Application of handling and roll stability performance measures for determining a suitable tractor wheelbase

J. H. WOODROOFFE and M. EL-GINDY, Vehicle Dynamics Laboratory, National Research Council, Ottawa, Canada

This study stemmed from earlier work done by the authors to resolve a problem arising from widespread changes to vehicle length limits in Canada. Regulations had been introduced which separately limit the combined trailer length and overall vehicle length, with a resulting indirect but important effect on the length of the tractors and, in particular, their wheelbases. The influence of the tractor wheelbase on the vehicle's handling performance and roll stability, not having been examined before, needed to be studied. Doing so revealed a *further* need to examine the performance measures by which vehicles are judged.

In this paper a set of handling performance measures, including suggested criteria, are presented for assessing vehicle design variables. The measures put forward are suitable for use with a single computer program adapted for use on a PC. It is controlled by a post-processor that calculates each measure according to predetermined manoeuvres and yields a pass/fail conclusion.

The objective of this paper is to present and discuss two of these performance measures and to demonstrate their application in the determination of acceptable wheelbases for tractors used in B-Train doubles. The "acceptability" is shown to depend on the characteristics of individual vehicle componentry, in this case the tractor's tandem-axle suspension roll stiffness, thereby revealing that weights-and-dimension-based assessments are by no means adequate to determine if a vehicle can meet reasonable minimum dynamic performance standards.

NOMENCLATURE

Del, δ	Average front axle steering angle	deg
F_z	Vertical load at an axle's tires	lbs
F_y	Cornering force at an axle's tires	lbs
g	Gravitational acceleration	ft/s^2
I_x	Roll moment of inertia of tractor sprung mass	in-lbs-s^2
I_y	Pitch moment of inertia of tractor sprung mass	in-lbs-s^2
I_z	Yaw moment of inertia of tractor sprung mass	in-lbs-s^2
K_u	Understeer coefficient	deg/g
K_{cr}	Critical understeer coefficient	deg/g
L	Tractor reference wheelbase	in
OM	Steering frequency	rad/s
U	Forward speed	ft/s
W_s	Tractor sprung weight	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 trailer	deg

1. 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 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 is a continuation of a previous study (El-Gindy and Woodrooffe 1990) which was sponsored by le Ministère de transports du Québec, through the Roads and Transportation Association of Canada. Its purpose was to understand better the effect that tractor wheelbase variations have on heavy truck performance so that this may be reflected appropriately in future weight and dimension laws. It examined the net effect of tractor wheelbase on the dynamic performance -- namely, handling; static and dynamic roll stability; friction demand; and offtracking -- of a B-Train and tractor/semitrailer. It also presented a novel handling performance measure for assessing the handling behaviour of a vehicle. This performance measure is based on a vehicle handling diagram and will be referred to as the "three-point measure."

R.D. Ervin and Y. Guy (1986) presented an extensive study on the dynamics of articulated vehicles, conducted under the Canadian Heavy Vehicle Weights and Dimensions Study, that examined the influence of various vehicle parameters on vehicle performance during low- and high-speed path-follow manoeuvres. However, the study did not address the influence of varying a tractor wheelbase on the dynamic performance of the overall vehicle.

A study reported by Fancher et al. (1989) showed that future transportation technology will involve developing heavy commercial vehicles with measurable and predictable levels of performance in safety-related manoeuvres. The study concentrated on vehicles weighing more than 36t (80,000 pounds) and used the same evaluation methods used in the Canadian Weights and Dimensions Study. New performance targets were chosen based on accumulated research experience including knowledge gained from the examination of trucks involved in fatal accidents.

A recent study (El-Gindy and Woodrooffe 1991) presented a review of existing performance measures, discussed their application, and proposed modifications to improve their effectiveness. Some new performance measures were put forward and improvements to existing measures were presented as well. In addition, an example of a pass/fail criterion for each measure was suggested. To present and rationalize these performance measures and to demonstrate their application, five common commercial freight vehicle configurations were selected: tractor-semitrailer, A-Train, B-Train, C-Train, and truck/full-trailer.

The objective of the current paper is to demonstrate the use of two of these performance measures in the vehicle *design and regulatory* process whereby the designer alters various tractor parameters, such as its wheelbase and its trailing tandem-axle's suspension auxiliary roll stiffness to satisfy the dynamic performance criteria, while keeping entirely within the dimensional, weight, and axle load constraints of Canadian regulations. In this paper the use

Heavy vehicles and roads: technology, safety and policy. Thomas Telford, London, 1992.

of two safety related measures, namely handling and roll stability is demonstrated on a Canadian 8-axle B-Train double, where the vehicles are identical except for the value of the auxiliary roll stiffness of the tractor's trailing axle, and except for the tractor wheelbase which is used as a compensating variable needed to make the overall vehicle pass the performance measures.

2. COMPUTER SIMULATION MODEL AND POST-PROCESSOR

The task of evaluating the handling performance and the dynamic roll stability measures were supported by the constant speed Yaw/Roll Model developed by the University of Michigan Transportation Research Institute (UMTRI). The outputs from the Yaw/Roll model were evaluated with a post-processor developed specifically for this purpose.

2.1 Yaw/Roll Model

The UMTRI Yaw/Roll model (Mallikarjunarao and Segel 1981), was developed for the purpose of predicting the directional and roll response of single and multiple articulated vehicles engaged in steering manoeuvres which may approach the limits of stability. In the model, the forward speed of the lead unit is assumed to remain constant during a manoeuvre, and therefore the longitudinal motion of each articulated unit cannot vary. Each unit is treated as a rigid body with five degrees of freedom: lateral and vertical displacement, and yaw, roll, and pitch rotation. The axles are treated as beam axles that are free to roll and to deflect vertically with respect to the sprung mass of the vehicle. This simulation model is used in this study to compute the handling and rollover stability performance measures, as means of assessing the vehicle's dynamic behaviour.

2.2 Post-Processor

It is necessary but not sufficient, for the practitioner studying vehicle behaviour, to obtain numerical values from a simulation model. Also needed are criteria by which to judge the behaviour, and thresholds to indicate whether the behaviour is in a desirable range or not. The criteria and thresholds emerging from this study are embodied in a post-processor program that analyzes various performance measure parameters. The Yaw/Roll simulation program produces an output file, known as an ERD file, in a format developed at UMTRI (Sayers 1989). The ERD file contains all the important simulation output data, together with some data describing the vehicle and the manoeuvre.

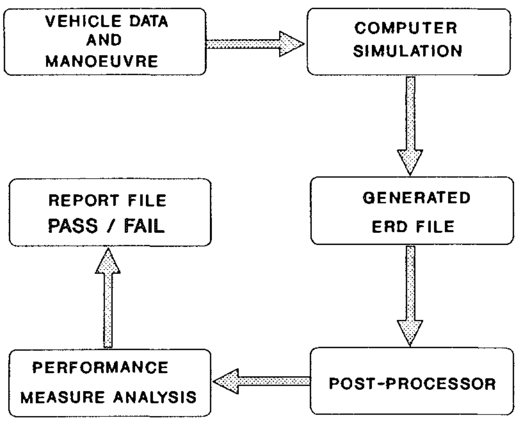

Figure 1. Method of approach

The post-processor identifies which manoeuvre was simulated and calls the relevant subroutine where calculations appropriate to that manoeuvre are carried out. The relevant channels in the ERD file are read, the analysis is performed, and the results are written to a summary file. An inspection file is produced to verify that the correct data had been used [Fig. 2]. The post-processor has been modified by NRC to deal with the new and the modified performance measures and their relevant manoeuvres.

3. VEHICLE CONFIGURATIONS

The 8-axle B-Train vehicle configuration used in this study is diagrammed in Figure 2. To generate curves showing the relationship between tractor wheelbase and vehicle performance, the tractor wheelbase is varied from a length of 3.76 m (148 in) (SHORT) to 5.69 m (224 in) (LONG). Two values of the tractor trailing tandem axle roll stiffness have been used to demonstrate an example of the influence of one design parameter on the determination of a tractor wheelbase; these values are 1920 N-m/deg (17000 lb-in/deg), referred as **B-Train #1**, and 9600 N-m/deg (85000 lb-in/deg), referred as **B-Train #2**.

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, has been considered. The changes in the basic tractor design parameters as a function of the tractor's wheelbase are as follows (Ervin and Guy 1986):

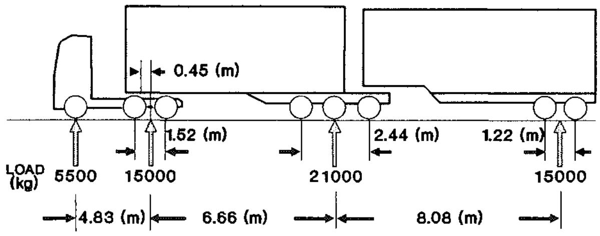

Figure 2. vehicle configuration

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

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

This formula assumes that the sprung weight of a baseline (190-inch (4.83 m) wheelbase) tandem-axle tractor is 11800 lbs (52500 N), implying a total tractor weight of 18000 lbs (80100 N), and that each additional 30 inches (0.76 m) of wheelbase corresponds to an additional 1000 lbs (4450 N) of sprung weight. Equation 1 shows that the tractor sprung weight increased linearly as the wheelbase increases.

Location of Centre of Gravity of Tractor's Sprung Mass: The longitudinal location of the sprung-mass centre of the baseline tractor (190-inch (4.86 m) wheelbase) is 55 inches (1.40 m) 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:

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

Equation 2 shows that as the wheelbase increases, the distances between the centre of gravity and all axles increase linearly.

Tractor Sprung Mass Moments of Inertia: The sprung-mass roll moment of inertia I_x (in-lbs-s^2) is determined from the tractor's sprung weight W_s, for each of the different wheelbases, assuming a constant value of 29 inches (0.74 m) for the radius of gyration of the sprung-mass, namely:

$$I_x = 2.178\ W_s. \quad (3)$$

The sprung-mass pitch and yaw moments of inertia, I_j (j = y, z), are 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. As the wheelbase increases, the roll moment of inertia slightly increases, while both pitch and yaw moment of inertia sharply increase. The height of the centre of the tractor sprung-mass is assumed to be a constant 44 inches (1.12 m) above ground level. The values of W_s, X, I_x, I_y, and I_z for short, 4.19 m (165 in), 4.83 m (190 in), and long wheelbase tractors are depicted in Table 1.

When the tractor wheelbase is varied in this study, the tandem-axle spread is held at 60 inches (1.52 m), and the location of the fifth-wheel is held at 17.75 inches (0.45 m) forward of the tandem-axle centreline.

As a result of the combined variations of the tractor parameters tied to wheelbase changes, the tractor's axle loads will also vary. The front (steering) axle load remains practically constant as the wheelbase increases, while the tandem-axle load increases significantly as the wheelbase increases. Consequently, the lateral force and aligning torque characteristics of the front axle's tires will not change, while the tandem-axle tire characteristics will significantly vary as the wheelbase changes. These changes in tire characteristics coupled with changes to other tractor parameters will have a significant effect on the vehicle's directional dynamics.

Throughout this study, it is assumed the trailers are fully loaded with a homogeneous-density cargo and their design parameters are not varied. Both of the B-Trains being examined in this study are fitted with identical tires. Variations of the tractor wheelbase resulted in the following vehicle configurations:

1. **B-Train #1**: This configuration has a tractor with wheelbase of 3.76 m (148 in), 4.19 m (165 in), 4.83 m (190 in), and 5.69 m (224 in); a fifth-wheel offset equal to 0.45 m (17.75 in); a tandem-axle spread of 1.52 m (60 in); and a tractor leading and trailing tandem axle roll stiffness of 1920 N-m/deg (17000 lb-in/deg) and 9600 N-m/deg (85000 lb-in/deg), respectively.
2. **B-Train #2**: This configuration has a tractor with 3.76 m (148 in), 4.19 m (165 in), 4.83 m (190 in), and 5.69 m (224 in) wheelbase, a fifth-wheel offset equal to 0.45 m (17.75 in), a tandem-axle spread of 1.52 m (60 in), and a tractor leading and trailing tandem axle roll stiffness of 1920 N-m/deg (17000 lb-in/deg) each.

Table 1. Basic tractor design parameters.

	Tractor Wheelbase			
Parameter	Short 148 (in) 3.76 (m)	165 (in) 4.19 (m)	190 (in) 4.83 (m)	Long 224 (in) 5.69 (m)
W_s (lbs) (000's)	10.4	11.0	11.8	12.9
X(in)	34	42.5	55	72
I_x (in-lbs-s^2) (000's)	22.7	24.4	25.7	28.2
I_y (in-lbs-s^2) (000's)	75.1	106.7	173.2	289.5
I_z (in-lbs-s^2) (000's)	75.1	106.7	173.2	289.5

4. PERFORMANCE MEASURES

As a result of previous studies (El-Gindy and Woodrooffe, 1990 and 1991) conducted by the Vehicle Dynamics Laboratory of the National Research Council of Canada, some modifications to the Heavy Vehicle Weights and Dimensions Study's performance measures have been considered. These changes were necessary to formulate measures suitable for evaluating the influence of tractor parameter variations on the dynamic performance of commercial vehicles. These studies also recommended a comprehensive review of all the existing performance measures to improve technical harmonization of commercial vehicle assessment. The measures that emerged from the general review are as follows:

1. Handling performance measure ("three-point measure")
2. Roll stability measures, including:
   (a) Load transfer ratio (LTR);
   (b) Rearward amplification (RWA);
   (c) Static rollover threshold (SRT).
3. Yaw damping measure
4. Friction demand (FD) measures, including:
   (a) Low-speed friction demand (LFD);
   (b) High-speed friction demand (HFD).
5. Lateral friction utilization measures (LFU), including:
   (a) Low-speed lateral friction utilization (LLFU);
   (b) High-speed lateral friction utilization (HLFU).
6. Braking performance

Only two of these performance measures, handling and roll stability, will be used in this paper as an example of how the remainder can be applied to vehicle design or regulation. The handling performance and the roll stability are chosen because of their relative importance, but for thorough design purposes, *all* of the performance measures should be applied to insure adequate dynamic performance of a given vehicle at both low and high speeds.

5.0 HANDLING PERFORMANCE MEASURE

El-Gindy and Woodrooffe proposed (1990 and 1991) some modifications to the definition and method of evaluation of the current performance measures established during the Canadian Weights and Dimensions Study. Among the principal changes was the incorporation of a handling performance measure referred to as the "three-point measure", which had emerged previously in the study on tractor wheelbase variations (El-Gindy and Woodrooffe, 1990). This measure replaced the handling measure used in the Weights and Dimensions Study because the latter was found to have several shortcomings, namely:

1. The previous handling performance measure included steering system compliance in the calculation of the understeer coefficient. This would require knowledge of the mechanical characteristics of the steering system between the driver and the steering axle of the vehicle, including the steering wheel ratio and the damping and elasticity characteristics of the steering system of the particular vehicle. The vehicle handling characteristic therefore would include the effects of a particular steering design. While this certainly is a requirement for the proper design of a particular vehicle, it detracts from the regulatory requirements of a more general systems approach to assessing the performance aspects of the overall vehicle's design. The proposed three-point measure concentrates on the understeer and oversteer characteristics attributable to the general vehicle configuration *per se*.
2. The previous criterion for assessment of the understeer coefficient was evaluated from the vehicle's handling diagram at a single lateral acceleration level of 0.25 g's. However, the complex but meaningful nature of the handling curve cannot be adequately judged from a single point on the curve. The handling diagram provides important information about the stability and control characteristics of a vehicle over the entire operational range of lateral accelerations. The three-point measure analyzes the handling diagram of a vehicle in a more complete way. It is constructed using $\{(L/R - \delta), A_y\}$, where δ is the front axle steer angle. This form of handling diagram excludes steering system compliances where representing the general vehicle system. The three-point measure is shown in [Fig. 3] and is defined as follows:

a) First point. The understeer coefficient, K_u, at 0.15 g's, should be held within a range from 0.5 deg/g (sensitivity boundary) to 2.0 deg/g (steerability boundary).

b) Second point. The level of lateral acceleration at which the vehicle transforms from understeer to oversteer should not be less than 0.2 g's.

c) Third point. The understeer coefficient, K_u, evaluated at a lateral acceleration of 0.3 g's, must be higher than the critical understeer coefficient, K_{ucr}. The safety margin is expressed by the oversteering sensitivity, as will be explained later. The critical understeer coefficient is defined as $-Lg/U^2$, where U is the vehicle speed (100 km/h), and g is the acceleration due to gravity (9.81 m/s^2).

Based on the definition of the handling diagram used in this study, the understeer coefficient, K_u, at a lateral acceleration level, A_y, can be obtained as follows:

$$K_u = d(\delta - L/R)/d(A_y) \qquad (6)$$

The understeer coefficient, K_u, is evaluated, theoretically, from a ramp-steer manoeuvre at a vehicle speed of 100 km/h. A ramp-steer rate of .03 deg/s at the front axle is used.

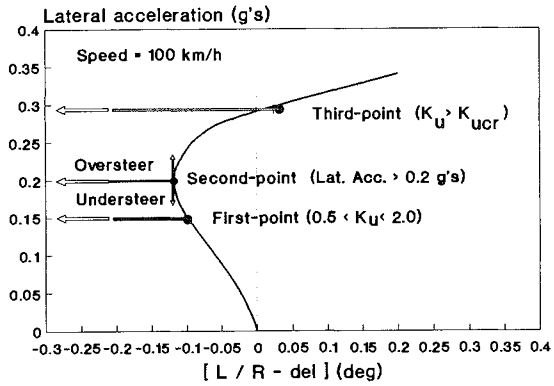

Figure 3. Representation of the NRC three-point handling performance measure

In addition to the "three-point measure" there are two measures designed to act as a "flag" if the vehicle is highly sensitive. A discussion of them is beyond the space limitations of this paper.

6.0 ROLL STABILITY MEASURES

The evaluation of roll stability includes the following measures:

a) Load Transfer Ratio: is defined as the ratio of the absolute value of the difference between the sum of right wheel loads and the sum of the left wheel loads, to the sum of all the wheel loads. For vehicles with trailer units de-coupled in roll, such as the second trailer of an A-Train, load transfer ratio calculations apply only within the independent units. On roll-coupled vehicles such as the B- or C-Train combinations, all vehicle units are included in the calculation. In all cases the front steering axle is excluded from the calculations because of its relatively high roll compliance. The LTR can be calculated as follows:

$$LTR = \sum ABS(F_{zr} - F_{zl})/(F_{zr} + F_{zl}) \qquad (2)$$

where F_{zl} and F_{zr} are the left and right side vertical loads at a given axle.

When assessing the effect of varying a tractor design parameter, the load transfer ratio is evaluated (El-Gindy et al. 1991) during an open-loop rapid-steering lane-change manoeuvre of 1.0 degree steering wheel amplitude and a period of 3.0 seconds (resulting in a steering-change frequency of 3.14 rad/s). The vehicle speed is 100 km/h. This is a change from previous studies that employ a path-change (closed-loop) manoeuvre. For the sake of comparison with the results published in the Canadian Weights and Dimensions Study, the load transfer ratio is also calculated during a path-change (closed-loop) manoeuvre of 0.15 g lateral acceleration with a time period of 3.0 s at 100 km/h. In both manoeuvres, the recommended target value is 0.6, above which the vehicle is considered to be unsafe from the standpoint of dynamic roll stability and it would fail to meet this requirement.

b) Rearward Amplification: the rearward amplification ratio is a frequency-dependent measure, defined as the ratio of the peak (positive or negative) lateral acceleration at the centre of gravity of the rearmost trailer to that at the centre of gravity of the lead unit (tractor or straight truck). Accordingly, it should be evaluated using the frequency response method to yield the critical steering frequency at which the peak rearward amplification ratio occurs. The method of evaluation can be summarized as follows: "The rearward amplification ratio of a vehicle should be obtained over a wide range of steering input frequency. The sinusoidal steering input has a 1 degree amplitude and a frequency range from 0.0 to 10.0 rad/s". The simplified yaw plane models developed by UMTRI were shown to provide valid results (Wong and El-Gindy 1985) for such an evaluation. For the sake of comparison, the rearward amplification ratio in this study is calculated during several steering lane-change (open-loop) manoeuvres, of 1.0 degrees with periods of 2.0, 2.5, and 3.0 seconds resulting in a steering frequencies of 3.14, 2.51, and 2.1 rad/s, respectively. The target value is chosen to be 2.2 at any of these frequencies, above which the vehicle would fail to meet this requirement.

c) Static Rollover Threshold: the static rollover threshold is defined in this study as the maximum lateral acceleration level in g's beyond which static rollover of a vehicle occurs. This is a modification of the traditional definition which assumes that evaluation will be done in steady turn; however using a tilt table to predict the rollover threshold is preferable (Preston-Thomas and El-Gindy, 1992). The static rollover threshold values can accurately determined using the UMTRI Static Roll Model (Mallikarjunarao, et al. 1982). The recommended minimum acceptable limit in Canada is 0.4 g's. The authors would recommend use of the Static Roll Model, where possible, instead of the Yaw Roll Model, because of its greater simplicity and ease of use.

7.0 DETERMINATION OF TRACTOR WHEELBASE

Based on application of both handling and roll stability measures and their target values, the process for determining an acceptable tractor wheelbase for B-Train #1 and B-Train #2 is described below.

7.1 Handling

An application of the proposed "three-point" handling performance measure for the B-Train double to the assessment of varied tractor wheelbases is shown in Figures 4a, 4b, and 4c. From these diagrams the following analysis can be made:

(a) Figure 4a (first-point measure) plots the understeer coefficient at a lateral acceleration of 0.15 g's and indicates a suggested acceptable range. For B-Train #1, as the tractor wheelbase increases up to 5.30 m the understeer coefficient reaches the lowest acceptable limit of 0.5 deg/g, while for B-Train #2, this limit is reached at a wheelbase of 5.60 m. Based on the first-point criterion, these two wheelbases are the maximum wheelbase limits for each vehicle.

(b) Figure 4b (second-point measure) plots the transition acceleration level as a function of tractor wheelbase. The minimum transition acceleration level is recommended as 0.2 g's. Note that the second-point measure is *very* sensitive to the change in the auxiliary roll stiffness of the tandem trailing axle, as can be seen from the markedly different shapes of the lines for B-Train #1 and B-Train #2. The transition acceleration of the B-Train #1 is highly affected by the increased lateral load transfer at the tractor tandem axles due to the increased roll stiffness which in turn reduces the effective cornering stiffness at these axles, compared to B-Train #2 which has a lower tractor's tandem-axle roll stiffness. The tractor wheelbase of B-Train#1 therefore must not exceed 5.02 m (197 in) or else the vehicle will exhibit oversteering below 0.2 g's, and would be unable to pass this measure. B-Train #2 exhibits no practical tractor wheelbase limitation to pass this measure.

(c) Figure 4c (third-point measure) plots the understeer coefficient at 0.3 g's and the critical understeer coefficient at 100 km/h as a function of the tractor wheelbase. The intersection point between them represents the stability boundary. Where the understeer coefficient is higher (closer to zero) than the critical understeer coefficient (both negative values), the vehicle is stable. Where the reverse applies, the vehicle is unstable. This figure therefore establishes that B-Train #1 and #2 will pass this measure across the entire range of wheelbases examined in this paper. Note however, that B-Train #1 has understeering coefficients closer to the critical values due to the reasons explained in point (b). In conclusion, this measure dose not impose a wheelbase limitation for either of these two B-Trains.

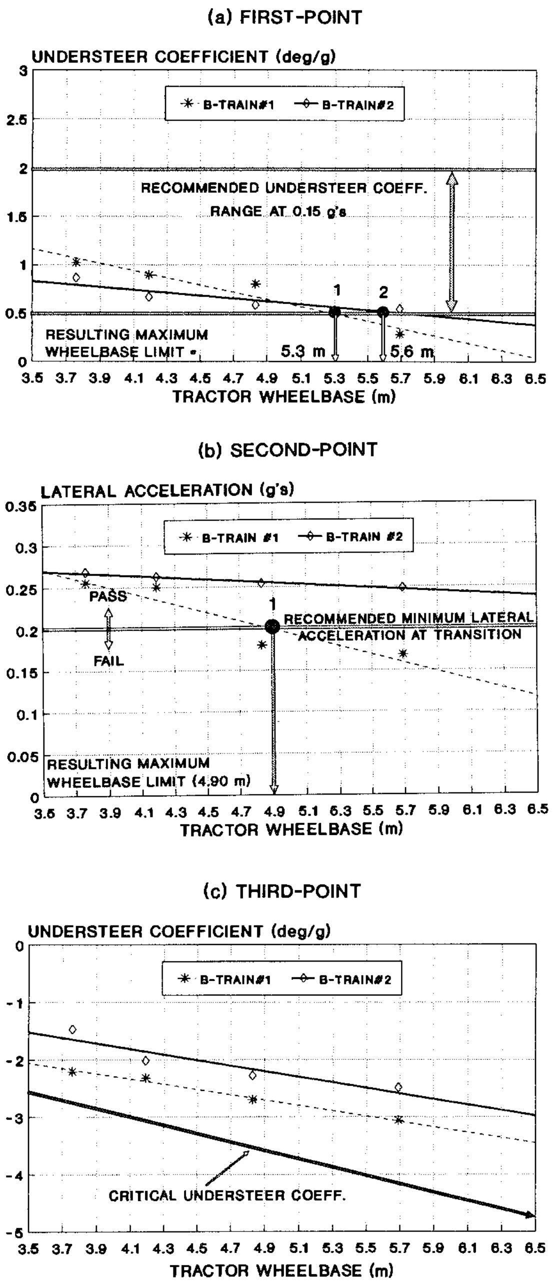

Figure 4. Determination of the tractor wheelbase based on the "three-point" measures

7.2 Roll Stability Measures

a) Load Transfer Ratio, LTR: The variation of the LTR as a function of the tractor wheelbase, evaluated using the rapid-steering lane-change manoeuvre for both vehicles, is shown in Figure 5. The short wheelbase tractor (3.76 m) exhibits the highest peak LTR, while the long wheelbase tractor (5.69 m) has the lowest. In general, lengthening the tractor wheelbase improves dynamic rollover stability. For B-Train #1 increasing the equivalent roll stiffness of the tractor tandem-axle reduces the dynamic load transfer ratio at the tractor's steering axle; however it increases the load transfer ratio at the vehicle's other axles. Based on the load transfer ratio definition, one can see that this degrades the dynamic rollover stability (i.e., by increasing the LTR). Note that in the final analysis, both B-Trains retained an LTR of less than the recommended maximum of 0.6, meaning that both vehicles pass this measure at all wheelbases.

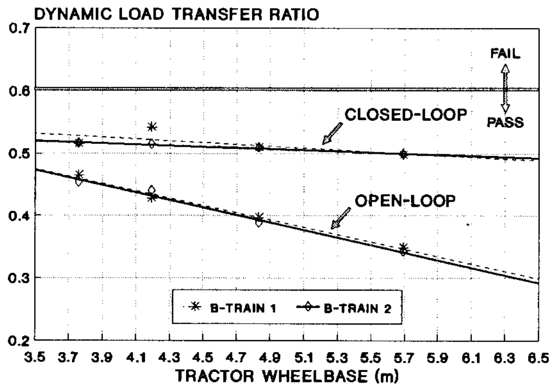

Figure 5. Determination of tractor wheelbase based on load transfer ratio measure

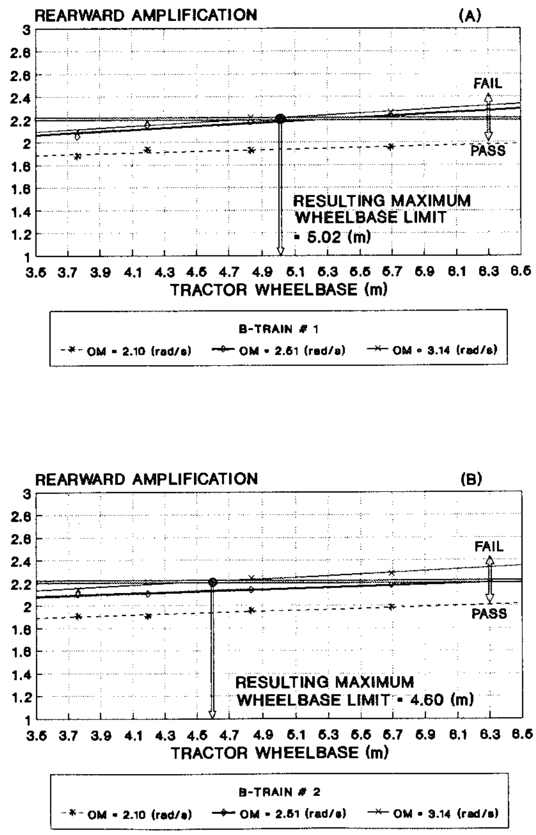

Figure 6. Determination of tractor wheelbase based on rearward amplification ratio measure

b) Rearward Amplification, RWA: The variation of the RWA for both vehicles is shown in Figure 6a and 6b as a function of the tractor wheelbase, using the rapid-steering lane-change manoeuvre at three steering frequencies, OM. In general, lengthening the tractor wheelbase causes the RWA to increase. The upper limit of RWA during this manoeuvre is recommended at 2.2 (El-Gindy 1992), above which the vehicle would be judged to fail the measure. Based on this performance target value, B-Train #1 and #2 will pass if their tractor wheelbases are no longer than 5.02 m and 4.60 m, respectively. See Figure 6.

c) Static Rollover Threshold, SRT: The static rollover thresholds, predicted by the Yaw/Roll Model during a quasi-steady manoeuvre at 100 km/h, are shown in Figure 7 as a function of the tractor wheelbase for the two B-Trains. From Figure 7 it can be seen that if the rollover threshold limit is 0.4 g's as recommended for Canadian usage, the tractor wheelbase should not be less than 3.78 on B-Train #1, but there is no upper limitation on the tractor wheelbase for B-Train #2.

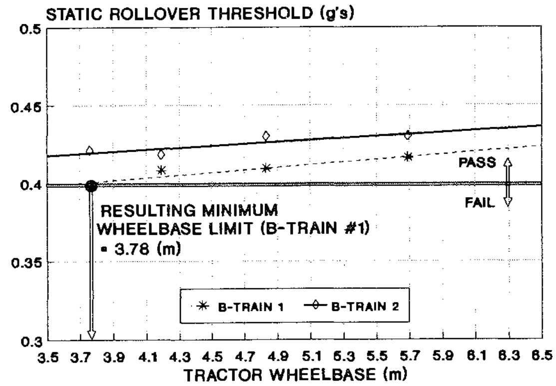

Figure 7 - Static rollover threshold measure

Once the handling (three-point) and the dynamic and static roll stability measures have been applied to establish the tractor wheelbase, the other performance measures (namely, low- and high-speed offtracking, friction demand and utilization, yaw damping, and braking performance) should be reviewed to see whether they further constrain the tractor wheelbases. Table 2 shows the recommended tractor wheelbase ranges for both B-Trains. If kept to these ranges, the vehicles will exhibit satisfactory stability and controllability, and dynamic roll stability.

Table 2. Recommended wheelbase.

PERFORMANCE MEASURES	B-TRAIN #1 TRACTOR WHEELBASE		B-TRAIN #2 TRACTOR WHEELBASE	
	MINIMUM	MAXIMUM	MINIMUM	MAXIMUM
HANDLING (THREE-POINT)	4.19 (m)*	4.90 (m)*	3.76 (m)*	5.60 (m)
LOAD TRANSFER RATIO	3.76 (m)	5.69 (m)	3.76 (m)	5.69 (m)
REARWARD AMPLIFICATION	3.76 (m)	5.02 (m)	3.76 (m)	4.60 (m)*
STATIC ROLLOVER THRESHOLD	3.78 (m)	5.69 (m)	3.76 (m)	5.69 (m)
RECOMMENDED WHEELBASE RANGE	$4.19 \leq$ WB ≤ 4.90 (m)		$3.76 \leq$ WB ≤ 4.60 (m)	

* Limiting value

CONCLUSION

The performance sensitivity of two eight-axle B-Train doubles has been examined as a function of variations in the tractor wheelbase, and tractor trailing tandem-axle roll stiffness. The roll stability and the handling performance measures have been used to select an acceptable tractor wheelbase range for each vehicle configuration.

The results of this study indicate that it is possible to select or modify a vehicle design parameter, such as tractor wheelbase, using vehicle handling and roll stability criteria. The performance measures of the type described in this paper appear to be suitable for use in the design of commercial vehicles and as a basis for the development of size and weight regulations. The technology now exists to enable practitioners in vehicle design and regulation to assess the performance of vehicles or classes of vehicles.

REFERENCES

Ervin, R.D. and Guy, Y., 1986. "The Influence of Weights and Dimensions on the Stability Control of Heavy Trucks in Canada - Part 2.", Technical Report, Vol. 2, Vehicle Weights and Dimensions Study, Roads and Transportation Association of Canada (RTAC), July.

El-Gindy, M. and Woodrooffe, J.H., 1990. "The Effect of Tractor Parameter Variations on the Dynamic Performance of a B-Train Double.", ASME Symposium on Transportation Systems, Winter Annual Meeting of the American Society of Mechanical Engineers, Dallas, Texas, ADM-Vol. 108, November 25-30.

El-Gindy M. and Woodrooffe, J.H.F., 1991. "Influences of Tractor Wheelbase, Tandem-Axle Spread and Fifth-Wheel Offset on Commercial Vehicle Dynamics", Technical Report, No. TR-VDL-003, Vehicle Dynamics Laboratory, National Research Council of Canada, Ottawa.

El-Gindy, M. and Woodrooffe, J.H.F., 1992. "Evaluation of the Constant Speed and Braking Performance of Heavy Commercial Vehicles", Technical Report, Vehicle Dynamics Laboratory, National Research Council of Canada, Ottawa (to be published).

El-Gindy, M., Woodrooffe, J.H.F., and White D. W., 1991. "Evaluation of the Dynamic Performance of Heavy Commercial Vehicles", ASME Symposium on Advanced Automotive Technologies, Winter Annual Meeting of the American Society of Mechanical Engineering, Atlanta, Georgia, DE-Vol. 40, December 1-6.

Fancher, P.S., Jr. and Mathew, A., 1989. "Safety Implications of Trucks Designed to Weigh Over 80,000 Pounds", SAE Paper No. 891632.

Mallikarjunarao, C., and Segel L. 1981., "A Study of the Directional and Roll Dynamics of Multiple-Articulated Vehicles", Proceedings of the Seventh IAVSD Symposium, pp 81-96, September.

Sayers, M., 1989. "Fortran ERD File Toolbox for the Macintosh", University of Michigan Transportation Research Institute. (UMTRI).

Wong, J.Y. and El-Gindy, M., 1985. "Computer Simulation of Heavy Vehicle Dynamic Behaviour: User's Guide to the UMTRI Models", Technical Report No. 3, Vehicle Weights and Dimensions Study, Roads and Transportation Association of Canada (RTAC), June.