7th International Symposium on Heavy Vehicle Weights & Dimensions
Delft, The Netherlands, June 16 – 20, 2002

INCLUDING PERFORMANCE MEASURES IN DIMENSIONS AND MASS REGULATIONS

John de Pont TERNZ, PO Box 97-846, S.A.M.C., Manukau City, New Zealand.
Peter Baas TERNZ, PO Box 97-846, S.A.M.C., Manukau City, New Zealand.
Don Hutchinson Land Transport Safety Authority, PO Box 2840, Wellington, New Zealand.
Dom Kalasih Land Transport Safety Authority, PO Box 2840, Wellington, New Zealand

ABSTRACT

Heavy vehicle dimensions and mass regulations exist primarily to maintain safety and to preserve the infrastructure. In terms of these aims, prescriptive regulations are a crude mechanism but they are straightforward to measure and thus compliance checking is relatively simple and cheap. Since the RTAC study, which led to the first of these symposia in 1987, there has been increasing interest by regulators and users in the potential use of performance-based standards either in lieu of or as an adjunct to dimensions and mass rules.

In New Zealand the performance-based approach was adopted very early on for vehicles operating under permit. Vehicles operating within the existing dimensions and mass limits are not required to meet any performance standards for stability. However, New Zealand's dimensions and mass regulations are currently under review with the new rule scheduled for implementation in July 2002. The new rule includes a requirement that all heavy vehicles have a static roll threshold (SRT) greater than 0.35g.

In order to keep compliance costs reasonable an analytical method for determining SRT has been developed and implemented as an internet-based calculator. The calculator has been designed so that the required user inputs are easily obtainable. Default values are used for the other vehicle parameters. For the most significant parameters the user has the option to replace the defaults with real data. Where a vehicle fails to meet the target SRT, the load reduction at current height and the load height reduction at current mass required to achieve the target are both calculated. Validation tests comparing the calculator results with those obtained from tilt table testing and from computer simulations have shown a remarkably good level of accuracy.

INTRODUCTION

Since the Road Transport Association of Canada (RTAC) study of the mid 1980s, which led to these symposia, there has been a growing interest by regulators and road users in the potential of using performance-based standards rather than prescriptive limits to control heavy vehicle dimensions and mass. Ultimately the aim of any mass and dimensions controls is to promote safety without unduly inhibiting the efficiency of road transport operators. Performance measures are more directly linked to safety outcomes than prescriptive limits.

Since the early 1990s New Zealand regulators have required stability performance assessments to be undertaken for vehicles with divisible loads requiring permits to exceed the prescriptive dimension and mass limits. Until now it was considered too costly to require stability assessments for vehicles complying with the prescriptive limits. However, the method for calculating Static Roll Threshold (SRT) described in this paper is straightforward and cost-effective enough to be able to be applied to the heavy vehicle fleet. It is part of the New Zealand Dimensions and Mass Rule 41001 (LTSA, 2001) scheduled to come into force in July 2002.

SAFETY AT REASONABLE COST

Road safety in New Zealand is the responsibility of the Land Transport Safety Authority (LTSA), which is a Crown entity charged with promoting land transport safety at reasonable cost.

The approach that is generally used to determine reasonable cost is broadly as follows:

- Surveys of road users (the public) are undertaken regularly to determine the amount that the average person is prepared to spend to save one life on the roads. Currently this figure is about NZ$2.8M. Values for serious injury, minor injury and property damage only crashes are also obtained. These values reflect the social cost of road crashes rather than the actual economic costs.
- The benefit of any proposed road safety counter-measure is then determined by multiplying the expected reduction in crashes by the social cost values associated with those crashes.
- The cost of the counter-measure is estimated by including all costs, not just those incurred by LTSA. Loss of productivity by road transport operations, for example, is a cost that is included.
- The decision on whether the counter-measure will be implemented then depends on whether the benefits exceed the costs.

Numbers of performance measures relating to stability and control have been developed. In New Zealand we have used Static Roll Threshold (SRT), Dynamic Load Transfer Ratio (DLTR), Rearward Amplification (RA), High Speed Transient Off-tracking (HSTO), High Speed Steady Off-tracking (HSO), Yaw Damping Ratio (YDR) and Low Speed Off-tracking (LSO). Within these measures there are variations in both the names used and the test procedures so care needs to be taken when comparing results from different studies. Evaluation of the performance measures can be through experimental measurement or by computer simulation. For most of the measures experimental measurement is relatively costly. Accuracy is very good provided the test procedures are followed rigorously. Computer simulation is generally cheaper but requires that the characteristics of the key components of the vehicle are modelled accurately. In New Zealand, computer simulation by approved analysts has been accepted by the LTSA as an acceptable method for evaluating performance measures.

The LTSA have generally required a performance assessment against all of the measures listed above and expected the vehicles to perform as well or better than the alternatives. Thus although the safety benefits may be unknown, there is reasonable confidence that there will be a safety benefit rather than a cost. The operator expects some economic gains from operating this vehicle. He or she knows the cost of applying for approval (including having the performance assessment done) and hence would not proceed if the expected economic benefits do not exceed the costs. The question of safety at reasonable cost resolves itself.

However, requiring all vehicles that meet the prescriptive requirements to also meet a performance standard is quite different. The productivity benefits are likely to be negative as some existing vehicles will not achieve the performance standard without a reduction in freight capacity. The reduction in social cost through reduced crashes must therefore offset both the cost of compliance and the loss in productivity. From this it can be shown that it is not viable to require all heavy vehicles to be assessed for their stability performance using computer simulation or experimental measurement because the compliance costs would be greater than the potential safety benefits using the social cost model outlined above. However, if some easily implemented low-cost method of estimating the performance measures to reasonable accuracy can be developed it may be possible to meet the reasonable cost criterion.

A recent study (de Pont et al, 2000, Mueller et al, 1999) determined relationships between some performance measures and relative crash rates for rollover and loss-of-control crashes involving heavy vehicles in New Zealand. Figure 1 shows the relative crash rate against SRT as determined by this study. From this figure we see that vehicles with poor SRT (less than 0.3g) have a crash rate about four times the average. The study also found that the 15% of the vehicle fleet with an SRT below 0.35g was involved in 40% of the rollover and loss-of-control crashes. This indicates that improving the performance of the poorest performing vehicles in the fleet should have a significant impact on the overall crash rate. Similar relationships were found for DLTR and HSTO. Not unexpectedly SRT and DLTR were strongly correlated so that improving one should improve the other.

SRT, which is the lateral acceleration at which rollover occurs during steady speed cornering, is one of the most fundamental of stability-related performance measures. Rather fortuitously it is also one where under certain assumptions an analytical solution is possible thus lending itself to the development of a simple low-cost algorithm for its determination.

THE STATIC ROLL THRESHOLD CALCULATION ALGORITHM

Figure 2 shows a cross-section view of heavy vehicle subject to a lateral acceleration, α. For this 2D vehicle model rollover occurs when $F_2 = 0$. It can easily be shown that, assuming small angles, this implies that:

$$SRT = \alpha = \frac{T}{2H} - \Phi$$

where T is the track width
H is the centre of gravity height
Φ is the total roll angle due to compliance

The most important term in this expression is the T/2H, which is often referred to as the static stability factor. It is an upper bound for SRT and represents what the SRT would be if the vehicle was a rigid body with no compliances. In practice, however, the compliances from the tyres, the suspension and the body itself are not zero and the actual SRT could be as much as 50% below the value given by T/2H.

Winkler et al (1992) proposed a three level screening approach to ensuring that vehicles achieved a minimum SRT level of 0.35g. At the first screening level vehicles with T/2H greater than 0.58 almost certainly have an SRT greater than 0.35g and are acceptable. Vehicles with T/2H less than 0.58 but greater than 0.46 are accepted if the tyres and suspensions on each axle meet certain minimum stiffness requirement. These requirements are load dependent. Finally at the third level of screening testing is required. Note again that if T/2H is not greater than 0.35 the vehicle cannot have an SRT greater than 0.35g.

For implementation in New Zealand we have used a mathematical solution based on the graphical approach developed by Chalasani (Winkler et al, 2000) to estimate the actual SRT. We assume that the tyres are linear springs acting through the mid-point of the tyre. The suspension is assumed to consist of linear springs at the spring hangers with a rotational spring at the roll centre to represent the auxiliary roll stiffness. The suspension springs are assumed to have lash. The vehicle body is assumed to be rigid. Because each vehicle unit is registered and certified independently the algorithm is only required to be applied to single vehicle units with, at most two suspension groups. (The issue of roll coupled combination vehicles will be discussed in the next section on implementation.) For each suspension group force and moment balance equations can be written with the rigid body requirement generating a further equation coupling the two ends of the vehicle.

As lateral acceleration is applied to the vehicle, the vehicle body rolls until the onset of lash or wheel lift-off at one of the suspension groups. The roll stiffness of the system then reduces. Further lateral acceleration causes more body roll until the next event occurs when the stiffness again changes and so on until rollover occurs. For each suspension group, there are three possible sequences of events:

1. wheel lift-off before the onset of suspension lash
2. onset of lash followed by wheel lift before maximum lash is reached
3. onset of lash, full suspension lash, followed by wheel lift-off.

The sequence of events for the two suspensions can interleave each other. As each event causes the system stiffness to change it is necessary to determine the correct sequence of events as well as solving for each point. At each event the corresponding lateral acceleration is calculated and the maximum value achieved is the SRT. This may not be the last event in the sequence. For many vehicles the point of no return occurs before the last wheel has lifted off.

The list of variables needed to solve these equations are, for each suspension group,:

- tyre stiffness
- tyre track width
- unsprung mass
- unsprung mass centre of gravity height
- suspension spring stiffness
- suspension track width
- suspension auxiliary roll stiffness

- suspension roll centre height
- sprung mass
- sprung mass centre of gravity height

IMPLEMENTATION

The above algorithm has been coded using javascript so that it can be run as a web-based application via the internet. In keeping with the principle of safety at reasonable cost, a number of assumptions were made to make the software easier to use and to reduce the potential cost of compliance.

Smaller trucks (with a GVM less than 12 tonnes) were exempted from the requirements. These vehicles operate primarily within urban areas and do not have a high rollover crash rate. As the calculator software is then only required for larger vehicles it was assumed that all vehicles operated at the maximum allowable width (2.5m) and the wheel track was calculated back from this based on the tyre configuration.

Three tyre configurations (dual, single and wide single) are permitted and three tyre sizes (22.5, 19.5 and 17.5). Generic tyre stiffness values are assumed for each tyre type and size. No allowance is made for low profile tyres or for variations in stiffness for different brands of tyre. There are two reasons for this approach. The first is that it simplifies the user input requirements and the second is that the tyres will be changed numbers of times during the vehicle's life and there is no easy way of ensuring that tyre-specific characteristics will be maintained at all times.

Input values for tare mass and payload mass at each axle group are required. Default values are used for the axle masses and wheel masses. These depend on the axle type, i.e. steer, drive or trailer axle and tyre size. From these values the unsprung mass is estimated by summing the axle mass and the wheel masses. By subtracting the unsprung mass from the tare mass, the empty vehicle sprung mass can also be estimated. The laden sprung mass is calculated from the payload mass and the empty vehicle sprung mass.

The unsprung mass centre of gravity height for each axle is set at the typical tyre radius for the tyre size. The error in centre of gravity height from ignoring low profile effects has a minimal effect on the final SRT value. For the empty vehicle sprung mass the centre of gravity height is estimated by adding a fixed value, which depends on vehicle type to the axle centre of gravity. Currently the values used are 0.56m for prime movers and 1.25m for trailers which are typical values given by Fancher et al (1986). Although these values are somewhat arbitrary for most typical vehicles they work quite well because they are combined with the payload centre of gravity to give an overall sprung mass centre of gravity and for most vehicles the payload contribution dominates. For prime movers the empty sprung mass component is more significant but the variations from the 0.56m value are small. For trailers, the 1.25m value is typical for a van body vehicle and is probably high for say a flat deck vehicle. However, the vehicles with lower empty sprung mass centre of gravity heights also tend to have low empty sprung masses so the error in the overall sprung mass centre of gravity is small. To estimate the payload centre of gravity the user specifies one of three load types, uniform density, mixed freight or other. For uniform or mixed freight he or she is required to enter the load bed height and the maximum load height. Uniform density assumes the load is uniformly distributed and thus the payload centre of gravity is midway between the load bed and the maximum load height. Mixed freight assumes a load distribution where 70% of the load mass is in the lower half of the load space and 30% is in the top half. This is a typical distribution given by Fancher et al (1986). The centre of gravity is therefore at 40% of the distance between the load bed and the maximum load height. The "other" category is for any load types not covered by the previous two categories. In this case the user is required to determine the height of the payload centre of gravity from the ground and input the value directly. Unusual vehicle bodies that do not fit the assumptions for empty sprung mass centre of gravity height outlined above can be handled by treating them as part of the payload using the "other" category.

The payload centre of gravity calculations do not currently take into account the case of moving loads such as partially filled liquid tankers or hanging animal carcasses. These cases can be approximated by determining the lateral shift in centre of gravity and reducing the effective track width accordingly but this has not been implemented in the current version of the software. Many liquid tankers in New Zealand are subject to the dangerous goods regulations because of the potential impacts of a rollover. These regulations are currently under

review but it is being proposed that these vehicles will need to meet an SRT limit of 0.45g when full and be required to have the tank subdivided into multiple sections to limit the proportion of the load that is free to slosh at any time. These two requirements together should ensure that even in the worst loading case the SRT will be better than 0.35g. Other liquid tankers such as dairy generally also have a very good SRT (> 0.45g) when full and thus even in the worst case sloshing of a partial load should still meet the 0.35g limit. The hanging carcasses load is not addressed but this involves only a small number of vehicles which have not been identified as having a stability problem.

The final inputs needed to solve the equations are the suspension characteristics. For each axle group, the user can select one of three suspension options, generic steel, generic air, or user specified. For the generic suspensions, default parameter values, which depend on the axle type are used. The current default values, which are shown in Table 1, are derived from Fancher et al (1986) and represent suspensions at the more compliant end of the spectrum. Thus the resulting SRT values are conservative. If the user-defined option is selected the user is required to input values for suspension parameters, which have to be either measured or obtained from the manufacturer. Some suspension manufacturers will readily make this information available while others regard it as commercially sensitive and are very reluctant to divulge it.

Once all the input data have been obtained or inferred, the calculator software determine the SRT and compares it against the target value of 0.35g. Originally it was proposed that the target value for trailers should be 0.4g, which, as can be seen from Figure 1, would give a greater safety benefit. However, estimates indicated that too many vehicles would need to reduce their load capacity and hence the economic losses from reduced productivity would be greater than could be justified by the safety benefits. If the vehicle fails to meet the SRT target, the calculator program iterates to calculate the reduced payload mass at the same load height and the reduced load height at the same payload mass needed to achieve the 0.35g SRT.

As mentioned in the previous section the algorithm has been developed to solve for the single vehicle unit with up to two axle groups because vehicles in New Zealand are certified and registered as individual units not as parts of a combination. This then raise the question of how to handle tractor units and semi-trailers, which normally operate as a combination and are roll-coupled so that both units together contribute to the SRT. Winkler (1992) has addressed this issue in tilt table testing by developing the concept of a virtual tractor for testing semi-trailers. In the calculator the same philosophy is used although the details differ. Tractor units, which carry no payload, are exempted from the SRT calculation requirement because without a trailer they will always meet the target value. Semi-trailers have their SRT calculated on the basis of the loads, suspensions and axles on the rear axle group alone. The philosophy is that if the rear axle group can withstand a lateral acceleration of 0.35g without rolling over it should not apply an overturning moment at the king pin till the lateral acceleration exceeds 0.35g. This is not necessarily correct because the total compliance is made up of several components (tyres, suspension, fifth wheel) and the relative magnitude of the different components is not necessarily the same for the semi-trailer axles and the tractor drive axles. However, given that any tractor could be coupled to the trailer it is not possible to determine whether this will be the case or not. In practice the approach seems to work reasonably well.

A demonstration version of the software was made available on the internet for three months during the public consultation phase of the dimensions and mass rule and was well received. The code is structured so that there is general public access to the calculator itself up to the point where the results are presented on the screen. There is then a login facility for certifying engineers to proceed further where a certificate containing all the vehicle data together with the parameters used in the calculation and the results is printed. Feedback from the consultation process suggested some modifications to the input data options but a general satisfaction with the ease of use of the calculator and the credibility of the results. A final version of the software is now being developed and will be available on the internet before the Dimensions and Mass Rule comes into force in July 2002. As with the demonstration version the general calculator part of the program will be freely and publicly available to all. Compliance with the legal requirement for heavy vehicles to achieve a minimum SRT level will be primarily determined by approved certifying engineers using the software on the internet.

VALIDATION

As outlined earlier, performance assessments have been undertaken by computer simulation for permit vehicles in New Zealand for over ten years. Most of these were done using the Yaw-Roll software from UMTRI. Yaw-Roll has also been used for a number of other studies in New Zealand to investigate potential changes in vehicle regulations. Results from some sample vehicles from these earlier studies together with a tilt table test on a log transport trailer have been used to validate the SRT calculator algorithm.

The tilt table test was conducted on a 4-axle full trailer with steel suspension loaded to 20 tonnes gross vehicle mass as shown in Figure 3. After the tilt table test the vehicle SRT was calculated using Yaw Roll software to simulate steady speed travel through a slowly tightening curve. The SRT was also calculated using the calculator software described above with both generic steel suspension option and with the "user defined" option using a linearised version of the manufacturer-supplied suspension data that was used for the Yaw Roll analysis. The results are summarised in Table 2. Using the generic suspension option, the calculator gives a slightly lower estimate of SRT which is conservative as it means the vehicle's actual SRT is probably better than this value. With user-defined suspension parameters, the calculator gives and SRT value which is the same as that measured using the tilt table test to within the measurement accuracy. Yaw Roll gives a value which is slightly higher than the measured value. This is probably because of the differences in the test conditions. Yaw Roll considers a moving vehicle in a tightening curve while the tilt table has the vehicle stationary on a slope. The SRT values obtained by all the methods are acceptably close to each other.

From records of previous simulation studies 11 vehicles were selected with calculated SRT values ranging from 0.32g to 0.62g. For each of the vehicles the SRT was recalculated twice using the SRT calculator program. The first time generic suspensions were used and the second time the "user defined" suspension option was used and the same suspension parameters as used in the Yaw Roll analysis of the vehicle were input into the calculator. Figure 4 shows a comparison between the Yaw Roll calculated SRT and the SRT calculator results when generic suspensions are used. For most of the vehicles the correspondence is very good. However, for two of the vehicles, the calculator gives significantly lower SRT values than Yaw Roll. This is not unexpected as the generic suspensions are at the more compliant end of the performance spectrum and thus if the actual suspension is significantly more roll stiff the true SRT will be significantly higher than estimated by the calculator. Figure 5 shows the comparison of the Yaw Roll results with the calculator when correct suspension parameters are input by the user. The correspondence is markedly better and there are now no outliers.

CONCLUSIONS

New Zealand has used stability performance assessment for many years for policy development and for issuing permits for vehicles with divisible loads to operate outside the prescriptive mass and dimensions regulations. It is now going a step further in requiring vehicles within the prescriptive mass and dimensions regulations to meet a stability performance level. As far as is known this is a world first.

The key to be able to implement this measure cost-effectively is having a low cost method for accurately and reliably estimating SRT. This paper describes the development of such a method and its implementation via an web-based software program called the SRT calculator. The user inputs that are absolutely required are generally known or easily measured by the vehicle operator. Increased accuracy can be achieved through the input of suspension parameters but for most vehicles this is not necessary.

The software will be publicly available and the demonstration version has already been used by some transport operators for parametric studies relating load height to stability for their particular operation. The calculator will be the primary tool for certifying that vehicles comply with the SRT requirement, although the option to use a tilt table test or a computer simulation as an alternative will exist.

The validation that has been undertaken shows that the calculator is consistent and sufficiently accurate.

Implementation of the new dimensions and mass rule with the SRT requirement is scheduled for July, 2002.

REFERENCES

de Pont, J. J., Mueller, T. H., Baas, P. H. and Edgar, J. P. 2000."Performance Measures and Crash Rates" In Proceedings of 6th International Symposium on heavy vehicle weights and dimensions (Ed, Borbely, C. L.) Saskatoon,Saskatchewan, Canada, pp. 1 - 10.

Fancher, P. S., Ervin, R. D., Winkler, C. B. and Gillespie, T. D. 1986. "A Factbook of the Mechanical Properties of the Components for Single-Unit and Articulated Heavy Trucks" US Dept of Transportation Report DOT HS 807 125.

LTSA 2001. "Land Transport Rule Vehicle Dimensions and Mass Rule 41001" Land Transport Safety Authority Yellow draft 77p.

Mueller, T. H., de Pont, J. J. and Baas, P. H. 1999. "Heavy Vehicle Stability Versus Crash Rates." TERNZ Report 48 p.

Winkler, C. B., Blower, D., Ervin, R. D. and Chalasani, R. M. 2000. "Rollover of Heavy Commercial Vehicles" The University of Michigan, Transportation Research Institute. SAE research report.

Winkler, C. B. et. al. 1992. "Heavy Vehicle Size and Weight - Test Procedures for Minimum Safety Performance Standards" The University of Michigan Transportation Research Institute Final technical report UMTRI - 92 - 13.

TABLES & FIGURES

Table 1- Suspension properties for generic suspensions.

Suspension Name and Model Number	Generic - steer axle	Generic steel	Generic air
Suspension spring stiffness (N/m)	185000	900000	350000
Suspension track width (m)	0.8	0.8	0.8
Total roll stiffness per axle (Nm/radian)	130000	520000	780000
Suspension lash (mm)	15	15	1000
Roll centre height from axle (m)	-0.02	0.2	0.2

Table 2 - Predicted SRT from Tilt table, Yaw Roll software and SRT calculator.

Tilt table test*	Yaw Roll	SRT Calculator Generic steel suspension	SRT Calculator User defined suspension
0.418 ± 0.006	0.428	0.407	0.415

*The tilt table results are based on seven tests and the range given is the 95% confidence interval for the mean.

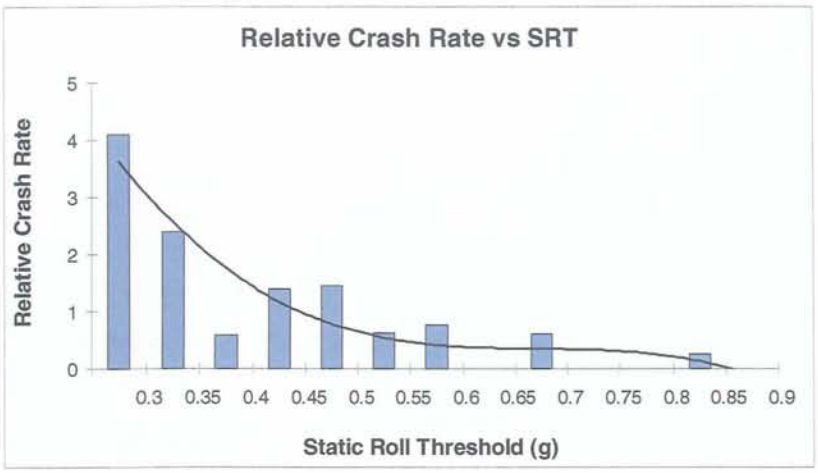

Figure 1 - SRT Relative Crash Involvement Rate.

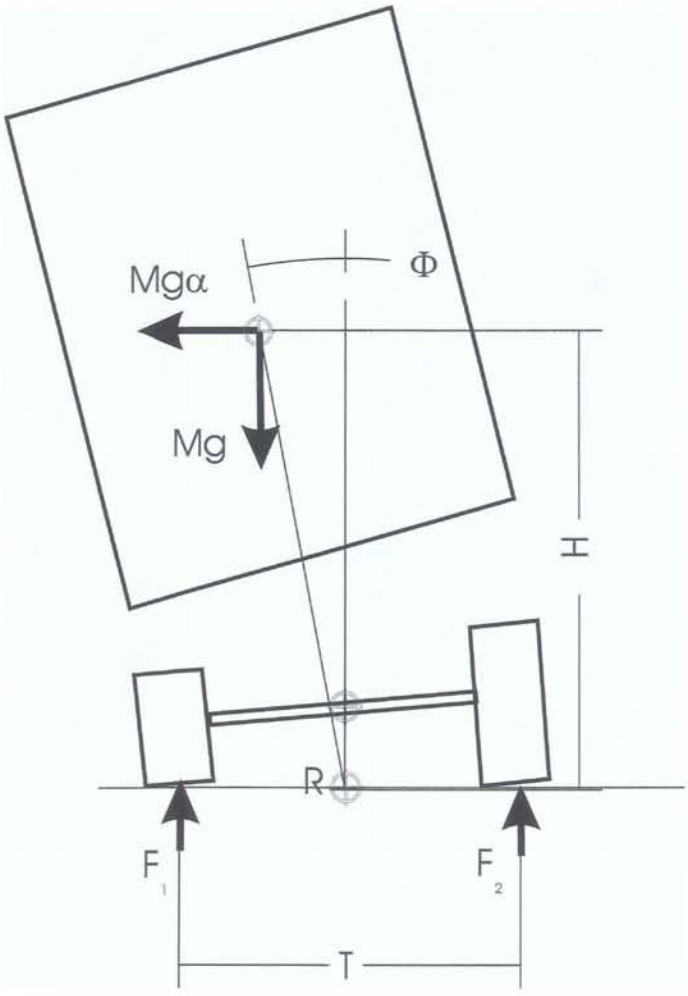

Figure 2 – Schematic showing heavy vehicle under lateral acceleration.

Figure 3 - Tilt table test on a 4-axle log trailer.

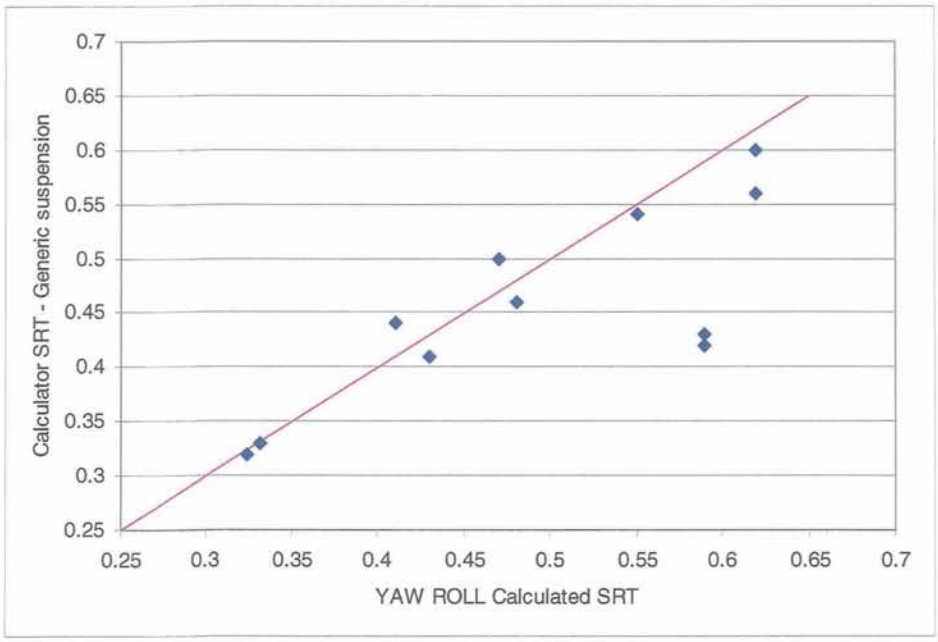

Figure 4 - Comparison of SRT Calculator using generic suspensions with Yaw Roll

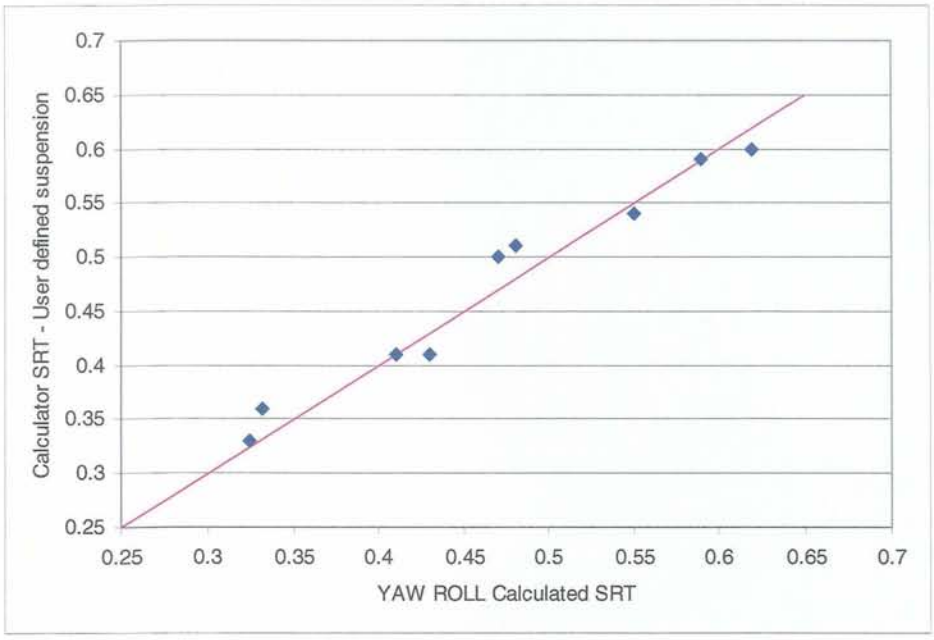

Figure 5 - Comparison of SRT Calculator using user defined suspension parameters with Yaw Roll