the ASHOO road test. It will be denoted 'load equivalent factor' LEF* in this context. Thus Eq.(6) becomes:

$$LEF^* = (\eta_1 \eta_H F_{stat} / 10t)^4 (1 + 6 \cdot v \cdot RI \cdot VDF) \quad (7)$$

As can be seen the LEF* consists of a quasi-static and a dynamic part. The dynamic part is the part of pavement distress, that is caused by roughness, while the quasi-static part denotes that part, that theoretically would occur in the absence of any roughness. This is a mathematical formulation of the fact, that with increasing roughness the loading and thus the damaging increases. The following equations can be set up:

$$LEF^* = LEF_{stat}^* + LEF_{dyn}^* \quad (8)$$

$$LEF_{stat}^* = (\eta_1 \eta_H F_{stat} / 10t)^4 \quad (9)$$

$$\frac{LEF_{dyn}^*}{LEF_{stat}^*} = 6 \cdot v \cdot RI \cdot VDF \quad (10)$$

It is noteworthy that Eq.(10) denotes the increase in pavement distress caused by the roughness. According to this equation the increase is a function of speed, roughness and vehicle dynamics. So we have got a rather simple means for comparing different trucks and suspension systems concerning their influence on road damage.

PROVING THE MATHEMATICAL APPROACH

The mathematical approach presented in Eq.(7) is based on a number of assumptions, specifically the assumption of linear system behavior of the trucks. In order to prove whether real trucks behave that way, four trucks, typical for European highways, have been modeled in detail. The three-dimensional multi-body systems consider the nonlinear behavior of steel (coulomb friction) and air springs as well as of the hydraulic dampers. The geometric and dynamic data are taken from state-of-the-art truck development. The four trucks are:

- a 2-axle truck, 18 ton gross weight, steel sprung
- a 3-axle truck, 25 ton gross weight, air sprung
- a 5-axle truck trailer, 40 ton gross weight, air sprung
- a 5-axle tractor-trailer, 40 ton gross weight, front axle steel sprung, rest air sprung

The four vehicles are shown in Figs. 3 to 6.

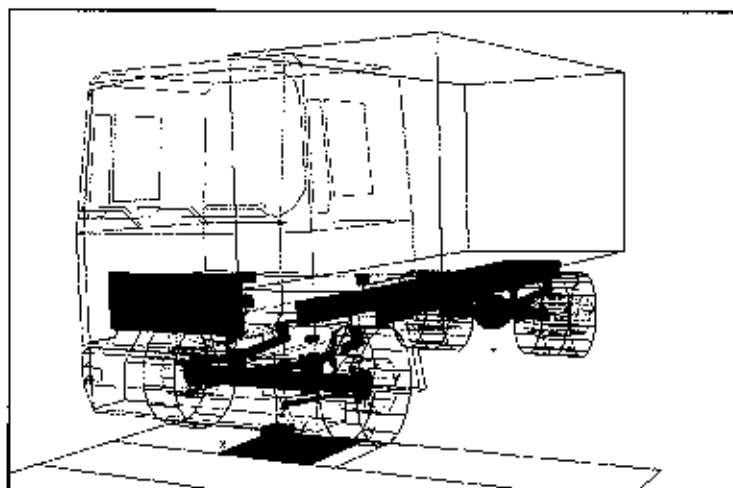


Figure 3 - 2-axle truck, 18 ton gross weight, steel sprung.

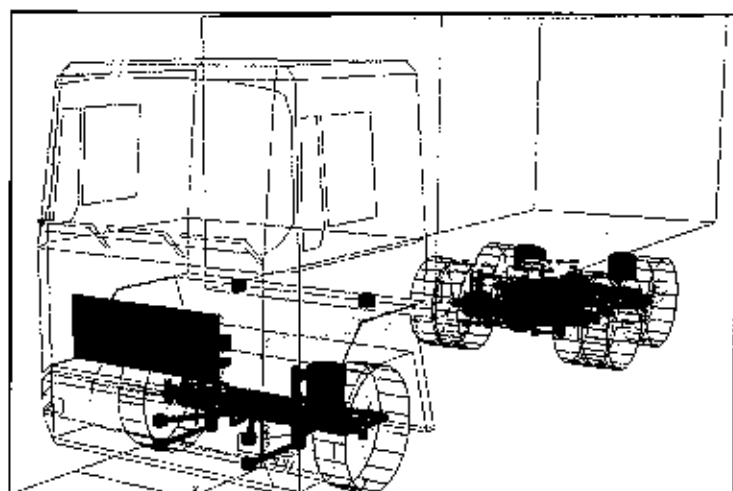


Figure 4 - 3-axle truck, 25 ton gross weight, air sprung

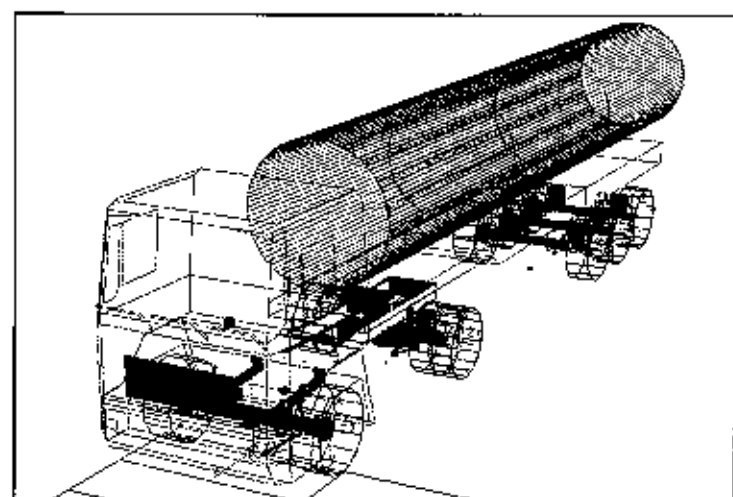


Figure 5 - 5-axle tractor-trailer, 40 ton gross weight, front axle steel sprung, rest air sprung

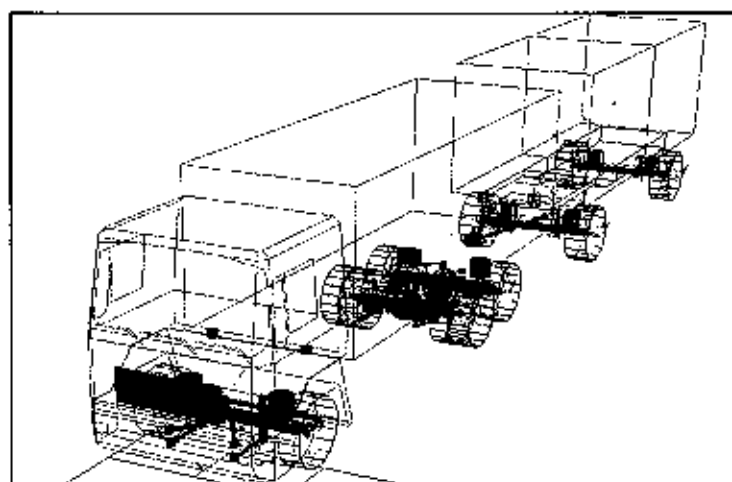


Figure 6 - 5-axle truck trailer, 40 ton gross weight, air sprung

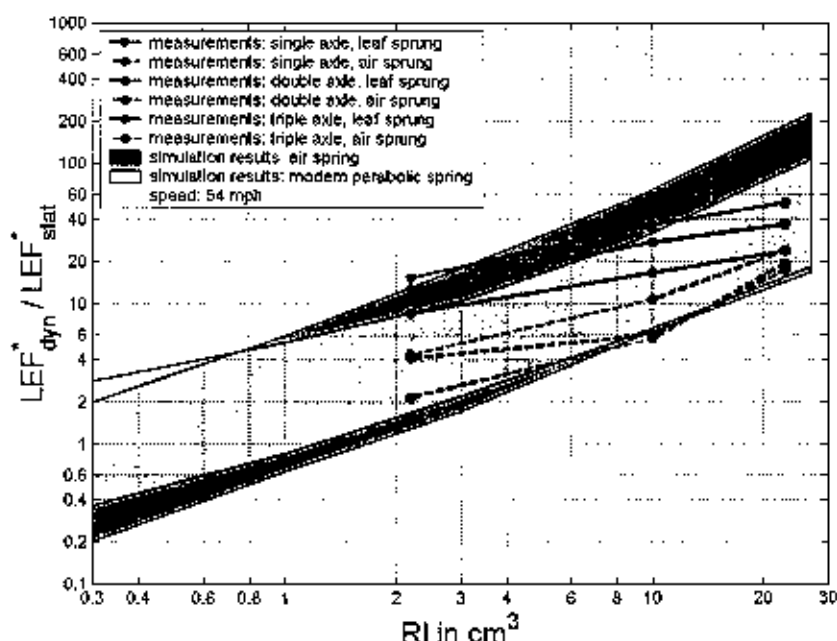


Figure 7 - Amount of pavement distress due to road roughness as referred to a very smooth road; comparison with results from (5).

Simulated test rides over a variety of different roads were performed in order to prove whether Eqs.(7) and (10) apply for non-linear dynamic systems as well. For that reason 20 different roads ($RI [cm^3] = 0,3 / 1 / 3 / 9 / 27$ and $w = 1,5 / 2 / 2,5 / 3$) were generated, which had to be driven by each of the vehicles with a speed of 24 m/s, i. e. 53,7 mph. The wheel loads were measured and, summed up to axle loads, raised to the fourth power and finally averaged. In comparison with the corresponding static axle loads, raised to the fourth power and averaged as well, the additional pavement distress due to the roughness of the road (which is denoted $LEF_{dyn}^* / LEF_{stat}^*$ in equation 10) could be determined for each of the axles. The results are shown in Fig. 7 summarized for leaf and air sprung axles respectively. They spread from 0,2 to 2 % for very even roads up to 20 to 200 % for very uneven roads, dependent on the type of axle and the waviness of the road. It is astounding that the leaf springs performed as well as the air springs, except in the case of very good roads, where the leaf springs tended to cause higher pavement distress. This is due to the Coulomb friction, which becomes apparent in some kind of 'stick-slip mechanisms' only on very smooth road surfaces. The leaf sprung axles were fitted with modern low-friction

parabolic springs. The result confirms the observation that modern well-maintained leaf sprung suspensions, operated under optimum loading conditions, can compete with air sprung suspensions. The rather "bad" come-off of the air sprung suspensions on the other hand was mainly caused by the front suspension of the 25-ton truck. In order to achieve good driving comfort the manufacturer decided to equip this type of truck with a rather low damped front axle, which in turn causes higher dynamic wheel loads and, as a result, higher pavement wear.

The question that was supposed to be answered by this chapter was: can the additional part of pavement distress that is due to the roughness of the road be expressed in terms of a linear function of the roughness as shown in Eq.(10)? Fig. 7 confirms this assumption: dividing the curves by the abscissa (i.e. the road roughness RI) yields in almost horizontal curves (not shown), which is a strong indicator for the linear dependency. Dividing them by $(6 \cdot v \cdot RI)$, see Eq.(10), yields in vehicle dynamics factors (VDF) between 45 and 450 s/m^4 , which very well agree with VDF s mentioned in the literature ranging from 60 to 400 s/m^4 [7]. Additionally, Fig. 7 contains the comparison with results found in the literature. The University of Hannover conducted a series of measurements [5], that confirms the theoretical results found here.

RESULTS FOR THE WHOLE VEHICLE

As 'load equivalent factors', LEF^* , can be determined for single axles, they can be determined for a whole vehicle. Corresponding to Eq.(10) a 'vehicle dynamics factor', VDF , for the whole vehicle can be defined. So Eqs.(8) to (10) apply to the whole vehicle as well, except for a little modification in Eq.(9) to consider the n axles of the truck:

$$LEF_{stat}^* = \sum_{i=1}^n (\eta_i \eta_{II} F_{stat,i} / 10t)^4 \quad (11)$$

Fig. 8 shows the results for each the four vehicles. The LEF^* consists of a 'static part', LEF_{stat}^* , caused by the static weight, and a 'dynamic part', LEF_{dyn}^* , caused by the road roughness (see Eq.8). The static part (Eq.9) is the value at the very left of the graph. The dynamic part (Eq.10) is a function of the roughness. Its slope is dependent on the static LEF^* , the speed and the 'vehicle dynamics factor', VDF .

As can be seen, the static LEF^* for the 18-ton truck is about 1, for the 25-ton truck it is about 1.5 and for the 40-ton vehicles it is about 2. This means that on a very smooth road a 25-ton truck (fully loaded) damages a road about 50% and a 40-ton truck about 100% more as compared to a 18-ton truck. This is astounding at first, when thinking of the assumption that the weight influences the road damage by the fourth power. But considering the tire configurations of the different axles (see factor η_i in Eq.9) the above mentioned results are yielded. With increasing road roughness the LEF^* increases as well, up to 100% in the case of the tractor-trailer on a poor road ($RI = 27 \text{ cm}^3$).

The shaded areas mark the results of the simulated test rides of the four trucks, covering a waviness of $w = 1.5$ to 3. They were calculated basically by using Eq.(5), i.e. multiplying the axle loads by the factor $(\eta_i / 10t)$ (the factor η_{II} was set to 1), raising them to the fourth power, averaging them over the road's length and summing up over all the axles of the truck. This was done for each of the trucks and each of the 20 different roads that were driven.

The four bold straight lines in Fig. 8 are the result of a fitting according to Eq.(7) with vehicle dynamics factors VDF of 70 s/m^4 (for the tractor-trailer) to 100 s/m^4 for the 18-ton-truck. From figure 5.1 it is obvious that up to a roughness of $RI = 10 \text{ cm}^3$ (which corresponds to a bad highway) the damaging influence of the four trucks can

well be described by this equation assuming an average VDF for a waviness between $w = 1,5$ and 3 . Because of it's importance the equation is repeated here:

$$LEF^* = LEF_{stat}^* (1 + 6 \cdot v \cdot RI \cdot VDF) \quad (12)$$

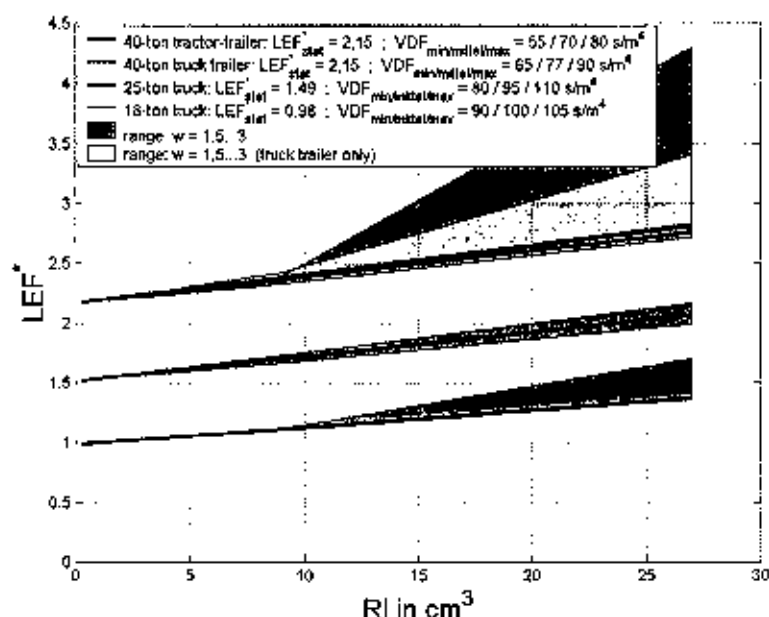


Figure 8 - Pavement Distress caused by different vehicles at a speed of 54 mph

For a larger roughness the spreading caused by the different waviness is too large in order to describe the behavior of the truck by an average VDF . Still, for a waviness of 2 (which is about the average waviness for highways) it was found that the equation very well fits the results of the simulated test drives over the whole range of roughness from 0,3 to 27 cm^3 .

Summarizing the findings, table 1 contains the LEF^* and the VDF , for both: a roughness up to 10 cm^3 (applicable to all kinds of roads) and a roughness up to 27 cm^3 and more (applicable only to roads with a waviness of about 2).

Table 1 - Static Load Equivalent Factor and Vehicle Dynamics Factor

	LEF_{stat}^* fully loaded	VDF in s/m^4 for $RI < 10^3 cm^3$	VDF in s/m^4 for $w = 2$
18-ton-truck	1	100	97,5
25-ton-truck	1,5	95	83
40-ton-truck-trailer	2,15	77	77
40-ton-semi-trailer	2,15	70	66

CONSIDERING PERIODIC PHENOMENA IN THE ROAD SURFACE

The examinations have been extended to road surfaces with combined irregular and periodic phenomena in that way, that some of the road profiles used before have been superimposed by periodic longitudinal profiles of different amplitude. The periodic profiles were

- sinusoidal profiles with amplitudes of 2,5 / 5 / 10 mm exciting the trucks in the range of 1...2 Hz and 10 Hz, which is about the frequency of the sprung mass and the axle respectively
- "tooth-saw"- profiles with average step heights of 2,5 to 20 mm corresponding to average "amplitudes" of 1,25 / 2,5 / 5 / 10 mm, representing concrete plates of 5 meter length

For this investigation five "basic" roads with a roughness index $RI = 0,3 / 1 / 3 / 9 / 27 \text{ cm}^3$ and a waviness $w = 2$ have been used. The driven speed was 53,7 mph as before.

The results of the simulated test rides are exemplary shown in Fig. 9 in case of the 40-ton tractor-trailer on the "saw-tooth" profile. It is noticeable, that the increase in LEF^* is independent of the roughness. This was found for all of the four trucks and all of the above mentioned excitations except for one: the excitation with 10 Hz in combination with the maximum amplitude of 10 mm. In this instance the LEF^* was highest for very low RI (i. e. almost pure sinusoidal excitation) and decreases with increasing roughness (i. e. increasing disturbances in the harmonic signal), which is understandable. In all the other instances the results can be mathematically expressed by basically adding a "P-value" to Eq.(12) representing the periodic part of the excitation:

$$LEF^* = LEF_{stat}^* (1 + P + 6 \cdot v \cdot RI \cdot VDF) \quad (13)$$

For the example in Fig. 9 the P-value is 0,06 and 0,2 for a step height of 10 and 20 mm respectively, otherwise zero (see legend). This means that pavement wear, measured in LEF^* , increases by 6 and 20 % respectively. The differences between simulated test ride results and the approximation through Eq.(13) are below 3% for a roughness index below 10 cm^3 and below 10 % for an RI up to 27 cm^3 . Fig. 10 summarizes the results. The completely air sprung vehicles (the 25-ton truck and the 40-ton truck trailer) turned out to be most sensitive to periodic excitations through concrete plates (about 5 Hz at 54 mph) and harmonic excitation with 10 Hz (resonance of the axle), while least sensitive to harmonic excitation with 1 to 2 Hz (resonance of the sprung mass). Excitations typical for concrete roads as well as harmonic excitations with 1 to 2 Hz caused a maximum increase in pavement damage of about 30 to 40%, while excitations with 10 Hz caused an increase of 50 to 250 %.

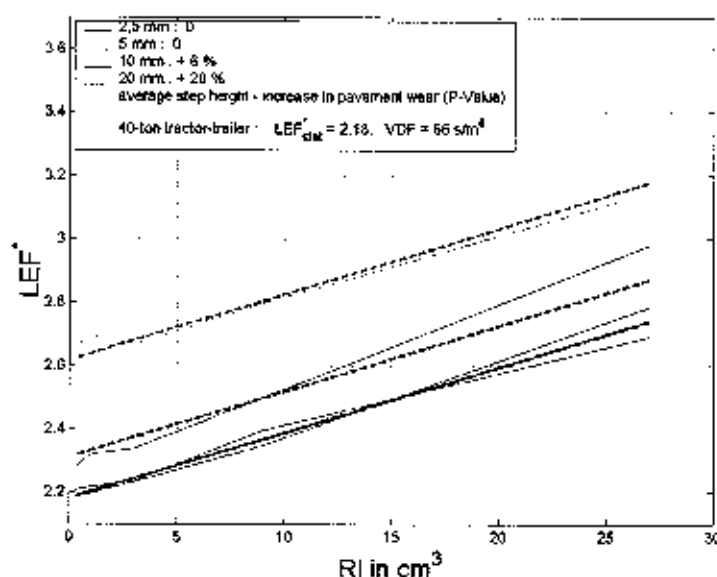
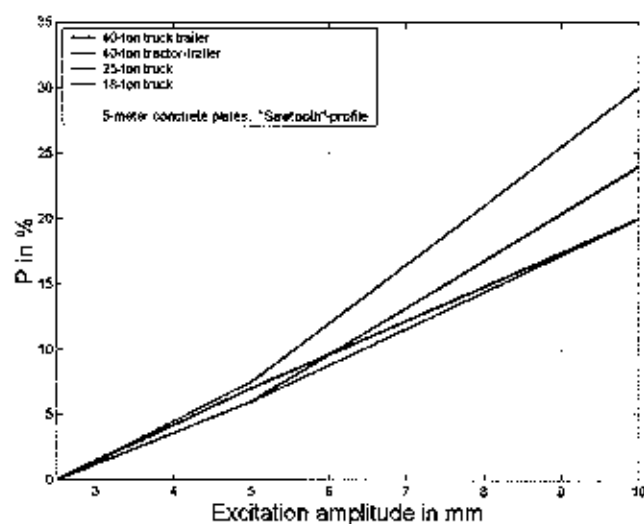
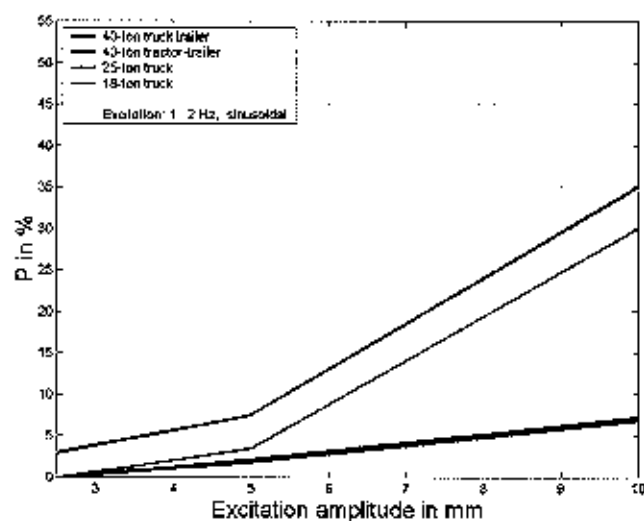


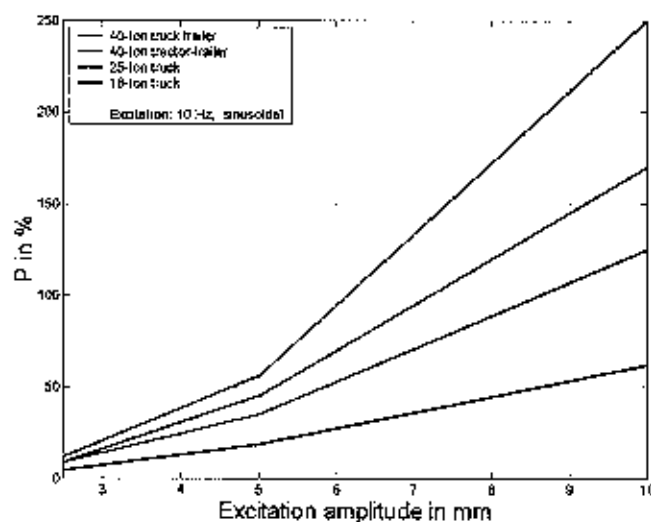
Figure 9 - The influence of „tooth-saw“ profiles on the road damage, 40-ton tractor-trailer



a)
'saw-tooth'-profile
concrete plates (5m)



b)
sinusoidal excitation
1...2 Hz



c)
sinusoidal excitation
10 Hz

Figure 10 - Increase in pavement wear by periodic excitation (P-value)

CONCLUSION

The intention of this paper is to shed light on the interaction between road roughness and road damage caused by static and dynamic wheel forces. For this purpose the first chapter deals with the description of the road in terms of the power spectral density (PSD). Furthermore it can be shown that a mathematical relationship exists between PSD and dynamic wheel loads. In a third step the static and dynamic wheel forces are being linked to road damage by using the "fourth power law", obtained from the results of the ASHOO Road Test. The result is an equation that allows to relate road roughness data to road damage. This equation has been extended to periodic phenomena later and proved by rather detailed non-linear multi-body systems representing four different trucks. For its importance to this paper it shall be repeated here:

$$LEF^* = LEF_{stat}^* (1 + P + 6 \cdot v \cdot RI \cdot VDF) \quad (14)$$

The equation says that pavement distress with respect to fatigue and cracking (measured here in terms of a 'Load Equivalent Factor', LEF*) can be understood as a process caused by a quasi-static (LEF_{stat}^*) and a dynamic contribution of the truck, the latter resulting from the roughness (indicated by the 'Roughness Index', RI) and periodic phenomena eventually contained in the road surface (expressed by 'P'). The dynamic part resulting from periodic phenomena turned out to be independent of the roughness except for excitations with very high amplitudes at resonance frequency of the axles. The dynamic part resulting from the roughness is, besides the RI, dependent on the vehicle speed, v , and the 'Vehicle Dynamics Factor', VDF, which describes the dynamic properties of the truck, actually the 'road friendliness' of the suspension system. The 40-ton tractor-trailer was found to have the most and the 18-ton truck the least 'road-friendly' suspension, 70 and 100 s/m³ respectively.

The VDF proves to be a suitable means being able to describe the dynamic properties of the truck in a distinctive way. It applies for an RI up to 10 cm³ and the maximum permitted speed. For roads up to RI = 10 cm³ (which corresponds to a bad highway) and maximum permitted speed roughness accounts for up to $(6 \cdot 24 \frac{m}{s} \cdot 10 \cdot 10^{-3} m^3 \cdot 100 \frac{s}{m^4}) = 15 \%$ of the pavement distress caused by the truck. For a higher roughness (here: 27 cm³) Eq.(14) only applies, if the waviness is about 2. Here a 40% increase is found. These results comply very well with results found in the literature [8]. For a waviness other than 2 roughness is partially found to have a much higher influence: in case of the tractor-trailer it can increase the pavement wear up to 100% (see Fig. 8).

These are by definition (see Eq.3) values that are averaged over the length of a road. Peak values are 25 % to 200% higher depending on the roughness [3].

If periodic phenomena are contained in the road surface, the pavement distress is even enhanced. Excitations in the range of the axle resonance turned out to have the highest influence on pavement wear. An amplitude of 5 mm can lead to a 50%-increase ($P = 0,5$) in pavement wear (mean value) and an amplitude of 10 mm in combination with a low RI (1 cm³) even enlarges the pavement distress by up to 250% ($P = 2,5$; see Fig.10). Excitations in the range of 1 to 2 Hz (resonance of the sprung mass) and "saw-tooth"-profiles encountered on concrete roads turned out to have only a tenth of that effect ($P = 0,06$ and $0,25$ for an amplitude of 5 and 10 mm respectively).

The findings presented in this paper allow to qualify vehicles with respect to their 'road friendliness' and their contribution to pavement wear in terms of fatigue and cracking. The formula developed doesn't claim to yield quantitative results regarding pavement damage (which depends on a wider range of parameters than considered here), but rather allows to compare and rate different vehicles and is supposed to give a better understanding of the influence of road roughness on pavement wear. In this way the contribution of this paper should be understood.

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PRODUCTIVITY OPPORTUNITIES WITH STEERABLE AXLES

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ABSTRACT

Setting aside the primary (front) steering axles fitted to trucks and prime movers, a wide variety of steerable axles are available for use on multi-axle vehicles. These steerable axles are designed for both trailing (unpowered) axles and driven axles. All of these steerable axle types address, in different ways, the fact that vehicle tyres operate in a sub-optimal way as soon as a vehicle unit is fitted with more than two axles and/or more than one "fixed" axle. This degradation in tyre and vehicle performance can be exhibited in:

- Increased tyre wear
- Increased vehicle swept path
- Increased pavement surface wear
- Increased resistance to forward motion (and increased fuel consumption)
- Potentially undesirable effects on vehicle steering control.

Steerable axles offer performance improvements for all classes of heavy vehicle and provide direct benefits to transport operators who choose to use them. Such performance improvements also open the way to productivity gains in road freight transport operations because longer or heavier vehicles may be enabled within the constraints of the infrastructure, traffic and safety.

Steerable axles may also adversely affect certain areas of heavy vehicle performance, depending on the vehicle configuration and the characteristics of the steerable axle. To address these issues, the dynamic performance of selected vehicles fitted with steerable axles was compared with that of currently-operating vehicle configurations and with the performance parameters being developed in the Performance Based Standards (PBS) project being carried out by the National Road Transport Commission and Austroads.

This paper describes the results of an Australian study of current steerable axle practices, performance effects on a range of vehicle configurations, safety and geometric impacts, productivity benefits, net economic benefits and regulatory implications.

1 INTRODUCTION

The National Road Transport Commission (NRTC) commissioned Roaduser Systems Pty Ltd to carry out a study of the benefits, costs and potential for length increases under PBS and performance effects of steerable axles. The study included investigation of regulations affecting the use of steerable axles, and whether there are any current impediments to the use of steerable axles.

The study included investigation of:

- Current practices with steerable axles (including current regulations affecting steerable axle use)
- Benefits of steerable axles for a wide range of vehicle configurations (as perceived by key stakeholders)
- Benefits of marginal semi-trailer length increases
- Potential for productivity increases (through increased length and mass) with the wider use of steerable axles throughout the Australian fleet (as defined by the prime constraints of low-speed geometric performance)
- Geometric and safety impacts of these initiatives; computer simulation assessment of a wide range of vehicle configurations fitted with steerable axles was carried out
- Productivity benefits, costs and net economic benefits of a range of initiatives deploying steerable axles.

Setting aside the primary (front) steering axles fitted to trucks and prime movers, a wide variety of steerable axles are available for use on multi-axle vehicles. These steerable axles are designed for both trailing (unpowered) axles and driven axles. All of these steerable axle types address, in different ways, the fact that vehicle tyres operate in a sub-optimal way as soon as a vehicle unit is fitted with more than two axles and/or more than one "fixed" axle. This degradation in tyre and vehicle performance can be exhibited in:

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2 CURRENT PRACTICES WITH STEERABLE AXLES

While steerable axles come in a range of generic types, the most common are "automotive type" steerable axles used on semi-trailers; this type of steerable axle is also available for rigid trucks and prime movers. Other types include linked-articulation axle group steering systems for semi-trailers.

Steerable axles are not currently in widespread use in Australia. Current users of automotive-type steerable axles on triaxle semi-trailers report improved tyre wear and improved swept path. Linked-articulation steerable axles are new and are not currently being used in road transport, but offer a large improvement in swept path performance.

Current Australian regulations mitigate against the use of automotive-type steerable axles on trailers because the rear overhang dimension may be exceeded if the rearmost fixed axle is replaced with a steerable axle. There are no current regulatory impediments to the use of steerable axles on rigid trucks, but little use is currently evident on this vehicle type.

The literature suggests that steerable axles on rigid trucks may in certain cases adversely affect vehicle handling and control; this is much less likely on trailers. Most of the research involving steerable axles and vehicle dynamics has been carried out on potential "problem" areas for steerable axles, such as rigid trucks and "C-dollies" for multi-combination vehicles.

3 PERFORMANCE EFFECTS OF STEERABLE AXLES

In addition to the known benefits of reduced swept path and reduced tyre wear, steerable axles also affect vehicle dynamic performance. Provided that steerable axles have at least a threshold level of self-centring, their effects on dynamic stability and tracking behaviour of the common Australian freight vehicle configurations are modest. Only in road trains of conventional configuration were dynamic performance impacts found to be of concern.

In the case of linked-articulation steerable axle group systems, the effects on improving swept path performance can be dramatic. In the case of an automotive-type steerable axle introduced into a triaxle group, there is a modest but worthwhile improvement in low-speed offtracking and swept path.

When an automotive-type steerable axle is introduced into a triaxle group of a typical Australian articulated vehicle (A123), there is a dynamic performance trade-off in a lane-change manoeuvre: the high-speed dynamic offtracking increases (a negative) and the dynamic stability increases (a positive). As illustrated in Figure 1, this trade-off becomes more marked as the self-centring stiffness of the steerable axle reduces.

4 STEERABLE AXLE POTENTIAL UNDER CURRENT REGULATORY REGIME

Steerable axles have the potential to improve access of vehicle combinations in the road network and into sites and depots. This has the greatest potential for B-doubles, where access is often tight and the use of steerable axles could provide substantial gains. Operators should give more consideration to the benefits of fitting steerable axles to B-doubles. Figure 2 shows a typical Australian B-double and the effect on swept path performance of fitting one steerable axle to each triaxle group; this graph shows the effect of overall length (OAL), when caused by increasing trailer s-dimension (distance from kingpin to centre of axle group) on swept path. The steerable axles reduce swept path by approximately 0.5 m; alternatively, they allow the vehicle to become over 1.0 m longer for the same swept path performance.

Steerable axles could also be fitted to rigid trucks, leading to R13 and R23 configurations with increased GVM and productivity for mass-limited loads. Although not currently impeded by regulations, these applications currently find little uptake and there are likely to be useful gains available to some operators.

5 STEERABLE AXLES DRIVING ROAD FREIGHT PRODUCTIVITY

Steerable axles potentially offer productivity benefits in (i) increased cubic capacity for road freight vehicles and (ii) increased gross mass for road freight vehicles (within existing axle mass limits). Regulatory review or the introduction of Performance-Based Standards (PBS) is needed to realise the potential productivity benefits. A PBS regulatory regime is currently being developed in Australia (1). It is anticipated that this will provide an alternative to the current prescriptive limits and will require demonstration of compliance with certain measures and standards pertaining to dynamic performance and infrastructure performance.

Increased cubic capacity is related to increased trailer length and increased length of rigid trucks, leading to increased "load length" of the vehicle or vehicle combinations. Such length increases will be constrained in the first instance by low-speed geometric performance considerations, and may be further constrained by considerations of dynamic performance (tracking and stability behaviour or infrastructure impacts).

Increased mass is a less direct consequence of the use of steerable axles, but could arise from:

- The ability to place additional axles on existing configurations without incurring unacceptable tyre scrub; for example, a quad axle semi-trailer with one steerable axle in the quad group
- The ability to introduce heavier axle groups spaced further apart (to maintain compliance with bridge formulae) and still retain acceptable low-speed geometric performance.

Generally speaking, requirements to ensure that dynamic performance of a vehicle combination is not degraded by the fitting of an automotive-type steerable axle are straightforward. This may be achieved via steerable axle standards such as requiring that the aligning stiffness of the automotive-type steerable axle should be at least equivalent to the medium stiffness value used in this study, or simply by requiring certain vehicle PBS to be achieved. There should also be a limit of one automotive-type steerable axle per triaxle group, and the steerable axle should be fitted in the rear position. Consideration should also be given to the need for any specific requirements for load sharing performance of steerable axles when incorporated in an axle group.

Figure 3 shows a potential 15 m semi-trailer with an axle group comprising two fixed axles and one steerable axle in combination with the longest prime mover which will allow the vehicle to comply with the Austroads General Access Swept Path Specification. This 15 m semi-trailer has an s-dimension of less than 10.0 m, but the distance from the kingpin to the load-bearing centre of the triaxle group remains at 10.0 m in order to preserve effective load distribution. This combination exceeds the current limit 19 m in overall length and the wheelbase of the prime mover is less restricted.

Figure 4 shows a 50 t quad axle (steerable) semi-trailer combination which meets the Australian general access bridge formula for road-friendly vehicles, is capable of optimum load distribution for a water-level load and utilises a standard-length 14.6 m (48 ft) semi-trailer. This vehicle has the following characteristics:

- Gross combination mass of 50 t
- Overall length of 18.75 m (with a conventional prime mover of moderate wheelbase)
- Semi-trailer length of 14.6 m
- S-dimension of 9.5 m
- Rear overhang of 4.35 m (in excess of current limit).

In addition to good swept path performance, this vehicle improves on the current A123 (with fixed trailer axles) in terms of dynamic stability (see Figure 5); at the same time, the gross mass increases from 45.5 t to 50 t with only a modest increase in tare mass. The presence of the steerable axle assists swept path performance and reduces pavement wear and tyre wear.

In Australia, 25 m/68 t B-doubles currently operate on designated routes. In the future, major road freight routes will be better defined and designed for large vehicle combinations. Figure 6 shows a "super B-double" which utilises two automotive-type steerable axles and increases both mass and cubic productivity within major freight route performance parameters. This vehicle meets the current B-double bridge formula, increases gross mass to 77 t and increases the total deck length from approximately 19.5 m to approximately 24 m. Swept path performance is on the limit for major freight routes and stability is significantly improved relative to current B-doubles (see Figure 7).

6 ECONOMIC BENEFITS

The wider deployment of steerable axles offers substantial financial and economic benefits in cases where productivity gains are able to be exploited with high-utilisation vehicles. The net benefits to the Australian economy depend on the take-up rate of such initiatives, and take-up can only be estimated.

The economic benefits of minor regulatory change in relation to improved access and reduced tyre wear are difficult to estimate. However, as the necessary changes are small and no significant costs to agencies have been identified, such changes are recommended.

The nett contribution of steerable-axle enabled vehicle combinations to PBS-based regulatory reforms in Australia is conservatively estimated to be M\$60 per year (Australian dollars).

7 CONCLUSIONS

The use of automotive-type steerable axles in Australian vehicle combinations offers significant productivity and economic nett benefits with relatively little impact on vehicle safety performance. Steerable-axle-enabled vehicle combinations offer certain improvements in dynamic performance, provided some straightforward controls on steerable axle characteristics are introduced.

Steerable axles fitted to trailers in optimised vehicle combinations offer increases in both payload cube and mass, taking into account the physical constraints of road space and bridge strength.

Steerable axles also offer benefits when fitted to current (non-optimised) vehicle combinations. Such benefits include improved access, reduced tyre wear, reduced pavement wear and reduced fuel consumption and emissions. These benefits are considered to be worthwhile, although were not able to be quantified in the study.

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TABLES & FIGURES

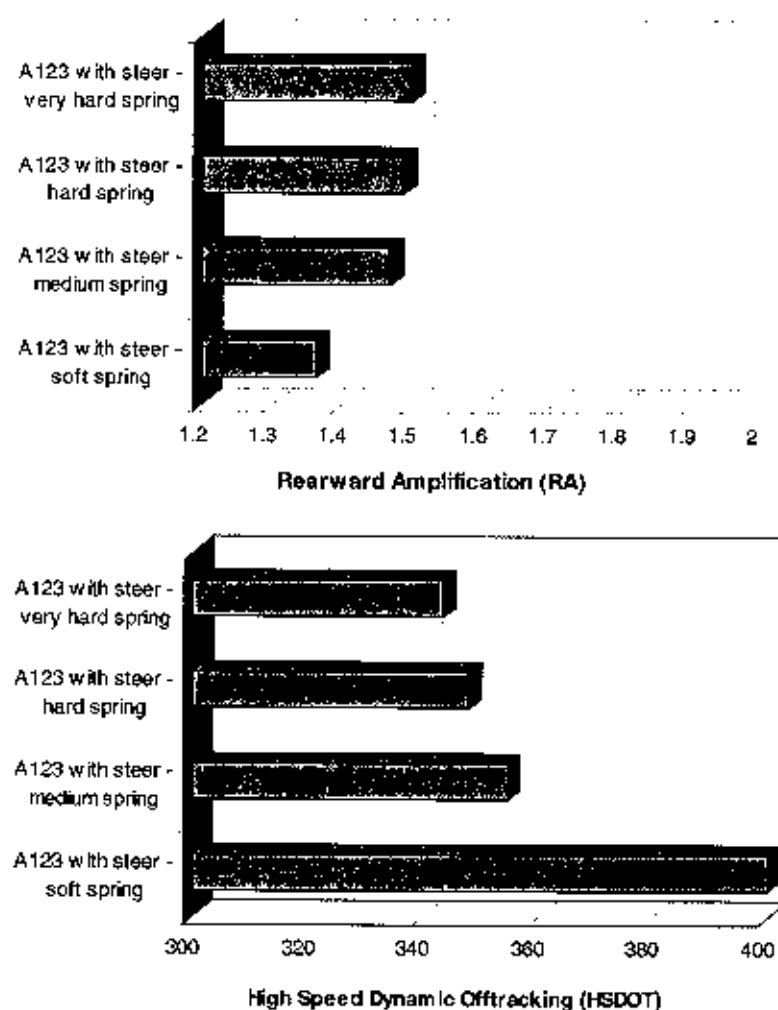
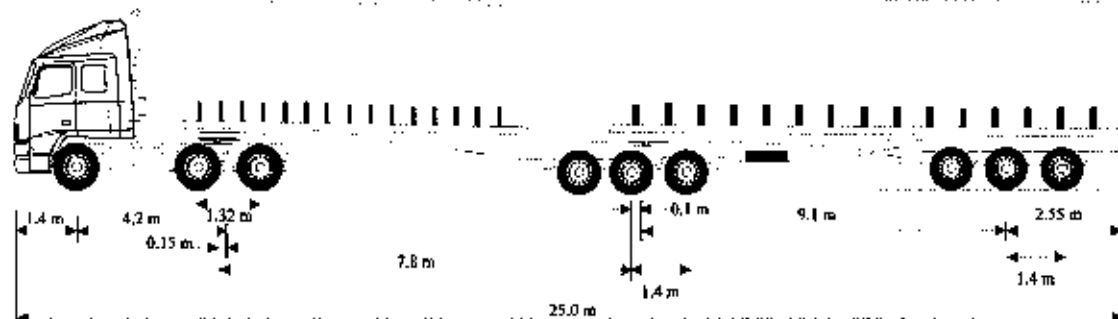
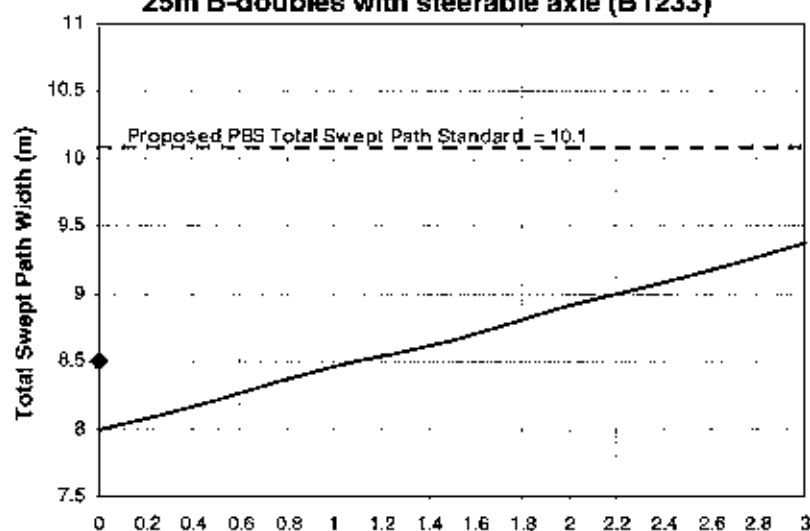


Figure 1 - Dynamic performance trade-off for A123 vehicle with steerable axle in rear position, as a function of steerable axle stiffness

Baseline B-Double (B1233)



25m B-doubles with steerable axle (B1233)



■ Baseline Result Additional OAL with increase in Trailer S-dimension (m)

Figure 2 – Effect of steerable axles on swept path of 25 m Australian B-double (B1233)

Candidate 15m Tractor-semi-trailer (A123) with a steerable axle fitted

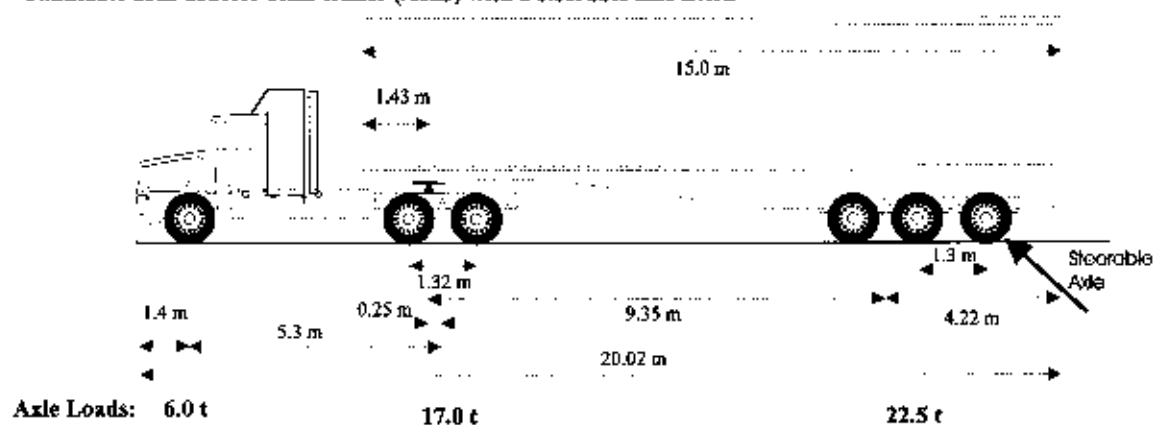


Figure 3 – Potential 15 m triaxle semi-trailer with automotive-type steerable axle in rear position

(A124) Tractor Semi-Trailer - (Increased mass capacity with additional steerable axle)

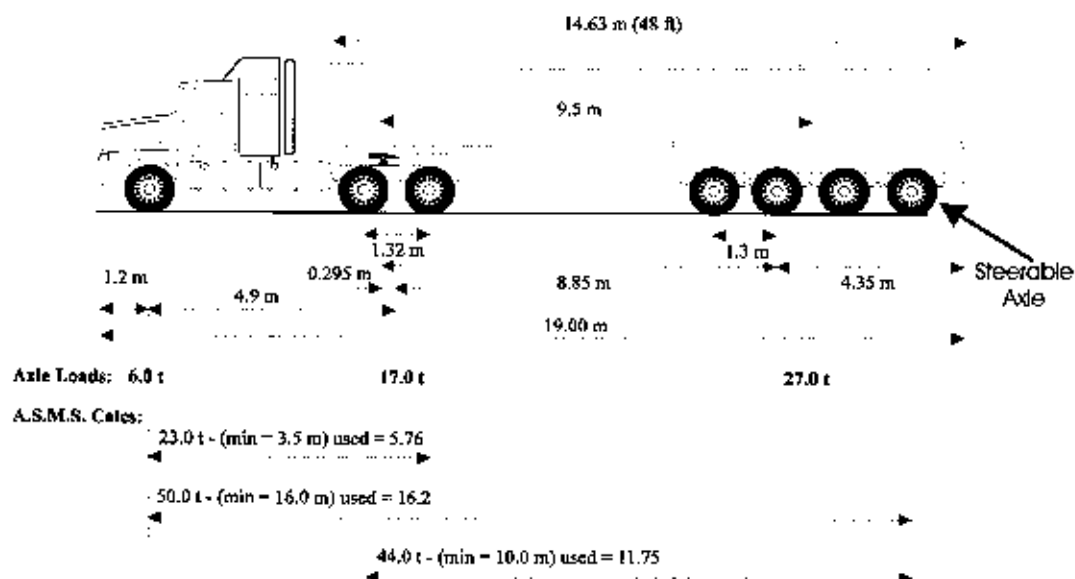


Figure 4 – Potential 50 t quad axle semi-trailer with automotive-type steerable axle in rear position (including axle spacing mass schedule analysis)

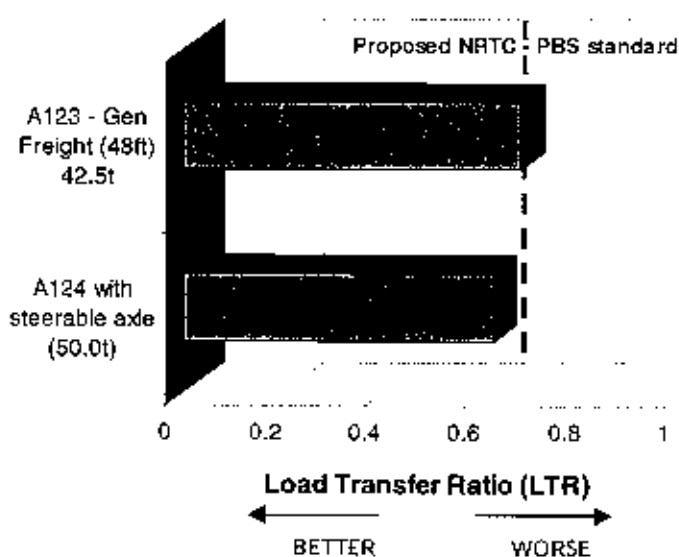


Figure 5 – Improved dynamic stability of A124 (with steerable axle) vs current A123

(B1234) Candidate B-Double - (Increased mass capacity achieved by adding 2 steerable trailer axles)

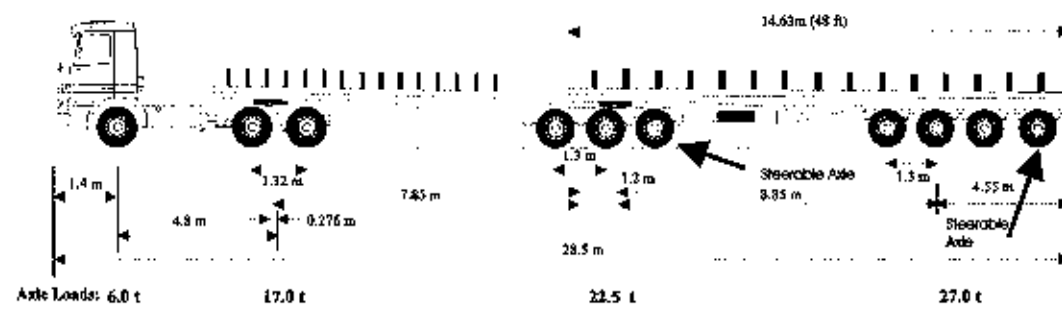


Figure 6 – “Super B-double” for major freight routes

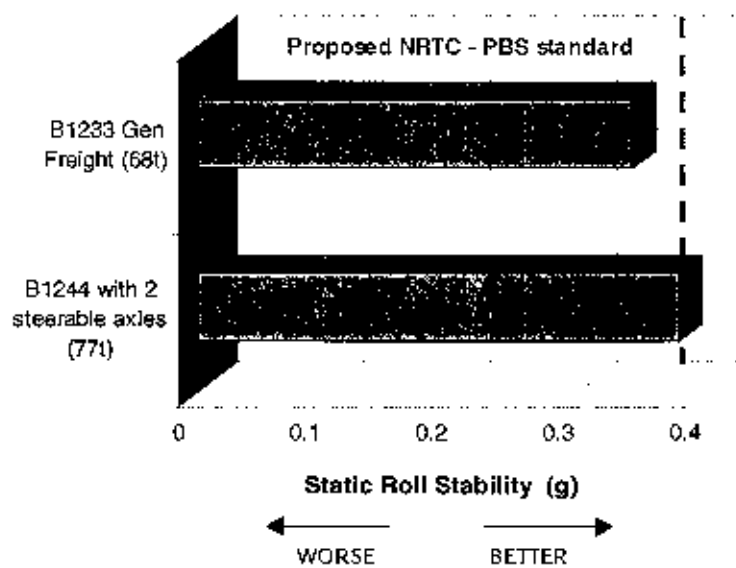


Figure 7 – Stability of “super B-double”

PUTTING THE DRIVER IN THE VEHICLE PERFORMANCE EQUATION WITH ON-ROAD TESTING

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ABSTRACT

The engineering performance of heavy vehicles is a critical element in their access to road systems. Increasingly, vehicle performance must be assessed before permits are granted, or regulations are changed, to allow more productive vehicles to operate on public roads. This is especially true for the larger and heavier vehicle combinations.

Heavy vehicle performance is usually assessed in open-loop manoeuvres under certain sets of conditions. Extensive use of computer simulation has been made for this purpose. However, in order to adequately assess larger vehicle combinations, or problem vehicles, it is often necessary to carry out dynamic testing under typical on-road operating conditions. This allows measurement of vehicle performance plus certain aspects of driver steering performance.

This paper will present the results of several recent studies carried out by Roaduser Systems where both vehicle and driver performance have been measured under actual on-road operating conditions. Results are presented for a range of vehicle configurations from tractor-semi-trailers through double trailer combinations (including B-doubles) to multi-trailer combinations.

The test results are presented in such a way as to encourage benchmarking of on-road vehicle performance and to encourage the use of some common measures for comparing vehicles from around the world.

1 INTRODUCTION

The productivity and safety of heavy vehicle operations, and the way in which they are regulated, depend on their engineering performance. Engineering performance describes vehicle behaviour in terms of its ability to start, stop and turn and its impacts on infrastructure including pavements and bridges.

Vehicle performance assessment has become a useful tool in a world where creep in vehicle limits, albeit based on rational periodic review, has run its course. Performance assessment has become essential in the issue of permits for larger and/or unusual vehicle combinations. Specialisation in road freight vehicles is also creating a myriad of vehicle configurations which defy prescriptive regulation because prescriptive limits rely in part on knowing the vehicle configuration in advance.

Heavy vehicle safety and community acceptance is a major public policy issue. While vehicle engineering performance directly affects a small but significant number of truck accidents (typically 10 % or less), it also has a pervasive effect on many other truck accidents which are currently attributed to "driver error"; mass accident investigation data currently offers poor insight into accident causation and countermeasures, but we do know that truck accidents are highly multi-factor, especially in urban areas. This means driver-vehicle-environment interactions often come into play.

Depending on whether we need a screening tool prior to letting a vehicle on the road, an alternative to prescriptive limits (performance-based standards) or means of investigating truck safety issues after they are manifested we require different types of performance measures. Some measures pertain to the vehicle only and some to the driver-vehicle and driver-vehicle-road interaction. It is prudent to avoid permitting or proliferating the worst-performing vehicles (according to vehicle-based measures) and it is equally prudent to minimise the potential for poor driver-vehicle-environment performance.

The latter is more difficult to measure. While a range of vehicle measures and road measures have been developed, driver measures are few and far between. The current Australian project to develop performance-based

standards (PBS), as an alternative form of regulation to prescriptive standards, has collected and refined available performance measures (1). Recent major Australian safety investigations (2,3) have attempted to apply performance measures to the identification of systemic safety issues related to truck on-road behaviour.

This paper presents stability and control performance measures which select from and go beyond the current "PBS" measures and have provided insight into on-road safety and acceptability issues. This is necessary because remarkably little physical testing of the PBS measures has been carried out.

2 ON-ROAD PERFORMANCE MEASUREMENT

On-road performance measurements have been carried out for a range of vehicle configurations from tractor-semi-trailers through doubles combinations (including B-doubles) to multi-trailer roadtrains. The performance measurements vary in scope and intent and include:

- Vehicle parameters – measures which are properties of vehicle sub-systems (such as suspensions) and are not affected by the manoeuvre or operating environment
- Vehicle performance measures – measures which are a property of the entire vehicle system, apply to a particular manoeuvre and are not intended to be affected by the operating environment
- Driver-vehicle performance/interaction measures – measures which are a property of the driver-vehicle interaction and reflect the operating environment.

2.1 Tractor-Semi-Trailers

Tractor-semi-trailers were tested (2) in response to complaints about unsafe handling. All testing was carried out under typical on-road operating conditions and the range of performance measures included:

- Vehicle parameters (steering ratio, suspension roll steer coefficient, front axle bump steer coefficient)
- Vehicle performance measures (understeer coefficient, roll gradient, response transfer functions)
- Driver-vehicle performance measures (rms steering angle, dominant steering frequency, rms responses (yaw and lateral acceleration)).

The following vehicle parameters were able to differentiate problem vehicles from benchmark vehicles:

- Suspension roll steer coefficient
- Suspension roll stiffness and roll centre height
- Front axle bump steer coefficient.

The following vehicle performance measures were able to differentiate problem vehicles from benchmark vehicles:

- Understeer coefficient
- Roll gradient.

The following driver-vehicle performance measures were able to differentiate problem vehicles from benchmark vehicles:

- Rms steering wheel angle
- Dominant steering frequency
- Rms tractor yaw rate.

This work showed that poor values of certain vehicle parameters (suspension and steering) flowed through to poor performance measures for the vehicle system (handling and stability) and these in turn caused the driver to apply more steering corrections at a higher frequency but the net result was still poorer control of the tractor (yaw rate).

2.2 Doubles Combinations

A range of doubles (see Figure 1) was tested on-road in response to regulatory agency concerns about stability and control for some configurations (3). Certain standard test manoeuvres were also carried out (lane-change and yaw damping).

The following vehicle performance measures were able to differentiate poor combinations from the other combinations:

- Roll gradient
- Response transfer functions
- Rear amplification ratio
- Yaw damping
- Rearward amplification.

The following driver-vehicle performance measures were able to differentiate poor combinations from the other combinations:

- Rms suspension roll angle
- Rms lateral acceleration of rear trailer.

This work showed that there are three key indicators of poor performance in doubles combinations:

- Suspension roll angles developed by all units (expressed as rms values); these are affected by suspension roll characteristics (represented in the roll gradient)
- Various measures of rear amplification
- Yaw damping.

Of these measures, suspension roll angle characteristics and rear amplification can be measured in normal travel or in stylised lane-change manoeuvres. Yaw damping can only be measured in a pulse steer manoeuvre, but this is readily carried out during normal on-road operation.

It should be noted that several measures of "rear amplification" were obtained:

- What was termed rear amplification (rms lateral acceleration of rear trailer divided by rms lateral acceleration of tractor, obtained during normal travel) – this can also be measured using yaw rates
- Rearward amplification as measured in the well-defined SAE lane-change manoeuvre
- Transfer functions between the lateral acceleration (or yaw rate) of the rear trailer and that of the tractor.

The latter transfer functions provide valuable insight into "resonances" in the response of multi-combination vehicles. Figure 2 shows the lateral acceleration transfer function for the B-double. It is apparent that the peak gain of 2.85 occurs at a frequency of 0.8 Hz; by comparison, the standard lane-change test of rearward amplification returns a gain of 1.11 at a frequency of 0.4 Hz. It was found that the standard test missed the peak gain for all doubles tested. Does this matter? Driver steering input has a dominant frequency in the range 0.2 – 0.6 Hz (2) and therefore high gains occurring above this frequency are of lesser concern (unless they coincide with suspension frequencies).

Extensive US research (4) also measured rearward amplification of multi-combinations under actual operating conditions. Measures which were found to be useful in relation to quantifying the stability-enhancing characteristics of improved trailer coupling arrangements included:

- The lateral acceleration transfer function gain described above
- The peak rearward amplification obtained from the transfer function gain
- Comparison of front and rear histograms of lateral acceleration (percent of time spent above a certain lateral acceleration, front versus rear)
- A measure termed the lateral acceleration experience of trailers relative to the experience of tractors; this was computed as the ratio of the percent of time spent above a certain lateral acceleration value (trailer over tractor) plotted against lateral acceleration.

2.3 Roadtrains

Candidate innovative roadtrains of GCM in excess of 100 t have been tested on-route alongside currently-permitted triple roadtrains to provide performance data to guide permitting procedures. Certain standard test manoeuvres have also been carried out (lane-change and yaw damping).

The following vehicle performance measures were able to differentiate poor combinations from the other combinations:

- Response transfer functions
- Rear amplification ratio
- Yaw damping
- Rearward amplification

The following driver-vehicle performance measures were able to differentiate poor combinations from the other combinations:

- Rms suspension roll angle of rear trailer
- Rms lateral acceleration of rear trailer.

A key finding for some roadtrains is that the response transfer functions may have more than one peak. Multi-trailer combinations tend to have a response peak in the lower-frequency range (0.2 – 0.5 Hz) plus a peak in the higher-frequency range (0.8 – 1.2 Hz). Figure 3 shows a typical example for a conventional Australian triple roadtrain. These transfer functions have important implications for performance measurement because:

- The lower-frequency peak may or may not coincide with the standard lane-change test (0.4 Hz)
- The rear amplification may vary rapidly at frequencies varying from 0.4 Hz (adversely affecting the accuracy of rearward amplification measurement)
- The higher-frequency peak tends to be higher for B-coupled combinations
- In some cases, the higher-frequency peak may coincide with the suspension bounce frequency, causing dramatic yaw oscillations under some road conditions.

It is also important to consider the effect of driver steering behaviour on vehicle behaviour, as well as the reverse: the effect of vehicle behaviour on driver steering behaviour. Driver behaviour is evidenced in the power spectrum of steering wheel angle or front wheel angle movements. Drivers normally have a dominant steering frequency of 0.2 – 0.3 Hz. Under demanding conditions or emergency manoeuvres, the dominant frequency increases to 0.5 – 0.6 Hz. It is therefore possible that the driver's emergency mode would excite an undesirable yaw resonance. However, observation of roadtrain drivers suggests that they are well aware of vehicle resonant frequencies and modify their steering behaviour to avoid these frequencies.

3 VOCATIONAL USE OF PERFORMANCE MEASURES

Performance measures which have proven vital in the make-up of problem vehicles, and which may be used to screen vehicles in a type approval sense are the following *vehicle parameters*: suspension roll stiffness and roll centre height, suspension roll steer coefficient and front axle bump steer coefficient.

Performance measures which have a proven track record in the identification of problem vehicles, and which may be used for pre-qualification regulatory purposes under a PBS regime include:

- The following *vehicle parameters*: suspension roll stiffness and roll centre height, suspension roll steer coefficient and front axle bump steer coefficient
- The following *vehicle performance measures*: Static Roll Stability, understeer coefficient, roll gradient, yaw damping, rearward amplification (in various forms including rearward amplification in the lane-change, response transfer function using lateral acceleration or yaw rate, peak response gain and peak frequencies).

Performance measures which have a proven track record in the benchmarking and comparison of vehicles, and which may be used for post-compliance regulatory purposes under a PBS regime include:

- The following *vehicle performance measures*: Static Roll Stability, understeer coefficient, roll gradient, yaw damping, rearward amplification (in various forms including rearward amplification in the lane-change, response transfer function using lateral acceleration or yaw rate, peak response gain and peak frequencies)
- The following *driver-vehicle performance measures*: rms roll angle of each unit, rms lateral acceleration of each unit.

Performance measures which have been able to pinpoint vehicles with stability-and-control safety deficiencies are the following *driver-vehicle interaction measures*: rms steering wheel angle, dominant steering frequency(ies), rms tractor yaw rate.

4 INTERNATIONAL BENCHMARKING OF PERFORMANCE MEASURES

With some important exceptions, relatively little research is available to link performance measures with actual safety outcomes on a statistical basis. One useful and practical way of assessing vehicle performance is to compare the various countries' performance measures applicable to various common categories of vehicle, which could be represented by:

- Most common vehicle configurations which have been generally acceptable over a period of time
- Innovative or special designs which are believed to have superior safety performance
- "Black sheep" configurations which are suspected to be inferior
- Larger and heavier configurations which are permitted for productivity reasons on selected routes or at reduced operating speeds.

With regard to commonly-accepted vehicle configurations, it would be useful to compare the tractor-semi-trailers which are in wide use in many countries with the truck-trailer combinations used widely in some European countries and to a lesser extent elsewhere. While vehicle performance measures for these two configurations may diverge, driver and environmental factors may lead to a convergence in driver-vehicle performance measures.

Important measures for comparison would include:

- Rms trailer lateral acceleration
- Rms trailer suspension roll angle
- Response transfer function
- Yaw damping
- Rms steering wheel angle, dominant frequency(ies)
- Rms truck/tractor yaw rate
- Truck/tractor understeer coefficient
- Roll gradient
- Average operating speed.

All of these measures may be obtained cost-effectively during on-highway operation with a realistic level of instrumentation, and with little risk.

By way of example, Australian data shows the following rms lateral acceleration of tractors:

- In the range 0.05 – 0.10 g in tractor-semi-trailer combinations
- In the range 0.02 – 0.04 g in multi-combinations (indicating greater driver caution).

The rms lateral acceleration of trailers has been found to be:

- In the range 0.03 – 0.04 g for B-doubles
- In the range 0.04 – 0.07 g for other doubles
- In the range 0.05 – 0.07 g for roadtrains (where speeds are somewhat lower).

7 CONCLUSIONS

The safety significance of heavy vehicle stability and control performance exists not only in relation to "good" and "bad" vehicles but also to the ability of the driver to interact with, and compensate for, aspects of vehicle performance.

Certain performance measures have been shown through extensive on-road testing to provide practical differentiation of vehicle quality. These measures include vehicle parameters, vehicle performance measures and driver-vehicle performance/interaction measures.

Consideration has been given to the most suitable of these practical measures for the following purposes:

- Type approval screening
- Pre-qualification in a PBS regulatory regime
- Post-compliance in a PBS regulatory regime
- Pinpointing stability-and-control safety deficiencies.

As definitive research linking vehicle performance to safety outcomes is scarce, more effort should be devoted to international benchmarking of practical performance measures.

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TABLES & FIGURES

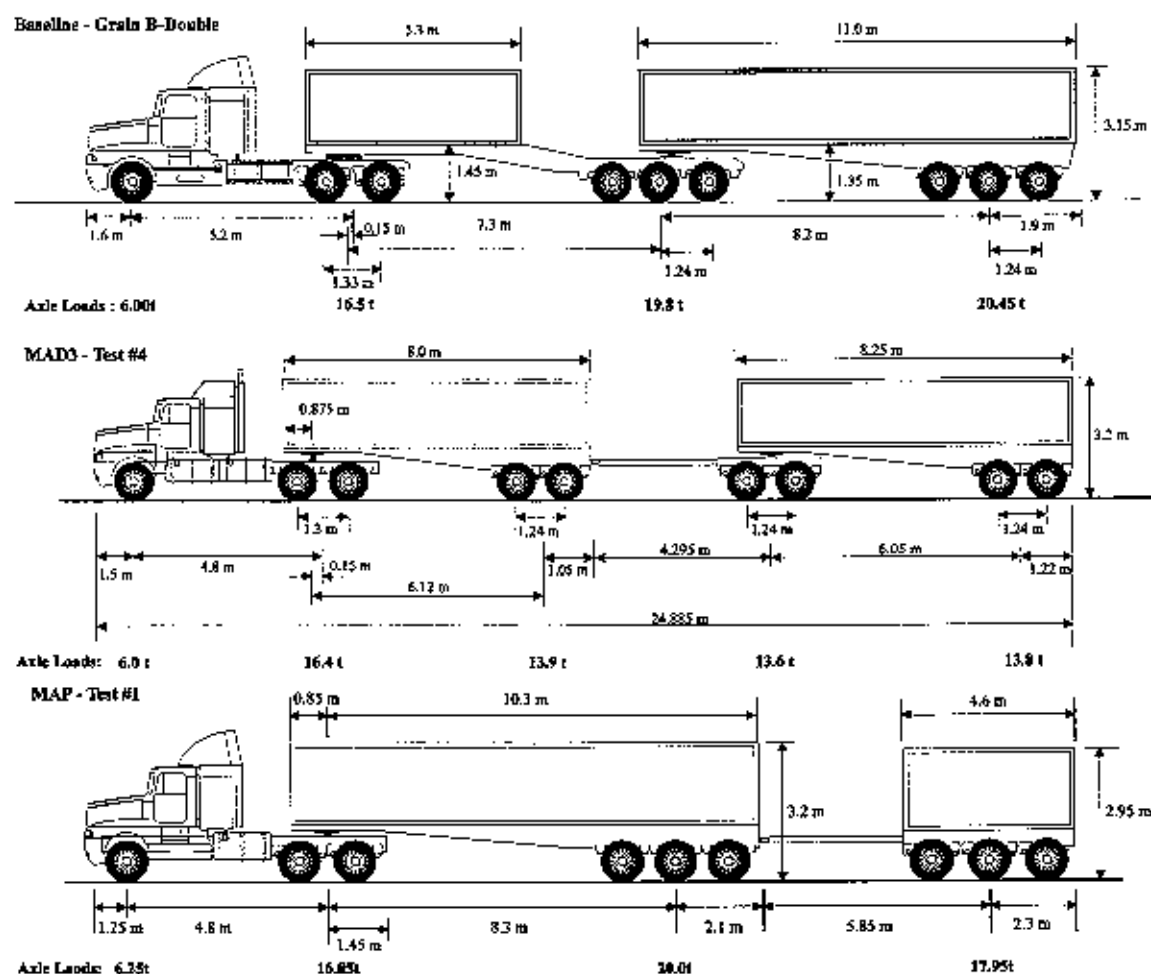


Figure 1 – Examples of doubles combinations tested under typical on-road conditions

B-Double - Transfer Function of Lateral Acceleration from Prime Mover to Trailer 2

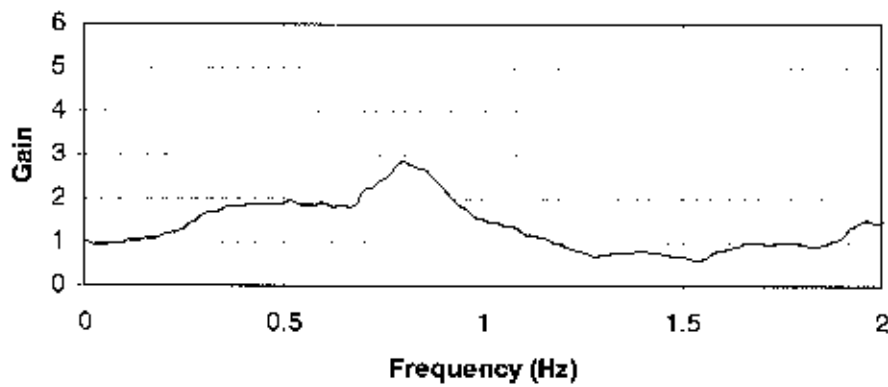


Figure 2 – Transfer function for B-double – obtained under typical on-road conditions

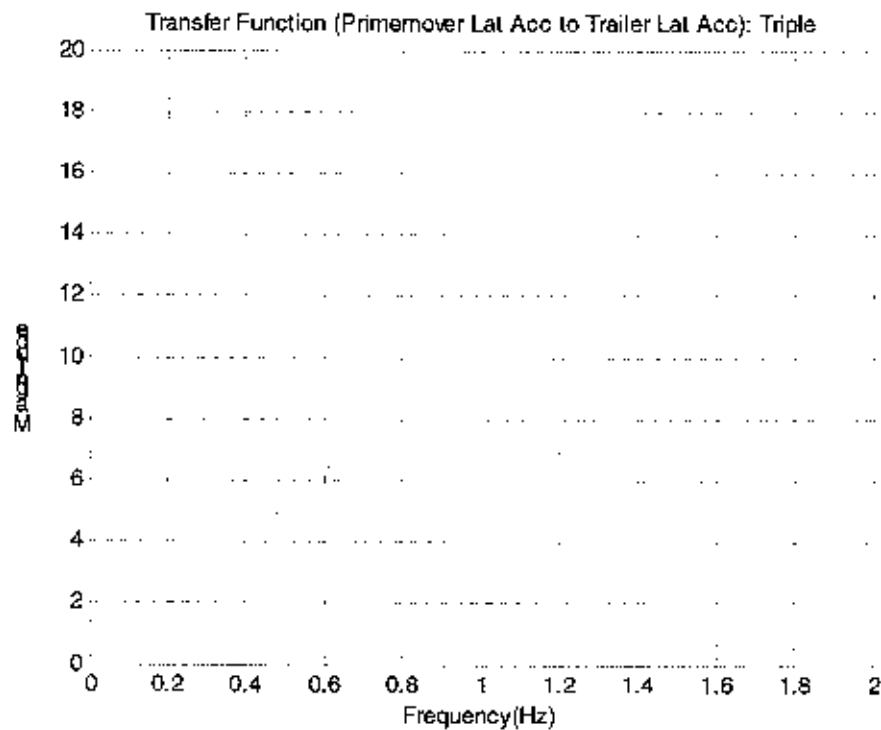


Figure 3 – Transfer function for triple trailer roadtrain – obtained under typical on-road conditions

A RAIL-ROAD HYBRID VEHICLE: DYNAMIC STABILITY ANALYSIS

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ABSTRACT

The paper presents an intelligent logistic system for the area of the Rhine-Schelde Delta (an area within a radius of around 250 km from Rotterdam) as an alternative for, and complement to, the conventional rail and road transport systems. The idea is to design a logistically adequate transport system which does not take up any more space than necessary, has the minimum environmental impact with regard to noise and uses little energy. The system combines the advantages of road transport (speed, flexibility and 'door to door' service) with the lower cost of rail transport for distances longer than 250 km.

The system makes use of vehicles, which, in addition to the usual road wheels sets, are equipped with rail wheel sets. The road wheels are actively steered, are equipped with brakes and possibly also with drives. The lateral control is electronic. The self-centring rail wheel sets can move vertically. In the low position they run on rails to reduce the support forces on the road wheels, thus significantly reducing the vehicle rolling resistance. The rail wheels have no flanges and are not used for braking, ensuring a low noise production.

The mechanical engineering aspects concern the mechanical behaviour of road-rail hybrid vehicles and the rail wheels. The dynamics of the vehicle are modelled and analysed in the software simulation package ADAMS. The results of simulation runs are compared to a model of a reference vehicle with identical mechanical and control parameters but without the rail wheels fitted. The simulations show that the system is robust and stable for the desired operating speed (approximately 55 km/h) of the system. In a succeeding phase, the suspension of the rail wheels and the wheel control system will be designed in detail. The design will be tested with the aid of a specific simulation model.

INTRODUCTION

This study derives from the research programme 'Intelligent Logistic Systems', that has been running within the TRAIL Research School since 1995. Various applications have been developed, including the intelligent transport system on the Maasvlakte. A generic control programming system has been designed, successfully tested in a laboratory with 25 mini robot-vehicles. The architecture of the system is based on distributed intelligence and, as such, is suitable to co-ordinate logistics in areas of any size. A challenging application could be a Transport System Rhine Delta (abbreviated TSR), connecting Maasvlakte and Zevenaar (at the German-Dutch border), with branches to Rijswijk (direction Amsterdam) and Moerdijk (direction Antwerp). Figure 1 shows the plan. On Maasvlakte the proposed TSR is integrated with the terminal transport system.

On the trajectory between Maasvlakte and Zevenaar a new railway is under construction, called the Betuwe Route. Instead of the traditional form, it might be possible to implement a rail-road hybrid design. In this the railway sleepers are extended to a length of 2.7 meter, on which driving strips can be mounted in parallel to the rails. In this set-up the Betuwe Route is both suitable for traditional freight trains but also for automated road vehicles. Along the Nieuwe Waterweg the line continues on dedicated traffic lanes on the national highway R15. The idea is to design a logistically adequate transport system, which does not take up any more space than necessary, has the minimum environmental impact with regard to noise and moreover uses little energy. In this context the paper is focussed on an intelligent rail-road hybrid vehicle, in particular on its functional aspects and its mechanical behaviour.

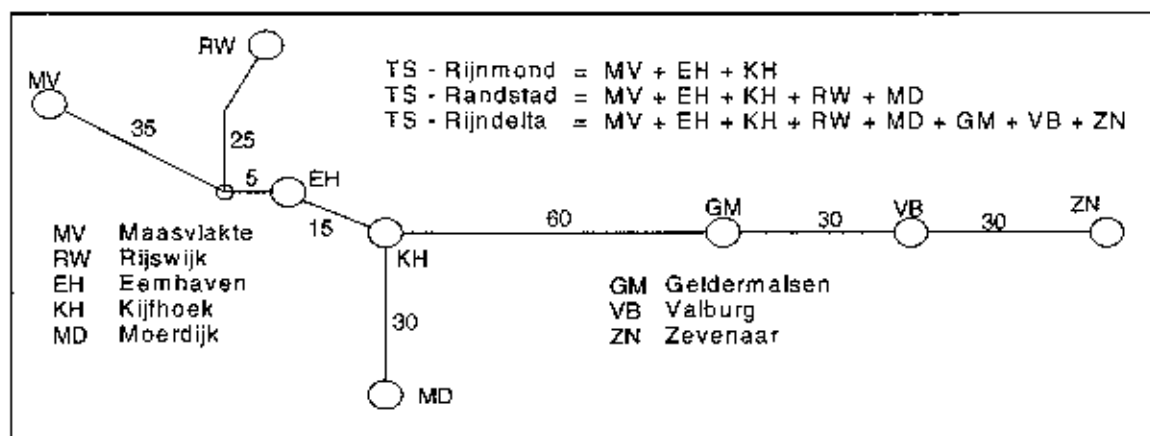


Figure 1: Dedicated infrastructure for freight transport in the Rhine Delta

Low rolling resistance when travelling on rails is favourable from the points of view of energy consumption and economy. With increasing scarcity of energy, this aspect is becoming increasingly important. It is thus desirable to create an energy economical, generally accessible and reliable transport system, with a high capacity, that can combine the advantages of both means of transport, avoid their disadvantages and is suitable for continuous operations.

A road-rail hybrid system profits from this favourable property by making use of vehicles, which, in addition to the usual road wheels sets, are equipped with rail wheel sets. The road wheels are fitted with pneumatic tyres, are actively steered, and are equipped with brakes and possibly also with drives. The lateral control is electronic. The rail wheel sets can move vertically, so that in the low position they can run on rails. This reduces the support forces on the road wheels. They are mounted on the chassis 'following' and are 'self centring'. The rail wheels are quite wide and have slightly conical flanges. They are not used for braking.

With raised rail wheels the vehicle functions as a vehicle running on pneumatic tyres. With the rail wheels lowered, the axle pressure is automatically controlled in such a way that a stable and slip-free movement is maintained. When braking, the vertical forces on the rail wheel sets are reduced as the braking force of the road wheel sets increases. The length of the braking distance thus is equivalent to that of a road vehicle, even when the rails are being used. The hybrid-track can thus facilitate intensive traffic with these hybrid vehicles. A road-rail-hybrid freight vehicle can be almost entirely constructed on the basis of proven technology. The concept is registered under the label P54316NL00. In this paper, a preliminary engineering verification has been carried out with the aid of mathematical simulation models for the dynamic-mechanical properties of the hybrid vehicle.

SIMULATION MODEL USED

General model approach

A dynamic simulation model of the hybrid vehicle is generated in the multi-body simulation program ADAMS. Figure 2 shows the model layout and the main vehicle components. After some primary tests, the following vehicle configuration has been selected for dynamic evaluation:

- The base vehicle is a 4 axle autonomously driven freight vehicle with a gross weight of approximately 40 tonnes. The location of the engine and the drive on the wheels has not yet been accounted for in the model. The centre of gravity height of the vehicle is at approximately 2 meters above ground level.
- Each of the 4 axles carries a load of 10 tonnes. In the schematic, single tires are shown. Due to the clearance with the rail infrastructure, the lateral dimensions of the tires must be minimal.
- Models are generated using a structured approach. Using high level vehicle modules and command input files, different models are made with variable number of axle suspensions or suspension modelling methods.

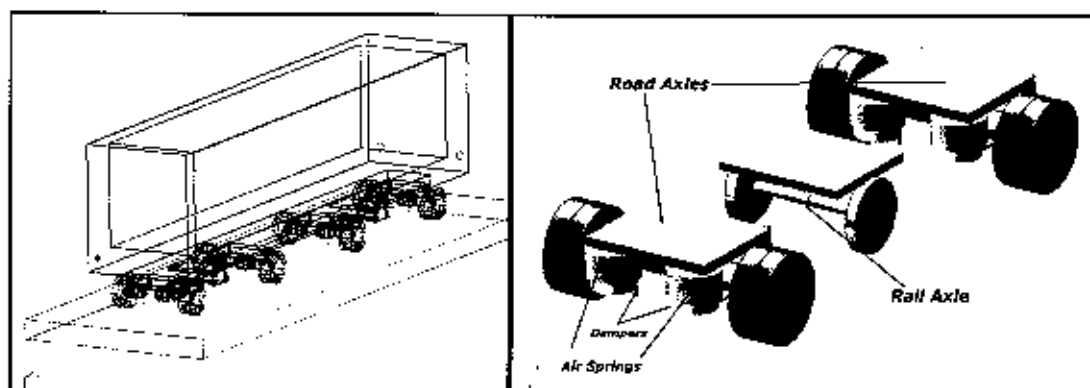


Figure 2: Overview and suspension components of the ADAMS model

- In the centre of the axles of both the front and rear road axle group a single axle rail bogie is mounted on the vehicle. The axle of each rail bogie can pivot about a vertical axis. Steering of each rail axle is controlled using a torsion spring-damper for which proper parameters are investigated in this research.
- The suspension of both road and rail axles is modelled using a generic air spring model. Additional roll stiffness (besides the contribution of the air bags in the suspension) is defined in the parabolic springs acting as trailing arms. Figure 3 shows an overview of the suspension model applied. Icons denote the different connection forces, constraints and the method for following an arbitrary shaped rail path in space. This set-up is not intended to be implemented in the final mechanical design.

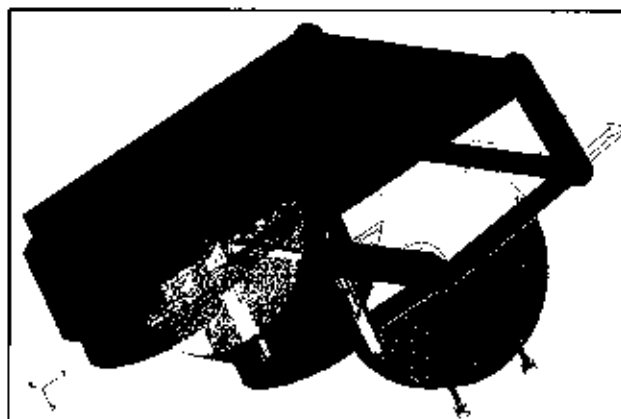


Figure 3: ADAMS model of a rail axle suspension

Axle steering method

The model has four road axles numbered 1 to 4 from the front to the rear axle. All four axles are equipped with an intelligent steering system. The front and rear axle steering angles are defined by a point follower mechanism. It applies a proportional control law on the lateral error signal (ϵ_s) of a virtual point of the vehicle chassis placed at a distance X_s in front of the axle body pivot. Different values of X_s are investigated for the front and rear axle to optimise the vehicle response to disturbances.

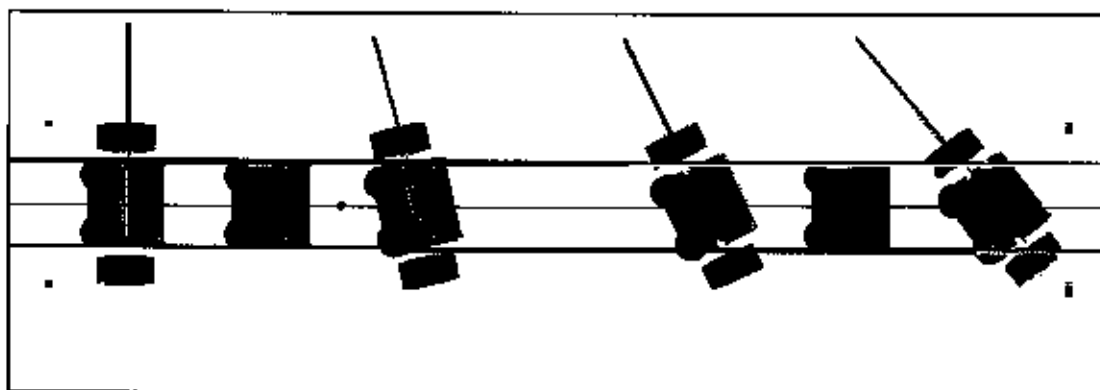


Figure 4: Ackerman steering for the four road axles

The resulting steering control signals $\alpha_{s[1,4]} (= \epsilon_{s[1,4]} / X_{s[1,4]})$ are introduced as control input signal on the front and rear axle. The physical properties of a steering system are accounted for using linear torsion spring-

dampers between axles and sub frames. For the input steering signals of axle 2 and 3 ($\alpha_{S(2,3)}$) it is assumed that the spin axes of all four axles intersect in the same point: the centre of rotation. The geometric parameters of the vehicle and the angles $\alpha_{S(1,4)}$ uniquely define the location of the centre and from that a relatively simple goniometric relation is used to calculate $\alpha_{S(2,3)}$. Due to the small steering angles, it was allowed to use a simpler method where $\alpha_{S(2,3)}$ are derived from linear interpolation of angles α_{S1} and α_{S4} . Figure 4 shows the result for the case that α_{S1} is non-zero.

Wheel load distribution method

As was already mentioned, all axles in the models apply air suspension. In combination with an active axle lift, the distribution of the load of each of the axles can be varied from zero up to half the vehicle weight (in case only 2 axles are on the ground). The initial pressure in all air bags is calculated based on the desired distribution. No dynamic axle load distribution is implemented. If needed, either the rail axles or the inner (axles 2 and 3) respectively the outer road axles can be lifted during a simulation run. An extra advantage of the air suspension in the model is the possibility of varying the vehicle ride height to investigate possible effects on system dynamics.

Contact models

Tire-road contact

The contact between the road axles and ground is modelled with handling type tire equations based on the Magic Formulae as defined by Pacejka and Bakker. The models can apply both a steady state and a transient algorithm for calculating tire slip forces. The tire adaptation tool MF-Tool, a product of the Delft Tyre software, was used to generate tire data files representing tires with a range in nominal vertical tire load. The three tire data sets are used to investigate dependency of lateral stability on tire properties.

Wheel-rail contact

The contact forces defined in the wheel-rail contact are defined by non-linear equations based on simplified slip characteristics. In the model, longitudinal and lateral slip is calculated between the rolling wheel and the rail surface; no spin slip is taken into account. The conicity λ of the wheels is assumed constant which reduces the complexity of the calculations. Wheel conicity, lateral displacement of the wheel y_w and the nominal wheel radius R_0 result in the instantaneous left and right wheel contact radii:

$$R_l = R_0 - \lambda y_w \text{ respectively } R_r = R_0 + \lambda y_w$$

Longitudinal creepage slip between left and right wheel and the rail $\kappa_{(l,r)}$ is defined as:

$$\kappa_{(l,r)} = (V_{x(l,r)} - R_{(l,r)} \Omega_w) / V_{x(l,r)}$$

With:

- $R_{(l,r)}$ instantaneous radius of the wheel at the location of the contact point,
- Ω_w angular velocity of the wheel set,
- $V_{x(l,r)}$ longitudinal velocity of the left and right wheel centres.

The lateral slip between wheels and rails is calculated using:

$$\alpha_{(l,r)} = V_{y(l,r)} / V_{x(l,r)}$$

With:

- $V_{y(l,r)}$ lateral velocity of the left and right wheel at the contact point.

Finally, slip quantities $\kappa_{(l,r)}$ and $\alpha_{(l,r)}$ are used to calculate the longitudinal and lateral slip forces:

$$F_{x(l,r)} = \Xi(\kappa_{(l,r)}) \quad \text{and} \quad F_{y(l,r)} = \Psi(\alpha_{(l,r)})$$

where Ξ and Ψ are saturation functions with physical properties. In these functions, the small slip force slope represents the contact area slip stiffness and the large slip saturation value ($F_{\{x,y\}} = \mu F_z$) represents the contact material friction coefficients. For the steel-to-steel wheel rail contact, longitudinal and lateral friction coefficients are assumed to be 0.2.

SIMULATIONS PERFORMED

Subdividing the different research tasks

The model described in the previous section has been simulated in a variety of parameter settings. The main goal of the simulations is to investigate the lateral stability of the hybrid road-rail vehicle system. The complete analysis task is divided in different groups of simulations, each with a different analysis task.

The following sub-stages were defined and investigated in the research:

1. A vehicle is suspended 100 percent on road wheels, the air pressure in the rail axles is set to zero and the axle lifts for the rail axles are activated so the rail wheels are raised approximately 0.15 meters. Simulations with those settings are performed to create a reference case. Settings of the vehicle parameters were tuned to define a proper reference case. Using this reference case, settings for the lateral disturbance are tested and defined to create similar load cases for all simulations. The sensitivity of the lateral behaviour in the reference vehicle was analysed by variation of overall settings such as vehicle speed and tire parameters.

Based on the reference vehicle settings, rail axles are engaged to analyse changes in lateral dynamic behaviour of the vehicle system. In the simulation runs performed, the load on the two rail axles is varied from 0 to 70 % of the complete vehicle weight.

2. In a second set of simulations, the parameters of the wheel-rail interface are set to create a cylindrical wheel-rail contact (i.e. with zero wheel conicity) with zero lateral and longitudinal friction coefficients. These simulations are labelled *cylindrical wheels on ice* runs, and act as a worse case scenario to answer a crucial first question:

What will happen to the lateral stability if a significant part of the complete vehicle weight is placed on a level frictionless surface (cylindrical wheels), where it essentially does not contribute to the vehicles' lateral stability?

The main goal of this part-research was to examine which effects will dominate the change in vehicle behaviour and to try and determine whether positive effects can be enforced while avoiding the negative effects. This part-research is essential as it is still an option to use the rail axles only for vertical suspension and not at all for lateral guiding of the vehicle. One method to achieve this is by allowing sufficient lateral play in the rail axle suspensions (i.e. up to approximately 0.20 m). The main advantage of this implementation is an absolute minimal weight of the rail suspensions.

3. Having obtained a first estimate of what can be achieved with the hybrid system, a study was done to optimise parameters of the steering system. At first, a simple proportional feedback is used in the steering control algorithm. Using optimisation on time domain results and rules of thumb for the control parameters, a basic PID control is defined for the front and rear steering axles. The improvements in lateral stability of the vehicle are analysed in the range from 0 to 70 % of vehicle load on the rail axles.
4. In a next series of runs, the parameters of the rail surface are varied to analyse which settings for the contacts are required to obtain an optimal matching of the forces accounted for by the rubber-road contact with the steel-to-steel contact of the rail axles. These parameters include the wheel-rail friction coefficient, which can be influenced in the vehicle design in a limited range.

Simulation settings and qualification method

The target vehicle speed is 55 km/h (or 15.3 m/s). The vehicle has a block shaped cargo area with dimensions (l, w, h) = (14.0, 2.80, 3.20) meters. The cargo area roof is at a height of 4.2 meters above the road surface. The vehicle starts at a given velocity and will coast down a straight piece of road-rail without extra driving power

activated. The method to maintain constant driving speed is by having zero rolling resistance and zero aerodynamic friction forces on the vehicle system.

The vehicle is simulated over a period of 30 seconds. At 5 seconds, a step shaped lateral wind gust is applied to the vehicle. The wind gust is modelled as a pure lateral point force located at 6.5 meter from the vehicle front at 2.7 meters above the road surface. The amplitude of the wind gust is 40 kN, which is 10 % of the gross vehicle weight. The wind force builds up to full value in 0.3 seconds, the dominant frequency of it being significantly higher than the roll mode of the vehicle suspension, which is approximately at 0.28 Hz for the fully road suspended vehicle. Thus, the wind force is assumed a step input so that different situations can be compared without being influenced by interference with the roll mode. As the wind force amounts up to 10 % of the gross vehicle weight, the steady state response of the vehicle is equivalent to the response to a 10 % camber angle in the road surface or cornering at 0.1 g lateral acceleration.

During the simulations, a number of signals and vehicle characteristics are calculated for analysis or post processing after the runs. Typically, of the response signals, peak value Y_{peak} (occurring between 7 to 8 seconds) and the value at the simulation end Y_{steady} (30 seconds) are used as objectives in further design studies.

For example:

Plotting Y_{peak} and Y_{steady} for $Y_{max,road}$ versus vehicle speed for 10 different runs with vehicle speed varying from 5 to 20 m/s quantifies the effect of vehicle speed on steady state and peak value of lateral response of the vehicle road axes.

A number of time signals were used as vehicle stability indicators:

- $Y_{max,road}$: Maximal lateral position of any road axle: the lateral deviations of the centre points of all four road axes are used to calculate the extreme lateral displacement of any of the road axes in any point in time. The maximal lateral deviation of any rail axle was also monitored but was found to be less critical than that of any road axle.
- Car_{roll} : The roll angle of the vehicle body measured in a plane perpendicular to the road surface.
- $F_{z,rail\%}$: The total sum of vertical rail forces as a percentage of the total vehicle rail forces. Obviously, the pressure in the suspension element air bags dictates the initial value. During the simulation, the signal monitors possible large changes that occur due to dynamic effects. This signal is mainly used to check correctness of the vehicle settings and to indicate unexpected effects such as grounding of wheels on lifted axles.
- $F_{y,road,max}$: The largest value of the lateral forces exerted by any of the road axle suspensions.
- $F_{y,rail,max}$: The largest value of the lateral forces exerted by any of the rail axle suspensions.

Typical simulation results

This section shows some typical results of the simulation runs. Time plots of vehicle responses are shown and explained to introduce the design study plots used in the remaining sections.

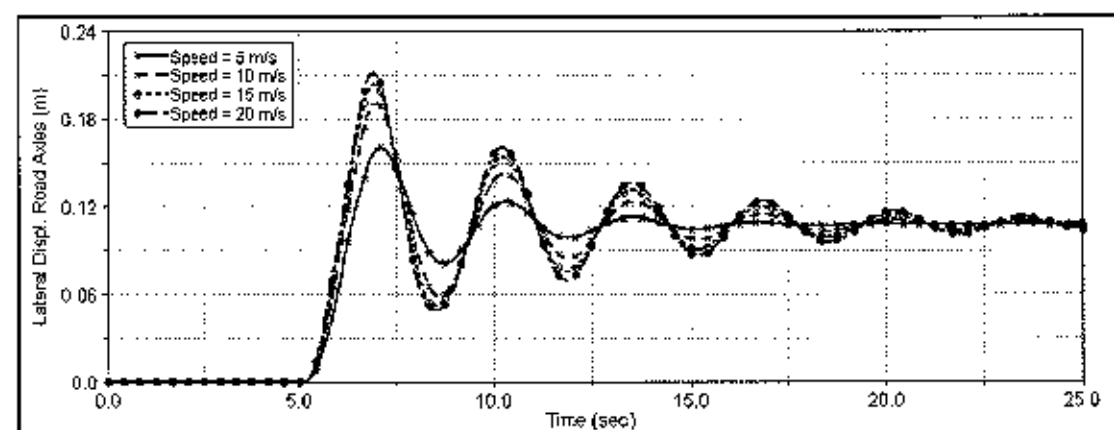


Figure 5: Vehicle speed variation at zero rail axle loads, maximal lateral displacement of any of the road axes

Figure 5 shows the largest value of the lateral deviation of all road axes. The results show that the steady state value is independent of speed and approaches a value of 0.10 meters. The peak value of the lateral axle displacement strongly increases with the vehicle speed; at 20 m/s the peak value is 0.21 meters.

Figure 6 shows the response of the vehicle roll angle, for this signal both the peak value and the steady state value hardly depend on vehicle speed. As expected, the overall system damping decreases with increasing speed.

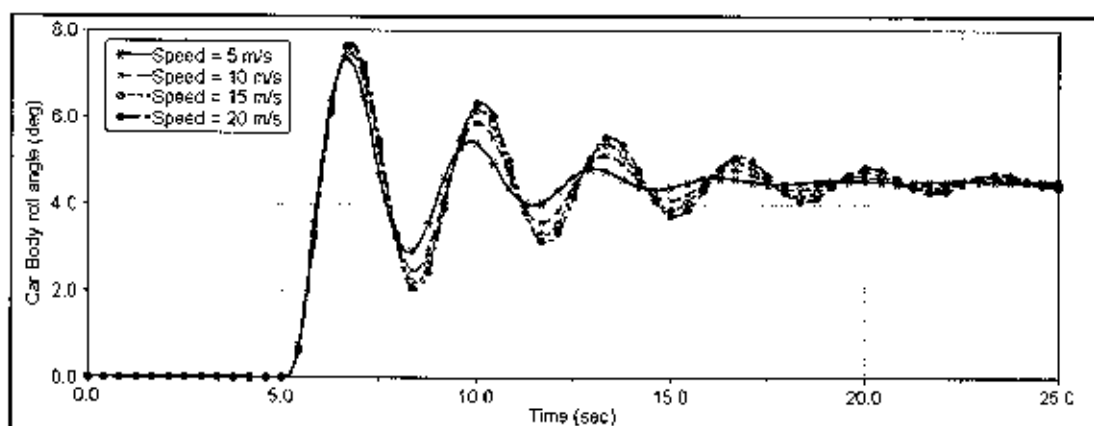


Figure 6: Vehicle speed variation at zero rail axle loads, car body roll angle

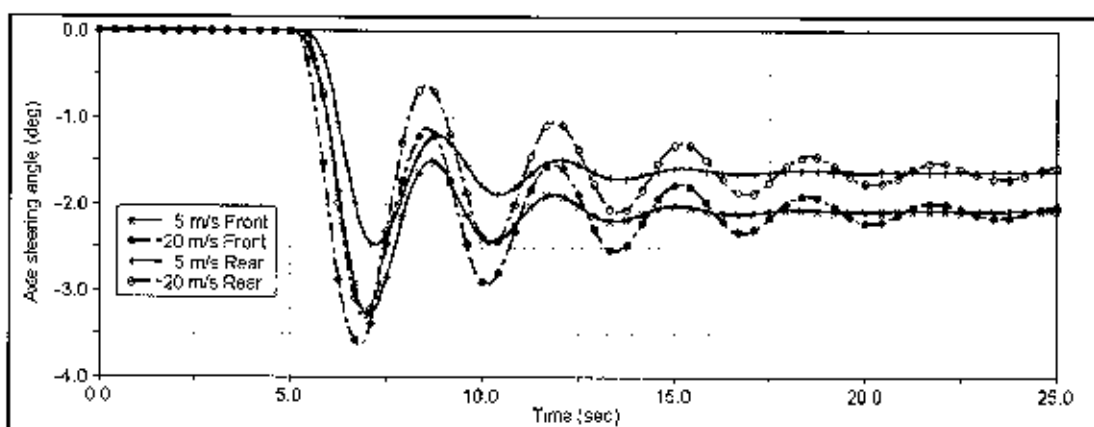


Figure 7: Vehicle speed variation at zero rail axle load, front and rear axle steering angles

Figure 7 shows the response of the front and rear steering angles. It appears that the steady state steering angles are independent of speed and are -2.1 and -1.6 degrees for the front respectively the rear axle. The peak values of the steering angles strongly depend on vehicle speed, the changes of the front and rear steering angles with speed appear to be similar.

RESULTS OF DESIGN STUDIES

In the simulation model, a macro is defined to lift certain axles (rail or road) depending on a design variable denoting the distribution of the axle loads. Using this model functionality, a continuous range of rail axle loads could be defined from 0 up to 100 % of the total vehicle load. The simulation results obtained with the reference model with lifted rail axes (0 % load on rail axes) are included in the remaining results as the minimal value of the scale denoting the percentage of load on the rail axes.

Results of cylindrical wheels with zero friction

In Figure 8, the main results are summarised of gradually transferring vehicle load from the rubber-to-asphalt wheel-road interface to a frictionless level wheel-rails interface. Figure 8 shows curves for the peak value, the

value at simulation end and overshoot values (= peak value - end value) for lateral displacement and car body roll angle. The interesting fact is that the lateral stability of the vehicle does not decrease immediately. This effect seems to be strongly related to the increase in roll stiffness of the complete vehicle. The car body roll angle, shown at the right side of Figure 8, decreases due to the extra roll stiffness introduced by the two rail axes touching ground at non-zero value of $Pz_{rail\%}$. In the design settings, the additional roll stiffness of the rail axes is identical to that of the road axes. The positive effect of extra vehicle roll stiffness causes both the steady state value and the peak value of the vehicle lateral displacement to decrease in a range for of $Pz_{rail\%}$ of 0 to 20. At higher rail axle load percentages, lateral stability decreases at an increasing rate. At roughly 40 % rail axle load the peak value of the lateral displacement equals the value at zero rail axle load.

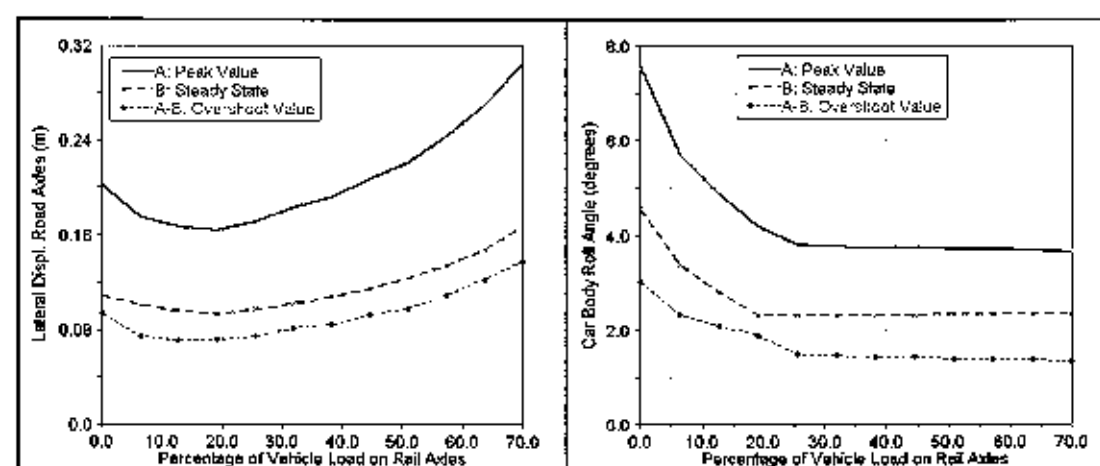


Figure 8: Effect of vehicle load transfer from road to rail axes (cylindrical wheels on ice)

To further investigate the effect of roll stiffness on lateral stability, the additional roll stiffness of the rail axes is varied. This stiffness results from the torsion stiffness about the vehicle lateral axis due to bending stiffness of the parabolic leaf springs connecting the air bags to the rail axle body. Figure 9 shows that a lower value of the additional roll stiffness deteriorates the lateral stability. At extremely high rail axle roll stiffness, the lateral deviation only increases above 60 % rail axle load. The left plot of Figure 9 presents the steady state value of the axle lateral displacement; the right plot gives the overshoot value. Both steady state response and overshoot improve with an increase of the axle roll stiffness.

At the relatively high lateral wind load applied a decrease of rail axle roll stiffness causes vehicle roll over at high rail axle loads. This explains the difference in results for case $C_{roll Rail} = 0.3 \cdot Road$ in Figure 9 above 55% rail axle load. This indicates that, to prevent rollover in all cases, the roll stiffness of the rail axes must be defined at a level higher than that of the road axes.

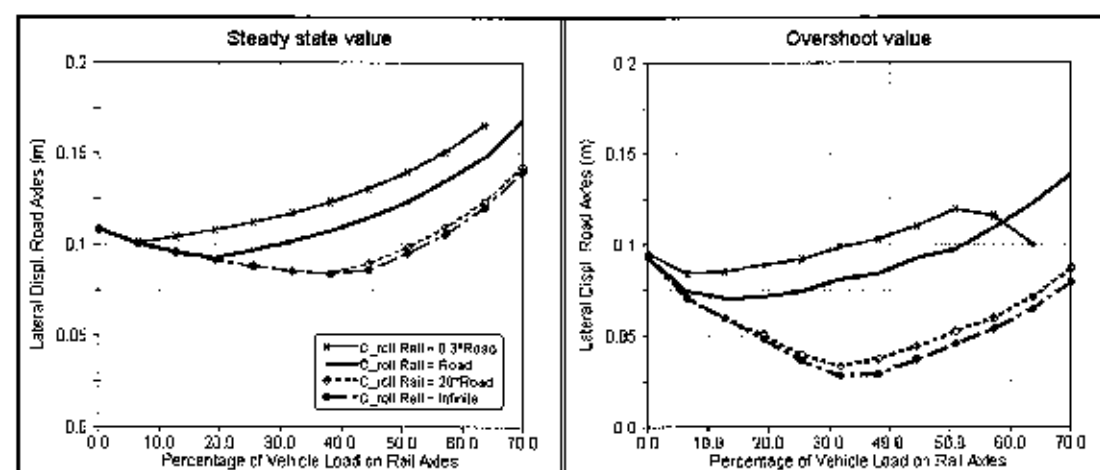


Figure 9: Effect of rail suspension roll stiffness on lateral vehicle stability (cylindrical wheels on ice)

Effect of wind excitation

A small investigation is performed on the influence of the wind force application. Results not shown indicate a linear relationship between wind force amplitude and lateral response peak and steady state values. For Figure 10, the position of the application point of the wind force position was varied in three different rail axle load settings. The three lines denote the peak values of the largest road axle lateral displacement. Especially at 0 and 30 % rail axle load, the effect of the wind force location is not large. At approximately 6m behind car front (1m in front of the car centre) we have a minimum. Furthermore, at 30 % rail axle load, the system lateral stability is still comparable to the reference vehicle with 0 % rail axle load. However, at 60 % rail axle load the peak lateral response increases significantly for all wind force positions.

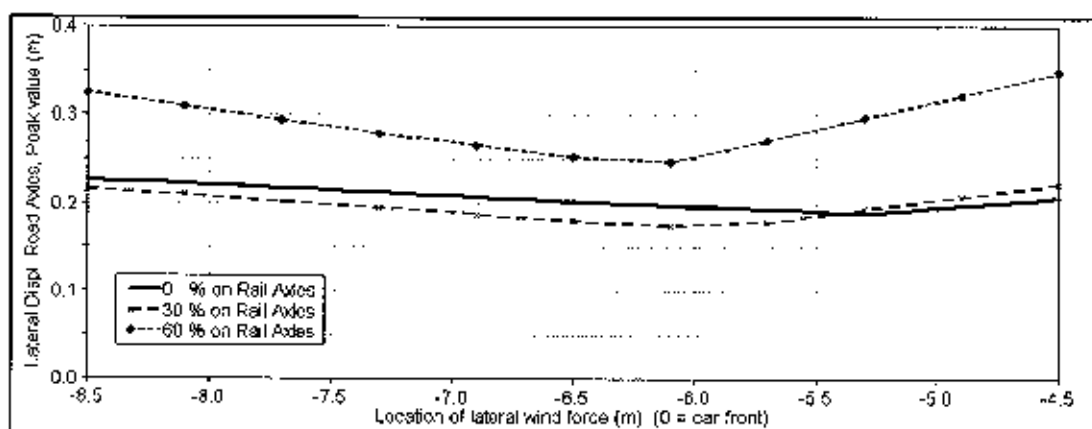


Figure 10: Effect of wind force application point (cylindrical wheels on ice)

Effect of tire parameters

An important parameter for road tires is the characteristic of the lateral slip stiffness versus the vertical load on the tire. Figure 11 shows three characteristics for different values of the tire nominal vertical load. This nominal load can be considered as the vertical load where a tire is designed to operate in. The dark vertical lines denote the average tire load in each tire at 0 and 60 % vehicle load on rail axes.

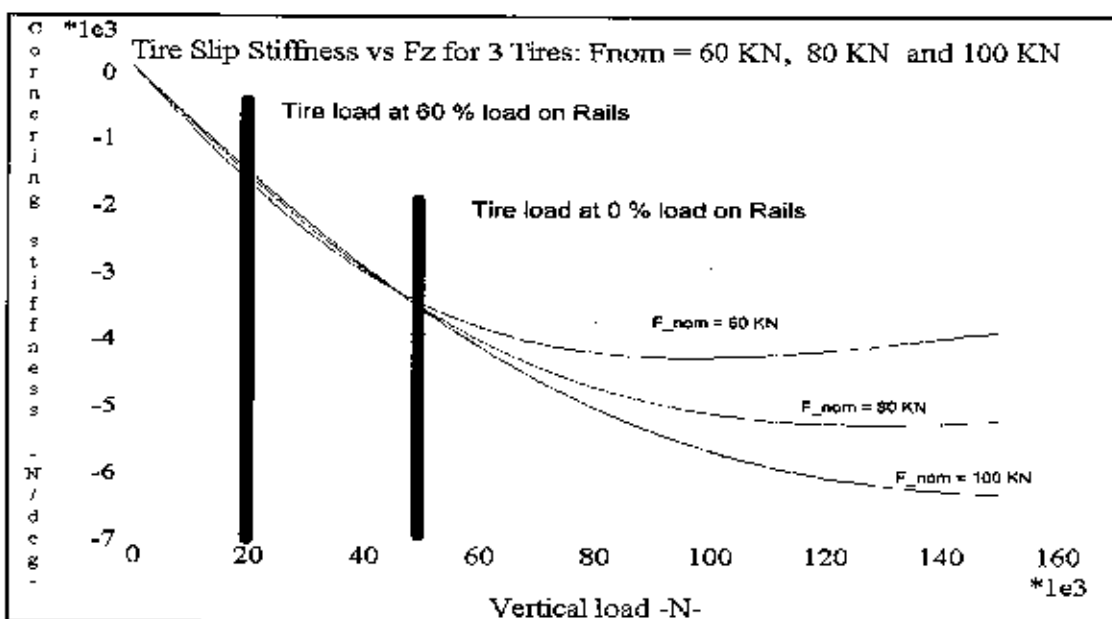


Figure 11: Three tire characteristics with variation of lateral slip stiffness versus vertical load

Due to lateral tire forces, (i.e. while curving or during a lateral wind) a wheel load transfer will take place from one side of the vehicle to the other. Due to this load transfer, the wheel loads left and right will be evenly distributed at

both sides of the solid vertical lines in Figure 11. The resulting decrease of the effective axle lateral slip stiffness depends on the amount of load transfer and the curvature of the tire slip stiffness versus tire vertical load. Figure 11 thus gives one explanation of the increased lateral stability at increased rail axle roll stiffness. A decrease in road wheels load transfer left-right reduces the decrease of axle sideslip stiffness, which increases the lateral vehicle stability.

Figure 12 shows the effect of changing the tire nominal loads. The tire with $F_{\text{nominal}} = 60 \text{ kN}$ has the strongest load transfer dependency and shows the largest lateral deviation at zero rail axle load. With an increasing rail axle load, this tire benefits most of the reduced road wheel load transfer. This means that the proposed system is robust with respect to the choice of the tires. In case of (small) vehicle overloads, the lateral deviations of the system will increase less than proportional compared to a properly loaded vehicle.

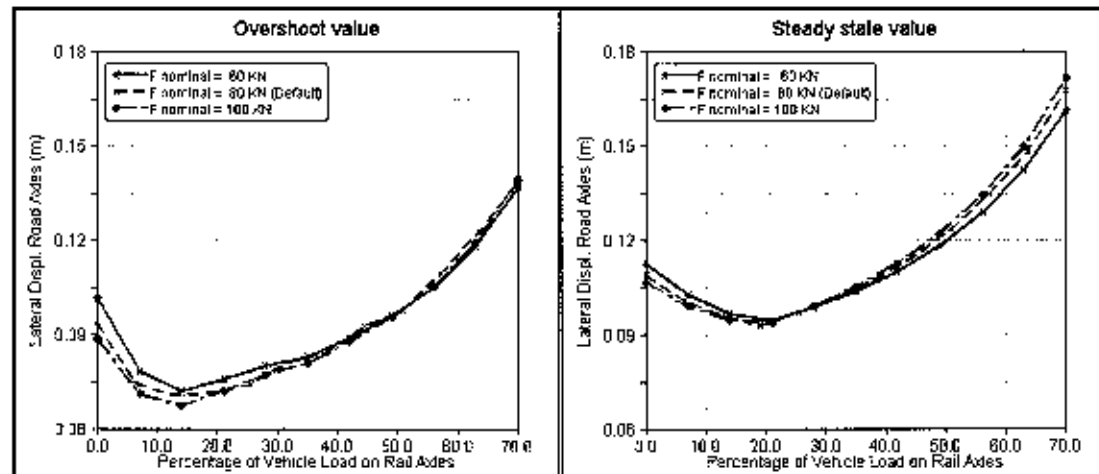


Figure 12: Effect of tire nominal load on vehicle lateral stability (cylindrical wheels on ice)

Implementation of steering PID control

Up to this point in the research, the parameters of the steering system have not been given special attention (only proportional control has been used with $P=1$). In this section a PID control is designed to improve the proportional steering feedback for the front and rear axles. At this stage no curving of the track has been investigated. In a straight track, the preview length parameter is equivalent to the reciprocal value of the proportional feedback parameter of the steering system. Therefore the preview length parameters of the steering system have been fixed to the design values. They will be considered in a later phase of the design and will have to meet conflicting criteria such as vehicle manoeuvrability and sensitivity to rail irregularities.

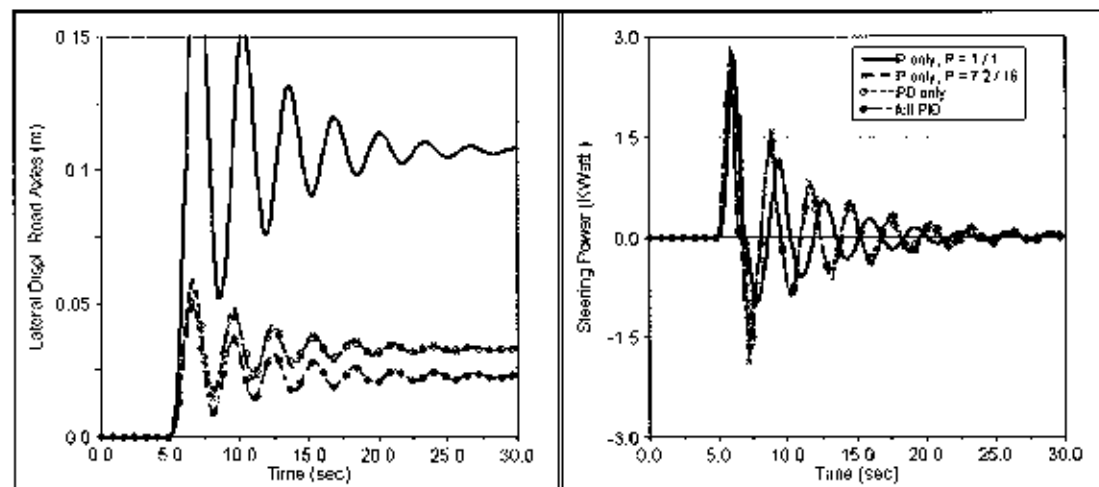


Figure 13: Effect of steering system PID on vehicle behaviour (cylindrical wheels on ice)

Using a practical iterative approach, reasonably optimal PID parameters for the steering system of front and rear road axles are assessed. This resulted in $(P/I/D) = (7.2/8.0/2.7)$ for the front axle and $(P/I/D) = (16.0/ 5.0/ 4.0)$ for the rear axle. Figure 13 shows the results of the vehicle lateral stability in a number of phases of PID design. It is clear that while the lateral deviation of the vehicle is drastically reduced, the power consumption of the steering system is hardly affected. The same wind gust as before (at 6.5 m) has been used. The results shown in Figure 13 indicate that the biggest lateral deflection of the road axles can be reduced to a peak value of 0.05m and a steady state value of 0.025m. The current design of the vehicle assumes a lateral play of the rail axles of approximately 0.1 meters. This means that even at this large lateral excitation, the amount of play in the rail axles is still sufficient. Therefore low lateral forces in the rail axles can be expected. In the next sections this will be verified.

Using rail axles for extra lateral guidance

The results of the simulation runs discussed before indicate that at proper system parameters the lateral stability of the vehicle can be controlled quite well. The results of the above simulations must be considered as worst case as no slip forces are being generated between wheel and rail. In this section, realistic wheel-rail frictional contact will be considered and the effect of conicity will be investigated. A further improvement of system behaviour is expected because the rail axles will now actively contribute in the lateral suspension as well. The simulations will be used to indicate the cost: how big are the lateral forces applied on the rail axles to obtain a certain improvement. This knowledge will be used in the design of the hybrid system.

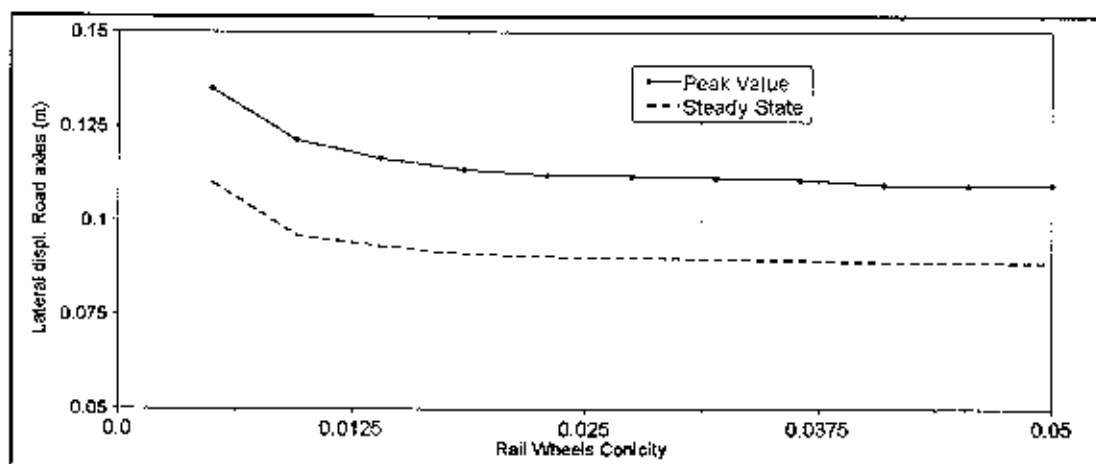


Figure 14: Simulation results of the vehicle on rail axles only, wind force decreased to prevent flanging

Simulations are performed with 100 % load on the rail axles and lifted road axles to find optimal rail vehicle settings. The rail forces are not able to create sufficient lateral forces to counteract 0.1 g, as flanging is not included in the wheel-rail contact. Therefore the amplitude of the lateral wind force is decreased to 4.0 kN. In the resulting design, a lateral stiffness is defined of $3.0e4$ N/m between the rail axle body and vehicle. Above 0.1 meters deflection, a hump stop stiffness of $3.0e7$ N/m is applied. Also, the system appears to be rather sensitive to lateral damping in the axle suspension. The optimal damping was found to be of the order of $1e4$ Ns/m. Figure 14 shows that limited conicity promotes stability. The results shown in Figure 15 indicate that beyond a conicity value of ca. 0.15 an increasingly violent oscillatory hunting instability occurs. To ensure inherent stability of the system (also when the steering control fails) a small value of the wheel conicity of say 0.01 may be selected.

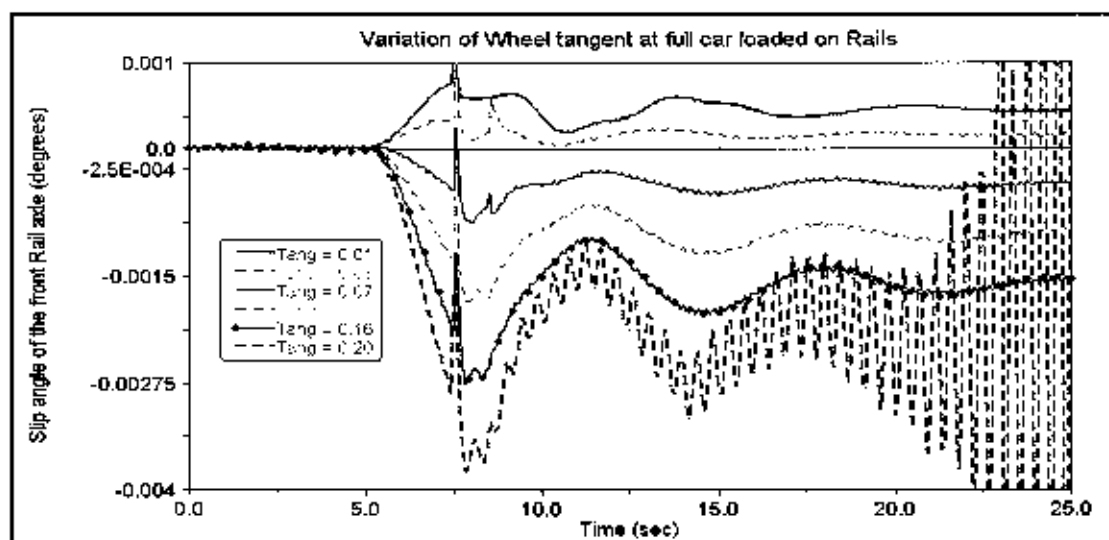


Figure 15: Increasing instability for the full rail vehicle at high wheel conicity values

Figure 16 shows the results of simulations with a variation of the percentage of load on the rail axes. Runs have been performed both with cylindrical wheels on ice and on a normal rail with variation of wheel conicity. The left plot of Figure 16 shows a minimal effect of conicity and wheel-rail friction. In all cases, the peak value of the lateral displacement decreases with increasing load on the rail axes. Note that with a proper PID steering control, the vehicle system is always more stable when using the rails than in the 100 % road case (extreme left side points in the plots).

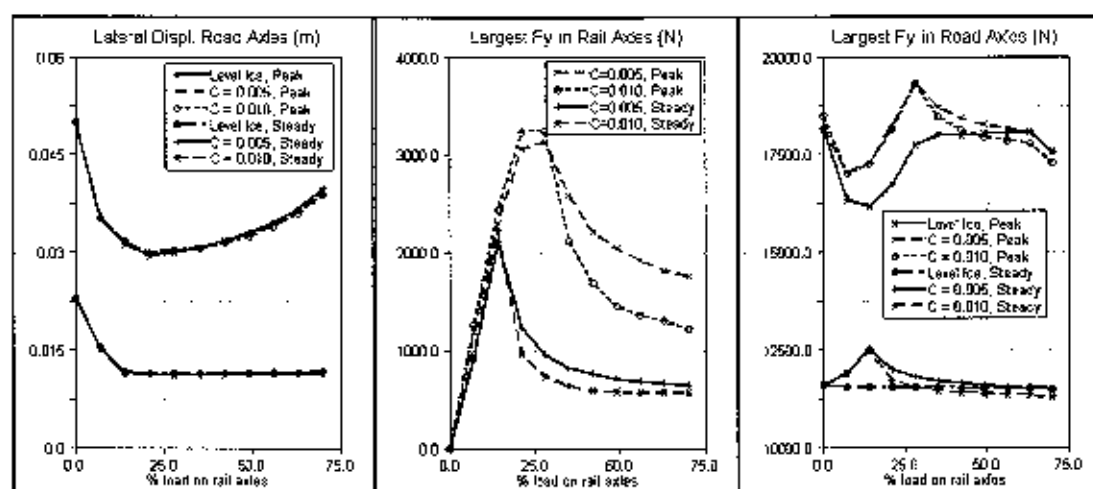


Figure 16: Effect of rail parameters on vehicle dynamics, optimal PID steering

The centre picture of Figure 16 indicates that lateral forces in the rail axes will remain small. The similar shaped curves for the rail axle lateral forces seem to be influenced by the increase of self centring behaviour at increasing rail axle loads. The lateral forces in the road axes show no dramatic changes (right-hand diagram).

Next, a design study is performed to analyse the effect of wheel conicity on the hybrid vehicle. Figure 17 shows the results of two design studies with variation of conicity from 0 to 0.015. In the studies, the rail axle load is set to 30% and 60% respectively. No significant changes in lateral axle displacements are found while varying the conicity (not shown). Figure 17 shows that both the road and rail axle lateral forces decrease at increasing load on the rail axes. Furthermore we see that at increasing conicity all lateral forces decrease slightly. At conicity higher than approximately 0.01, rail lateral forces will start to increase again. This is an extra reason to set the conicity to 0.01.

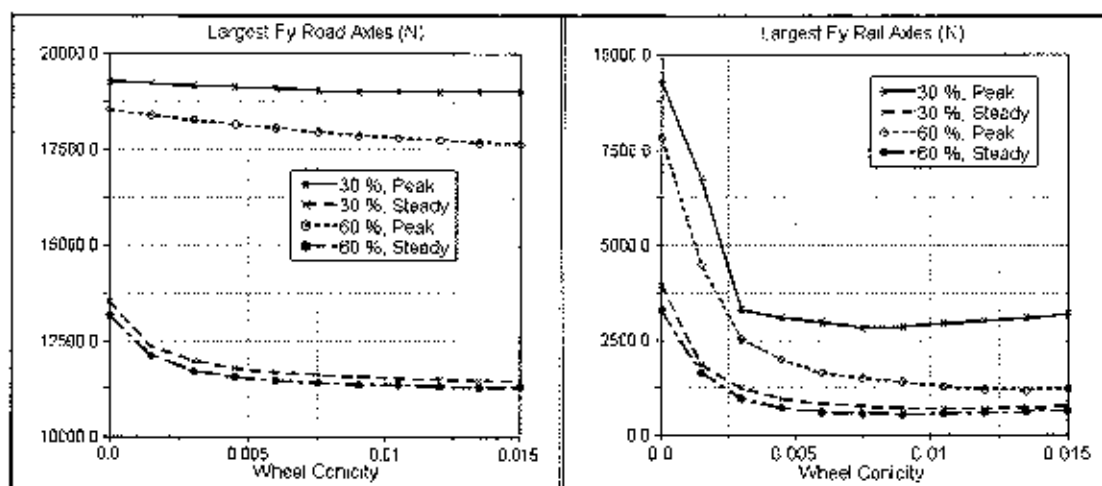


Figure 17: Effect of wheel conicity and rail axle load on road and rail lateral forces

The effect of lateral play in the rail axes on rail forces and lateral deflection is obtained by varying the deformation at bump stop engagement between 0.01 and 0.1 meters. No significant changes are found at higher deflections. The results in Figure 18 (conicity = 0.01) show that at 30 % load on the rail axes, rail axes require a lateral play of 0.06 meters. At play values below 0.06 meters, large peak forces occur both for the rail and the road axes. At 60 % load on rail axes, the self-centring effect of the rail wheels reduces the required amount of play to a value of 0.02 meters.

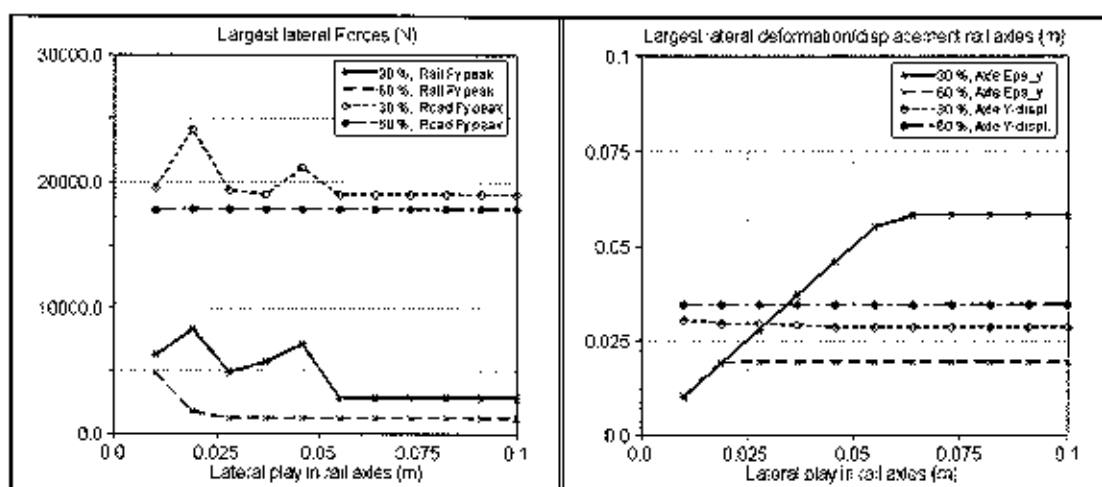


Figure 18: Effect of lateral play in rail axes on lateral forces and deviations

Overall conclusions from these simulations are that for lateral excitations up to the level applied in these simulations, a lateral play in the rail axes is advised of more than 0.075 meters (0.15 meters in total). Novel rail axle suspensions may be required to further support the good lateral stability encountered here at low lateral forces in the rail axes. Further research into this area is beyond the scope of this paper.

Using the defined rail axle settings, simulations were performed with speed varied between 5 and 20 m/s. Figure 19 shows the resulting peak- and steady state values of the lateral axle displacement. All results of preceding steps in the design are included. The top labelled curves denote the vehicle with 60 % rail axle load on wheel with conicity 0.01 and only P=I steering control. Note the speed dependency in the peak response for this curve. Due to the PID steering control, the peak response and steady state values are decreased by a factor of three to four while the speed dependency is fully suppressed. The curves labelled C=0.0 indicate runs with cylindrical wheels on ice, they completely overlap with the results labelled C=0.01. The curves with PID control all show a stable behaviour, indicating only minor dependency on rail axle load, rail wheel conicity and friction between wheels and rails.

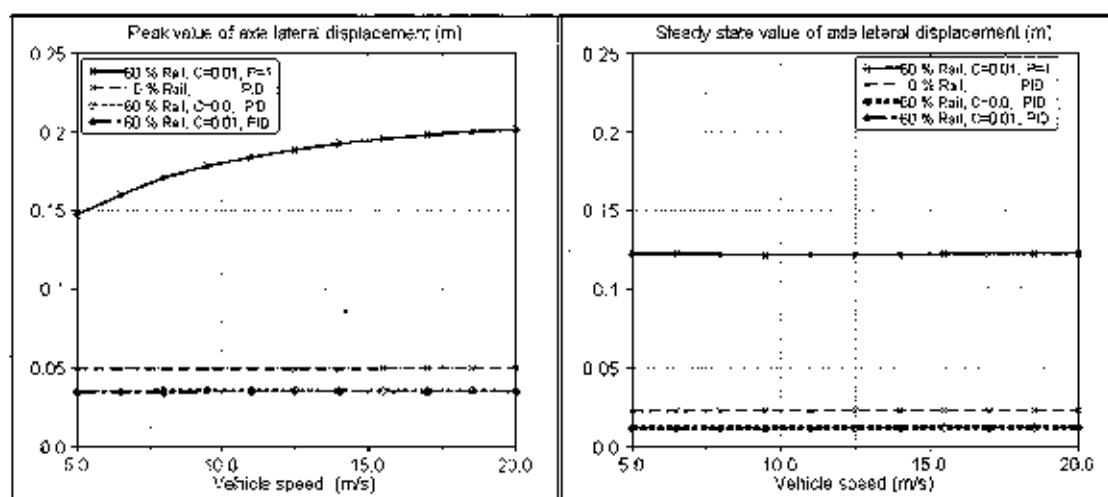


Figure 19: Effect of vehicle speed on lateral stability, variation of steering PID, axle load and rail contact

CONCLUSIONS

From the analysis and simulations described, the following conclusions may be drawn:

- A system has been presented and described with high potential for novel use of (partly) existing infrastructure. The presented system combines the benefits of a rail vehicle (low rolling resistance) with the benefits of a road vehicle (flexibility in manoeuvring).
- For the expected range of operating velocities of the hybrid vehicle the dynamic simulations have not indicated major dynamics problems. The vehicle system, based on a standard set of vehicle parameters demonstrates a predictable response to large lateral disturbances.
- A large number of dynamic simulation runs is performed to investigate the effects of reducing the weight on rubber tires versus the weight on rail-wheels. Conclusions of these simulations are:
 - Using a straight forward steering feedback mechanism (with proportional control only) it is possible to control the lateral stability of the vehicle.
 - The lateral stability of the vehicle is dominated by the increase of roll stiffness due to the rail axles suspensions. Extremely large roll stiffness values can even improve vehicle stability up to a large amount of road axle unloading.
 - In general, the system proposed seems to behave robustly with respect to tire mismatch problems.
 - The steering system improves dramatically by implementation of a reasonably simple PID control strategy. Applying one single strategy for the different vehicle settings already gives a large reduction of the vehicle lateral displacements. Further improvement may be possible with more advanced control strategies (self learning, case dependent control settings).
 - The extra lateral stabilisation of the vehicle system from lateral guidance of the rail infrastructure (through conicity) appears to be relatively small. In combination with the need for an extremely light rail axle construction, this leads to the proposal to design rail axes systems that do not account for any lateral guidance forces at all.
- An automated approach for modelling complex vehicle systems is strongly advised for performing a complicated full-vehicle analysis as described in this paper. Working in a macro oriented manner with adequate separation of model data (design variables) and model topology/functionality (model components) enables users to perform large and complex simulations at a high quality level.

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REVERSE ENGINEERING OF A TRANSIT BUS FOR F.E. CRASHWORTHINESS ASSESSMENT

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ABSTRACT

A reverse engineering process was used to develop a finite element (FE) model of a transit bus for the crashworthiness and impact analysis. The process involved disassembly and digitization of the bus components, building finite element meshes, and merging all parts into one, complete bus model. Before tear down process began, a global coordinate system was established by marking reference points, and taking measurements with a portable coordinate measurement equipment. All structural components were identified, labeled and removed from the bus structure. Connections between all structural parts were identified. After disassembly, geometric entities as lines, points and curves were scanned for each component. The geometric data obtained was transformed to the MSC PATRAN pre-processor, where finite element grids were generated, modified and merged. Individual parts were assembled. FE model of the transit bus also includes data from mass and thickness measurements, and material properties obtained from laboratory tests conducted for selected material samples.

The transit bus model is still under development; it will be subsequently validated, modified and improved. Connections between vehicle components are modeled by merging nodes, applying node constraints and spot welds. Nonlinear, dynamic finite element analyses are performed using LS-DYNA, an explicit, 3-D, dynamic, nonlinear finite element code. Rollover, rear and side impacts of the complete bus model will be studied to evaluate crashworthiness of the bus structure and safety of passengers. The current results will be presented on the Symposium.

INTRODUCTION

Florida Department of Transportation acquires over 300 buses and vans for public transportation each year. A large part of this fleet is designated to transport disabled passengers, often in wheelchairs. Due to the unique safety requirements for these vehicles on one hand, and the lack of strict standards for crashworthiness evaluation, on the other, there is a need for an advanced study of structural integrity and dynamic behavior during accidents. These vehicles, and especially their body structures, may have mechanical properties, such as component connections or new composite materials, which make them more vulnerable to a specific damage. In extreme cases, this situation can result in more risk during accidents for disabled passengers who, in contrary, should require special care and additional protection.

The paper presents the first phase of the research conducted to evaluate the crashworthiness of transit buses through computer impact analysis (simulated, virtual crash-testing). It illustrates a process of building a finite element model for a Ford Eldorado Aerotech 240 transit bus (see Figure 6). Because of potential liability, motor vehicle manufactures are reluctant to share their proprietary FE models with universities and other roadside research organizations. Since design data was not available for the bus selected by research sponsors, the reverse engineering was performed in order to create a reliable FE model. The process required vehicle teardown and digitizing of all major, structural parts of the bus.

A typical engineering process begins with an engineering concept, which often follows with an analysis and design. Desirable material properties are studied and selected materials are used for part production. Finally, all required parts are assembled in a final product or a system.

Reverse engineering proceeds in the opposite direction. It begins with a final product, which is disassembled into individual parts. The parts are taped, scanned, digitized, and mapped into computers for further analysis of the

entire design and manufacturing process. Reverse engineering assists engineers in further improvement of their product. Here, the process involves disassembly and digitization of the bus components, creating finite element meshes, and merging parts into a complete bus model. Although nowadays the manufacturing process is frequently aided with computers to reduce possible errors, the final product is not always free of flaws, which could be introduced at any stage of the engineering process. A reverse engineering process is therefore needed to: identify all potential problems in all stages of the product manufacturing process, and assess an impact of different engineering options/design and manufacturing imperfections on the performance of the final product.

It is expected that the final computer model of the Eldorado Aerotech transit bus may consist of 50,000 to 100,000 finite elements. In addition to the size of the model, the crashworthiness studies require using non-linear, 3-D, explicit, dynamic, finite element codes capable of handling contact-impact problems with high-strain rate and non-linear, large deformations including plastic deformation and failure. The model developed will be used in the future to evaluate safety of different buses during accidents of various types including side- and rear- impacts, and rollover incidents.

TOOLS AND TECHNOLOGY

Since design data was not available for the selected bus, an actual vehicle was used to acquire geometric data needed to create a virtual model. The process applied in this study is commonly used in crashworthiness research community [1], [5] for passenger vehicles. It involves disassembly and digitization of the vehicle components, creating finite element meshes and merging parts into a complete bus model. A portable measurement device called a digitizing arm [3], presented in Figure 1, was used to collect most of geometric data. The equipment allows for mapping complex geometrical entities such as 3D curves and surfaces into AutoCAD computer format. This device consists of three arms with rotary transducers in each joint. The signals from transducers are sent to the computer, which determinates the position of the probe tip end in x-y-z coordinate system. The device and the software AnthroCAM used to operate it, allow for digitizing of points, scanning curves and construction of such geometric entities as poly-lines.

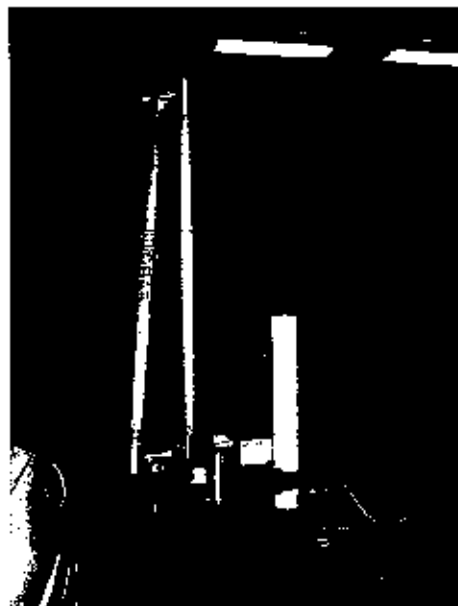


Figure 1 – Portable digitizing device: FARO Arm

Geometric data obtained using FARO arm is exported in format of IGES files, which are transferred to PATRAN pre-processor [4]. The finite element model is developed in PATRAN and input files dedicated to LS-DYNA are created. LS-DYNA is an explicit finite element code for analyzing 3-D, non-linear, large deformation, dynamic response [5]. Extensive library of material models, multipoint constraint and spot weld options, and contact-impact algorithms allow for efficient analysis of contact-impact problems with failure.

Figures 2 through 5 show an example of a reverse engineering process starting with digitizing (Figure 2) and completed with computer simulation (Figure 5). A segment of a front bumper of the transit bus, presented in Figure 2, was selected as an illustrative example. Figure 2 shows tapes marking curves on the actual bumper, which were scanned using digitizing arm. Resulting poly-lines are presented in Figure 3. Figure 4 shows finite element model created using quadrilateral FE shell elements. Fully integrated shell element with 5 Gauss integration points through the thickness was applied. Bilinear elastic – plastic material model with properties for steel was used. Figure 5 presents results for computer simulation of a high velocity impact of the bumper with a rigid wall. Figure 5 shows contours of von Mises effective stresses induced during the contact with a rigid wall (legend of stress fringes remains invisible in picture). Presented simple example illustrates the reverse engineering procedure of building a finite element model. Computer simulation obtained using LS-DYNA program provides numerical data including time histories of displacements, velocities and accelerations for selected points. Contours of strains and stresses can be plotted to assess structural integrity of the system as a function of time during impact.

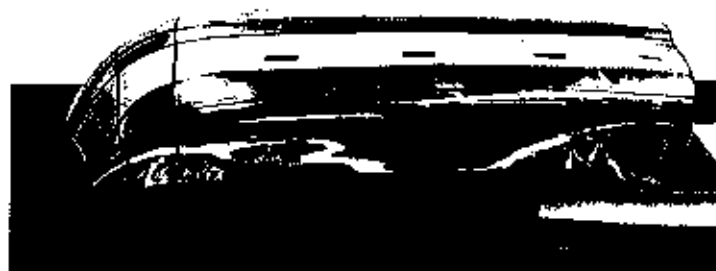


Figure 2 – Segment of a transit bus bumper with tapes marking curves for digitizing.



Figure 3 – Digitized poly-lines.



Figure 4 – Finite element model.



Figure 5 – Contours of von Mises effective stresses.

DIGITIZING AND DISASSEMBLING OF FORD ELDORADO BUS

The User Coordinate System (UCS) should be determined at the beginning of the geometric data acquisition, before taking any measurements. The User Coordinate System set up for the bus is presented in Figure 6. It was established by digitizing a set of 240 reference points marked with labels on the bus body (see Figure 6). Figure 7 shows computer image (top view) of digitized reference points and device positions. Reference points are used (after error distribution) to determine a current position of the digitizing arm. Since Faro Arm is relatively small comparing to the bus structure, the device had to be moved around the bus from one location to another. The first set of reference points was also used to digitize additional points marked on all structural parts before their removal. The total number of reference points exceeded 700.

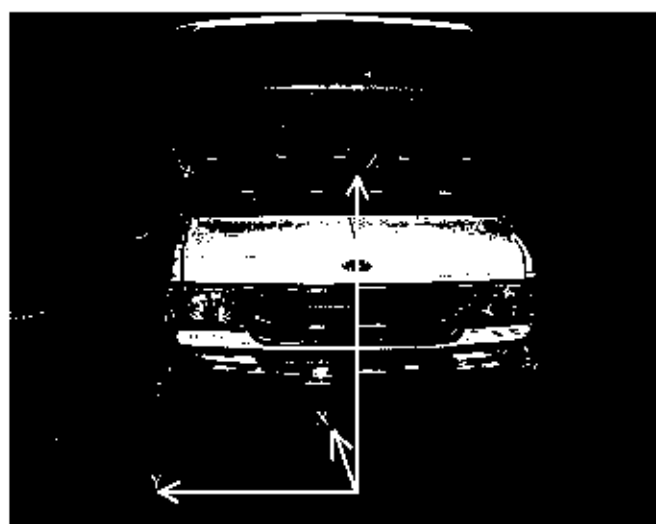


Figure 6 – User Coordinate System established for Eldorado Aerotech transit bus.

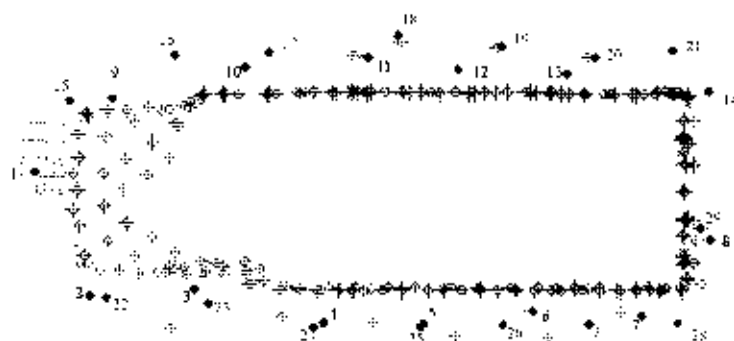


Figure 7 – Computer image of reference points and 29 device positions (top view).

The model development requires vehicle teardown and digitizing of all major, structural parts of the bus. All structural components were identified, labeled and removed from the bus structure. Figure 8 shows disassembled

bus structure. At least three reference points were digitized for each part before its removal to determine the component position in the global coordinate system. Connections between structural components were identified. After disassembly, geometric entities like lines and curves were scanned for each component. Geometric data obtained was transferred to MSC PATRAN, where finite element grids were generated, modified and merged. An interim set of the geometric quantities, obtained for exterior bus surface, is depicted in figure 9. Digitizing part of the research resulted in data acquisition of major geometric quantities including reference points and poly-lines, which are mapped into the computer model.

Geometric data is subsequently used as an input for building finite element meshes. Figure 10 presents a process of building finite element grids using digitized poly-lines, shown earlier in Figure 9. Development of finite element grids for some bus components is presented in Figure 11. Finite element (FE) model will include data received from mass and thickness measurements and material properties obtained from destructive strength tests



Figure 8. Disassembled Ford Eldorado Aerotech 240 transit bus.

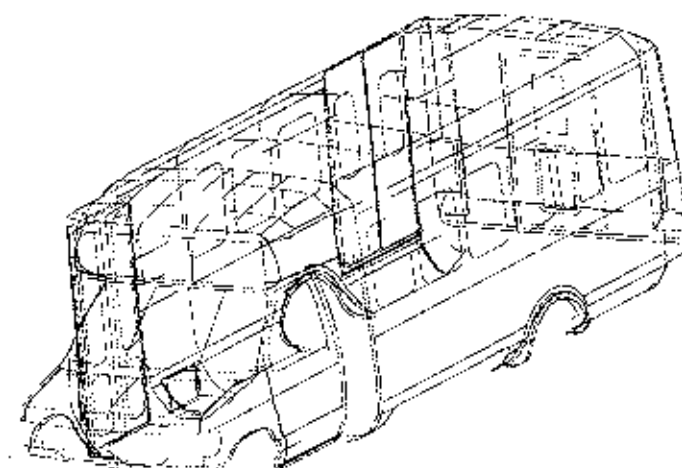


Figure 9. Poly-lines scanned for exterior bus components.

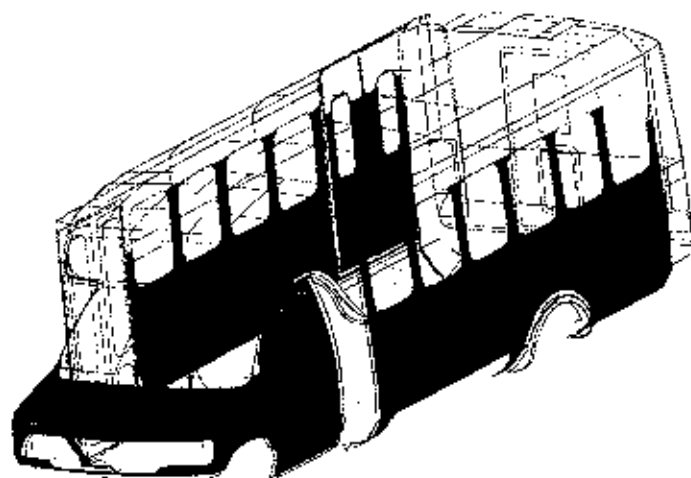


Figure 10. Development of finite element meshes.

performed for several material samples. The first validation step of the model will be based on the weight comparison between FE models and the parts that they represent. The parts are to be merged using pre-processors. Connections between vehicle components will be modeled by: merging nodes, applying constraints and spot welds. Inertia property testing will be performed for the actual bus using a tilt table technique. This data is used to validate mass distribution for the whole model, its center of gravity, and moments of inertia. Several preliminary tasks will be performed, including static and dynamic testing of structural samples, and testing and improving of the methodology for data acquisition.



Figure 11. Finite element meshes for some of hood, fenders and driver door.

Nonlinear, dynamic finite element analysis is performed using LS-DYNA, an explicit finite element code. After model validation phase is completed, rear and side impacts are planned to be simulated to evaluate safety and integrity of the bus structure.

It is expected that this cutting-edge technology will assist engineers in designing safer and more economical transit buses. Subsequently, research results from this study will help improving safety standards for transit buses.

ACKNOWLEDGEMENTS

This research was sponsored by the Florida Department of Transportation. The authors wish to acknowledge the assistance and support from the Project Managers; Mr. Robert Westbrook and Mr. Paul Johnson. The tear down of the bus was professionally performed by the QMG Consulting Inc. Thanks are due to Mr. Paul Queen and Mr.

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TRANSLATION OF MEASURED VEHICULAR WEIGHTS INTO DESIGN LOADS TO BE USED FOR BRIDGE ENGINEERING

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ABSTRACT

Measurements performed in 1978, 1994, 1996, and 1999 at three locations in The Netherlands were used to derive a statistical model for the axle loads and vehicular loads. The paper discusses the procedure that was used to fit statistical distributions to the measured data and to derive design loads for both axle and vehicle loads. Moreover, a trend analysis of the design loads is presented. Finally, the influence of special permit vehicles on the design load was evaluated.

INTRODUCTION

During the last ten years, TNO Building and Construction Research has carried out an investigation into the modelling aspects of traffic loads on bridges, see for instance Vrouwenvelder & Waarts (1993) and Vrouwenvelder et al (1999). The Dutch ministry of Transport, Public Works and Water Management commissioned the investigations. The aim of the investigations was to find a design procedure, which guarantees a structure with a prescribed level of safety with respect to the various limit states for the intended lifetime, and to replace the old code VOSB (1963).

According to present day standards, the required level of safety is expressed as an acceptable probability of failure or a target reliability index β in combination with a reference period of time for which the probability should be applied. For the Ultimate Limit State (ULS) the anticipated life time of the structure has been set equal to 100 years. For this period a reliability index $\beta = 3.6$ has been chosen according to the current Dutch building regulations (NEN6700). This value corresponds to a probability of 10^{-4} . For sake of comparison: the informative annex of the Eurocode requires a reliability index $\beta = 3.8$. For the service-ability limit states with respect to static deflections and vibrations an occurrence rate of once per year on the average was thought to be appropriate.

DERIVATION OF DESIGN LOADS

For the design of bridges with respect to strength or serviceability a designer needs axle and vehicular loads for the details and distributed lane loads for the design of principal construction parts like the main girders or the cable stays. Derivation of these loads involves two important uncertainties:

- Extrapolation of (short time) measurement results to design loads;
- Future trends in traffic loads.

A set of measurements done in 1978 (one location: Rheden) and in 1994, 1996 and 1998 at three locations (Moerdijk, Arnhem and Eindhoven) in The Netherlands were used to derive a statistical model for the axle loads and vehicular loads. The measurements lasted some 2 to 7 days each, registering about 10,000 to 80,000 heavy vehicles per measurement. This large set of data enabled an extensive statistical analysis. Furthermore it was expected that the trend in vehicular load could be analysed in view of the long period between the Rheden (1978) and later measurements. This paper will only consider the ultimate limit state. Furthermore only part of the study is presented, i.e. concerning the axle and vehicular loads. Given the vehicular load, the distributed lane loads can be derived. The procedure for this is already explained in Vrouwenvelder & Waarts (1993) and will not be further discussed in this paper.

PROBABILISTIC DESIGN PHILOSOPHY

For the fundamental case, where there is one load effect parameter S and one resistance parameter R , the basic design requirement in present-day codes is given by:

$$R_d > S_d \quad (1)$$

The index d indicates "design value". The value of S_d should follow from the requirement that the probability of " $S > S_d$ " in the reference period is equal to:

$$P\{S_d > S\} = \Phi(-\alpha_s, \beta) \quad (2)$$

Where

- Φ = standard normal distribution;
- α_s = influence coefficient for the load;
- β = target reliability index.

In the code one may use any combination of characteristic value and partial safety factor that leads to S_d . To find α_s in equation (2) one should in principle perform a complete reliability analysis. For reason of simplification, according to the appendix of ISO 2394, a standardised value of $\alpha_s=0.7$ is adopted. This means that for the ultimate limit state the design load has a probability of exceedance of $\Phi(-\alpha_s, \beta) = \Phi(-0.7*3.6) = \Phi(-2.5) = 0.0062$ in 100 years. If all uncertainties in the load model are of an ergodic (not time-invariant) nature, the value S_d has a return period equal to:

$$T_L = t_{ref} / \Phi(-\alpha_s, \beta) \quad (3)$$

Non-ergodicity may be caused, for instance, by statistical systematic uncertainties, by uncertainties in the traffic or vehicle models, and so on. So, if the load process is ergodic, this corresponds to a load effect with a return period $T_d = 100/0.0062 = 16\,000$ year. For the serviceability limit state, the β value equals 0.0 and so the value of α_s does not matter. The design value simply equals the maximum load effect to be expected in one year.

AXLE LOAD

Distribution type

A statistical analysis was carried out to fit probability distributions to the measured axle loads. On the basis of these probability distributions design values for the axle loads could be derived. The statistical analysis was carried out in two parts. The first part, which is discussed in this section, concerns the derivation of the distribution type. The second part, i.e. the assessment of the distribution parameters, is addressed in the next section. The largest dataset (Moerdijk '98) was used to derive the distribution type. Because of physical circumstances it is supposed that the distribution type is equal for all measurements (locations and dates). Figure 1 shows the frequency distribution of the measured Moerdijk '98 data; Figure 2 shows the cumulative frequency on logarithmic scale.

Both figures are based on 284,881 measured axle loads over a period of 7 days. The shape of the distributions are characteristic for all data sets: at least one mode at about 40 kN, a main mode around 60 kN and a "shoulder" that starts above about 90 kN. These frequency distributions cannot be described by one single standard distribution type. As we are interested in the extreme values, we aim to describe only the tail of the distribution (above a suitable threshold value y_0) with a standard distribution.

There is no other information available except the measured data. Figure 2 shows relative frequencies of exceedance ranging down to about 10^{-6} . The probability of exceedance of the design load is 10^{-11} (the target probability level of 0.0062 mentioned in the previous paragraph divided by the number of vehicles per reference period of 100 years). So, the cumulative frequency distribution needs to be extrapolated over about five orders of magnitude. As illustrated in Figure 3 the result of the extrapolation may highly depend on the choice of the standard distribution type. The four distributions shown Figure 3 result in a wide scatter of design loads. It is assumed that only analysing these four distributions describes the various extrapolation possibilities sufficiently.

The determination of the distribution type is based on the Bayesian approach. For each of the four distribution types the probability is calculated based on the measured data (posterior probability $P\{i|y\}$):

$$P\{i|y\} = \frac{P\{y|i\}P\{i\}}{\sum_{i=1}^4 P\{y|i\}P\{i\}} \quad i = 1, \dots, 4 \quad (4)$$

where

i = the distribution type (see indices in Figure 3);

y = measured data;

$P\{i|y\}$ = the posterior probability of the data being distributed according to distribution type i ;

$P\{y|i\}$ = likelihood of data y in distribution i ;

$P\{i\}$ = the prior probability of distribution i .

Distribution parameters and design loads

The design load is derived from extrapolating the tail of the frequency distribution. The result depends on the threshold load y_0 . Two considerations lead to the best choice of the threshold load y_0 . The higher y_0 , the better the statistical distribution describes the tail of the frequency distribution. On the other hand: a higher y_0 leads to less data and therefore a larger uncertainty in the fitted distribution. The optimal threshold load y_0 balances the two considerations.

The cumulative distribution in Figure 6 shows a small kink above 100 kN. A threshold load lower than 100 kN is therefore supposed to be inappropriate. The design load is presented against the threshold load y_0 in Figure 5 to deduce the most suitable value for y_0 . The figure also shows the uncertainty margins belonging to the design load for the various threshold loads. The increase of the uncertainty margins with increasing y_0 is based on the decrease of used data.

To calculate the uncertainty margins, 20 samples of axle loads were randomly drawn from the Weibull distribution shown in Figure 6. Each sample had the same size as the measured data set (284,881 axle loads). Subsequently on each of the samples a Weibull distribution was fitted for various threshold loads. This resulted for each threshold load in 20 fitted Weibull distributions from which 20 design loads were calculated. The uncertainty margin per threshold load was estimated as +/- two times the standard deviation of the associated 20 design loads.

Figure 5 shows that a threshold load of 125 kN is the optimum choice for the Moerdijk '98 axle loads. Higher threshold loads lead to some variation in design values, but this variation can be interpreted as statistical fluctuation as the values remain within the uncertainty bands. Apparently, the distribution found at a threshold load of 125 kN also gives an acceptable fit to the data above higher thresholds. If a lower threshold value had been selected, the drawn horizontal line in Figure 5 would have moved up, together with the uncertainty bands, to match the design value (circle) associated with that threshold. Due to this shift, the uncertainty bands would no longer have covered some of the design values for higher thresholds.

This procedure is repeated for the nine other measurements. Figure 7 shows the design loads. It is remarkable that all '94 to '99 measurements result in a higher design load than the old "Rheden '78" measurement. It is also remarkable that the '94 to '99 measurements show no trend in axle load. The analysed data give no basis to conclude whether there is a trend in axle load (based on '78 data compared to '94-'99 data) or that the difference can be explained by the difference in measurement location.

VEHICULAR LOADS

Distribution type, parameters and design loads

The frequency distribution for the vehicle loads resulting from the Moerdijk '98 dataset is shown in Figure 8. The distribution has a similar shape as the axle load distribution, except for one thing: there seems to be a pronounced heavy tail starting above 650 kN. About 30 vehicles form this tail. In a first analysis, the data points in this heavy tail were considered as outliers produced by measurement errors. The considerations were:

- these data points formed only a very small fraction of the data (30 in 80 000)

- the axle configurations of the vehicles were not recognised by the vehicle classification system
- Moreover, a possible explanation for the extremely heavy loads could be that two vehicles at close distance were interpreted as one single vehicle by the measurement system.

On the basis of this assumption, distributions were fitted to the data in a similar way as described in section 6 for the axle loads. For the Moerdijk '98 data this resulted in a Weibull distributions as shown in Figure 9 with a corresponding vehicle design load of 1100 kN. Figure 9 shows an overview of the design loads obtained for other locations and measurement periods. As for the axle loads all recent measurements ('94-'98) result in a higher design load than the older "Rheden '78" measurement. The recent measurements on its own do not show a trend in vehicular load. Whether the difference in design load between the '78 data and '94-99' data can be explained as trend in load or as a result of different measurement locations can not be concluded from this data.

Special permit vehicles

At the time of a second analysis, data had become available on permit applications for heavy vehicles. In The Netherlands, it is compulsory to apply for a permit for vehicles with a weight in excess of 1000 kN. Figure 10 shows the frequency distribution of the weights of vehicles, which were granted a permit over the period 1995-2000. The distribution is scaled to enable a comparison with the measured data. In the light of this information, the assumption that the heavy tail of the measured frequency distribution results from measurement errors needed to be reconsidered. Indeed, this tail connects smoothly to the distribution of the permit data. The axle configurations of the vehicles responsible for this tail were scrutinised one by one and it appeared that although these are not standard vehicles, most of them have realistic axle configurations. Hence, it was concluded that the earlier assumption of measurement errors could possibly be unjustified.

To explore the consequences of the heavy tail being real, a distribution was also fitted to the distribution including this tail. The procedure differed from the procedure for the axle loads in two respects. First, it was considered that the fit and the resulting design load should be based on uncontrolled traffic only, i.e. vehicles without a permit. A frequency distribution for the vehicles without permit was obtained by subtracting the frequency distribution of the permits from the frequency distribution of all measured vehicle loads. The result is shown in Figure 11. On this frequency distribution the fit was carried out. Second, two separate populations of vehicles were distinguished, each with its own probability distribution. The probability distribution of the first population, the 'standard' vehicles, is the fitted distribution as obtained in the first analysis. The probability distribution of the second population, the 'special' vehicles, was calculated with a similar fitting procedure, but applied to the vehicles in the heavy tail only. The probability distributions for both populations of vehicles are shown in Figure 11. This figure also shows that if the heavy tail is real, a vehicle design load of 1600 kN should be considered instead of the 1100 kN that was derived in the first analysis. To enable better founded conclusions about the tail of the frequency distributions for vehicle loads, additional measurements are planned, in which traffic information including axle and vehicle loads will be recorded during a period of one year on various locations in The Netherlands.

CONCLUSIONS

The distribution type and the threshold value for the tail of the frequency distribution heavily influence the extrapolation from measured data to design load. Both items should be given extensive attention when deriving design loads for bridges. A trend in axle or vehicular load can not be observed from the '78 to '99 measurements. The various locations do not show a consistent view. Loads in the dataset that are extremely high should not be regarded as outliers from the distribution too easily. Data in the Netherlands leads to a first impression that this data are real and should not be excluded from the statistical analysis.

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TABLES & FIGURES

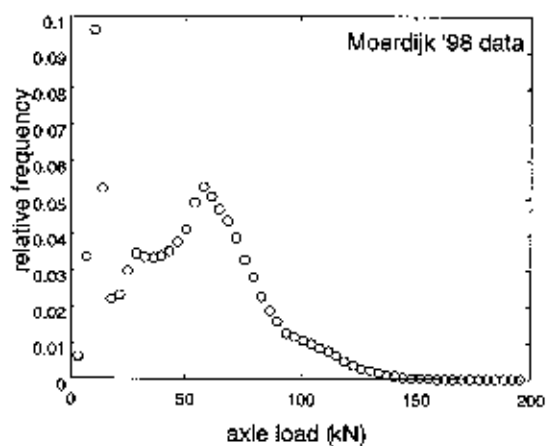


Figure 1 - Frequency distribution of axle loads

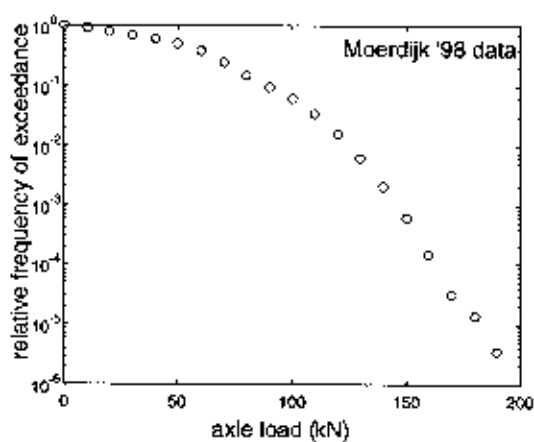


Figure 2 - cumulative frequency of axle loads

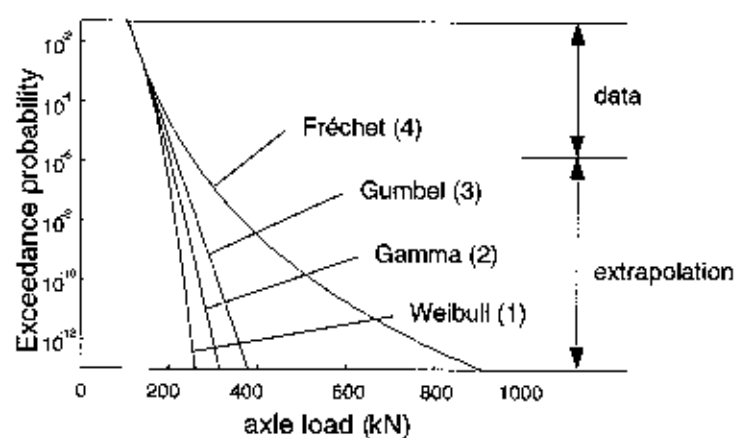


Figure 3 - Typical extrapolations depending on the distribution type

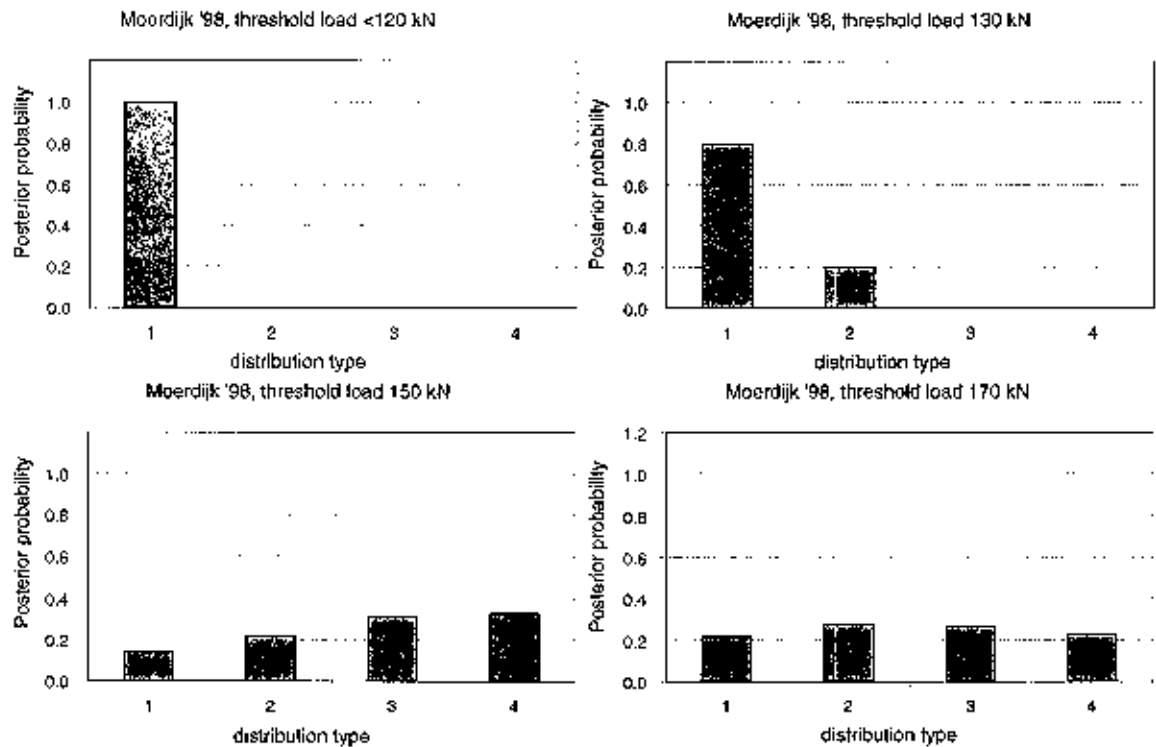


Figure 2 - Posterior distributions of distributions

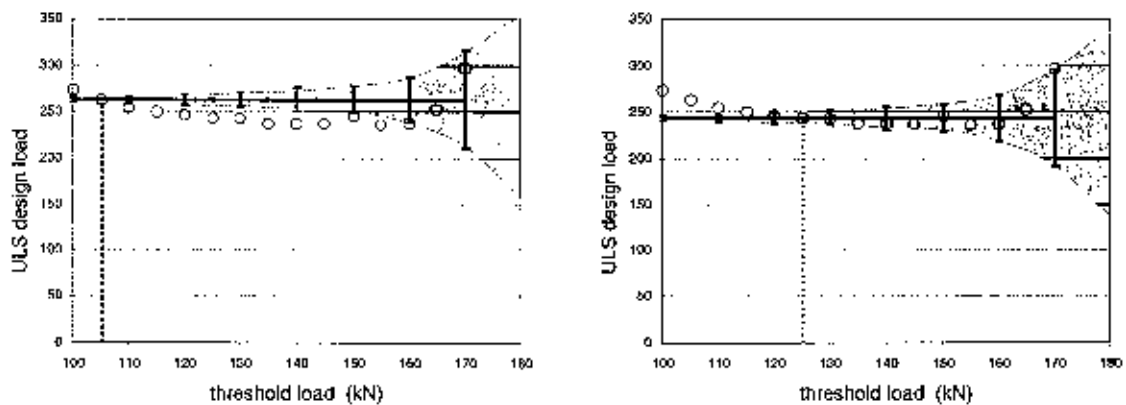


Figure 5 –Left figure: The circles show the calculated design loads as a function of the threshold value y_a . The horizontal line is the hypothetical 'true' design load, calculated from the data at the selected threshold load of 105 kN. The grey area shows which deviations from the 'true value' may occur if the design load would be estimated using a higher threshold value. Apparently, the threshold value at 105 kN yields a biased design load: many of the circles fall outside the grey area, i.e. their deviation from the horizontal line cannot be explained from statistical uncertainty.

Right figure: The horizontal line is the hypothetical 'true' design load, calculated from the data at the selected threshold load of 125 kN. Now all circles fall inside the grey area, which indicates that the design load, calculated at a threshold of 125 kN is unbiased.

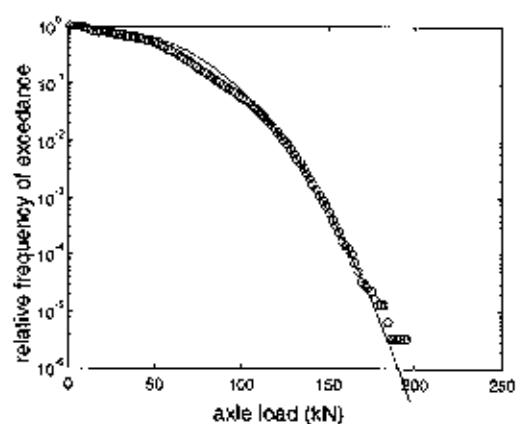


Figure 6 - The circles show the cumulative distribution and the fitted Weibull distribution at a threshold load of 125 kN

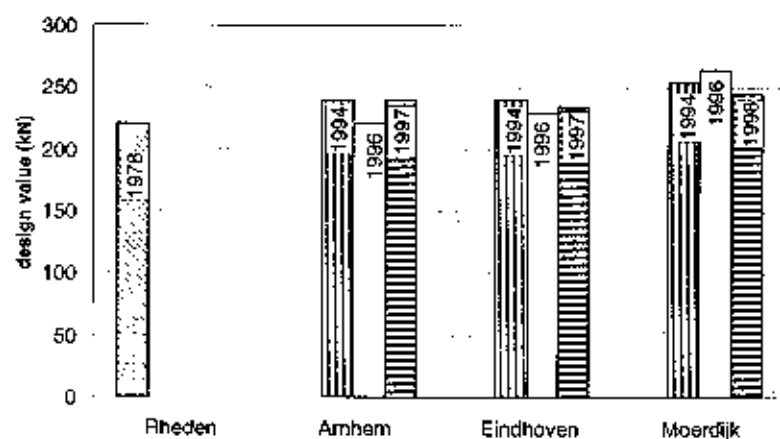


Figure 7 - Design axle loads for the various locations and positions in time

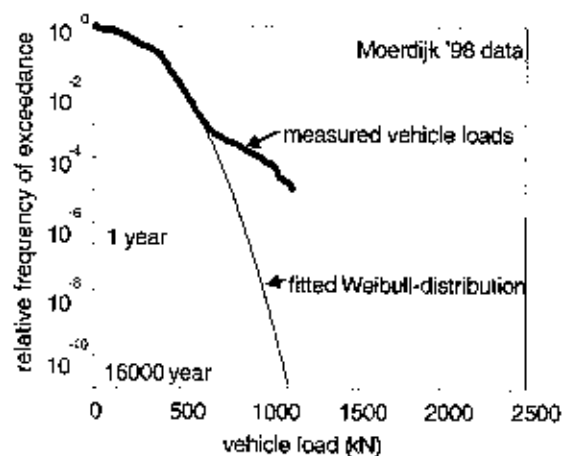


Figure 8 - Frequency distribution of vehicle loads from Moerdijk '98 data and extrapolation in the first analysis

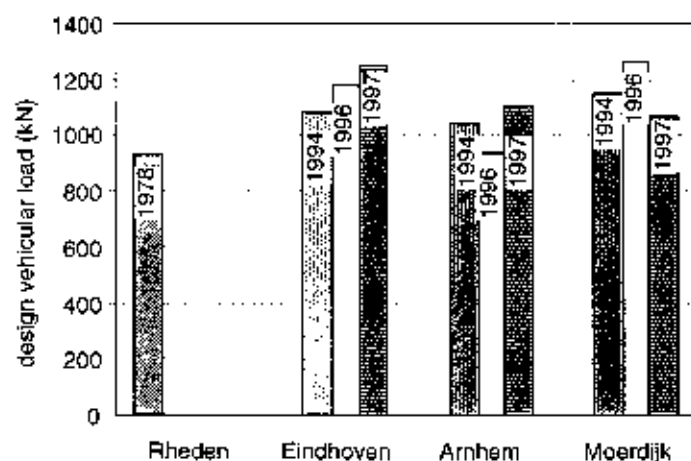


Figure 9 - Design vehicular loads (return period 16000 years) for all measurement locations

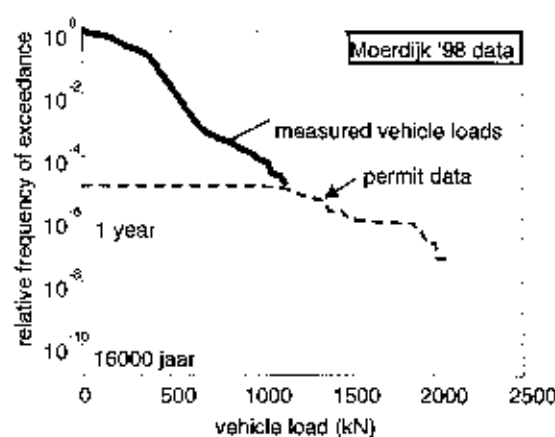


Figure 10 - Comparison of the measured data with permit data.

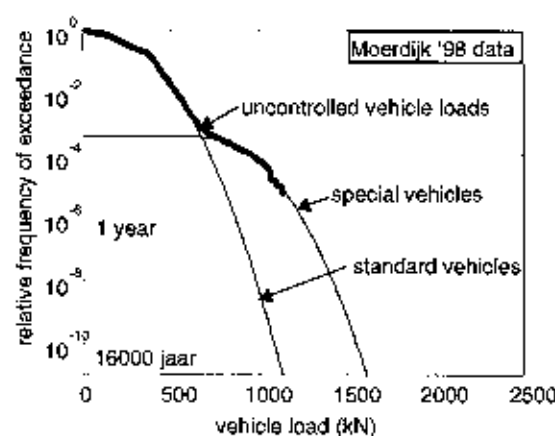


Figure 11 - Fitted distributions on both the population of 'standard' vehicles and the population of 'special' vehicles

HEAVY VEHICLE WHEEL SEPARATIONS: EXPLORING THE CAUSES

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ABSTRACT

Large truck wheel separations have resulted in 6 fatalities in the Province of Ontario, Canada since 1995. As a result of public concern, and by concluding that the wheel separations were the result of sub-standard maintenance practices, the Province of Ontario enacted severe penalties in an effort to reduce the problem. Despite the new measures and a significant effort from industry to improve wheel maintenance practices, wheel separations continue to occur in Ontario at a rate of 7 reported incidents per month. This paper examines the heavy truck wheel separation problem and discusses some unique technical factors that may be influencing the integrity of wheel systems.

INTRODUCTION

Wheel separation on a large truck is defined as the loss or detachment of a wheel while the vehicle is in motion. The cause of wheel separation includes:

- failure of the wheel unit, usually from metal fatigue resulting in the tire and rim detachment from the vehicle
- failure of the fastener system that secures the wheel to the brake drum, resulting in loss of the wheel and rim and in the case of dual wheel assembly, loss of both wheels
- failure of wheel bearings resulting in loss of the wheels, hub and brake drums

The mass of large truck wheels is significant. A set of dual wheels and brake drum is approximately 350kg. This represents a substantial body mass, which has the potential to cause severe damage to other vehicles and objects particularly when traveling at highway speed.

WHEEL TYPES

Until the 1980's, the most common heavy truck wheel design was known as the spoke system. The tires were mounted on simple rims supported by steel spoke wheels and secured by lugs, studs and bolts. These wheel systems were susceptible to wobble due to run out error that occurred during mounting or induced by shock loads that occurred as a result of rough roads and potholes. Disc wheels were developed to overcome these problems and to provide less massive wheel systems. The early disc wheel systems were known as stud-piloted wheels. These wheels used complex two-piece threaded fastener systems that independently secured the inner and outer wheels of a dual wheel set. The studs locate and supported the wheel about the rotational axis resulting in high stresses at the studs and wheel stud holes. The high stresses manifested in fatigue failures of the studs and wheel discs.

The most common wheel system used today is the hub-piloted wheel. An exploded view of the hub-piloted wheel is illustrated in Figure 1. The hub-piloted wheel is a disc wheel with 8 or 10 stud holes in a concentric bolt circle that allow the wheel to be fastened to the hub. The wheel has a precision cut hole at the axle centre, which engages on curved extension or seating pads extending from the hub. The engagement of these parts precisely locates the wheel about the centre of the axle, and the clamping force generated by the fasteners immobilizes the wheel. This has the affect of reducing the vertical load support shear stresses at the studs, which was one of the weaknesses in the fastener system used for the older generation stud-piloted wheels.

Figure 1. Exploded view of a hub-piloted wheel system.

ONTARIO DATA

In the province of Ontario, the Ministry of Transportation compiles wheel separation data. (1). The database tracks the location of the wheel separation within the province, the vehicle type, the cause of the wheel separation and the type of wheel. The database reveals that approximately 60% of all reported wheel separations in the Province of Ontario are related to fastener failures. This paper therefore will concentrate on the issue of fastener failures.

A review of the data to determine the frequency of wheel separations is found in Table 1. The data indicate that there are approximately 7 reported wheel separations in Ontario per month. (The number of unreported wheel separations is unknown.)

Table 1. Wheel Separations in the Province of Ontario

Year	Total Number of Separations	% Fastener Failure	Fastener Failures		
			% Hub Piloted	% Stud Piloted	% Spoke
2000	83	57%	64%	23%	17%
1999	79	62%	46%	31%	15%

Note: The values expressed in percentage do not add up to 100 as some of the wheel separations involving fasteners were not identified to a particular wheel type.

The data also indicate that the most common failure mechanism is associated with the fastener system and the most common wheel system failure is the hub-piloted wheel. It is important to note that the hub-piloted wheel is by far the most common wheel system used today and the comparatively high failure rate is a reflection, at least in part, of this fact.

One serious limitation in the Ontario Ministry of Transportation wheel separation database is the lack of specific information pertaining to fastener failures. It does not provide any detail or description of the failure mode. For example, it does not distinguish between fasteners that failed catastrophically or fasteners that simply backed off the studs.

ANALYSIS OF PREVIOUS RESEARCH

There are very few independent studies on wheel separation described in the literature. Three Canadian publications were found. One is published by the Professional Engineers Ontario entitled "Wheel Separation on Tractor-Trailers" (2) the other is entitled "Heavy Truck Wheel Separations: An In-Depth Study of Real-World Incidents" published by Transport Canada (3). The third is a paper (4) that was presented to The Fourth International Conference on Accident Investigation, Reconstruction, Interpretation and the Law; August 13-16, 2001; Vancouver. It discusses the legal details and framework used in Ontario as a counter measure to the wheel separation problem and reviews the technical issues associated with wheel separation. In addition to these papers, several industry publications exist in the form of safety and service manuals.

Woodrooffe & Associates studied the details from 13 cases reported in the Transport Canada paper. Of the 13 cases, 6 of the wheel separation incidents were the result of non-wheel related component failure such as failure of the wheel bearing, structural failure of the hub or axle spindle failure. One incident involving a stud piloted wheel was the result of failure of the wheel structure, most likely the result of metal fatigue. The remaining 6 wheel separation cases were the result of wheel fastener detachment. They all involved hub-piloted wheels where fasteners detached or loosened from the studs.

WHEEL FASTENER DETACHMENT

Wheel fastener detachment is defined as the wheel nuts unwinding fully from the studs and allowing the wheel to separate from the vehicle. In this failure mode, the stud remains intact and the wheel nuts are missing. This appears to be a common failure mechanism among cases recorded by Transport Canada. It appears that once the wheel clamping force diminishes to the point where relative motion between the hub and the wheel can occur, the rotational action of the wheel relative to the fasteners, results in the unwinding of the wheel nuts from the studs. Transport Canada case HFVS-96-01 most closely resembles a specific case, which was first reported by Woodrooffe & Associates. In the case HFVS-96-01, the truck was traveling in the westbound driving lane of a four-lane highway when the wheels of the left side of the last axle of the trailer separated. The fastening system on the steel hub piloted wheels was the common 10-stud system used on most large vehicles. The report states that the nuts backed off the studs completely while the vehicle was traveling and eventually the wheel separated.

Brake maintenance had been carried out on the failed wheel assembly just three days prior to the separation. The repair invoice indicated that the brake linings had been replaced on the left side of the axle. The driver's logbook showed that the vehicle had traveled 571 km since the last service. The report states "The driver indicated that he did not feel anything unusual with his vehicle, and was unaware of the wheel separation. He also stated that on the day of the incident, he had checked the fasteners prior to starting his trip, having borrowed a torque wrench from another transport driver." The case investigated by Woodrooffe & Associates was almost identical to this Transport Canada case. A brake job was done on the wheel in question two days prior to the incident. The vehicle traveled approximately 580 km from the point of service to the location where the wheel separation occurred.

Another important observation found when reviewing the Transport Canada cases is that of the 6 cases resulting from wheel fastener detachment, 4 of the wheels had been removed for service within one week prior to the incident. Of the 4 wheels that had been removed for service, 3 had been removed to allow for brake replacement. It is unknown whether the wheels from the other two incidents had been removed for service. This detailed analysis reveals that in the sample of wheel separation incidents reported by Transport Canada, all of the incidents involving wheel fastener detachment were of the hub-piloted design and most of these were of wheels that had been recently removed for service. Based on this limited sample it appears that hub-piloted wheels are more susceptible to wheel fastener detachment than any other wheel type. In addition, because at least 50% of these cases involved wheels that had been removed to allow for brake replacement, there may be factors related to brake replacement procedures that influence hub-piloted wheel fastener security.

Transport Canada case HFVS-96-01 most closely resembles a separate case (5) examined in detail by Woodrooffe & Associates. The truck was traveling in the westbound driving lane of a four-lane highway when the wheels of the left side of the last axle on the trailer separated. The fastening system on the steel hub-piloted wheels was the common 10-stud system used on most large vehicles. The nuts backed off the studs while the vehicle was traveling and eventually the wheel separated. The Ottawa urban bus transit authority (OC Transpo) experienced an incident (6) where a wheel separated from an urban bus as a result of fasteners backing off the studs.

TWO PIECE FLANGE NUTS

Hub-piloted wheel systems in North America use two-piece flange nuts comprised of a hexagon nut mated to a hardened washer. SAE J1965 specifies a minimum - maximum torque/tension relationship for two-piece flange nuts. To meet the minimum requirements there must be at least 133 kN (30,000 lb) of tension in the stud when 500 N-m (370 ft-lb) of torque is applied to the flange nut (7). To meet the maximum requirement, there must be less than 276 kN (62,100 lb) of tension in the stud when 678 N-m (500 ft-lb) of torque is applied to the flange nut.

As shown in Figure 2, the clamping force generated by the flange nut must fall within the limits prescribed by the SAE. Fasteners that fall outside of the torque limits are not acceptable.

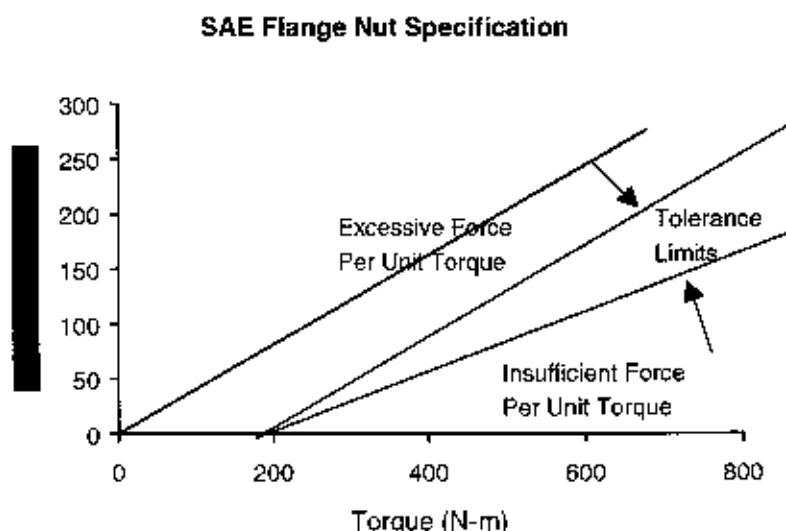


Figure 2: SAE Torque Clamping Force Performance Requirement

Anecdotal tests have shown that as the flange nuts age or when they are re-used, their torque/clamping force characteristics can diminish in the order of 50%. This means that when these flange nuts are tightened to a specified torque value, the achieved clamping force can be as little as 50% of the design value. Under these conditions the fasteners may not meet the SAE specifications and cannot produce sufficient clamping force to meet design requirements. Given that hub-piloted wheels depend exclusively on clamping force to prevent the wheel from separating from the hub, it is clear that such a reduction in torque/clamp force characteristics represents a significant risk to wheel separation. More work is required to better understand the torque/clamp force performance of these fasteners as a function of age and repetitive re-use so that industry can be more knowledgeable about wheel separation risk factors.

Heat Effects

After experiencing a wheel separation from an urban bus due to the loss of flange nuts, the Ottawa transit authority (OC Transpo) conducted an internal investigation on a new bus using a dynamometer (6, 8, 9). There was a suspicion that the problem may be linked to temperature change associated with braking. A moderate braking cycle was used to raise the temperature of the wheel assembly simulating normal brake duty cycles for urban busses. As the temperature increased, the flange nut required more rotational displacement to achieve the identical torque level. When the wheel cooled after being torqued at high temperature, the effect of thermal contraction resulted in the fastener being significantly over-torqued. From this experiment, it was concluded that as the temperature of the wheel assembly increases, the clamping force diminishes significantly. Given that urban busses routinely experience very high wheel assembly temperatures due to frequent brake applications, the bus company increased the installation torque level from the manufacturer's recommended value of 644 N-m (475 ft-lb) to a range of 813 – 881 N-m (600 – 650 ft lb) as a counter measure against heat related wheel separations. OC Transpo also specified that torquing of the flange nuts must not take place when the wheel assembly is hot in order to prevent excessive clamp force and stud stress when the assembly cools.

The same problem will occur when wheels are installed or re-torqued at very cold temperatures. As the weather improves the change in temperature will result in a reduction in the clamping force.

Re-Torquing of Wheel Nuts

There are very few independent reports focusing on truck wheels. No literature was found that provided technical information that supported or refuted the 80 km to 160 km re-torque requirement. There was no variation in the re-torque specifications among the various wheel manufacturers. It appears that this number is arbitrary and may have been chosen by wheel manufacturers as a convenient way to at least give some rough parameters governing re-torquing intervals. There is no information available that would allow an independent party to determine if this distance requirement is an absolute requirement, a reasonable requirement or a requirement of convenience. This is an important consideration because the requirement to stop a truck at 80 km to 160 km from its origin presents a logistics challenge (there are many routes where there are no repair facilities within such a departure distance). The Professional Engineers Ontario report "Wheel Separations on Tractor-Trailers" recognized this limitation by stating that "checking the wheel nut torque after 100 km of travel can realistically only be done by the driver using a bar wrench, which is acceptable. All that is required is to ensure that there is no perceptible loosening of the nuts. This is the only practical method for checking wheel nut torque on the road." It must be stressed that despite these challenges, re-torquing of wheel fasteners after initial installation is an important task, the issue is, at what distance from original service should this be done.

There is no comprehensive source on industry practice that speaks to truck wheel re-torquing. In the absence of such a source, the best that can be done is to rely on anecdotal experience. The Canadian truck transport industry is conscious of the importance of re-torquing wheel flange nuts after the wheel has been installed. The zone of operation appears to be the main factor influencing the distance traveled before re-torquing. The probability of wheel attachment problems increases on rough and mountainous roads due to high wheel vibration levels and high braking demands. This results in variations among carriers in the distance traveled before re-torquing occurs. Transport resource haulers operating in northern sectors on unpaved rough roads, instruct their drivers to re-torque the wheels before traveling more than 250 km after a wheel has been installed. The re-torquing task is normally done by the driver at the side of the road with a standard wrench and extension bar rather than a torque wrench. These companies believe that the rough terrain is much harder on the wheel systems thereby requiring shorter re-torquing intervals. Line haul companies operating on better quality roads generally re-torque at their terminal following the first trip segment. The author estimates the re-torque distance intervals to be in the order of 400 km to 700 km.

In the US, on the basis of anecdotal evidence, the practice of re-torquing appears to be quite unusual. Of the US carriers contacted by the author, not one of them practices re-torquing of the wheels.

LEGAL ISSUES

The Ontario Highway Traffic Act deals with wheel separations in a simplistic way. If a wheel separates, the offender is guilty of an offense and may be fined up to \$50,000. The Act makes it an absolute liability offense, that is, one where there is no defense, beyond the claim that the wheel did not separate. It is not open to the accused to prove that he or she took all reasonable measures to prevent the separation, which is the so-called defence of due diligence. From a forensic engineering perspective the inability to mount a defence means that comprehensive investigations into the cause of the wheel separation are not carried out and therefore the natural discover mechanism that is active in most court cases, ceases to exist. This deprives the engineering community of one of its most powerful retrospective analysis opportunities where potential design problems can be identified as a result of documented failures. Under the absolute liability rule where no defense is possible, the legal system inadvertently suppresses investigations of cause, which does not benefit potential defect discovery. Therefore in the absence of rigorous analysis, the legal system may be perpetuating the very problem that it is attempting to correct.

The notion of absolute liability for wheel separation is based on the assumption that truck wheel systems are properly designed and their failure mechanisms and causal factors are fully understood. It also implies that all wheel separations are preventable and are the result of poor maintenance practice. While poor maintenance is undoubtedly associated with some wheel separation cases, the evidence presented in this paper clearly show that these assumptions are not valid for all cases.

DISCUSSION

The Transport Canada study concludes with the following statement. "The fact that wheel separations are occurring over all wheel types, suggests that the expectations of current wheels are being exceeded, and that we may have reached the limits of current designs."

The Ontario Professional Engineer's report states the following: "It is said that the hub-piloted system is now used on all new tractor duals in Ontario, and on a high percentage of new trailer duals. This is evidence that the industry is making efforts to improve the situation. However, since this new wheel type has come into usage only recently, there is insufficient data to assess statistically the anticipated improvement or potential failure mechanisms."

Both of these independent reports express uncertainty about truck wheel systems. The Transport Canada report suggests that current wheel systems may have reached the limit of their design and the Ontario Professional Engineers report cautions against unknown failure mechanisms related to hub-piloted wheels. The significance of these observations and the possible problems with hub-piloted wheels constitutes an emerging issue.

Warnings contained in wheel manufacturers safety and service manuals clearly state, in prominent warning boxes, that dirt and contamination including thin layers of paint (greater than 0.0035") can result in wheel separation. According to the wheel manufacturers, these seemingly innocuous factors can have a profound effect on road safety. One can only conclude that safety factors inherent in the design of all other vehicle related parts are greatly diminished in the current design of the hub-piloted wheel system. It would appear that the owners and drivers of these vehicles are being asked to take responsibility for a design that is at its limit or is insufficiently robust to compensate for such factors.

The Professional Engineers of Ontario report concludes that "liberal axle loading allowances and eccentric axle configurations have undoubtedly contributed to wheel separation" and the report recommends that "vertical and lateral wheel loads should be examined on the basis of Ontario weight and dimension regulations; the effect of these loads on trailer performance and highway deterioration should be examined." This analysis was never conducted.

Ontario axle loads are among the highest in North America. All trucks operating on the US Interstate System are limited to a tandem axle group weight of 15,455 kg (34,000 lb) or 1,982 kg (4,250 lb) per wheel. In Ontario, the allowable tandem axle load for the vehicle in question is 17,900 kg (39,380 lb) or 2,238 kg (4,923 lb) per wheel. Therefore, the wheels in Ontario are loaded up to 13% greater than those in the US. Given that there are at least 10 times as many trucks operating in the US than in Canada, it can be concluded that the major market for wheels is in the US. It may be that the relatively small jurisdiction of Ontario inadvertently allows the hub-piloted wheel system to be operated more closely to its design limit and therefore increases the risk of wheel separation. It may also follow that the higher wheel loads permitted in Ontario exacerbate the sensitivity of the hub-piloted wheel system to such factors as paint thickness.

CONCLUSIONS

1. Despite harsh penalties and the enforcement of absolute liability legislation for wheel separation incidents, wheel separation continues to be a significant problem within the province of Ontario.
2. The majority of the wheel separations are caused by fastener failures on hub-piloted wheels.
3. The legal method that the province of Ontario has used to deal with the wheel separation problem is poorly structured as it is based upon the assumption that the wheel systems are without fault and any failure is always the result of failure to exercise proper care and due diligence. Detailed investigations of the specific cases presented in this paper challenge this view.
4. The evidence presented in this paper clearly shows that truck wheel separation mechanisms and causal factors are not fully understood and that the hub-piloted wheel system is vulnerable to the loss of flange nut fasteners that back completely off of the studs.
5. Wheel fastener integrity and the risk of wheel separation are affected by hub-piloted wheel system temperature variations associated with vehicle braking or installation at very cold temperatures.

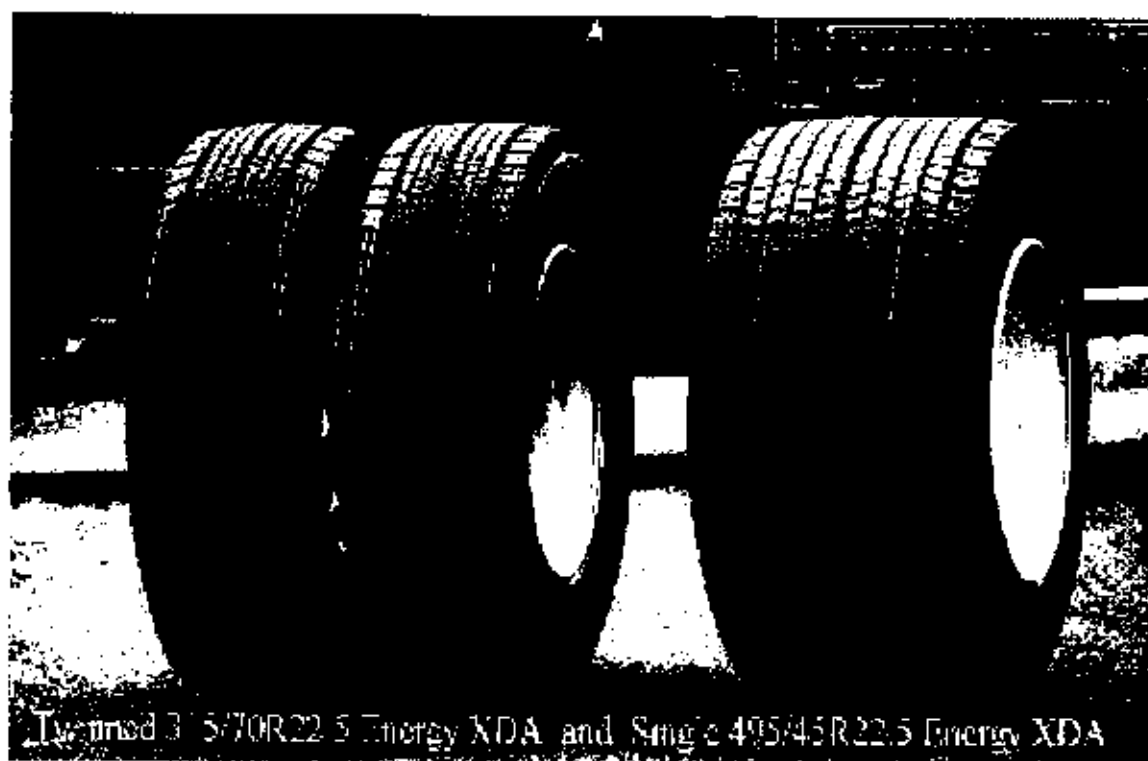
6. For hub-piloted wheels, it may be necessary to vary the applied wheel fastener torque to obtain the proper clamping force depending upon operating thermal variations, and the condition of flange nuts.

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COST 334 EFFECTS OF WIDE SINGLE TYRES AND DUAL TYRES

EXECUTIVE SUMMARY



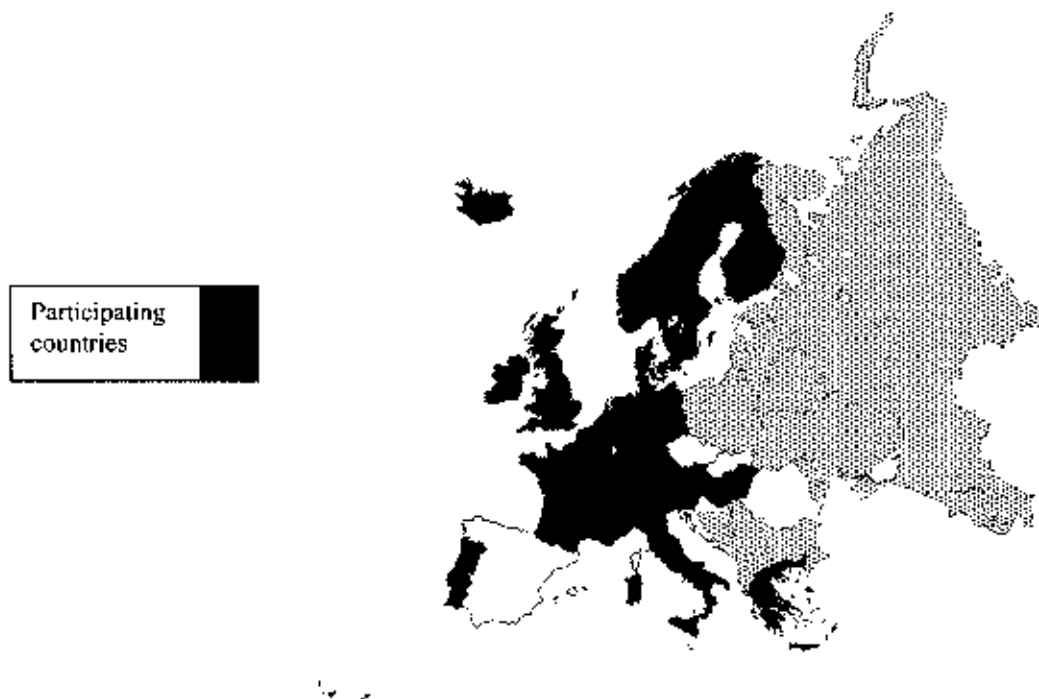
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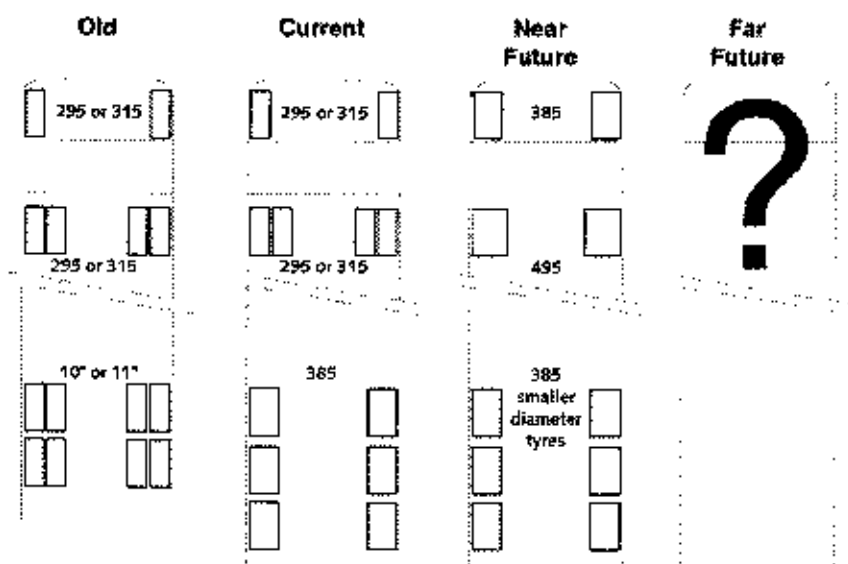


ISO Code	Country	Institute / Organisation	Acronym
AT	Austria	Institut für Strassenbau und Strassenerhaltung Technische Universität Wien	ISTU Wien
BE	Belgium	Belgian Road Research Centre European Tyre and Rim Technical Organisation	BRRC ETRTO
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EXECUTIVE SUMMARY

The trend for some years on heavy goods vehicle weights and dimensions regulation has been to increase allowable weights. Generally, such increases have been to maximum gross vehicle weights. In order to take advantage of increased gross vehicle weights, the total number of axles on the vehicle has increased. Other developments have taken place, however, in both the tyre and vehicle industries and these have enabled heavy goods vehicle operators to become efficient contributors to the European economy, and that will no doubt continue to do so.

These developments may be summarised in the following figure.



From the "Old" to the "Current" situation, for example, the number of trailer axles increased from 2 to 3. As a result of economic pressures, vehicle operators later took advantage of developments to their existing tyre fitments that improved performance and reliability. Such tyres produced significant advantages to the operator: these included reduced tare weights (and therefore increased payloads), reduced fuel consumption due to lower rolling resistance, and reduced tyre maintenance costs.

In the "Near Future" scenario, further tyre developments will be used to again reduce tare weights and rolling resistance, probably on the drive axle, and to decrease tyre maintenance costs by rationalising the number of different tyre fitments used on a vehicle. An example of the latter is the trend to use of wide single tyres on steering axles, which the use of power steering on most vehicles makes possible. This reduces the need to stock or carry on the vehicle combination more than two tyre sizes, brings about savings in tyre wear, and allows re-use of front tyres on the trailer after re-capping. The use of technologically advanced materials in tyre construction, generally to reduce rolling resistance, has enhanced these savings.

In the "Far Future" scenario, it is anticipated that the tyre industry in particular, and to a lesser extent the vehicle manufacturing industry, will make additional advances that will enable further economics to be made in the road transport sector. However, these economics could occur at the expense of additional road maintenance costs or other disadvantages.

During the period of these technological advances, there have also been other developments that significantly affect the interaction of vehicle and road, often to the detriment of the road and its function. For example, the increase in road traffic over the past 30 years, for both passengers and goods, has been dramatic, often exceeding predictions. Despite considerable capital investment in road infrastructure over the same period, road capacity is frequently severely tested, particularly on major routes. Road strengthening to meet the increased numbers and weights of heavy vehicles has taken place, more recently using advanced materials for surfacing and underlying layers.

Simultaneously, the trend towards increased use of wide single tyres instead of dual tyres, coupled with more "canalising" of goods vehicles due to increased traffic, have raised the stresses to which pavement surfacings have been subjected. A degree of overloading of a small proportion of heavy goods vehicles in most European countries adds to these stresses, with the result that pavement deterioration takes place more rapidly than expected.

Against this background, of unquantified but perceived reduced costs to vehicle operators, increased costs to road owners, benefits in safety, and reduced costs in environmental aspects, COST 334 has carried out a programme of work to quantify these costs and benefits, and propose ways in which they may be balanced.

The conclusions of the work relate to:

- Tyre fitment and pavement wear
- Tyre fitment and vehicle handling
- Tyre fitment and environmental effects
- Tyre fitment and vehicle operating costs
- Overall economic effects of the use of different tyre fitments.

On the basis of the conclusions drawn from the work, it was also possible to make recommendations on the development of future tyre fitments, including current sizes, and on the use of the results obtained in pavement design.

CONCLUSIONS

Tyre fitment and pavement wear

In examining the conclusions of the work in this area, it is useful to define some of the terms used.

Footprint	The area of the contact patch between the tyre and the road surface.
Tyre fitment	Refers to the tyre as a wide single fitment, or as a dual tyre assembly.
Primary Road Network	Defined, for the purposes of COST 334, as the network of principal roads in a country or state, generally comprising motorways (autoroutes, autostrade, etc) and other principal roads, state owned or otherwise. This network provides the major links between large urban areas and key national long-distance routes.
Secondary road network	Defined, for the purposes of COST 334, as the network of secondary roads in a country or state, generally comprising those roads owned by state, regional or local authorities, and acting as links between primary routes, but excluding some rural roads.
Primary rutting	The mode of pavement deterioration by which rutting occurs principally in the bituminous layers, mainly occurring in the pavement surfacing of the primary network.
Secondary rutting:	One of the modes of pavement deterioration of the secondary network occurring in the subgrade or granular layers of the pavement.
Fatigue cracking	Cracking in the bituminous or cement bound material originating at the bottom of the respective layers, due to fatigue of the material by a great number of repetitions of bending due to wheel loads. (Such cracking does not include surface cracking and cracking due to thermal cycling.) Fatigue cracking is one of the modes of deterioration of the secondary network.
Thin pavement	A pavement with a thickness of bituminous layers of 100 mm or less.
Medium pavement	A pavement with a bituminous thickness of around 200 mm.
Thick pavement	A pavement with bituminous thickness of around 300 mm or more.

Terms used in modelling of pavement wear in relation to tyre parameters:

"Width" (for width-based model)

In the context of tyre parameters, this term is used to describe the footprint width for wide base singles. For dual tyres the width is taken as twice the footprint width of the individual tyres. (All width values consider footprint (tyre contact area envelope) width, not tyre section width.)

"Total Width" (for total width-based model)

In the context of tyre parameters, this term is used to describe the footprint width for wide base singles. For dual tyres the Total Width is taken as twice the footprint width of the individual tyres plus the spacing between the footprints of the dual. This spacing is of the order of 100 mm for all dual tyre fitments.

Axle Wear Factor (AWF)

The Axle Wear Factor (AWF) is a dimensionless factor that relates the damage contribution of a single passage of an axle fitted with tyres of a specific tyre width and inflated to a given cold inflation pressure, carrying a given axle load, to the damage contribution of a single passage of an axle with the reference tyre (295/80R22.5 dual fitment with a cold inflation pressure of 650kPa) carrying the reference load of 10 tonnes.

Tyre Configuration Factor (TCF)

A factor describing the pavement wear attributable to different tyre fitments and sizes, when compared with an arbitrarily selected reference tyre, at equal load. The TCF includes factors for specific tyre characteristics, in relation to their performance regarding dynamic force transmissibility, and potential load imbalance. The chosen

reference tyre, with a TCF of 1.0, is the most commonly used drive axle dual tyre, namely a 295/80R22.5, under maximum recommended loading conditions.

Load imbalance

Differences in tyre load between the tyres of a dual wheel assembly.

Dynamic Loading

The effect by which vehicle loads applied to the road surface increase and decrease in response to pavement unevenness and other factors. Dynamic loading is strongly influenced by vehicle suspension type.

The work of COST 334 was confined to bituminous pavements; for concrete pavements, it is expected that there will be only small influences on pavement wear due to differences in tyre configuration. For bridges, viaducts, etc. no specific conclusions were drawn.

General

1. Large differences in relative pavement wear exist among dual tyre assemblies and among wide-base single tyres. Therefore, a single factor for the difference between wide-base single and dual tyres is not applicable. Comparisons between pavement wear effects can only be made if the detailed characteristics of the tyre fitments are taken into account.
2. The pavement wear effects of different tyres vary according to the types and thickness of pavement, as well as their associated distress modes. For this reason COST 334 developed the concept of the Tyre Configuration Factor (TCF). The TCF of a tyre expresses the amount of pavement wear, depending on the pavement thickness and distress mode considered, relative to an arbitrarily chosen reference tyre. In use, the higher the TCF value, the higher the pavement wear (with the same axle loads, suspension type, etc.).
3. The TCF formulae developed from the work enable the quantification of the pavement wear effects of current and future different tyre fitments and sizes. The derivation of TCF formulae for all pavement thicknesses was not possible in all cases, however, because of insufficient data.
4. On the basis of the TCF formulae, the main influencing factors for pavement wear are the width (see Conclusions 5 and 6) and size of the tyre-pavement contact area, and the ratio of the actual inflation pressure over the recommended inflation pressure for the actual load (hereafter referred to as the pressure ratio).
5. It was found that the thinner the pavement, the stronger was the influence of differences in tyre configurations on pavement wear.

On the tyre concept (one or two contact areas) and the tyre width parameter:

6. For primary rutting (mainly on thick and medium pavements) the main width parameter is Width, being the footprint width for wide base singles, and for dual tyres twice the footprint width of the individual tyres. (All width values consider footprint (tyre contact area envelope) width, not tyre section width.) As a consequence, for this distress mode, pavement wear due to wide base single tyres or dual tyre assemblies does not differ significantly, when the axle load, tread pattern width, contact area, tyre diameter and pressure ratio are equal.
7. For secondary rutting and fatigue cracking on thin and medium pavements the main width parameter is the Total Width of the footprint of the tyre assembly. [For dual tyre assemblies this includes the distance (100mm) between the footprints of the individual tyres.]. As a consequence, single and dual tyre assemblies will produce equal TCF values indicating equal pavement wear, when the Total Width is equal (all other factors being equal). Usually, however, for the same axle load, current dual tyres will have a greater Total Width than a current wide single tyre.
8. For secondary rutting and fatigue cracking on thick pavements there is little difference between different fitments and sizes of tyres, as the pavement wear is dominated by the overall magnitude of the load carried in these cases.

On size of contact area:

9. In addition to its width, the length of the tyre-pavement contact area was shown to be influential in the cases of primary rutting on thick (and probably thin and medium) pavements and fatigue on thin and medium pavements. Combined, this signifies the influence of the size of the tyre-pavement contact area, and hence the average contact stress. Sensitivity analysis showed that a decrease of 10% in contact area results in a 9- 39% increase in pavement wear for these cases. No similar conclusion could be drawn for secondary rutting because of a lack of data.
10. The tyre diameter can also be taken as an indicator for the contact area length and the related pavement wear. A reduced tyre diameter will lead to increased pavement wear (when all other tyre parameters remain constant). This is important in the context of a trend towards the use of smaller-diameter tyres in Europe, to allow the lower platform heights that will accommodate volume-limited loads to be carried, rather than mass-limited loads.

On tyre inflation pressure and contact stress distribution:

11. The tyre inflation pressure is not a direct parameter in the TCF formulae. For the same load and tyre, higher inflation pressures generally result in a smaller tyre-pavement contact area, and thereby increased surface stress in the pavement. As a consequence, higher inflation pressures generally result in higher pavement wear, especially on thin pavements.
12. The ratio of actual to recommended inflation pressure was shown to be influential for the cases of primary rutting on thick (and probably medium) pavements and secondary rutting on thin and medium pavements. An inflation pressure 10% higher than that recommended for the actual tyre load results in about 15% increase in pavement wear. In such a case of over-inflation, the contact stress distribution is non-uniform and the load is concentrated on a smaller area.
13. The detailed contact stress distribution within the contact area is probably relevant for distress modes whose origin is at or close to the pavement surface, such as ravelling (loss of aggregate in the pavement surfacing) and surface cracking. Although COST 334 established good techniques for the measurement of these distributions, insufficient data was obtained to draw robust conclusions.

On the effect of dynamic loading and load imbalance

Dynamic loading occurs as a vehicle passes over a road surface. A road with poor longitudinal profile will produce increased dynamic loading, compared with the same vehicle passing over a smooth road.

Load imbalance occurs when each tyre of a dual tyre assembly is inflated to a different pressure (where the tyres are of the same type), or when each tyre has a different diameter (as is the case when the tyres used are worn to different degrees)

14. By comparison with other effects, tyre fitment does not significantly affect the dynamic loading of the road pavement.
15. By comparison with other effects, the effect of load imbalance between tyres on a dual assembly was found not to significantly affect pavement wear or other aspects.

On TCF values for current common tyre fitments and possible future tyre fitments

As stated earlier, TCF values vary according to the pavement thickness and distress mode under consideration. For practical use, values for the current common and possible future tyres (rim sizes 19.5 and 22.5 inches) were determined for the European primary road network (based on primary rutting in the bituminous layers of thick pavements) and the European secondary road network (based on a weighted average of the three distress modes on medium pavements, namely primary rutting, secondary rutting and fatigue cracking). Most road freight in Europe is carried on the primary networks, however, and greater importance is attached to these.

16. Common current and possible future dual tyre assemblies for towed axles have TCF values for primary roads ranging from 1.5 to 1.7 and for secondary roads TCF values of 1.3 to 1.5. Current common and possible future wide base single tyres for towed axles have TCF values for primary roads ranging from 1.5 to 2.2 and for secondary roads TCF values ranging from 2.2 to 3.6. On average the use of current common or possible future wide base singles on towed axles, instead of dual tyre assemblies, increases the contribution of these axles to pavement wear on primary roads and secondary roads by 17% and 97%, respectively.
17. Common current and possible future dual tyre assemblies for driven axles have TCF values for primary roads ranging from 0.9 to 1.3 and for secondary roads TCF values ranging from 0.9 to 1.2. The prototype extra-wide base single tyre 495/45R22.5 for use on drive axles has a TCF value of 1.2 on primary roads and 1.6 on secondary roads. On average, the use of wide base singles on driven axles, instead of common current dual tyre assemblies, increases the contribution of these axles to pavement wear on primary roads and secondary roads by 17% and 64%, respectively.
18. Conventional single tyres for steering axles have TCF values for primary roads ranging from 2.8 to 4.0 and for secondary roads TCF values ranging from 5.0 to 8.0. Current common and possible future wide base single tyres (from the 385 - fitment and wider) for steering axles have TCF values for primary roads of 1.9 to 2.2 and for secondary roads TCF values of 2.8 to 3.6. On average the use of current common and possible future wide base singles on steering axles reduces the contribution of this axle to pavement wear on primary and secondary roads by 36% and 45% respectively.
19. Conventional single tyres for steering axles are relatively more damaging than the common dual tyre assemblies for driven and towed axles, and wide single tyres for towed axles. This is partly alleviated by lower loads on the steering axles, but in practice the steering axle still may cause more pavement wear than a driven or towed axle.

On Axle Wear Factors for the different axle types fitted with current common and possible future tyres.

Based on the TCF value of tyres, the damage contribution of a single passage of an axle can be calculated using the appropriate formula, taking into account the actual axle load. This damage contribution is expressed as the number of passages of the reference tyre with the reference load of 10 tonnes, that gives the same amount of damage. This number is called the Axle Wear Factor (AWF).

For the current common and possible future tyre sizes (for rim sizes 19.5 and 22.5 inches), AWF values were determined for different axle loads for the European primary road networks.

20. Current common and possible future tyre assemblies for the driven axle, either with duals or wide base singles, have, at a load level of 11.5 tonne, AWF values ranging from 1.2 to 1.7. Current common and possible future tyre sizes for the towed axle, either with duals or wide base singles, have, for their respective maximum allowable load levels (between 8 and 10 tonne), AWF values ranging from 1.1 to 1.9. This range of values is very similar to that for the driven axle. Finally, current common and possible future tyres for the steering axle, fitted with either conventional singles or wide base singles, have, at their respective maximum allowable axle load (between 6.5 and 9 tonne), AWF values ranging from 1.4 to 1.9. This range is marginally higher than that for the driven axle. That the lower level of the axle loads on the towed axles is not reflected in lower AWF values, is explained by the fact that generally, relative to the axle load, wider tyres are used on the driven axle. The marginally higher AWF values of the steering axle, though having a much lower load in comparison with the driven axle, is explained by the fact that on the steering axle, all load must be transferred by two tyres, whereas for the driven axle, four tyres are usually used.

Tyre fitment and vehicle handling

In the area of the effects of tyre fitment on vehicle handling, some definitions of effects are useful.

Vehicle handling	Those characteristics of the vehicle that control the ability of the driver and vehicle to carry out driving manoeuvres safely and efficiently.
Lateral stability	For a single unit vehicle the stability is related to the under-steering/over-steering characteristics. The under-steering or stability gradient, expresses how the steer angle is related to the lateral acceleration in a steady-state turn. If the stability gradient is negative there is a critical speed when the vehicle is unstable.
Longitudinal stability	The factor governing the ability of the vehicle to stop, and therefore controlling the stopping distance of the vehicle.
Off-Tracking	Off tracking is the lateral deviation between the path of the centre-line point of the front axle of the vehicle and the path of a centre-line point of some other part of the vehicle. If a single number is given, it refers to the maximum off-tracking.
Rearward Amplification	The Rearward Amplification is the ratio of the maximum value of the quantity of interest of a following vehicle unit to that of the first vehicle unit during some kind of manoeuvre. The quantity of interest may be, for instance, lateral acceleration or yaw velocity.

The results of COST 334 work in the area of vehicle handling are based on the experimental testing of selected vehicle types in a range of test conditions. These are supported by theoretical work using widely accepted simulation models of vehicle behaviour, which were validated by full-scale tests. The vehicles considered were:

- A rigid truck, conforming to ISO 3833
- A rigid truck and trailer conforming to ISO 3833
- An articulated vehicle (tractor and semi-trailer) conforming to ISO 3833

On the basis of the work carried out, it was concluded that:

21. In general, the use of wide single tyres on drive axles, when compared with dual tyres, improves vehicle under-steering in the direction of increased lateral stability, for the vehicles and tyres tested.
22. In the case of the 2-axle rigid truck, the handling behaviour brought about by the use of the wide base single tyre on the drive axle also reduced the lateral acceleration lag of the vehicle during a given driving manoeuvre.

One of the benefits previously claimed for the use of wide single tyres is that the wider spring base they provide leads to increased vehicle stability, i.e. reduced risk of "roll-over". This conclusion supports this view, but suggests that the improvement is mainly due to the increased lateral stiffness of the tyre.

23. In relation to sudden tyre defects (punctures) in the drive axle tyres, no increased risk due to the use of wide single tyres was established by the simulated tests carried out.

Tyre fitment and environmental effects

COST 334 addressed the matter of the environmental effects of the use of wide single and dual tyres in two ways. First, their effects on tyre-road noise emissions was assessed, and second, their effects on gaseous emissions were assessed by consideration of changes in the rolling resistance (and thereby fuel consumption) of each tyre fitment.

24. In relation to tyre-road noise emissions, the work found no significant overall difference between the noise generated by each tyre fitment, when used on drive axles or trailer axles.
25. The use of the wide single tyres generally reduced fuel consumption, and thereby reduced total CO₂ emissions.

Tyre fitment and vehicle operating costs

Vehicle Operating Costs (VOC) are significantly affected by the choice of tyre fitment. Thus, tyres with reduced rolling resistance lead to reduced fuel costs, and tyre fitments lighter than those used previously bring about potential increases in payload, with associated economic benefits. The work of COST 334 in the area of VOC made use of the "Past", "Current", and "possible Future" scenarios identified by the Group as being appropriate in Europe. The work also examined more particular situations, such as the possible use of prototype wide single tyres on the drive axles of heavy goods vehicles. As a result of this work, it was concluded that:

26. The choice of tyre fitment on the driven axle of a 40 tonne gross vehicle weight truck-semi-trailer vehicle can affect the VOC by up to 1%, as a result of 2% changes in fuel consumption.
27. A further saving of 1% is available to truck operators from the use of lighter tyres and wheels, when these weight savings are translated into increased payload.
28. VOCs are strongly influenced by taxes, particularly fuel taxes.
29. When VOC is considered without taxes, differences between the past and current tyre configurations selected by COST 334, and described in the Figure at the beginning of this Chapter, may be up to 3% in VOCs. A further 0.5% saving is available between the current and possible future tyre configurations.

Overall economic effects of the use of different tyre fitments

The work of COST 334 in the various areas noted above was consolidated into an overall view of the effects of the use of wide single tyres and dual tyres by aggregating their effects in financial terms, for those European countries for which data was available. In order to achieve this, costs and values needed to be attributed to each separate effect, and this was done using, wherever possible, widely accepted figures. The estimation of overall costs and benefits was applied to the three situations considered by COST 334, namely:

- a) the past situation, before the introduction of wide base single tyres.
- b) the present situation in which wide base single tyres are widely used on the towed axles (trailer and semi-trailers), of vehicles, but not on the driven axles of motor vehicles.
- c) a possible future evolution in which motor vehicle steering axles would be fitted with wide base single tyres of the size currently used on towed axles, and driven axles would also be equipped with wide base single tyres - the all-wide-base tyre vehicle. (The necessary tyre sizes for fitting drive axles are not yet commercially available, but sufficient data were generously provided by the tyre manufacturing industry.

Other possible future configurations were evaluated, to enable comparisons to be drawn.

The overall results of the estimation of total costs and benefits were as shown in the following Table.

Table: Impact of the tyre and vehicle fitments on the economic stakeholders, haulage operators, governments and society (All figures in Million).

	Period (vehicle and tyre fitments)	Pavement Maintenance Costs	Tax Income	Vehicle Operating CostsB	Non Pavement Costs - Gov	Non Pavement Costs - Ope	Costs Diff. Related to Present
Operator	Past			2648		785	2686
	Present			Ref		747	Ref
	Future			-881		699	- 929
Government	Past	6084	+572		4474		- 384
	Present	6095	Ref		4275		Ref
	Future	6026	- 483		4107		246
Evolution of Society Costs	Past	6084	572	2648	4474	785	2302
	Present	6095	Ref	Ref	4275	747	Ref
	Future	6026	- 483	- 881	4107	699	- 682

NB :

- The non pavement costs attributed to the governments are the consequences of the compensation of the HGV polluting emissions. The ones attributed to the hauliers are the consequences of the tyres and wheels recycling.
- The difference between past and present situation is due to both the tyre characteristics improvement and the evolution in maximum authorised vehicle weight (38 to 40 tonnes).
- With the "diameter model" for the TCF computations, the final differences for the Society would be very comparable : 2360 and -779 M instead of, respectively, 2302 and -682 M in the before mentioned table .

Comparing the past (a) to the present (b) situation:

30. The benefits of the use of wide single tyres on heavy goods vehicles, in terms of reduced vehicle operating costs and reduced gaseous emissions, are significantly greater than the additional pavement maintenance costs they cause.

Comparing the present (b) situation to the possible future (c) situation:

31. On the same basis, it was found that the use of a possible future all-wide-base-single tyre configuration on the same 5-axle vehicle would lead to a new societal cost reduction, when compared with the present tyre configuration on that vehicle.

Comparing the present (b) situation with other possible future situations:

Other Possible Future situations were also investigated by COST 334, namely:

- Using 425/65R22.5 tyres on the towed axles, instead of the present 385/65R22.5. The reference 80 series tyres would be kept on the motor vehicles. This would increase the tread width and reduce the tyre inflation pressure on the towed axles and lead to a reduction of pavement wear and pavement maintenance costs. However, it would increase the rolling resistance and the tyre and wheel weight, with a negative impact on the vehicle operating costs and the environmental costs.
- Retaining standard 70 or 80 series tyres on the front axles, while using wide-base single tyres on the driven and standard wide base single tyres on towed axles. This would allow conclusions to be drawn on the effect of the introduction of prototype wide base single tyres on driven axles.

32. These results showed that:

32.1 Replacing current 385/65R22.5 by 425/65R22.5 on the towed axles of the HGV would be less damaging to the pavements, but globally more expensive at the EU level.

32.2 Retaining the 70-series tyres on steering axles, with 495/45R22.5 wide base single tyres on the driven axles, and standard wide base single tyres on towed axles of the vehicles is more damaging to

the pavement by about 70 million . but globally less expensive at the EU level, by about 500 million .

33. Building on the Tyre Configuration Factor, further work was carried out to include other economic aspects of tyre performance, namely rolling resistance and mass of the tyre-wheel assembly.

On the current Legal and Regulatory setting

A comprehensive review of the technical standards, Directives and Regulations that might be suitable for the implementation of the results of COST 334 showed that:

34. There is no current technical standard or piece of legislation that accurately reflects the findings of this COST action.
35. Introducing or amending a technical standard to incorporate the findings of this COST action, without the force of legislation, is only likely to have a limited impact in preventing or reducing road pavement damage.

Introducing legislation at the European level that is binding on all Member States both nationally and internationally would be the most effective way of preventing or reducing road pavement damage throughout the EU.

RECOMMENDATIONS

In the opinion of COST 334, some of the recommendations made will need to be implemented into current or future legislation, while others may be voluntary. In either case, it is useful at this point to highlight some of the main points of the current regulatory framework affecting the use of different tyre fitments.

Current Legal and Regulatory setting

On the basis of the review carried out, and the conclusions it reached, the following recommendations are made for implementation of the results of COST 334.

At the European Level

- q The UNECE tyre construction Regulation 54 and the EU tyre construction Directive 92/23/EEC should be amended to include the findings of COST 334.
- q Directive 96/53/EC on the maximum authorised dimensions and weight of vehicles on an international journey should be amended to include a link between the findings of this COST action and the maximum gross and axle weights of lorries and buses.
- q Directive 97/27/EC should be amended to recognise the findings of this COST action in the same way that it already recognises a specification for road friendly suspension.
- q Any future European Vignette system should include the findings of this COST action.

At the National level.

- q To encourage road friendlier vehicle designs Member States should consider linking the findings of this COST action to national maximum vehicle weight limits.
- q To encourage improved vehicle designs Member States should consider ways of introducing or linking fiscal measures to the findings of this COST action.

Specific Recommendations

1. *On the use of tyre parameters in road pavement design.*
The development of the Tyre Configuration Factor allows discrimination between different tyre fitments based on the corresponding damage they cause to road pavements. It is recommended, therefore, that the TCF should be used by national road authorities in the design process to better estimate the damaging effect of the traffic that roads are designed to carry.
2. *On the application of the Tyre Configuration Factor to tyre design and use*
The results of the COST 334 work show that the use of a limit on TCF can be used to guide the design of new tyre sizes, and the further development of existing tyre sizes. It is recommended, therefore, that limiting values of TCF be placed on new and developing tyre fitments.

The limits to be used should be as follows:

Table: Proposed TCF limits for axle types in relation to applied axle load (rounded to nearest 0,05)

		Axle load in tonne						
		6,5	7,0	7,5	8,0	9,0	10,0	11,5
TCF limits								
<i>Proposed limit*</i>								
all axle types: ie. Steering, driven and towed	AWF=1,65	3,90	3,35	2,95	2,60	2,05	1,65	1,25

* with possible exceptions for tyre sizes having a low market share and for special purpose vehicles.

3. On Maximum Designed Operating Tyre Inflation Pressure

In addition to the proposed limits on TCF value of the tyre, it is also recommended that a maximum limit be placed on the manufacturer-recommended inflation pressure of the tyre (measured cold) according to the allowable load level of the specific axle on which the tyre is mounted. This will ensure that the TCF limits cannot be inadvertently exceeded by the use of increased inflation pressure.

The proposed maximum designed operating tyre inflation pressure (measured cold) is 9 bars.

4. On further work to improve the Tyre Configuration Factor

With further work on a number of tyre parameters, to establish reference values, etc., as discussed in Chapter 7.10, a more comprehensive index could be established in addition to the Tyre Configuration Factor. It is therefore recommended that such work is encouraged and supported. The aspects to be included could cover, but need not necessarily be limited to, rolling resistance, noise, and mass of the tyre-wheel assembly.

5. On the specific issue of the proposed amendment to European Directive 96/53/EC

This Directive deals (in Annex 1, parts 2.2.4.2, 2.3.2, 2.3.3 and 3.5.3 of the Directive) with the maximum authorised dimensions in national and international traffic, and the maximum authorised weights in international traffic, for certain road vehicles used within the Community. Among other things, it limits the use of wide single tyres on axles other than towed axles stating that for tandem axles of motor vehicles, "the allowable load is 19 tonnes if the driving axle is fitted with twin tyres and air suspension." A member state proposal to modify the Directive to allow the use of wide single tyres on such axles was deferred until COST 334 had reported.

The COST 334 view on this specific issue is as follows. On the grounds of pavement wear alone, for which this part of the Directive is intended, the modification should not be permitted. However, on other grounds, namely the benefits to vehicle operators, and the potential reduction in gaseous emissions, the modification should be permitted.

The recommendation, therefore, is that the use of wide single tyres should be allowed for 19 tonnes on a tandem axle of a motor vehicle, provided that the proposed Axle Wear Factor is complied with for each axle of the motor vehicle.

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