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

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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

1.0 INTRODUCTION

There is a perception among truck drivers that the stability and controllability of highway tractor-trailers is significantly affected by variations in the tractor wheelbase. Tractor wheelbase in turn is indirectly affected by provincial weight and dimension laws in Canada; these laws effectively have the potential to determine the mean wheelbase of tractors in the national trucking fleet, and thereby also the dynamic performance of the overall vehicles.

The research described in the paper has been sponsored by le Ministère de transports du Québec, through the Roads and Transportation Association of Canada, to better understand what effect tractor wheelbase variations have on heavy truck performance so that this may be reflected appropriately in future weight and dimension laws.

In addition to tractor wheelbase variations, the study also examines the effects of tractor tandem axle spread and the effects of variations in the fifth wheel (tractor trailer articulation point) position.

The baseline vehicle chosen for this study is the Canadian B-train illustrated in Figure 1. This vehicle tends to utilize tractors of shorter wheelbase than those commonly in use with other trailers because of the influence of weight and dimension laws. The B-Train is also the most dynamically favoured of multi-unit articulated vehicles in Canada and has been accorded the highest regular gross vehicle load limits.

To the authors' knowledge, the only previous work in this area was done by R.D. Ervin and Yoram Guy (1986). They conducted an extensive study of the dynamics of articulated vehicles, examining the influence of various vehicle parameters on vehicle performance during low- and high-speed path-follow manoeuvres. However, it did not address the influence of tractor wheelbase on the peak friction demand, dynamic load transfer ratio, transient offtracking, or handling, nor did it address the influence of tractor tandem-axle spread on the dynamic load transfer, transient offtracking, or handling.

The present study examines the net effect of tractor wheelbase, tandem axle spread, and fifth wheel position on the dynamic performance of this class of vehicle, the B-Train; and, by extension of tractor semi-trailer units as well.

The computer simulation models used in this study were developed by the University of Michigan Transportation Research Institute (UMTRI). They will be briefly explained in the next section.

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.

In these models several assumptions are made, most notably:

(a) The cornering forces and aligning moments generated at the tire/road interface are assumed to be linear functions of the slip angle of the tire.

(b) The motion of the vehicle takes place on a horizontal surface with uniform friction characteristics.

(c) Pitch and roll motions of the sprung masses are sufficiently small to neglect.

3.0 VEHICLE CONFIGURATIONS

The eight-axle B-Train vehicle configuration [Fig. 1] has been used in this computer simulation study. The parameter variations that were studied are:

(a) Tractor wheelbase (in the study this is varied from 3.759 m (148 in) to 5.689 m (224 in)).

(b) Fifth wheel offset of the tractor (in the study this is varied from 0.0 to 0.451 m (17.75 in) ahead the tractor's tandem-axle centreline).

(c) Tandem-axle spread of the tractor (in the study this is varied from 1.219 m (48 in) to 1.829 m (72 in)).

The authors have taken into account the influence of changing the wheelbase of the tractor on its other design parameters, namely its weight, centre of gravity location, and moments of inertia in yaw, roll, and pitch. The changes in the basic tractor design parameters as a function of the tractor's wheelbase are as follows (Ervin and Guy, 1986):

(1) <u>Tractor's Sprung Weight</u>

The sprung weight (units of lbs) of a conventional tandem-axle tractor with wheelbase L (units of in) is determined by the formula:

$$W_S = 11800 + 1000[(L - 190)/30]. \qquad (1)$$

This formula assumes that the sprung weight of a baseline (190 in wheelbase) tandem-axle tractor is 11800 lbs (implying at a total tractor weight of 18000 lbs), and that each additional 30 in of wheelbase corresponds to an additional 1000 lbs of sprung weight.

(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:

$$X = 55 + (L - 190)/2. \qquad (2)$$

(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:

$$I_x = 2.178\ W_s \qquad (3)$$

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

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

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.

Table 1. Basic Tractor Design Parameters

Parameter	Short 148 (in) 3.76 (m)	Middle 190 (in) 3.83 (m) (B.L.)	Long 224 (in) 5.69 (m)
	Tractor Wheelbase		
W_s (lbs) (000's)	10.4	11.8	12.9
X (in)	34	55	72
I_x (in-lbs-s^2) (000's)	22.7	25.7	28.2
I_y (in-lbs-s^2) (000's)	75.1	173.2	289.5
I_z (in-lbs-s^2) (000's)	75.1	173.2	289.5

4.0 PERFORMANCE MEASURES

The effects of tractor parameter variation on the dynamic performance of each vehicle configuration are assessed against the following performance measures.

4.1 Load Transfer Ratio (LTR)

This ratio is the absolute value of the difference in total right/left loads to their sum (Ervin and Guy, 1986).

$$LTR = \Sigma|F_L - F_R|/\Sigma(F_L + F_R) \tag{5}$$

where

- Σ indicates summation over all of the vehicle's axles except the tractor steering axle (Ervin and Guy, 1986).
- F_L and F_R are the left and right vertical load at each axle except the tractor steering axle.

The LTR measure serves as an indicator of the proximity to total wheel liftoff and can thus be used to distinguish between the dynamic rollover tendency of different vehicle configurations were subject to the same manoeuvre.

In this study the various vehicle configurations were examined under the same rapid steering manoeuvre. In the simulations, the time history of the steering wheel angle shown in Figure 3 was used as input to the Yaw/Roll Model. Figure 3 also shows that the amplitude of the steering

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):

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

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.

4.4 Handling

The steady-state handling performance of various vehicle configurations were compared by applying ramp-step steering input with different fixed steering wheel angles at a constant speed of 90 km/h [Fig. 6]. After the response of the vehicle reached a steady state, the lateral acceleration and yaw rate of the tractor as functions of the front-wheel steering angle was determined, which permitted the construction of the handling diagram for the tractor. The handling diagram illustrates the relationship between the lateral acceleration of the tractor and the understeer coefficient ($\ell r/U - \delta$), where ℓ is the reference wheelbase of the tractor, r is the steady-state yaw rate, U is the forward speed, and δ is the front wheel steering angle. The handling diagram for each vehicle configuration is constructed with data generated by the UMTRI's Yaw/Roll model. The understeer/oversteer characteristics of the vehicles can then be evaluated over a wide range of lateral accelerations.

5.0 RESULTS

The results are discussed in this section. It should be emphasized that in this study only one basic design parameter (i.e., wheelbase, tandem-axle spread, and fifth wheel location) is varied at a time, while the other parameters are held constant at the baseline vehicle's values. The following discussion looks at the trends of each performance measure as a function of changes to the three basic design parameters.

5.1 Load Transfer Ratio (LTR)

Figure 7 shows the Load Transfer Ratio (LTR) time history of the baseline vehicle (B.L.) during the rapid lane-change manoeuvre. From this curve, the peak value of the Load Transfer Ratio is taken; this and comparable peak LTR values from each of the other vehicles being studied are used to compare the various vehicle combinations. A high value of LTR indicates that the vehicle possesses low dynamic roll stability.

Figure 8 shows that the short wheelbase tractor (3.759 m) exhibits the highest peak LTR and that the long wheelbase tractor (5.689 m) has the lowest. Figure 8 also clearly demonstrates that for the range of wheelbases studied, the most significant increase in the LTR occurs between the middle wheelbase tractor (4.826 m) and the short wheelbase tractor.

Of the total increase in peak LTR that occurs when the wheelbase is reduced from the long wheelbase to the short wheelbase, only 29 percent occurs between the long wheelbase and the middle wheelbase; and fully 71 percent occurs between the middle wheelbase and the short wheelbase.

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 <u>trend</u> of whether the peak LTR increas~~ing~~es 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 offtracking. They do however, slightly affect the high-speed offtracking; moving the fifth wheel forward to the tractor's tandem-axle centre causes a slight reduction of the high-speed offtracking.

5.3 Friction Demand

- **Low speed**

Figures 13 and 14 show the friction demand at the baseline vehicle's tractor drive axles during the low-speed steering manoeuvre and the rapid lane-change manoeuvre [as specified in Figs. 3 and 5], respectively. From these curves, the peak values of the friction demand are taken; they and comparable peak values from each of the other vehicles being studied are used to compare the various vehicle combinations.

Figures 15A, 15B, and 15C show the influence of tractor wheelbase, tandem-axle spread, and fifth wheel offset on the peak value of low-speed friction demand. Increasing from the middle wheelbase to the long wheelbase causes a reduction in the peak friction demand from 0.07 to 0.06, while decreasing from the middle wheelbase to the short wheelbase causes an increase in the friction demand from 0.07 to 0.09. Increasing or decreasing the tandem-axle spread, [Fig. 15B] has a similar effect on the low-speed friction demand. Altering the fifth wheel location within the range examined does not significantly affect the peak low-speed friction demand.

The results indicate that increasing the tractor's wheelbase and the tandem-axle spread can result in a reduction of the friction demand at the drive axles of the tractor, which in turn reduces the low-speed jackknifing risk on a slippery road surface.

- **High speed**

Figures 16A, 16B, and 16C show the influence of varying the tractor parameters ~~have~~ on the friction demand during a rapid lane change manoeuvre. In general, the friction demand at high-speed (90 km) is much greater than that at low-speed. (Note that this manoeuvre could not be performed on icy or low-friction surfaces, with, say, 0.2 friction coefficient.)

The figures show that increasing from the middle wheelbase to the long wheelbase results in a reduction of the friction demand from 0.31 to 0.28, while reducing from the middle wheelbase to the short wheelbase results in an increase in the friction demand from 0.31 to 0.34.

Figures 16B and 16C show that the axle spread and fifth wheel offset have much smaller effects on the high-speed peak friction demand than does the tractor wheelbase.

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 wheelbase results in an increased understeer coefficient of -0.08 degrees. Moreover, the change is non-linear; shortening the wheelbase further by the same amount increases the understeer coefficient by -0.13 degrees, or 1.6 times as much as before.

The handling diagrams also reveal that, as the wheelbase is shortened, the rate of change between understeer to oversteer becomes more abrupt, which is an undesirable handling characteristic.

Figures 19 and 20 show the influence of the tandem-axle spread on the steady-state lateral acceleration response and the handling characteristics. As the axle spread increases, the understeer gradient of the vehicle increases for a lateral acceleration level less than 0.15 g's. The transition point between understeer and oversteer increases slightly with increased spread.

The small axle spread of 1.219 m and the large axle spread of 1.829 m both produce a high rate of change between understeer and oversteer. The large axle spread has an added complication of a very unusual transition curve.

The medium drive axle spread of 1.524 m produces a smooth transition curve and demonstrates a more desirable handling characteristic.

Figures 21 and 22 show that the position of the fifth wheel has a very significant effect on the vehicle handling. Position variations can change the characteristic of the handling curve dramatically from understeer to oversteer. As the fifth-wheel offset is increased (moved forward) from 0 m to 0.451 m, the understeer of the vehicle increases and the lateral acceleration at which the transition takes place increases. The slope of the oversteer portion of the curve was better for the fifth-wheel offset of 0.254 m than for either of the other positions studied. At the zero fifth-wheel offset position, the handling curve reveals a very reactive vehicle with virtually no understeer. This is a strong indication of high jackknife potential.

6.0 CONCLUSIONS

Tractor wheelbase variations have a first-order effect on the stability and control behaviour of tractor-trailer combinations. Increasing the length of tractor wheelbase improves the general stability of the vehicle. Offtracking performance of the vehicle combination diminishes with length, and represents the only negative performance factor within the limits of current industry practice regarding choice of tractor wheelbase. The upper bound of tractor wheelbase can best be evaluated and controlled using offtracking measure.

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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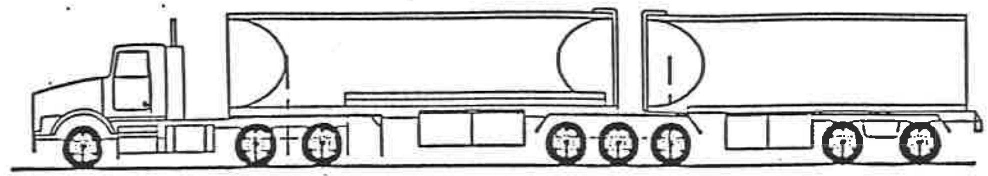

Fig. 1 - B-Train double

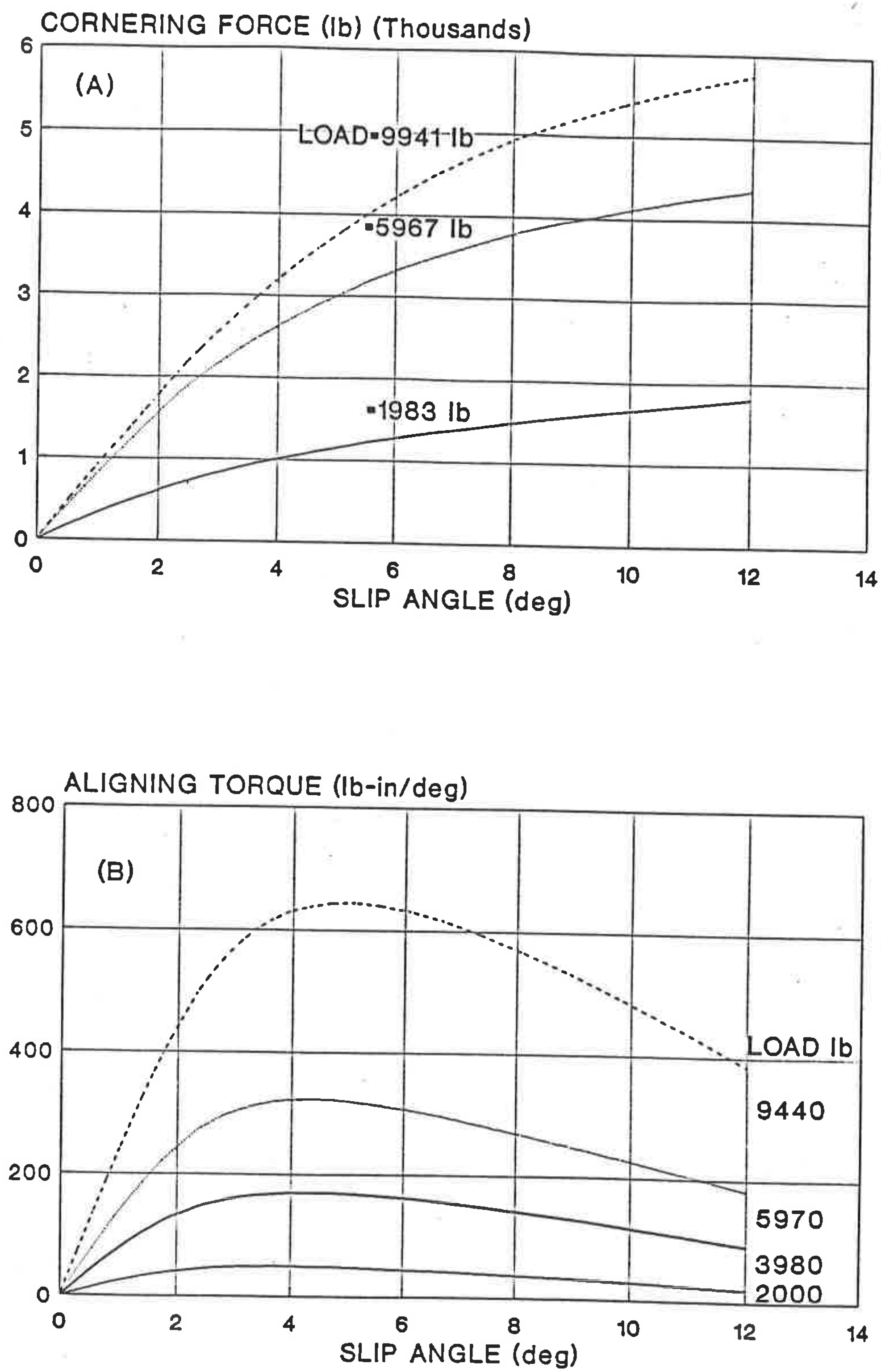

Fig. 2 - Tire characteristics

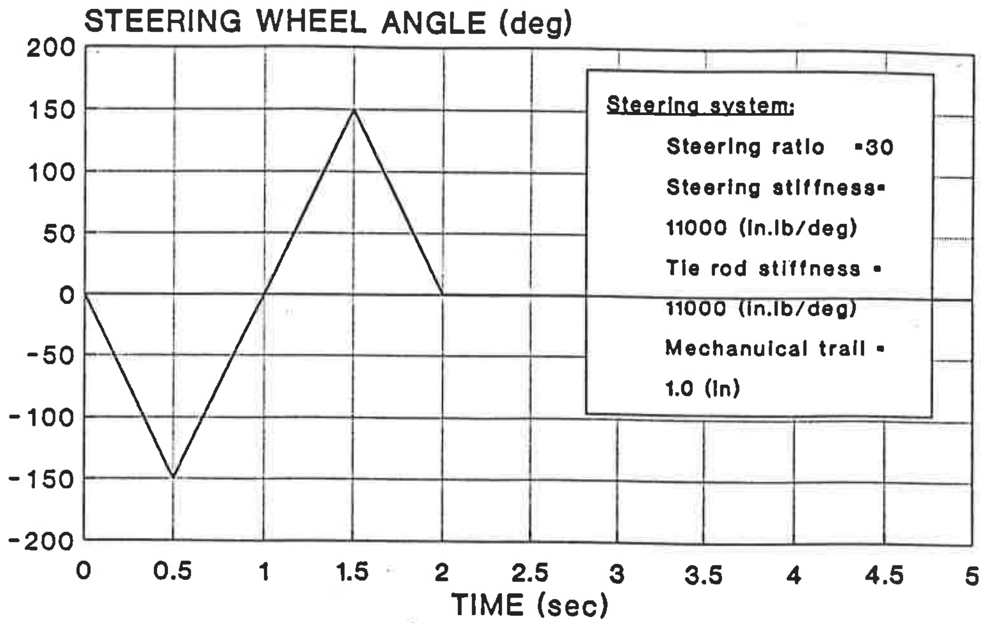

Fig. 3 - Steering wheel angle versus time during rapid lane-change manoeuvre

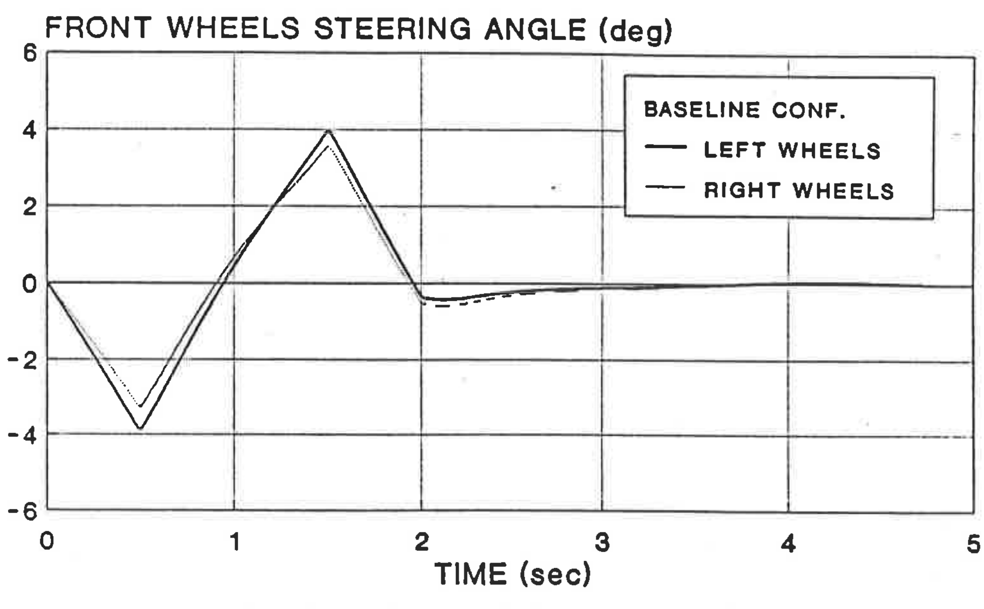

Fig. 4 - Left and right front wheels steering angle during rapid lane-change maneouver (speed = 90 km/h)

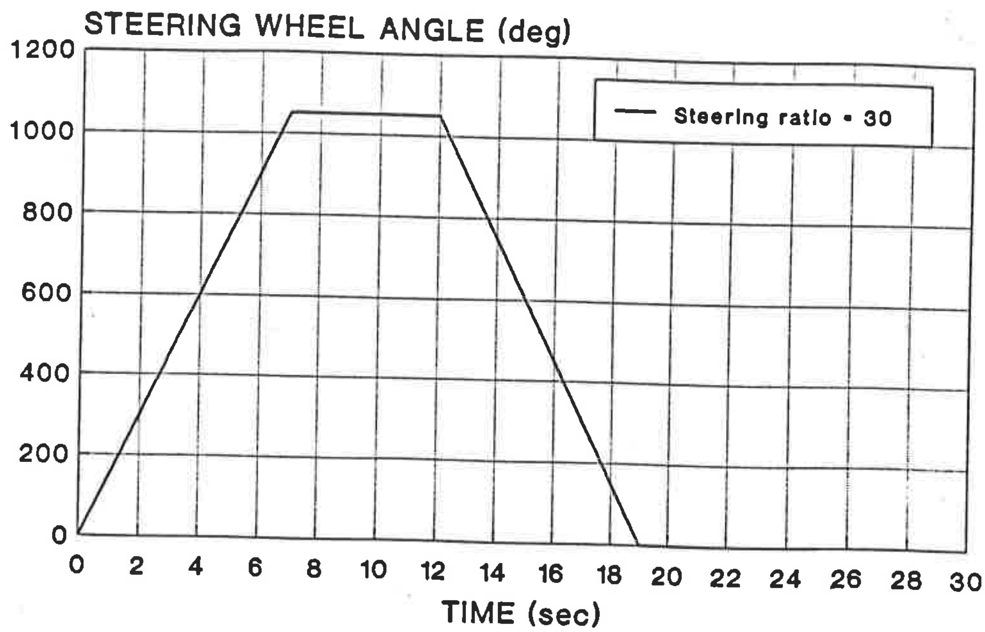

Fig. 5 - Steering wheel input during low speed maneouver (5 km/h)

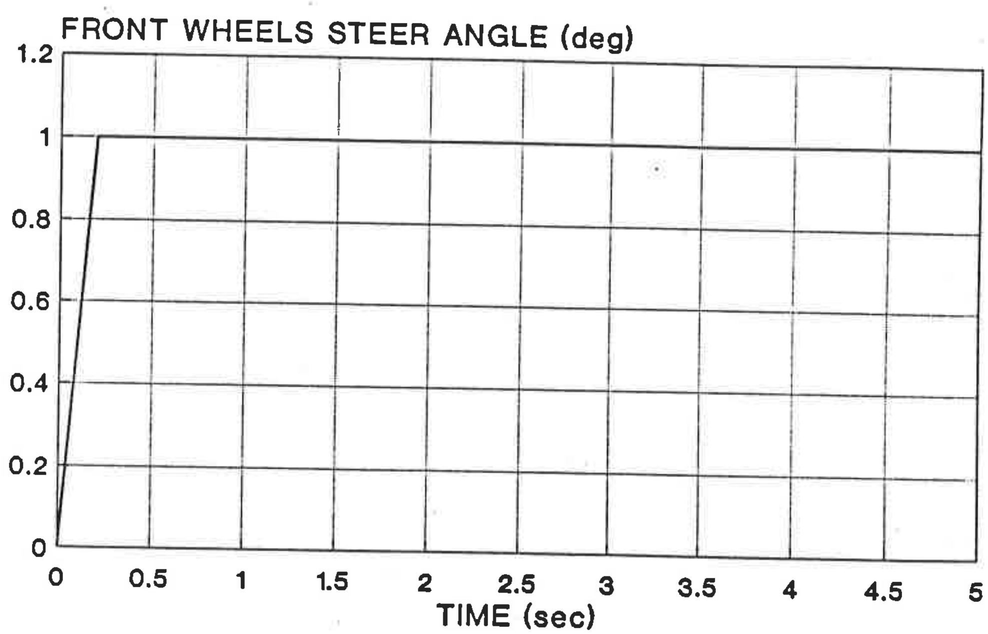

Fig. 6 - Ramp-step steering input at 90 km/h

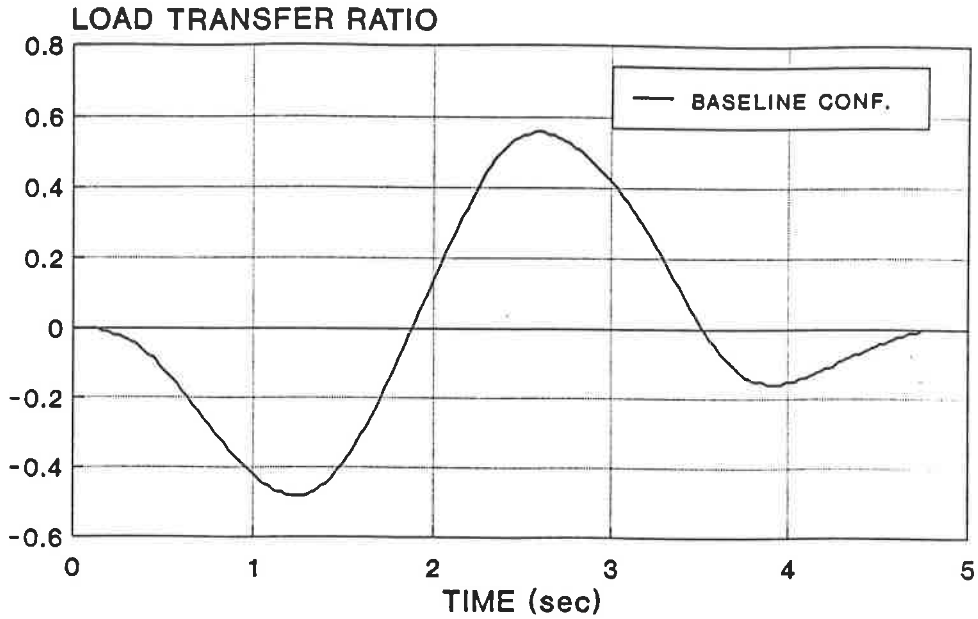

Fig. 7 -Load transfer during rapid lane-change manoeuvre (speed = 90 km/h)

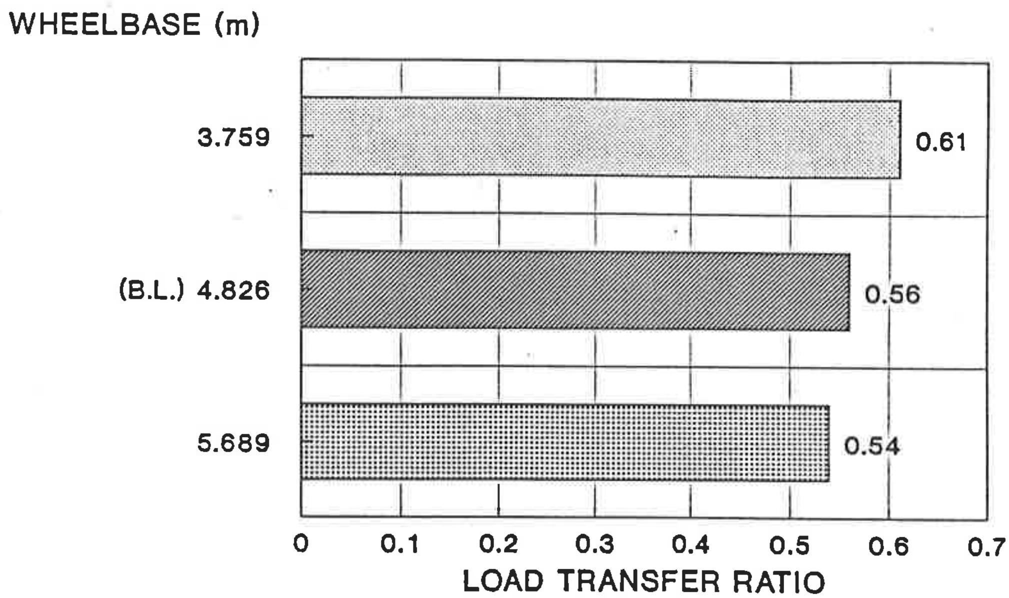

Fig. 8 - Influence of tractor wheelbase on load transfer ratio during rapid lane-change manoeuvre

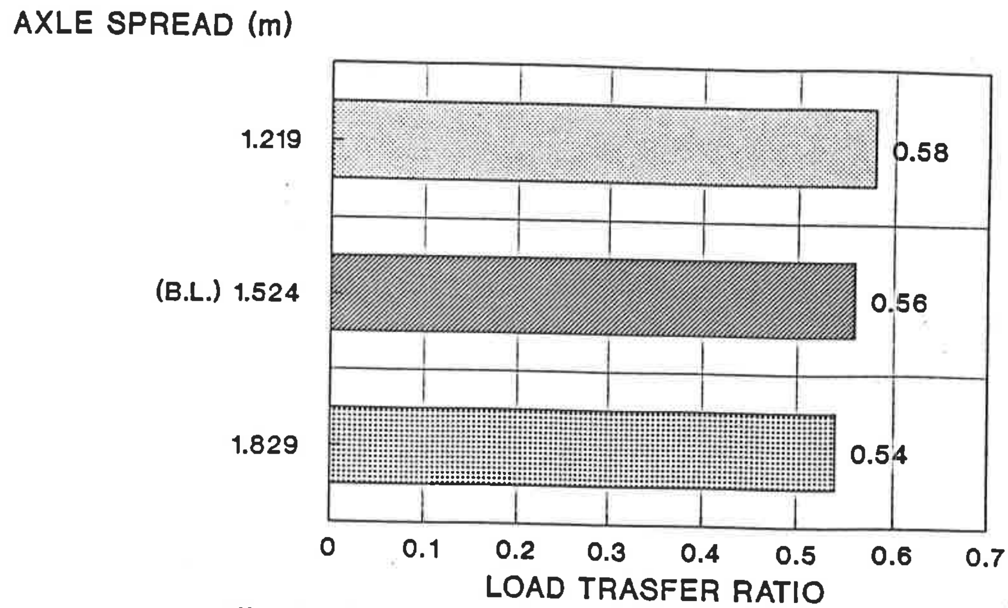

Fig. 9 - Influence of tractor's tandem axle spread on load transfer ratio during rapid lane-change manoeuvre

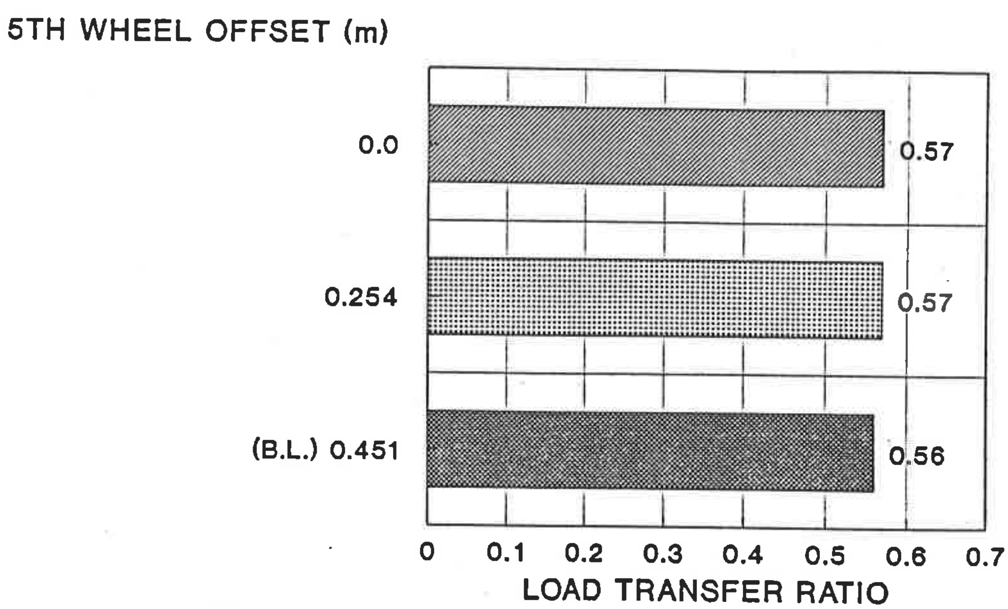

Fig. 10 - Influence of 5th wheel offset on load transfer ratio during rapid lane-change manoeuvre

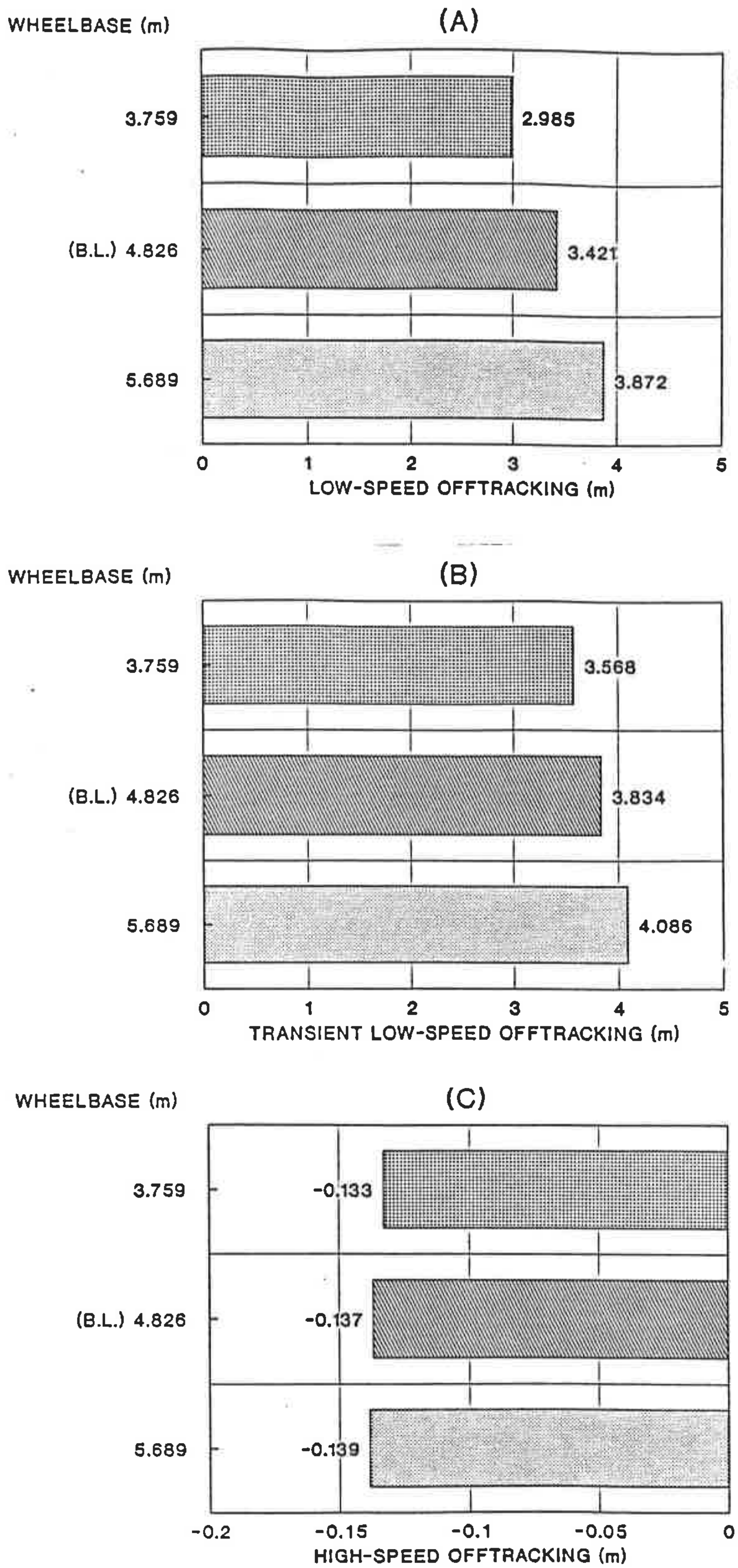

Fig. 11 - Influence of tractor wheelbase on offtracking

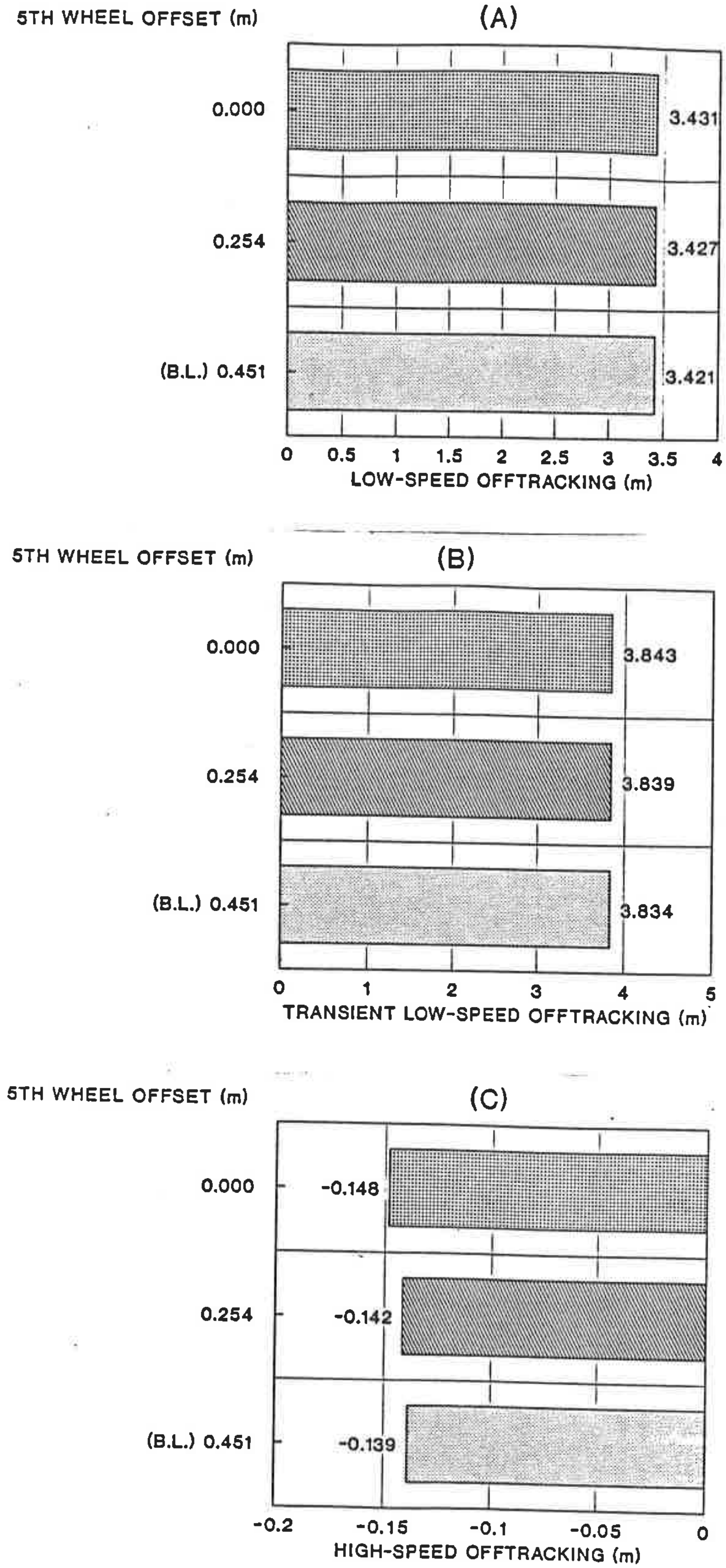

Fig. 12 - Influence of fifth-wheel offset on offtracking

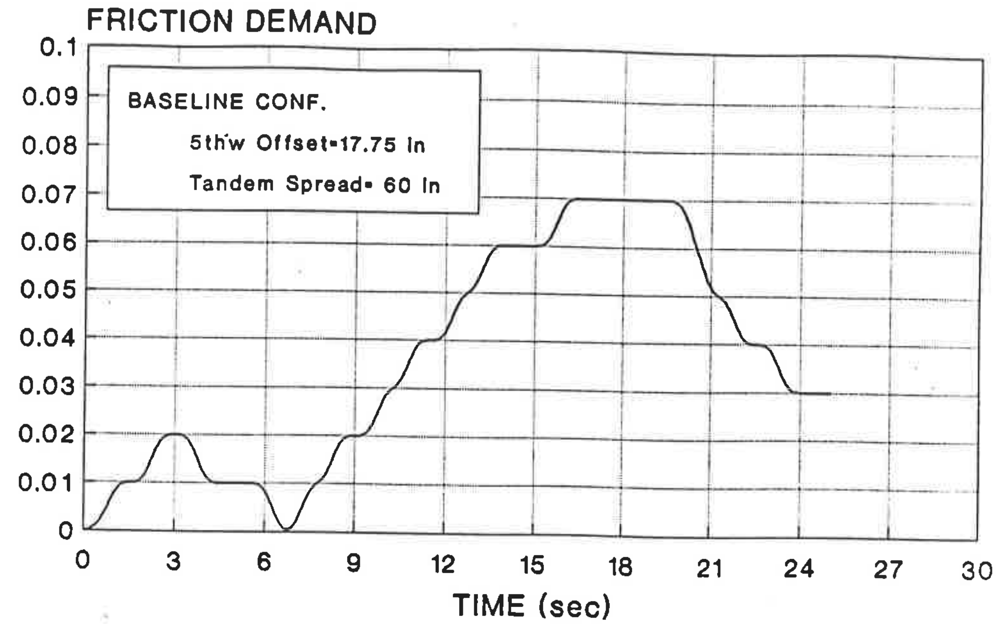

Fig. 13 - Friction demand at drive axles
during low-speed, tight turn manoeuvre

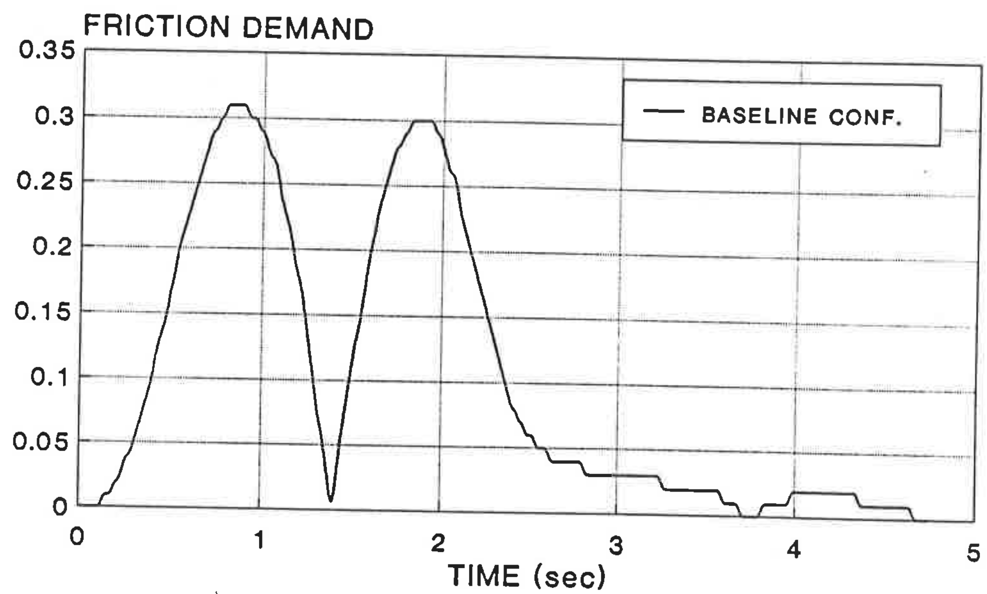

Fig. 14 - Friction demand at drive axles
during rapid lane change manoeuvre
(speed = 90 km/h)

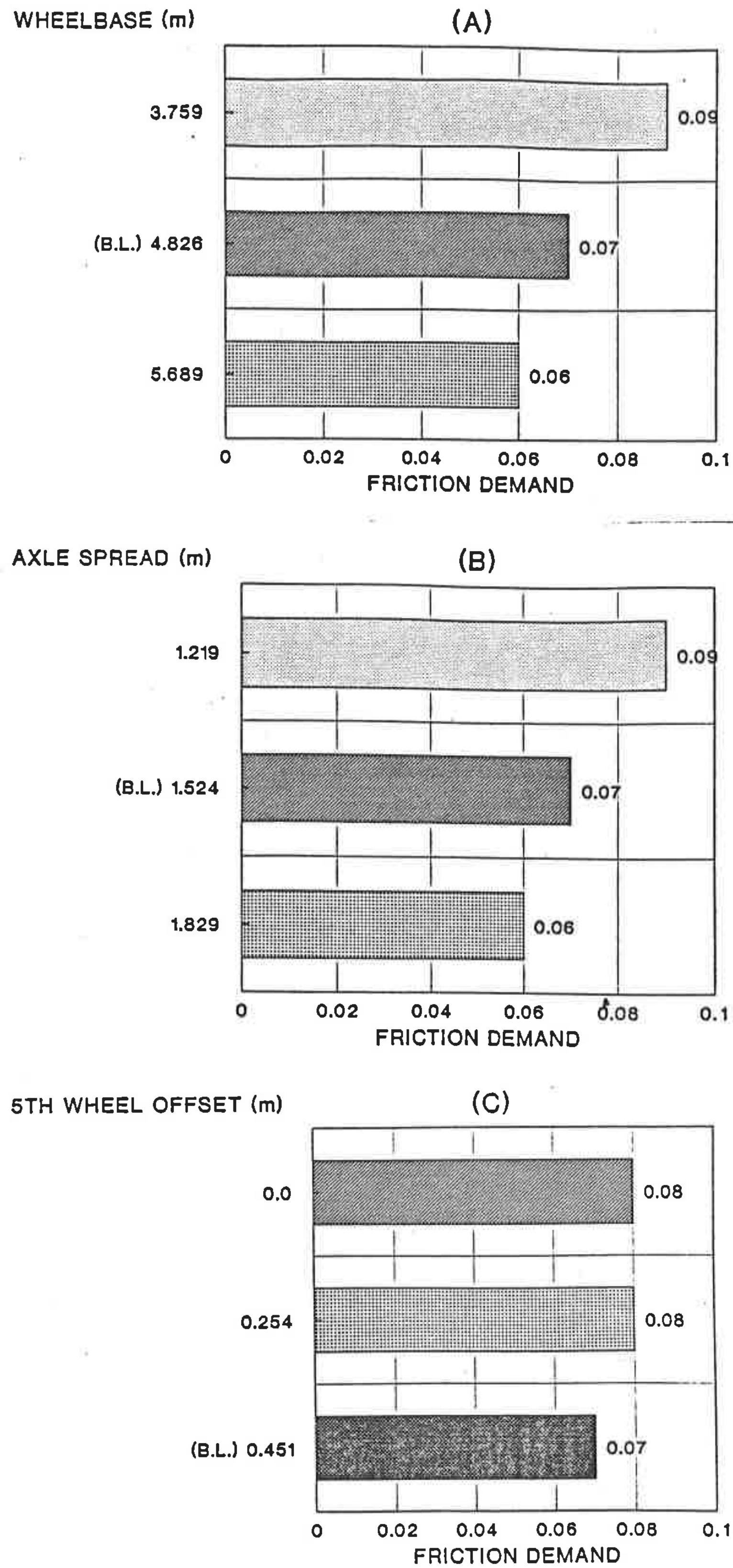

Fig. 15 - Influence of tractor parametes on friction demand during low-speed manoeuvre (5 km/h)

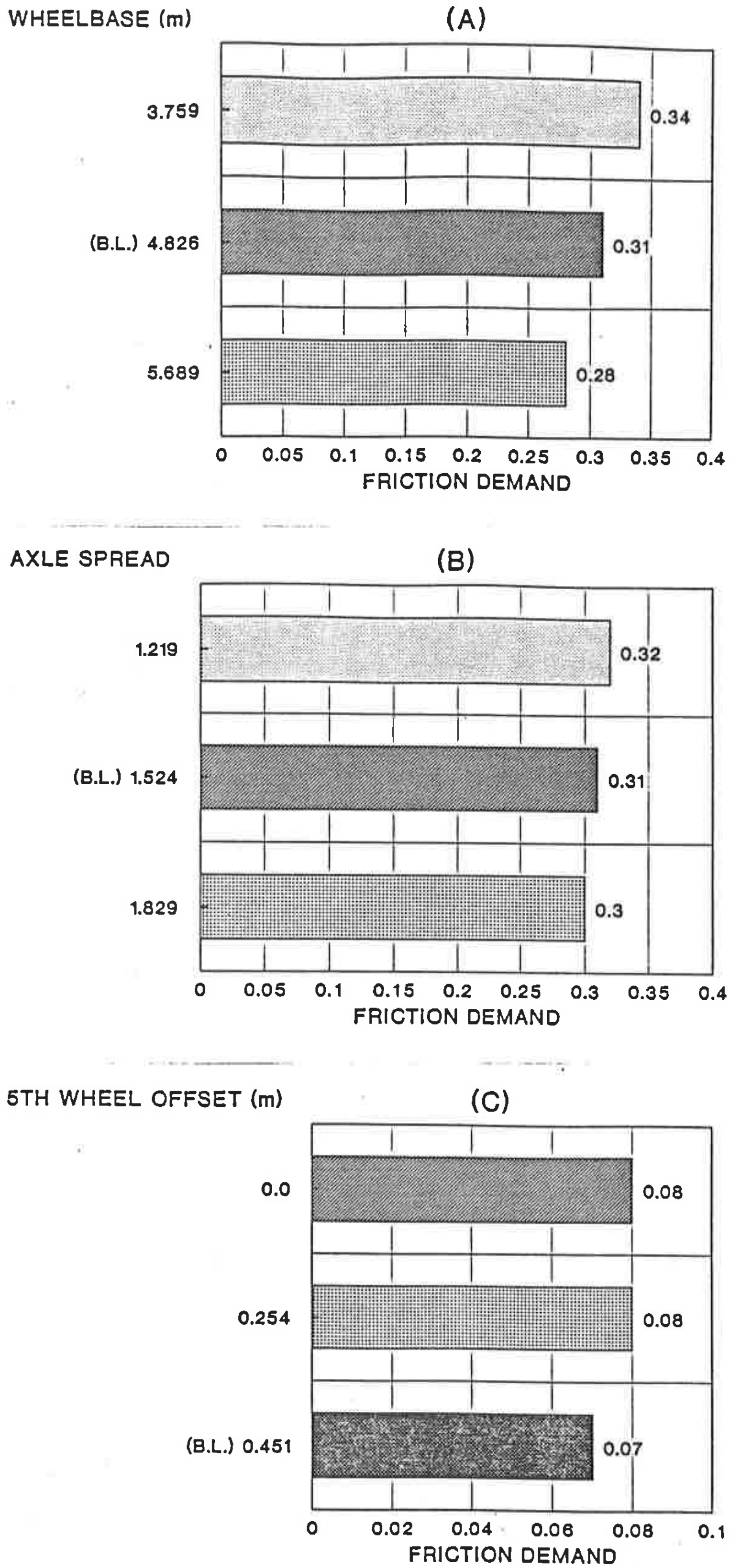

Fig. 16 - Influence of tractor parametes on friction demand during lane-change manoeuvre (90 km/h)

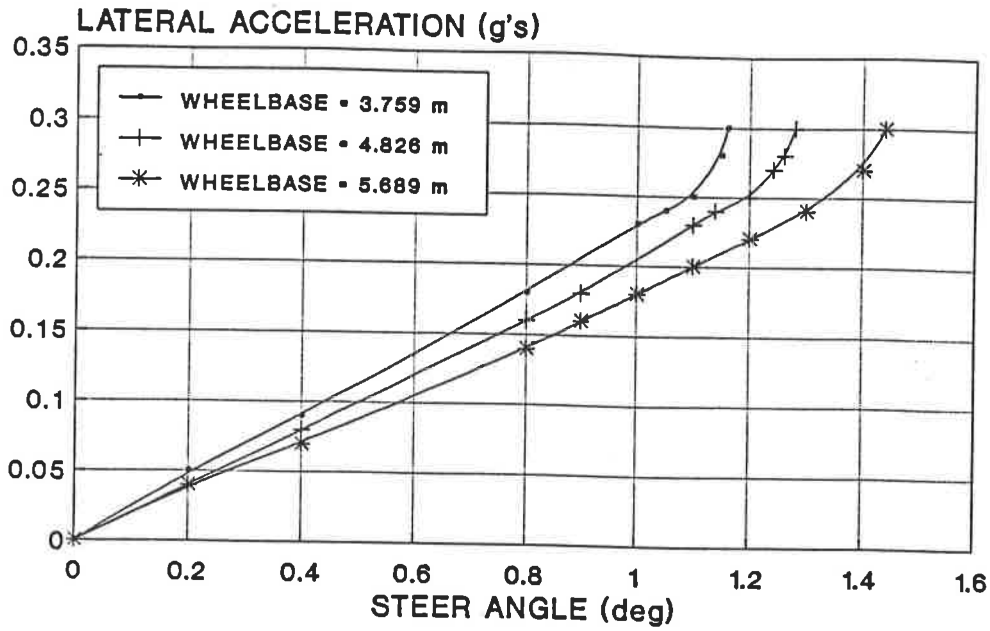

Fig. 17 - Influence of tractor wheelbase on steady-state lateral acceleration response at 90 km/h

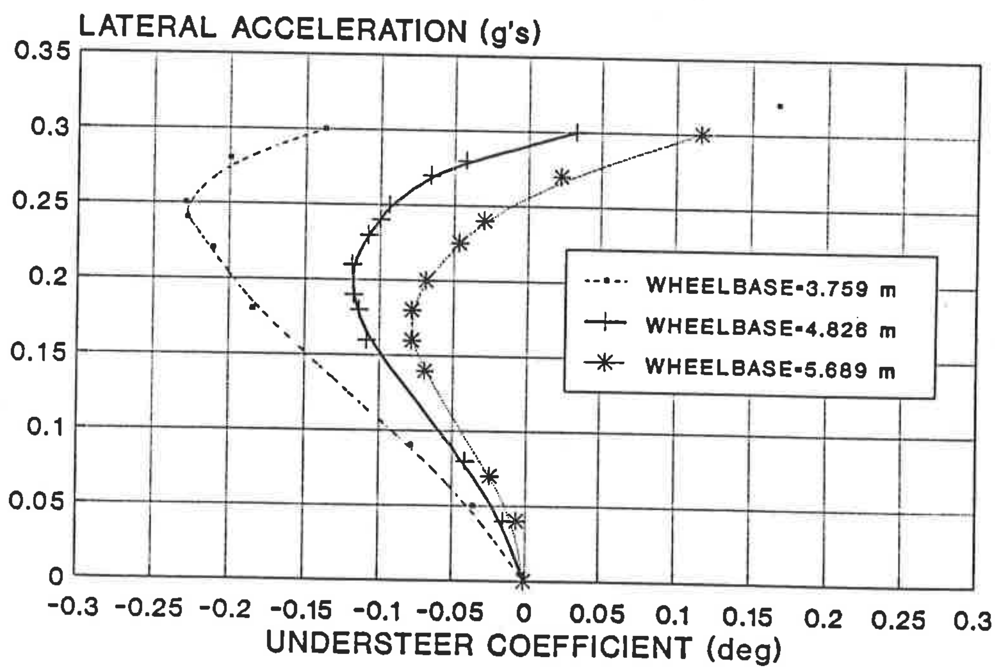

Fig. 18 -Influence of tractor wheelbase on handling performance at 90 km/h

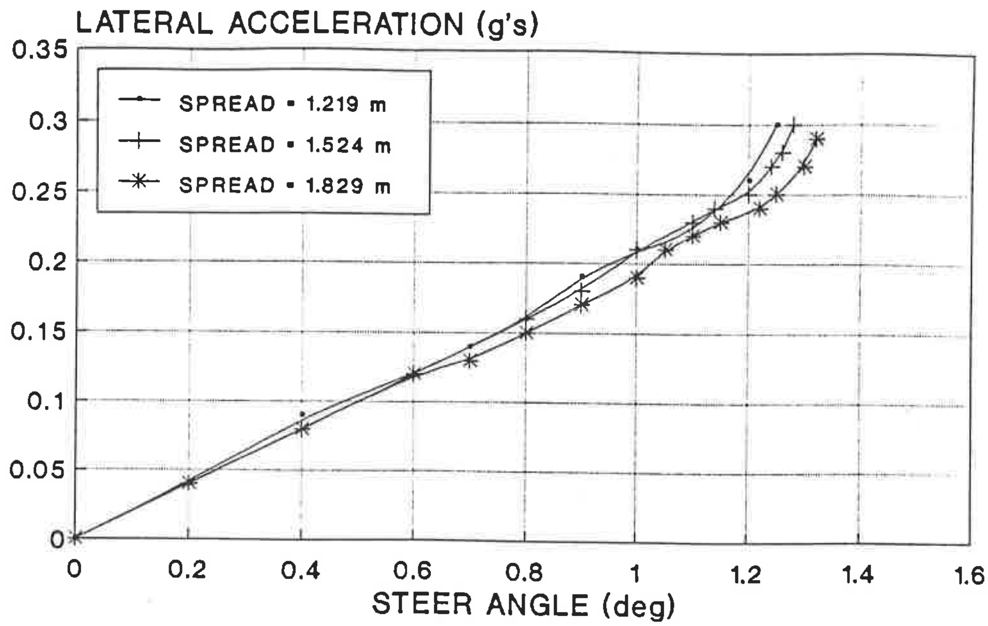

Fig. 19 - Influence of axle spread on steady-state lateral acceleration response at 90 km/h

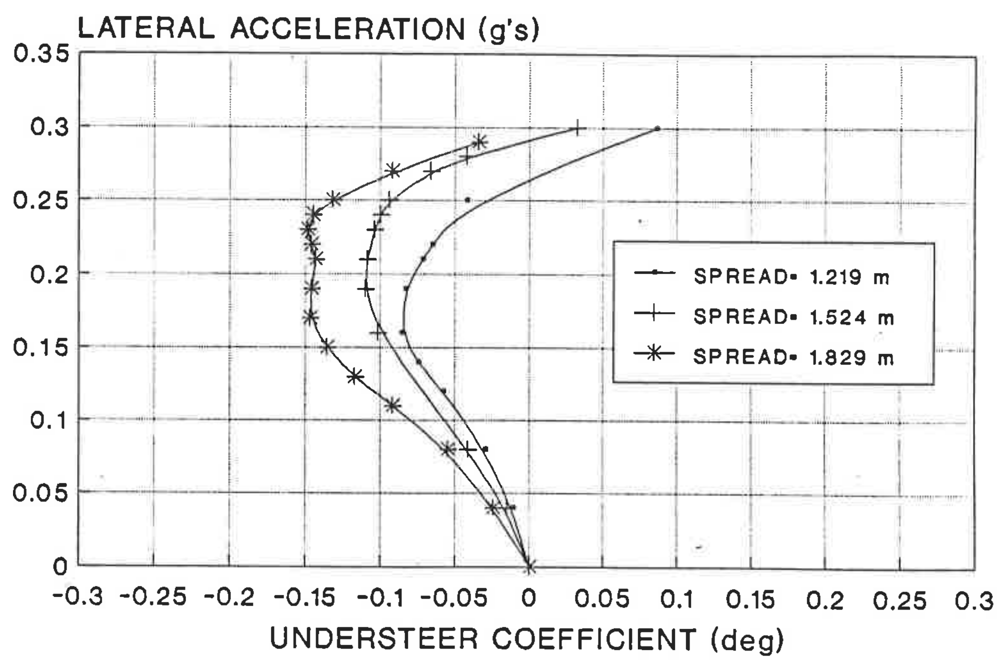

Fig. 20 - Influence of axle spread on handling performance at 90 km/h

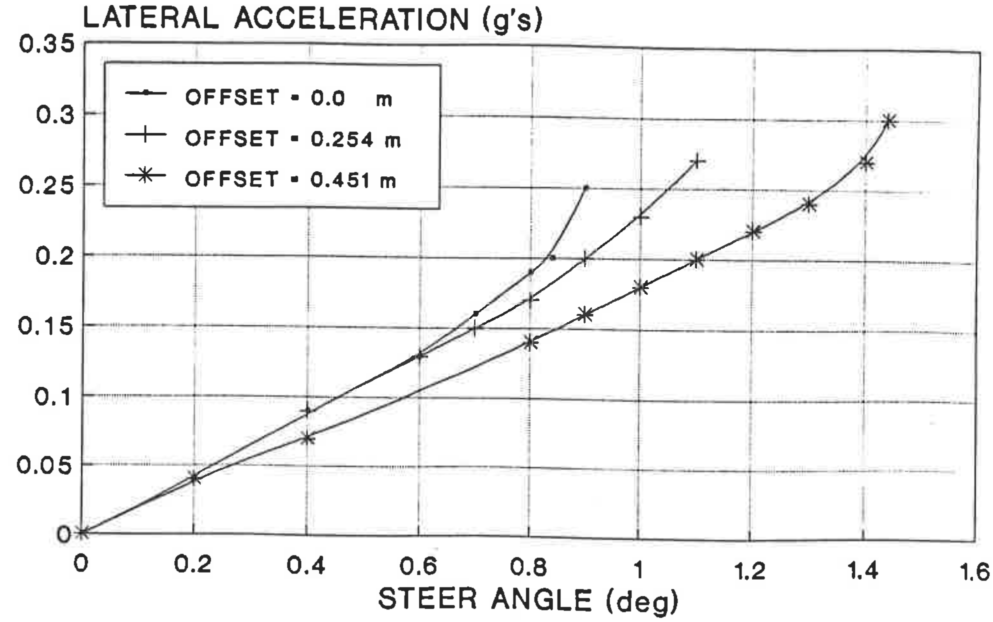

Fig. 21 - Influence of fifth-wheel offset on steady-state lateral acceleration at 90 km/h

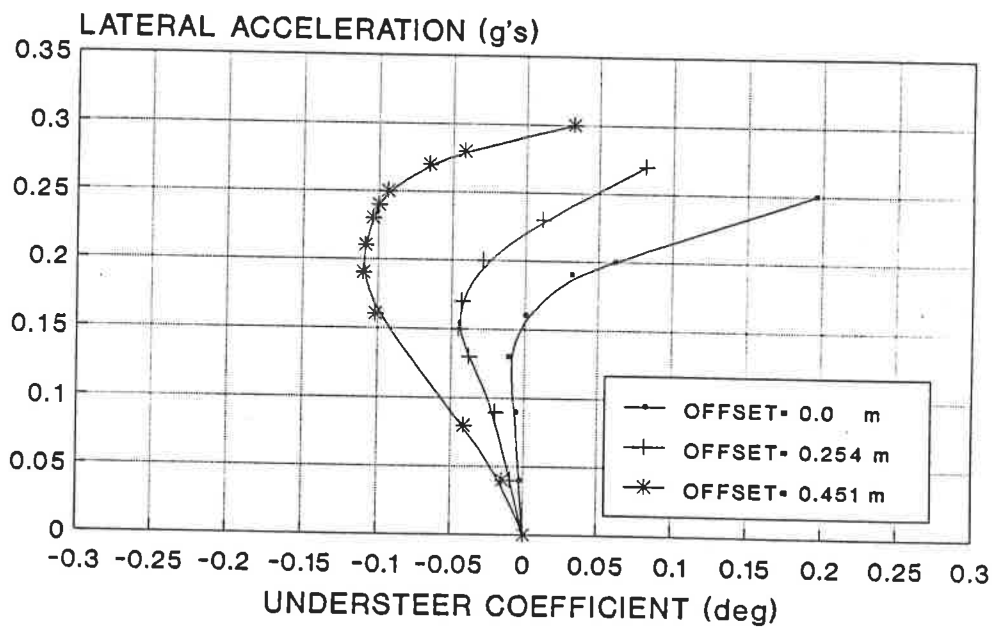

Fig. 22 - Influence of fifth-wheel offset on handling performance at 90 km/h