Evaluation of the Dynamic Performance of some Truck-Trailer Combinations

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6 June 1993

Introduction

The province of Alberta, at the request of sectors of industry, is considering options to allow truck-trailer and tractor-semitrailer-pony trailer combinations to operate at higher gross weights than those specified in the 1991 Memorandum of Understanding on Interprovincial Weights and Dimensions (M.o.U.). This report summarizes briefly an analysis of the dynamic performance of these proposed vehicle configurations against the performance standards which were the basis of the M.o.U.

Truck and Tandem-tandem Full Trailer

The truck-trailer combination consisted of a 3-axle straight truck and a 4-axle full trailer. The truck had a single front steering axle and a tandem drive axle. The truck's 4.88 m (16 ft) long dump box sat on a 5.08 m (200 in) wheelbase with a 1.53 m (60 in) tandem drive axle spread. The front and rear overhangs were 1.27 m (50 in) and 0.61 m (24 in) respectively. The full trailer consisted of a tandem axle A-dolly and a tandem axle semitrailer. The dolly provided a 5 m (197 in) inter-axle spacing from the truck, and had a 1.22 m (48 in) axle spread. The semitrailer was 8.53 m (28 ft) long with a 6.55 m (258 in) wheelbase and a 1.53 m (60 in) axle spread. The semitrailer's front and rear overhangs were each 0.61 m (24 in), and it was attached to the dolly through a conventional fifth wheel.

Three vehicle gross weights were examined in this analysis. The first case, 53500 kg, is as specified in the M.o.U., with the truck front and drive axles loaded to 5500 kg and 17000 kg respectively, and the trailer tandems each loaded to 15500 kg. The second simply increased the trailer tandem axle loads to 17000 kg, for a gross weight of 56500 kg. The final step increased the truck front axle load to 7300 kg, the maximum allowed in Alberta, for a gross weight of 58300 kg.

Note that the request suggested that a trailer wheelbase of 8 m should be used. However, it does not appear possible to reach this wheelbase within a box length of 18.5 m, except by use of a short wheelbase truck with perhaps a 4.26 m (14 ft) long box, which is shorter than most typical equipment currently in use. The analysis was therefore conducted using a 6.5 m trailer wheelbase, compatible with the M.o.U. Cases were also run using the longer trailer wheelbase, in violation of the box length restriction. Note that an 8 m wheelbase results in a trailer box about 10.4 m (34 ft) long, which is very long for an end-dump trailer. The typical Ontario tri-axle dump semitrailer is about 10.1 m (33 ft) long on an 11 m (36 ft) long frame. This is widely regarded by industry as too long for safe dumping in all conditions, although it must be noted that some of the problems are exacerbated by the biassed loading necessary for axle weight compliance with a vehicle design that compromises axle loads in pursuit only of gross weight. I would suggest that an 8 m trailer wheelbase would be undesirable, even if it would be feasible within the other dimensional limits. A longer wheelbase certainly promotes stability, which should improve highway safety. However, the resulting longer box does reduce stability while dumping, which makes it easier for the truck to tip over, which is an off-highway industrial accident that can have just as serious consequences as the highway accident.

Gravel was used to load the vehicle to its desired axle weights, which is compatible with the assumption of the study on which the 1991 M.o.U. was based. However, cases were also run with a lumber as the payload, to provide a high centre of gravity.

Table 1 shows the computed performance indices of the truck-trailer combinations. Results of this analysis indicate that the steady state performance of the truck-trailer combinations was not a concern. The rollover threshold of the three configurations was much higher than the desirable level of 0.4 g, due to the relatively low centre of gravity of the dense payload. The high speed offtracking exceeded the recommended maximum of 0.46 m by a small margin. Both the friction demand and low speed offtracking meet their respective performance criteria. However, transient performance of these three vehicle configurations did not meet the recommended performance criteria as shown in Table 1. Even with the trailer wheelbase increased to 8 m, the lateral load transfer ratio and transient offtracking were only improved to 0.627 and 0.884 m respectively and were still higher than the recommended values.

Table 1/ Performance of truck-trailer combinations

GVW (kg)	Trailer Wheelbase (m)	Load	Performance measure High speed Offtracking (m)	Load Transfer Ratio	Transient Offtracking (m)
53500	6.5	Gravel	0.576	0.675	0.896
56500	6.5	Gravel	0.601	0.715	0.966
58300	6.5	Gravel	0.604	0.708	0.948
53500	8.0	Gravel		0.627	0.884
53500	6.5	Lumber		Rollover	Rollover
Standard			**< 0.46**	**< 0.6**	**< 0.8**

Tractor-semitrailer-pony Trailer

The tractor-semitrailer-pony trailer combination consisted of a 3-axle tractor towing a tandem axle semitrailer and a tandem or tridem pony trailer with a wheelbase of 8.5 m. The truck had a 5.08 m (200 in) wheelbase with a 1.53 m (60 in) tandem drive axle spread. The semitrailer was 9.1 m (30 ft) long with a 7.1 m (280 in) wheelbase and a 1.53 m (60 in) axle spread. The pony trailer had a wheelbase of 8.5 m, and was attached to the semitrailer by a pintle hook with a hitch offset of 1.4 m. The vehicle was within 25 m overall length and 20 m box length, and the inter-axle spacings exceeded the minimum values specified in the M.o.U.

Three vehicle gross weights were examined in this analysis. The first case, 53500 kg, is as specified in the M.o.U., with the tractor front and drive axles loaded to 5500 kg and 15500 kg respectively, the semitrailer tandem loaded to 15500 kg, and the pony trailer tandem loaded to 17000 kg. The second simply increased the tractor and semitrailer tandem axle loads to 17000 kg, for a gross weight of 56500 kg. The final step replaced the pony trailer tandem with a narrow spread tridem, and increased the gross weight of 60500 kg. Gravel was used to load the vehicle to its desired axle weights, which is compatible with the assumption of the study on which the 1991 M.o.U. was based.

Table 2 shows the computed performance indices of the tractor-semitrailer-pony trailer combinations. Results of this analysis indicate that the steady state performance of the truck-trailer combinations was not a concern. The rollover threshold of all vehicles was much higher than the desirable level of 0.4 g, due to the relatively low centre of gravity of the dense payload. The high speed offtracking exceeded the recommended maximum of 0.46 m by a small margin. Both the friction demand and low speed offtracking meet their respective performance criteria. However, transient performance of these three vehicle configurations did not meet the recommended performance criteria as shown in Table 2. The tandem axle pony trailer failed the lateral load transfer ratio performance standard for both weights, whereas the tridem pony trailer was able to meet it at the lower weights, though failed at the highest weight. These results are consistent with earlier studies of these configurations, where only the use of a fifth wheel was effective in moderating the rearward amplification of the pony trailer.

Table 2/ Performance of tractor-semitrailer-pony trailer combinations

GVW (kg)	Trailer Axles	Load	High speed Offtracking (m)	Performance measure Load Transfer Ratio	Transient Offtracking (m)
53500	Tandem	Gravel	0.554	0.655	0.826
56500	Tandem	Gravel	0.578	0.667	0.869
53500	Tridem	Gravel	0.527	0.592	0.768
56500	Tridem	Gravel	0.555	0.604	0.817
60500	Tridem	Gravel	0.596	0.729	0.921
Standard			**< 0.46**	**< 0.6**	**< 0.8**

Discussion

The study which led to the addition of truck-trailer combinations to the 1991 M.o.U. examined a range of straight trucks and truck-trailer combinations. All of them were already in use somewhere in Canada, but not all of them were necessarily in use in all provinces at that time. It found that the dynamic performance of both truck-pony trailer and truck-full trailer combinations was marginal, in the same manner that the dynamic performance of A-trains is marginal. Since the proposed additions to the M.o.U. would allow truck-trailer combinations to operate in all provinces, it was a matter of some concern that vehicles having marginal performance would be introduced into provinces where they did not already operate. On that basis, the vehicles were configured in the 1991 M.o.U. with restrictions on the box length and gross weight, similar to the restrictions on these parameters in the 1988 M.o.U. for the A-train. The objective was the same, allow these vehicles to operate, but with sufficiently restrictive conditions that the B-train would remain the vehicle of choice for heavy haul.

Unfortunately, the B-train is not well suited for all commodities. While B-train designs have been developed that allow straight-through loading of both trailers for dry vans, loads that need end-dumping or refrigeration seem to require that the trailers be uncoupled for loading and unloading at this time.

It is clear that truck-trailer combinations could offer gross weights competitive with the B-train, or even greater in provinces where a higher front axle load than 5500 kg is allowed for straight trucks.

The dynamic performance of truck-trailer combinations is marginal, as shown in the earlier study, and above. These vehicles are principally sensitive to the evasive manoeuvre, where the load transfer ratio would indicate a likelihood of trailer rollover in an emergency lane change. The load transfer ratio for the trailer exceeds the performance standard even with a low centre of gravity load like gravel. This parameter is sensitive to both trailer payload, and trailer payload centre of gravity height. There is nothing in vehicle regulation at this time to prevent any vehicle being operated with any particular centre of gravity height, so these vehicles could carry loads of grain or lumber with a much higher centre of gravity which significantly degrades the load transfer ratio. The load limits in the 1991 M.o.U. were derived simply as a matter of policy so that no truck-trailer could have a higher gross weight than the tractor-semitrailer or A-train with a corresponding number of axles. The load allowed on the trailer was also restricted so that the truck would have to be fully loaded to achieve the maximum gross combination weight. This minimizes the load on the trailer, which tends to minimize the load transfer ratio. However, there is still no direct control on centre of gravity height, and it would be possible to operate these vehicles with loads that would tend to promote a high load transfer ratio. The thinking, though, was that the majority of this equipment is used in the construction industry, with a moderate centre of gravity height for most loads.

The M.o.U. was intended to promote uniformity in regulations between provinces, with those regulations based on objective measures of vehicle performance that are presumed to be related to safety. It allows provinces to continue to allow more generous weight and dimension limits than existed prior to 1988, but it was never intended that this provision would allow provinces to implement new M.o.U. vehicles at higher limits than the M.o.U.

That having been said, it is also known that Alberta allows a 7300 kg load on the front axle of straight trucks, with suitable limits on tire and axle ratings. The performance measures used as a basis for configuration of all vehicles under the M.o.U. are based on the power unit following a prescribed path. The actual path of a truck that follows the specified path has negligible difference for different loads on the truck, so for the particular truck-trailer combinations of interest here, where the responses of concern are all related to the trailer, these responses will be essentially independent of the load on the truck, but highly dependent on the load on the trailer. It would be possible to allow the gross weight of truck-trailer combinations to be increased above the values set in the M.o.U., provided all that load was added to the truck, and the trailer load limits were maintained. This would not affect vehicle performance as evaluated by the measures standards that are the basis of the M.o.U., to the extent that truck performance itself was not compromised. As noted in the original study, an increase in gross weight from 22500=kg to 24300 kg for a load of gravel is not a concern. From an operational point of view, the truck-trailer would still be able to operate into other provinces, but possibly at a reduced gross weight. However, in both cases, the trailer would be identical, and identically loaded.

Nevertheless, it is necessary to return to some earlier points. Truck-trailer combinations have marginal performance, and as such, should not be encouraged. They were purposely given no gross weight advantage over tractor-semitrailers and A-trains with the same number of axles, though they may in most provinces have an advantage in axle capacity. There is clearly a strong case that no more load should be allowed on trailers. However, as noted above, it would be possible to add load to the truck with no significant reduction in combination vehicle performance. This would then give these vehicles a gross weight advantage over tractor-semitrailers and A-trains, which would could make them a preferred configuration over the tractor-semitrailer for some commodities. This is quite different from choice of one of these vehicles for purely operational reasons.

The tractor-semitrailer-pony trailer was also studied as part of the work that led to the 1991 M.o.U., and was also found to be deficient in dynamic performance. Unlike truck-trailers, this configuration was not in widespread use in many provinces, so it was clear that this configuration should not be added to the M.o.U. as this would simply introduce a vehicle of marginal performance into national use, which is counter to the objectives of the M.o.U. It was suggested, rather, that the category of A-train could be broadened to include any vehicle of inadequate performance, so that the M.o.U. would provide a clear deterrent against use of vehicles that fail to meet the performance standards. However, this was not adopted. The terms of the M.o.U. do allow provinces to be more generous than the M.o.U., but this was envisaged as am means to allow provinces to avoid rollback of current more generous allowances than to add new configurations and increase allowances beyond the M.o.U.

Conclusions

The dynamic performance of the truck-trailer combinations described in the 1991 M.o.U. is marginal, and any increase in trailer load reduces that performance. However, if the truck has some available axle capacity, load can be added to the truck to the extent that it does not seriously degrade truck performance.

The dynamic performance of tractor-semitrailer-pony trailer combinations is marginal, and this configuration cannot be recommended except possibly at weights that would be unattractive for bulk commodities.

Any addition of new configurations of marginal performance, or increase in allowable gross weight above the values given in the M.o.U., is a blow to the integrity of the M.o.U.

Influences of Tractor Wheelbase, Tandem-axle spread and Fifth-Wheel Offset on Commercial Vehicle Dynamics

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Condensed Version of

Technical Report

1990/03

ABSTRACT

The impetus for this study has come from the prevailing debate on overall vehicle length limits. Regulations have evolved in Canada which limit both box length and overall vehicle length, with an indirect but important effect on the length limit for tractors and, in particular, their wheelbases. The study focuses on the influence of variations in tractor parameters that have a first-order effect on vehicle performance. The UMTRI Yaw/Roll Model and simplified low- and high-speed offtracking models are used for the simulation work. Only parameters which have potential for control by regulation, such as wheelbase, tandem axle spread, and fifth wheel location, are varied in the simulations. The influence of these parameters on the vehicle's dynamic performance is assessed against selected performance criteria, namely, friction demand, handling, dynamic rollover stability, and offtracking.

NOTATION

F_z	Vertical load on tires	lbs
F_y	Cornering force at a tire	lbs
G	Gravitational acceleration	in/s^2
I_x	Roll moment of inertia of tractor sprung mass	$in\text{-}lbs\text{-}s^2$
I_y	Pitch moment of inertia of tractor sprung mass	$in\text{-}lbs\text{-}s^2$
I_z	Yaw moment of inertia of tractor sprung mass	$in\text{-}lbs\text{-}s^2$
L	Tractor reference wheelbase	in
W_s	Tractor sprung mass	lbs
W_f	Equivalent partial sprung weight supported by front suspension of tractor	lbs
W_r	Equivalent partial sprung weight supported by rear suspension of tractor	lbs
X	Longitudinal position of the tractor sprung-mass centre of gravity with respect to front axle centre	in
μ	Friction demand at tractor drive axles	-
Γ	Articulation angle between tractor and first tractor	deg

2.0 COMPUTER SIMULATION MODEL

2.1 Yaw/Roll Model

The UMTRI Yaw/Roll Model (Winkler, et al., 1981), was developed for the purpose of predicting the directional and roll response of single and multiple articulated vehicles engaged in steering manoeuvres which approach the rollover condition. It should be noted that the model does not permit the simulation of braking manoeuvres. However, it does permit the analysis of unconventional vehicle layouts. The equations of motion are developed in such a fashion that it is possible to use the model for simulating vehicles with:

(a) Any number of units and articulation points.

(b) Any placement of wheels and tires.

(c) Any of the hitch mechanisms and constraints that are presently used in heavy-duty commercial vehicles.

In the model, the forward velocity of the lead unit is assumed to remain constant during the manoeuvre. The longitudinal motion of each sprung mass is therefore not allowed to vary, and so each is treated as a rigid body with five degrees of freedom: lateral, vertical, yaw, roll, and pitch. The axles are treated as beam axles that are free to roll and to bounce with respect to the sprung mass to which they are attached.

2.2 Simplified Offtracking Models

As with the Yaw/Roll Model, the simplified offtracking models used in this study were developed by UMTRI (in 1988). These models examine three different aspects of offtracking performance of multiple unit vehicles, namely:

(a) Low-speed steady-state offtracking

(b) Low-speed transient offtracking

(c) High-speed steady-state offtracking.

Each of these aspects is examined in a constant-radius turning manoeuvre where the radius is defined by the user. For the steady-state options, the vehicle is assumed to be turning continuously and to have achieved a steady-state response. For the low-speed transient option, the manoeuvre includes a straight line "entry" and straight line "exit" to the constant radius turn. The total arc, or angle, of the turn is defined by the user. The paths of the centreline of each axle and of the rearmost extremity of the vehicle are determined by the model.

(2) Location of Centre of Gravity of Tractor's Sprung Mass

The longitudinal location of the sprung-mass centre of the baseline tractor (190 in wheelbase) is 55 in behind the front axle centreline. The generalized relationship for tractors of wheelbase L, with longitudinal distance X of the sprung-mass centre of gravity behind the front axle, is estimated by the relationship:

$$X = 55 + (L - 190)/2. \tag{2}$$

(3) Tractor Sprung Mass Moment of Inertias

The sprung-mass roll moment of inertia I_x (in-lbs-s^2), is determined for each of the different wheelbases from the tractor's sprung weight W_s, assuming a constant value of 29 in for the radius of gyration of the sprung-mass, namely:

$$I_x = 2.178\ W_s \tag{3}$$

The sprung-mass pitch and yaw moments of inertia, I_j ($j = y, z$) is determined by the empirical formula:

$$I_j = [(W_f + 0.4\ W_r)X^2 + 0.6\ W_r(L - X)^2]/G, \tag{4}$$

where W_f and W_r are the equivalent partial sprung weights supported by the front and rear suspensions; and X and (L - X) are the absolute values of the distances from the sprung-mass centre of gravity.

The height of the centre of the tractor sprung-mass is assumed to be a constant 44.0 in above ground level.

Table 1 shows the values of W_s, X, I_x, I_y and I_z as a function of the tractor wheelbase, L. A commonly-used tractor is arbitrarily chosen as a baseline vehicle (B.L.) for purposes of this study.

When the tractor wheelbase is varied around the B.L. value, the other B.L. values are held constant; i.e., the tandem axle spread is held at 60 in, and the location of the fifth wheel is held at 17.75 in forward of the tandem axle centreline.

Throughout this study the vehicle is assumed to be fully loaded and the design parameters of the trailers were not varied. The vehicle was fitted with tires with cornering characteristics shown in Figure 2.

angle
wheel is 150° and the steering input during the lane-change manoeuvre is completed within 2 s. This represents an average left and right front-wheel steering displacement of about 4° amplitude as shown in Figure 4. Very small differences in the front-wheel steering pattern for the various vehicle configurations are attributed to the compliances of the various steering systems. The vehicle speed was held constant at 90 km/h during the manoeuvre.

4.2 Offtracking

Maximum steady-state and transient low-speed offtracking values are calculated for the vehicles during a 90° turn, where the radius of the turn is 13.7 m (45 ft) to the centre of the front axle of the tractor.

High-speed offtracking is defined as the lateral offset of the path taken by the trailing axles of a vehicle combination from the path taken by the tractor's steering axle in a steady turn. The offtracking is calculated during steady turning of a radius of 365.3 m (1,200 ft) at a speed of 88.6 km/h (55 mph).

4.3 Friction Demand

The friction demand is defined as the non-tractive friction levels between the tires and the road surface at the rear of the tractor. The friction demand is the absolute value of the ratio of the resultant shear force arising simply due to curvilinear travel divided by the cosine of the tractor/trailer articulation angle to the vertical load imposed on those tires, F_z. The instantaneous friction coefficient, μ, demanded at the rear tires of a tractor is given by (Ervin and Guy, 1986):

$$\mu = \left|(\Sigma F_y \;/\; \mathrm{Cos}\ \Gamma)/\Sigma F_z\right| \tag{6}$$

It should be noted that the absolute value of μ has been used to avoid negative values when the cornering force, F_y, is negative.

In this study the peak value of the friction demand is determined under two manoeuvring situations, namely a 90 km/h rapid lane-change steering manoeuvre, and a low-speed, tight steering manoeuvre. These manoeuvres will show the influence that tractor parameter variations have on the high- and low-speed jackknifing tendency of the vehicle on low friction road surfaces.

The high-speed friction demands for various vehicle configurations are calculated using the rapid lane-change manoeuvre described in Section 4.1. The low-speed friction demand is calculated during the steering manoeuvre shown in Figure 5.

In general, lengthening the tractor wheelbase can result in improved dynamic rollover stability. In particular, increasing the wheelbase from 4.826 m to 5.689 m results in a small improvement of the dynamic rollover stability (i.e., reducing the peak LTR).

Figure 9 shows the influence on load transfer ratio of changing the tractor's tandem-axle spread. There is a steady increase in the peak LTR when the axle spread is decreased. Over the range of spreads studied, the total increase in peak LTR is 7.2 percent.

Figure 10 shows the influence on peak LTR of the fifth wheel offset with respect to the tractor's tandem-axle centreline. Changing the location of the fifth wheel anywhere within the range of 0.0 m to 0.451 m ahead of the tandem-axle centreline does not significantly affect the LTR.

Parenthetically, it should be noted that when the peak LTR increases, the lateral acceleration rearward amplification of the last articulated unit also increases.

It should be emphasized that these results are partly dependent on the choice of manoeuvre to which the vehicle is subjected. In determining the peak LTR, a steer-input manoeuvre has been used (see Fig. 3); however, if a path-follow manoeuvre had been used instead (see Ervin and Guy, 1986), the calculated values for peak LTR would be somewhat different but nevertheless the trend of whether the peak LTR increases or decreases with each independent variable (tractor parameter) is expected to remain the same.

5.2 Offtracking

The influence of tractor wheelbase variations on the transient low-speed and high-speed offtracking is shown in Figures 11A, 11B, and 11C, respectively. Long wheelbase tractors exhibit greater offtracking than short ones. Increasing from the middle wheelbase to the long wheelbase results in an increase in the low-speed offtracking by 13.2 percent; the transient low-speed offtracking by 6.6 percent; and the high-speed offtracking by 1.5 percent. Reducing from the middle wheelbase to the short wheelbase results in a decrease in the low-speed offtracking by 12.8 percent; the transient low-speed offtracking by 7 percent; and the high-speed offtracking by 3 percent.

It should also be noted that the transient low-speed offtracking is significantly higher than the steady-state low-speed offtracking. This underscores the importance of considering transient offtracking when studying the effect of wheelbase changes on a vehicle; an increase in the transient low-speed offtracking may cause unexpected road geometry interference.

Figures 12A, 12B, and 12C show the influence on offtracking of the fifth wheel offset. Variations in the offset from 0.0 to 0.451 m do not significantly affect either the low-speed or the transient low-speed

It should be emphasized that these results are very dependent on the choice of manoeuvre to which the vehicle is subjected. In determining the peak friction demand at low speed, a steer-input manoeuvre has been used [see Fig. 5]; however, if a path-follow manoeuvre had been used instead (see Ervin and Guy, 1986), the calculated values for peak friction demand would be different, and, moreover, even the trend of whether the peak friction demand increases or decreases with each independent variable (tractor parameter) will be reversed. This trend reversal is particularly pronounced where the tractor parameter being studied is the tractor's tandem-axle spread. The pattern of trend reversals is expected to be true for high speed friction demand as well.

5.4 Handling

As mentioned in Section 4.4, the steady-state performance of the vehicle configurations is based on:

(a) Steady-state lateral acceleration response to fixed steering inputs at a constant speed of 90 km/h.

(b) Understeer coefficient at a given steady-state lateral acceleration using the "Handling Diagram".

Figures 17 and 18 are plots of the steady-state lateral acceleration versus steer angle at a speed of 90 km/h and the handling diagrams for vehicle combinations with varying tractor wheelbases.

From these diagrams the following observations can be made:

(a) Short wheelbase tractors generate higher lateral accelerations for a given front steer angle than long wheelbase tractors. A short wheelbase tractor will become yaw divergent (directionally unstable) at a smaller steer angle than a long wheelbase tractor. For the vehicles examined in this study, the short wheelbase tractor became yaw divergent at a steer angle of 1.15 degrees, and the long wheelbase tractor became yaw divergent at 1.45 degrees.

(b) The handling diagram shown in Figure 18 reveals that the lateral acceleration at which the transition from understeer to oversteer occurs increases as the wheelbase decreases. This is largely attributed to increased slip angles at the steering axle of the tractor as the wheelbase is reduced.

It is also clear from the handling diagram that the magnitude of the understeer coefficient, within a wide range of lateral acceleration (from 0.0 to 0.25 g's), increases with decreasing wheelbase. For example, at lateral acceleration of 0.25 g's, shortening the tractor from the long wheelbase to the middle

As the tractor wheelbase diminishes, vehicle stability and control diminishes. Understeer increases, and the transition between understeer and oversteer becomes more abrupt, which requires more active driver input.

The sensitivity of the vehicle handling response increases as the wheelbase diminishes, particularly in the range between 4.826 m (190 in) and 3.759 m (148 in). The longitudinal position of the fifth wheel relative to the center of the tractor tandem-axle group has a first-order effect on vehicle handling. For the B-Train examined in this study, positioning the fifth wheel at the center of the tandem-axle group is undesirable. At this position the handling curve reveals a very reactive vehicle with virtually no understeer, which is a strong indicator that the tractor would be more susceptible to jackknife. The data suggest that there may be an optimum setting for the fifth wheel. For the B-Train double a positive fifth-wheel setting of 0.254 m (10 in) produces slightly better vehicle handling characteristics than a setting of 0.451 m (17.75 in). It can be expected that the influence of the fifth-wheel position will be greater as the tractor wheelbase diminishes.

Of the three parameters varied, the tractor's tandem-axle spread had the smallest influence on vehicle handling. The data suggest that there is an optimum tandem-axle spread. It is clear that large spreads, in the order of 1.829 m (72 in) or more, are less desirable than smaller spreads. The intermediate spread of 1.524 m (60 in) was found to be the most favorable axle spread. The influence of tractor tandem-axle spread on tractor handling will depend on tractor wheelbase. The shorter the wheelbase, the greater is this influence.

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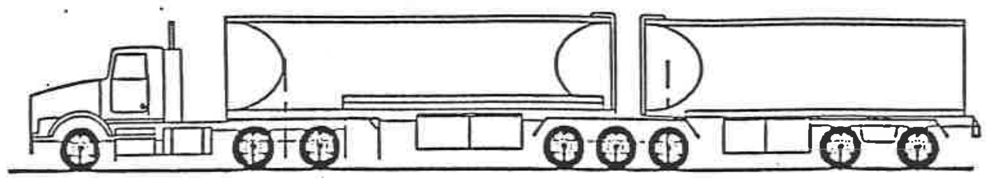

Fig. 1 - B-Train double

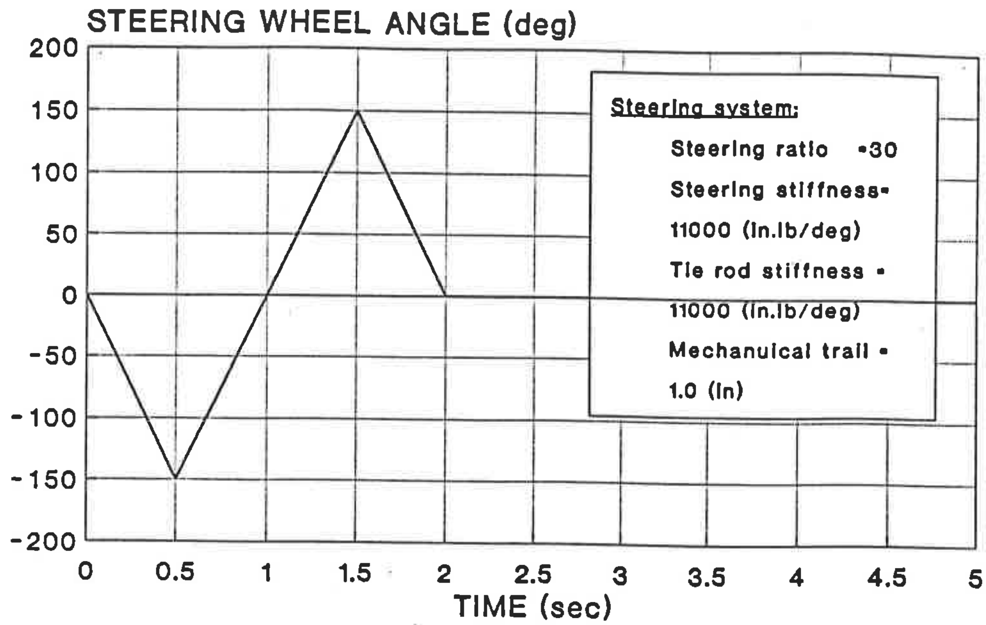

Fig. 3 - Steering wheel angle versus time during rapid lane-change manoeuvre

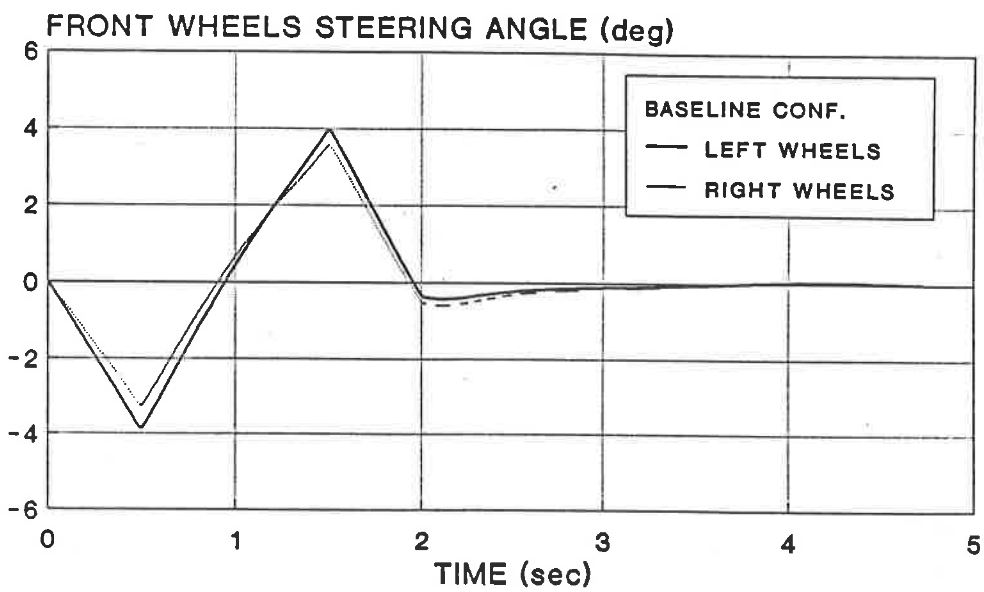

Fig. 4 - Left and right front wheels steering angle during rapid lane-change maneouver (speed = 90 km/h)

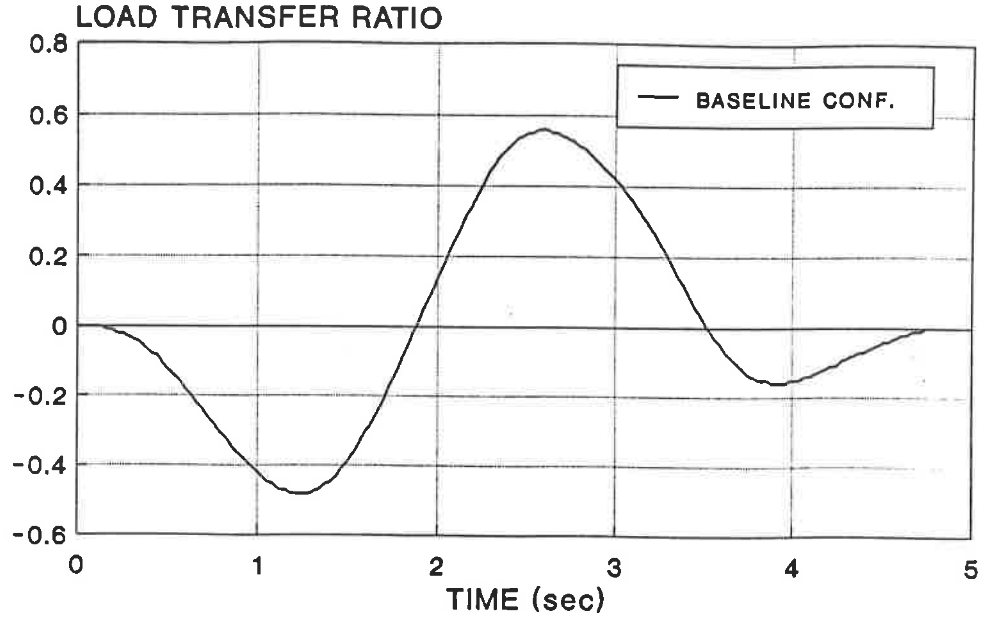

Fig. 7 -Load transfer during rapid lane-change manoeuvre (speed = 90 km/h)

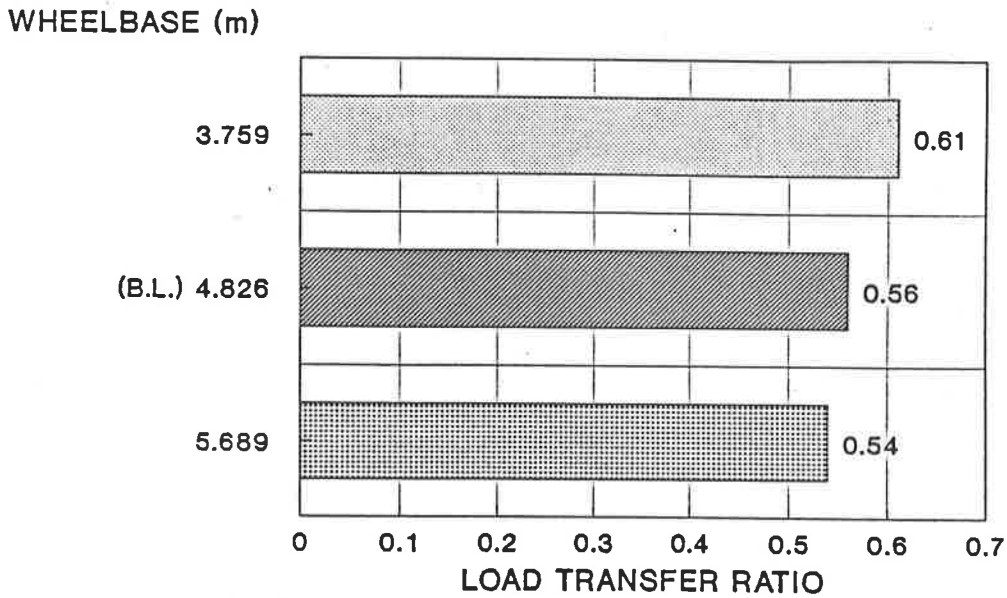

Fig. 8 - Influence of tractor wheelbase on load transfer ratio during rapid lane-change manoeuvre

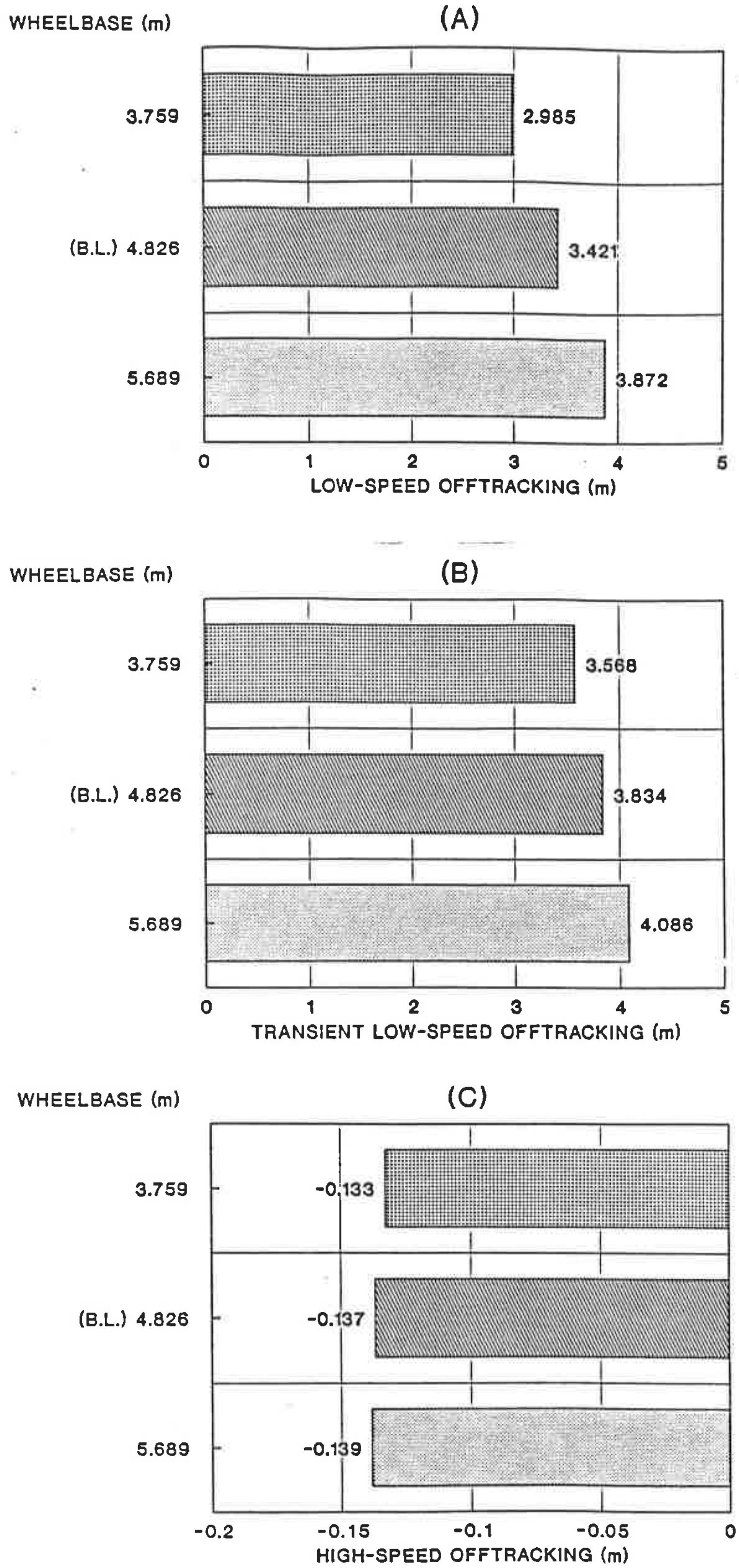

Fig. 11 - Influence of tractor wheelbase on offtracking

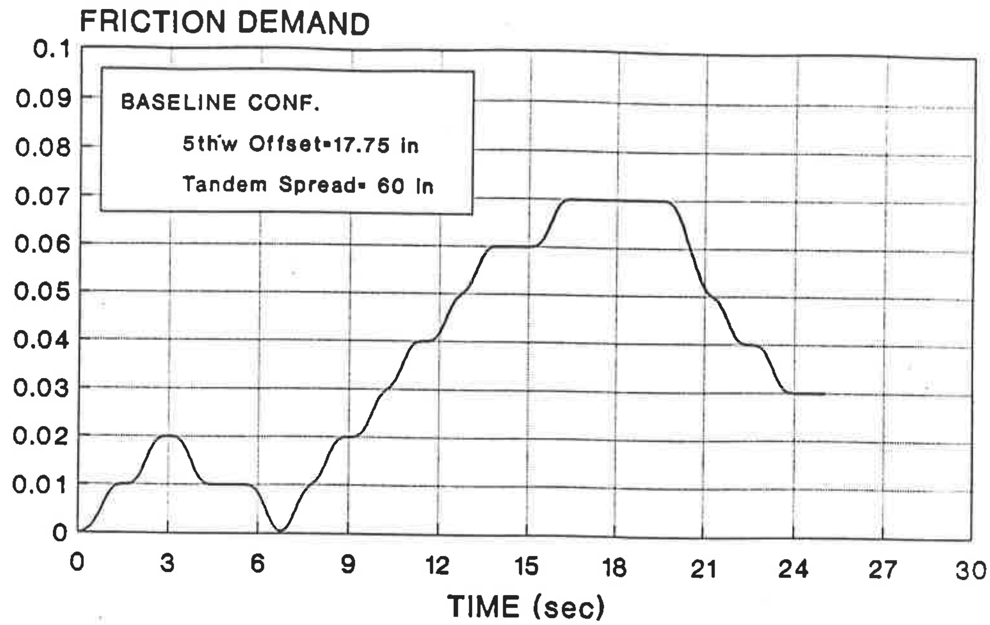

Fig. 13 - Friction demand at drive axles during low-speed, tight turn manoeuvre

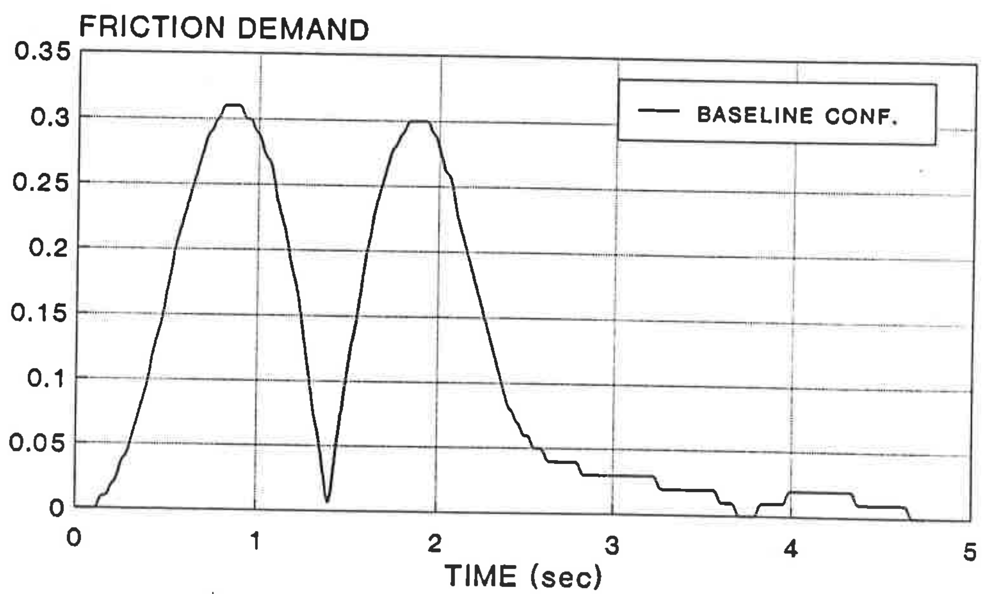

Fig. 14 - Friction demand at drive axles during rapid lane change manoeuvre (speed = 90 km/h)

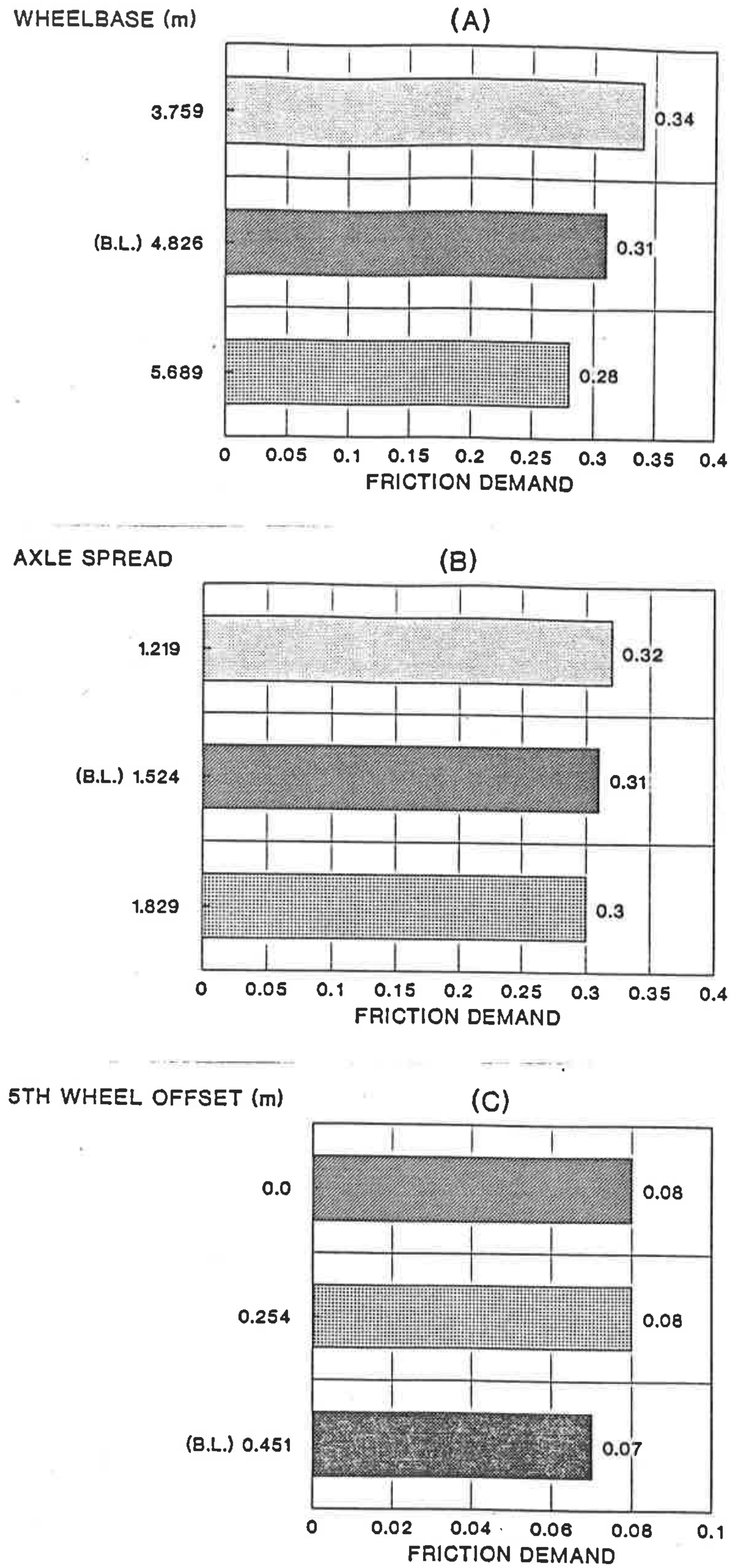

Fig. 16 - Influence of tractor parametes on friction demand during lane-change manoeuvre (90 km/h)

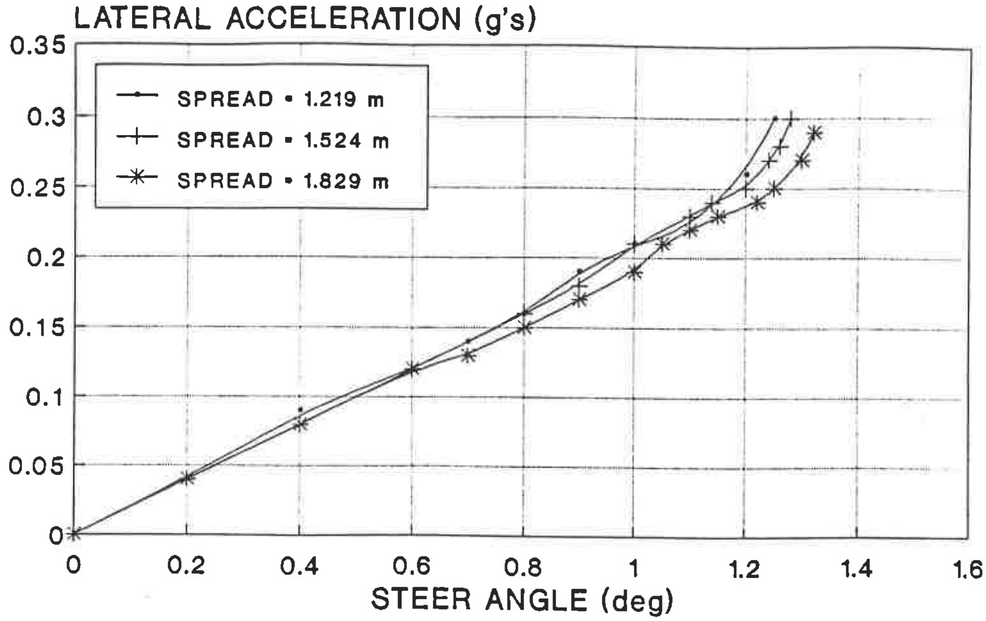

Fig. 19 - Influence of axle spread on steady-state lateral acceleration response at 90 km/h

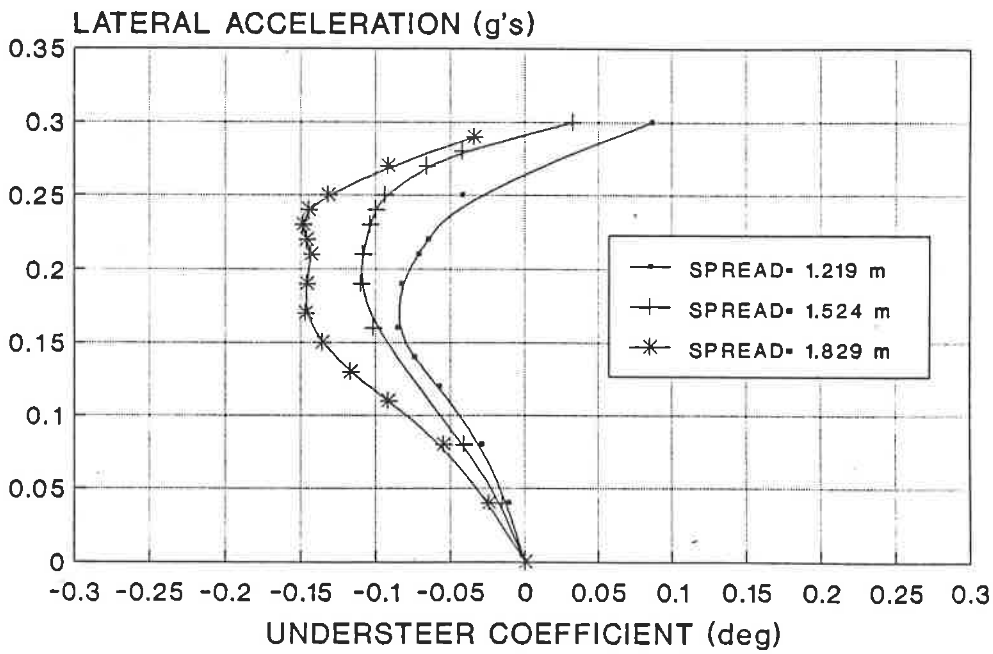

Fig. 20 - Influence of axle spread on handling performance at 90 km/h