COMPARISON OF THREE PROGRAMS FOR SIMULATING HEAVY-VEHICLE DYNAMICS

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ABSTRACT

Australia and New Zealand are developing a performance-based standards system of heavy vehicle regulation as an alternative to the current prescriptive systems, which is intended to encourage and foster innovation in road transport. Expected outcomes of the new regulatory regime are improved road safety, better use of the infrastructure, reduced environmental impacts and enhanced productivity. Computer-based modelling will play a central role in both the development and initial demonstration of innovative vehicles and concepts. Some Stakeholders have expressed the concern that performance predictions from computer-based models may not be reliable and may substantially differ with software package and with the Service Providers that use them. To address these concerns three computer-based modelling packages were compared: ADAMS, The University of Michigan Transport Research Institute (UMTRI) constant velocity Yaw/Roll model and AUTOSIM. Models of two heavy vehicles were created by two Consultants using the three packages. The same input datasets were provided to each Consultant and simulations were performed using the same test manoeuvres. Time histories of a wide range of variables from the simulations were compared as well as numeric values from a selection of performance measures. Simulations that provide a direct measure of vehicle responses to precisely defined steer inputs (openloop control) showed excellent agreement and the results were generally more consistent than simulations requiring steer controllers and closed-loop path following. With due care, acceptable agreement between modelling packages can be achieved. Recommendations are made that reduce the variability between models in path following tasks to acceptable levels.

1. INTRODUCTION

1.1 Brief Review of Heavy Vehicle Dynamics Modelling

Realistic models and simulations of heavy vehicle dynamics have been developed over a period of almost 40 years (Segel, 1993). Early simulations were based on mathematical models whose equations were laboriously and painstakingly derived by hand and solved using purpose-written computer programs. The development of these simulation programs and verification of their outputs against experimental tests was undertaken at considerable expense in government funded research programs and through the involvement and support of equipment manufacturers and the transport industry.

In the late 1970s engineers began using newly available computer programs that automated the process of creating the mathematical models and solving the equations (Orlandea and Chace, 1977). These programs are generically referred to as general-purpose multi-body dynamics programs, allowing detailed models of literally any mechanical system to be created and simulated rapidly with relative ease (Schiehlen, 1986; Kortüm and Sharp, 1993; Richard and Gosselin, 1993; Sharp, 1994).

Multi-body simulation programs are now widely used throughout the automotive industry, and they are also used to solve dynamics problems that arise in areas such as biomechanics, machine dynamics, rail vehicle dynamics, robotics and in spacecraft dynamics and control.

In the field of vehicle dynamics, particularly in Australia, New Zealand, Canada and the USA, three computerbased programs have emerged as being the most popular and widely used for simulating heavy vehicle dynamics. They are ADAMS[®] (Ryan, 1993; Mechanical Dynamics, 2002)¹, The University of Michigan Transportation Research Institute (UMTRI) "Constant Velocity Yaw/Roll Model" (Mallikarjunarao, 1981; Gillespie and MacAdam, 1982) and AUTOSIMTM (Sayers, 1990; Sayers, 1993; Mechanical Simulation Corporation, 2002)².

1.2 Modelling Applications in Performance-Based Standards

In a major initiative in heavy vehicle regulation, Austroads³ and the National Road Transport Commission are developing performance-based standards as a national alternative to the current prescriptive system (National Road Transport Commission, 2002). Expected outcomes of the new regulatory regime are improved road safety, better use of the infrastructure, reduced environmental impacts and enhanced productivity.

Given that performance-based standards are intended to encourage and foster innovation in road transport, computer-based modelling will play a central role in both the initial demonstration and the development of innovative vehicles and concepts. However, some Stakeholders have expressed the concern that performance predictions from computer-based models may not be reliable and may differ significantly with software package and with the Service Providers that use them.

This paper attempts to address these concerns by comparing three computer-based modelling packages – ADAMS, UMTRI's constant velocity Yaw/Roll model and AUTOSIM – in a range of manoeuvres.

2. METHODOLOGY

2.1 Verification of Models

The literature confirms that models created with each of the modelling packages considered in this paper have been previously validated against experimental tests (Orlandea and Chace, 1977; Gillespie and MacAdam, 1982; Lam, 1986; El-Gindy and Wong, 1987; Sayers, 1990; Sayers and Riley, 1996; Latto, 1999; Prem et al, 1999; Sweatman and McFarlane, 2000; Schade and Hamill, 2000; Elliot et al, 2001).

A review of a selection of verification studies was carried out as part of a separate study and is reported in Prem et al (2001b). The review generally confirmed that an acceptable level of agreement can be achieved between experimental data and responses predicted with any one of the three modelling packages considered in this paper. However, while this has been demonstrated, it is important to note that the level of agreement achieved between test data and simulations very strongly depends on the quality of the test data and on factors that are related to the models. These factors include the level of detail included in the model, accuracy of the component models, accuracy and integrity of the input data (dimensions and mechanical properties), and the manoeuvres performed.

It should be further noted that obtaining accurate test data that can be used for validating computer-based models is a non-trivial and exacting task. It necessarily covers issues such as the use of correct instrumentation and its proper installation and operation, pre-processing and recording of signals from a wide range of different transducers, the conduct of controlled tests under less than ideal in-the-field conditions – which at times may involve performing near-limit manoeuvres – and recording all inputs and disturbances to the vehicle (steering, roadway and environmental) that may influence the measured responses.

2.2 Testing the Modelling Systems

Given that the modelling packages had been validated against experimental tests, the aim was to devise a series of simulations, or test problems, in a heavy vehicle dynamics context, that would test and explore the boundaries of the modelling packages as well as the Consultants (Service Providers) using them.

Roads and Transport Dynamics (RTDynamics) performed the modelling in ADAMS, Transport Engineering Research New Zealand (TERNZ) used AUTOSIM and the UMTRI Yaw/Roll model.

The following sequence of activities outlines the process, discussed in further detail in the subsequent sections of this paper:

¹ ADAMS is an acronym for Automatic Dynamic Analysis of Mechanical Systems. In addition to its widespread use in Australia and North America, ADAMS is also used widely in Europe and Asia.

² The AUTOSIM acronym is derived from "AUTOmatically generate SIMulation codes".

³ Austroads is the association of Australian and New Zealand road transport and traffic authorities.

- Two reference vehicles were developed and a comprehensive list was prepared of the parameters required to create the vehicles in the computer-based modelling package;
- Standard manoeuvres were developed that would be simulated in each modelling package these manoeuvres are based on the ones presented in Prem et al (2001a);
- iii) The reference vehicles were created in each of the modelling packages, the standard manoeuvres were simulated, and a range of outputs from the simulations was compared and the level of agreement between them was determined.

3. VEHICLES MODELLED, INPUT DATASETS AND MANOEUVRES

3.1 Reference Vehicles

Two reference datasets describing two separate heavy vehicles were created. The datasets describe a generic Bdouble and a generic truck/trailer combination. Sufficient information was sourced and prepared that would fully describe the vehicles and enable computer-based dynamic models to be created in each of the packages. The two vehicles chosen for this exercise were considered appropriate for the purposes of this work because they contain features that are present not just in a significant proportion of the Australasian heavy vehicle fleet, but also in heavy vehicles used throughout the world. Where possible, hardware elements were included in the models, such as non-linearity in suspensions components and tyres, to ensure a range of features within the modelling packages and capabilities of the Consultants could be tested and demonstrated.

It is important to note that the models were developed to test a certain range of base level and advanced computer modelling features and skills pertaining to heavy vehicle dynamics simulation. The models do not necessarily test all of the capabilities or features of the modelling packages that were used or the Consultants using them.

Perspective views of the two reference vehicles are given in Figs 1(a) and 1(b). Full descriptions of all the parameters used in the models are provided in Prem et al (2001b), which also contains the complete set of values for each parameter of each reference-vehicle⁴. The parameters used in the models are briefly described below.

3.2 Input Datasets and Steer Control

3.2.1 Geometry, Mass Properties, Suspensions and Tyres

The input datasets for the two reference vehicles are presented and fully described in Prem et al (2001b). These include:

- geometric layout and key dimensions;
- mass properties of bodies⁵ (centre-of-gravity CG location and mass moments of inertia);
- couplings (turntables, pin-couplings and drawbar);
- · characteristics of the suspensions (springs, dampers, roll centre heights and roll steer coefficients); and
- tyres characteristics.

The force/deflection characteristics of the suspension springs used in the models, for example, are shown in Fig 2(a) and 2(b). These were derived from the data presented in Ervin and Guy (1986). The force/deflection values selected for the vehicle are simply the average of the compression and extension envelopes from the selected spring tables. As can be seen in Figs 2(a) and 2(b), with the exception of the steer axle, the springs are highly non-linear and there is a substantial increase in stiffness for deflections that are outside the range ± 30 mm.

The springs do not include inter-leaf coulomb friction (hysteresis). It was not included because no documentation could be found that described how it had been modelled in UMTRI's Yaw/Roll program. Further support for this decision came from a similar exercise conducted by Sayers and Riley (1996) in which various Yaw/Roll models were compared with models of the same vehicles created in AUTOSIM. They concluded that it would be difficult to get agreement with UMTRI's Yaw/Roll program (particularly with regard to transient responses) unless the damping due to hysteresis has been set to zero.

In the models identical tyres were used on the steer, drive and trailer axles. The tyre side force characteristics due to slip angle are based on data presented in Ervin and Guy (1986); the corresponding data for aligning moments were obtained from El-Gindy and Kenis (1998). These data are plotted in Figs 3(a) and 3(b) and describe tyre side

⁴ Prem et al (2001b) can be downloaded from the National Road Transport Commission website at http://www.nrtc.gov.au

⁵ For simplicity all parts used in the models were assumed to be rigid bodies.

force and aligning moment characteristics over a range of vertical loads and varying levels of slip angle in the absence of longitudinal slip.

Side force generation due to inclination (camber angle) has not been included in the models because there is no provision for this feature in UMTRI's Yaw/Roll program; though it is a feature within ADAMS and it can also be included in AUTOSIM. Further, there is also no provision in UMTRI's Yaw/Roll program to include relaxation length, and for this reason the tyre relaxation length parameter in the respect ADAMS and AUTOSIM models was set to zero. This means that the build-up of tyre side force and aligning moments is assumed to occur instantaneously, and the side force and aligning moments will always be at their maximum values for the prevailing slip and vertical load conditions.

3.2.2 Steer Control

Directional control of the models is achieved through the steer controller. Depending on the manoeuvre specified, steer control is intended to accomplish either a specific, precisely defined sequence of steer angle input (open-loop control), or it will guide the vehicle along a prescribed path in the same way that a real driver would steer the vehicle along a path (closed-loop control).

The characteristics of the steer controller for closed-loop control were not specified in the dataset. It was assumed that suitable controllers were either included as a feature of the specific software package (see, for example, Gillespie and MacAdam, 1982) or it was created from first principles, as was the case with the simulations in ADAMS.

3.3 Manoeuvres Simulated

Previous similar studies have used lane-change manoeuvres, steady (or quasi-steady) turns, and step steer inputs (Gillespie and MacAdam, 1982; Lam, 1986; El-Gindy and Wong, 1987; Sayers and Riley, 1996). These simulations were designed to test for specific performance attributes thereby revealing different aspects of the vehicle models and/or controllers.

The lane change, for example, is a very useful manoeuvre but it is not a reliable way of comparing vehicle models because identical vehicle models with different steer controllers may yield different responses. Hence, the controllers may have an influence on how well the responses of the vehicles match.

Steady turns, or quasi-steady turns, are also useful for comparing vehicle models but they also have limitations; typically, when the steady conditions have been reached all of the terms in the equations-of-motion that are related to the transient response have reduced to zero. These terms may be the only sources of differences between the response of the models which would not be revealed in the steady turn test⁶.

For the present study the following two types of simulations were performed:

- i) one to characterise vehicle-only responses under both transient and steady state conditions involving an openloop control strategy requiring the precise application of a predetermined steer sequence and observation of vehicle response;
- the other to provide information about the combined driver/vehicle responses under both high- and low-speed conditions using closed-loop steer control in two well-defined path following tasks.

In total, four simulations were devised that would test and compare a range of features in the models. Pulse steer and step steer inputs were used in the two simulations that employed open-loop steer control, and closed-loop control was used in path following tasks of a high-speed lane change and a low-speed 90° turn. These are described below under the two general headings of open- and closed-loop control.

⁶ When a dynamic system reaches a steady state condition many terms in the equations that govern its motion will have decayed to zero and thereby become numerically insignificant. By contrast, all of the terms in the equations of motion contribute to the transient response and matching the transient responses of two dynamic systems is considered a more challenging task than matching the steady state responses.

3.3.1 Open-Loop Control

a) Pulse-steer (transient response)

The pulse steer simulation is identical in concept to the test described in Prem et al (2001a) for estimating yaw damping. From an initial straight path, and a speed of 100km/h, the vehicle is subjected to a short duration input of steering; a very rapid application of steer is applied from the straight-ahead position over a short time-period and then quickly returned to the straight-ahead position over a time period of similar short duration. Once the steer has been applied the vehicle is then allowed to follow its own course without any further control action or external disturbances.

In response to the "pulse" of steering applied to the steered tyre, the towing unit (prime-mover or truck) will change its direction of travel (its heading) and the towed units (trailers) will follow. For a stable vehicle, and provided the subsequent motion is not extreme, the transition to the new heading is accompanied by vehicle oscillations that will progressively reduce in amplitude. For an unstable vehicle the oscillations may increase in amplitude culminating in rollover. This type of disturbance is commonly used to quantify the vibration characteristics of dynamic systems (Doeblin, 1980), including passenger cars and heavy vehicles (International Standards Organisation, 1988; International Standards Organisation, 2000). When applied to a heavy vehicle it will provoke a response from all of the key modes of oscillation; lateral/directional, yaw, roll and pitch.

The pulse steer input that was developed for this application is based on the haversine function described by Eqn 1, providing a smooth ramp-up and ramp-down transition.

$$\delta = \begin{cases} \frac{\delta_0}{2} + \frac{\delta_1}{2} + \frac{\delta_1 - \delta_0}{2} \sin \left[\left(\frac{t - t_0}{t_1 - t_0} \right) \pi - \frac{\pi}{2} \right] & t_0 < t < t_1 \\ \delta_1 & t \ge t_1 \end{cases}$$
(1)

where:

t = time (s) $t_0 = \text{commencement of steer application (s)}$ $t_1 = \text{termination of steer application (s)}$ $\delta = \text{steer angle (deg)}$ $\delta_0 = \text{initial value of the steer angle (deg)}$ $\delta_1 = \text{final value of the steer angle (deg)}$

A steer pulse of 10° and 0.5 s duration was chosen, which is significantly larger in amplitude and longer in duration than the 3.2°, 0.1 s pulse specified for yaw damping estimation described in Prem et al (2001a).

The much larger steer input used in this study was designed to ensure the vehicle models were well into the nonlinear operating regions, particularly the suspensions and tyres, in order to test the capabilities of the modelling packages at the boundaries and over the range of conditions that might realistically occur in performance-based standards related applications.

The pulse was also carefully designed to ensure that:

- the component models were not operated outside of the range of the real-world data that were used to create them, these include the measured tyre characteristics that are shown in Figs 6(a) and 6(b) and the suspension characteristics illustrated in Figs 4(a) and 4(b); and
- modelling assumptions were not violated, such as the small angle approximations for pitch and suspension relative roll angle used in UMTRI's Yaw/Roll program.

b) Step-steer (steady-state response)

The step steer simulation is very similar in form to the pulse steer except that the steer angle, instead of returning to zero, is held at its maximum value for the duration of the simulation. Once the transients have settled after the initial application of steer, the vehicle enters a steady turn and follows a circular path on a constant radius.

The steer angle input of 10° that was used for the pulse steer, if sustained, is much too large for the steep steer test and would cause the vehicles to rollover. A steer angle of 1° was found to be acceptable, making this manoeuvre consistent with the baseline manoeuvre developed by Sayers and Riley (1996).

3.3.2 Closed-Loop Control

Inertia forces and the prevailing vehicle dynamics are directly influenced by speed; at near-zero speed the inertia forces will be close to zero and the steer controller will be presented with a system that has different characteristics to one travelling at speed. For simulations involving closed-loop steer control it was therefore considered important to investigate the influence of speed on the closed-loop responses.

a) SAE lane change

The first of the closed-loop simulations is the SAE lane change (Society of Automotive Engineers, 1993). In the standard SAE lane change the vehicle is driven at 88km/h along a straight path approximately 100m in length. The vehicle is required to execute a lane change manoeuvre and the path of the centre of the steer axle not deviate by more than ± 150 mm from a precisely prescribed path over a distance of 61m. The lateral displacement of the lane change manoeuvre is 1.46m. These dimensions were chosen to give a peak lateral acceleration at the steer-axle of 0.15g.

b) Low-speed turn

In the low-speed turn the centre of the steer axle is required to follow a path comprising straight approaches to an 11.25 m radius 90° circular arc (Prem et al, 2002). This corresponds to the outside front wheel following a path of radius 12.5 m. A constant vehicle speed of 10 km/h is specified.

4. COMPARISON OF SIMULATION RESPONSES

The responses from the simulations described earlier are compared in this part of the paper. Only a selection of the key outputs and variables from each simulation are presented and compared. The key findings are discussed and where appropriate they have been compared with the results from other similar studies.

4.1 Open-Loop Control

4.1.1 Pulse-Steer (Transient Response)

a) B-double

The first variables considered, shown in Figs 4(a) and the first plot in Fig. 4(c), are tyre slip angles, the corresponding vertical forces (the lateral forces are not shown), and the tyre aligning moments, respectively.

These plots show that there is very close agreement of outputs from the three modelling systems with only small differences evident in the vicinity of the three largest peaks. The differences are particularly apparent in the tyre self-aligning moments, which are shown in Fig. 4(b). These differences are discussed in more detail shortly.

The initial peak in slip angle of about 9°, developed at the steer tyre and evident in the three plots, is due to the 10° pulse-steer input. This peak, and the subsequent slip angles developed by the other tyres in the ensuing motion, is well within the slip angle range (0 to 12°) of the measured tyre data shown in Fig. 3(a). Further, the vertical loads on the tyres do not exceed the range of the measured tyre data, and when considered together with the slip angle data it confirms the tyres are well into the non-linear operating region but not outside the range of the component data.

As discussed above, the main differences in the tyre forces and self-aligning moments occur at the peaks, with ADAMS typically predicting larger side forces (not shown) and aligning moments than either Yaw/Roll or AUTOSIM. This can be attributed to the way the tyre data are used in the component models within each modelling package. In Yaw/Roll and AUTOSIM linear interpolation/extrapolation is used (Gillespie and MacAdam, 1982; University of Michigan Transportation Research Institute, 1997), whereas the tyre models available within ADAMS use continuous and smooth analytical functions, which are based on tyre behaviour theory, to estimate the intermediate values (Bakker et al, 1987; Pacejka and Bakker, 1993). For slip angles (and

vertical loads) between the measured points the side force and aligning moment values from the ADAMS tyre models will generally be slightly greater than the corresponding values from either Yaw/Roll or AUTOSIM, as illustrated in Figs 5(a) and 5(b). With a greater number of input measured data points used in the component model the match between the models may be expected to improve.

The models created in Yaw/Roll and AUTOSIM assumed that pitch motion of the sprung masses are small. The pitch angles were checked (both for the B-double, and for the truck/trailer described later) and they were found to be less the 0.3° thereby confirming that the small angle assumption had not been violated.

The suspension forces are next compared in the second plot in Fig. 4(b). The suspension forces reach a maximum of about 55,000N confirming they are within the range and in the non-linear region of the component models that are based on the suspension data shown in Figs 2(a) and 2(b). Fig. 4(b) shows there is also close agreement between the outputs from the three modelling systems, except for the slightly larger amplitude 2Hz oscillation in the drive axle suspension, which appears in both the Yaw/Roll and the AUTOSIM outputs; it may be a by-product of the assumption of small pitch angles used in these two models but not used in the ADAMS model.

Fig. 4(c) shows there is little difference in the lateral accelerations and the roll angles, thus confirming there is close agreement between the three modelling systems. The small differences that do exist are likely due the differences in the tyre models discussed above.

Due to the very abrupt nature of this manoeuvre, it can be seen in Fig. 4(c) that the prime mover momentarily reaches a peak lateral acceleration of 0.45g. In a steady turn if this value were sustained it would be sufficient to cause rollover. However, because the prime-mover is coupled in roll through the turntable to the unit behind it, and because the event is short in duration, a significant roll angle is prevented from developing. Similarly, the lateral acceleration of the second trailer reaches a peak that is slightly greater than 0.35g, and the roll coupling to the unit ahead of it also prevents a large roll angle developing.

b) Truck/Trailer

The truck/trailer is a dynamically more active system than the B-double, comprising two sprung masses that are not coupled in roll but connected through a pin coupling. Its motions are much more complex than that of the B-double providing a greater challenge for the modellers and testing additional features of the modelling packages.

The simulation outputs for the truck/trailer are compared in Figs 6(a) and 6(b). These include tyre slip angles, yaw rates, lateral acceleration and roll angles of the sprung masses.

The first observation that can be made is that the responses of the truck/trailer are not as smooth as those of the Bdouble and take much longer to reduce in amplitude; there being residual oscillatory motion even after 10 seconds of run time.

Fig. 6(a) shows the greatest differences in slip angle, exhibited by the trailer, is small, being no more than about 0.5°.

Yaw rates, lateral accelerations and roll angles are shown in Figs 6(a) and 6(b). These variables also indicate there is close agreement between the models. The differences that are apparent mainly occur around the peaks, and there is a small difference in the zero crossings at several locations.

Lateral acceleration, which is a key measure for vehicle stability, is compared in Fig. 6(b) showing that all three units are subjected to high levels in this manoeuvre; approximately 0.4g for the truck and about 0.45g for both the dolly and the trailer. Overall the agreement between the models is considered to be good.

4.1.2 Step-Steer (Steady-State Response)

a) B-double

While the pulse steer compares the transient responses of the models over a broad range of lateral accelerations and frequencies, the step steer simulation was designed to compare the performance of the models in a steady turn at one specific level of lateral acceleration corresponding to 1° of steer angle.

The results for the B-double step steer simulations are compared in Fig. 7, which show the lateral acceleration and the roll angles of the three sprung masses. There is very close agreement across all three models and both variables, and the maximum difference in the steady state values are 3.0%, 2.0% and 2.1%, respectively, for roll angle, yaw rate (not shown) and lateral acceleration.

In a similar exercise Sayers and Riley (1996) compared various Yaw/Roll models with models of the same vehicles created in AUTOSIM (aka TruckSim). Except for a small difference in speed (96.5km/h compared with 100km/h in this paper) the step steer test specified in this paper is identical to the one used by Sayers and Riley (1996).

The models created by Sayers and Riley (1996) were designed to match their Yaw/Roll models, and some of the modelling assumptions they used were adopted in this study⁷. For the same reasons given in Sayers and Riley (1996), hysteresis has been excluded from the suspension spring models in this study, though the non-linear load-deflection characteristics of the suspension springs was retained; Sayers and Riley (1996) further simplified their models and made the suspension springs linear.

Figs 9(a) and 9(b) taken from Sayers and Riley (1996) show the lateral acceleration and roll angle responses of the sprung masses of a 5-axle tractor and semi-trailer to a 1° step steer input. The lateral acceleration transients for the tractor sprung-mass shown in Fig. 9(a) are nearly identical, expect for the sharp peak that is centred on 1 s, greater for one of the modelling programs (which one is not known) by about 15%. When the transients have reduced in amplitude and the steady state turn conditions have been established, the AUTOSIM model is seen to predict larger sprung mass lateral acceleration by roughly 4.5%, which is reflected in prediction of larger sprung mass roll angles by about the same amount, as shown in Fig. 9(b).

When compared to the results presented in Sayers and Riley (1996), there is closer agreement between the models in this study under fewer modelling assumptions and with a more complicated vehicle.

Nevertheless, Sayers and Riley (1996) make the point that some simplifications had been made that introduce small errors. They conclude that the Yaw/Roll equations contain a subtle error related to the coupling that accounts for about 2% of the difference from the equations derived by AUTOSIM. This may additionally account for some of the differences in the outputs from this study.

b) Truck/Trailer

The same variables used to compare the B-double step steer responses were used for the truck/trailer. The lateral accelerations and roll angles are shown in Fig. 8. The agreement in the outputs between the modelling packages for the truck/trailer is not as close as for the B-double. Even though the small amplitude oscillations matched almost exactly, there is a small constant offset between the program outputs; Yaw/Roll and AUTOSIM consistently showing lower values for the variables considered.

While the tyre models may account for part of this discrepancy, there are other differences between models such as the detail and treatment of the pin coupling in Yaw/Roll. In addition, the error in the hitch equations referred to in Sayers and Riley (1996), discussed above, may also account for some the observed difference.

The differences in the steady state values for roll angle are 2.8% for the truck and 8.5% for the trailer, and for yaw rate and lateral acceleration the differences are, respectively, 7.0%, and 7.5%. Given there is uncertainty of a few percent about the output from Yaw/Roll this match is still considered to be acceptable.

4.2 Closed-Loop Control

4.2.1 SAE Lane Change

a) B-double

The simulations presented in this part of the paper deal with the manoeuvres that require use of the steer controllers. The steer controller can be considered a mathematical equivalent of a driver trained to perform a specific task. Just as there are differences in steering performance between drivers, there will also be subtle

⁷ The assumption of small pitch angles that is used in the Yaw/Roll program was also used in the AUTOSIM models created by TERNZ but not in the ADAMS models created by RTDynamics.

differences between the performances of steer controllers created by different modellers. The task of the steer controller in the SAE lane change is to follow a prescribed path.

In the SAE lane change test procedure, the path of the vehicle, measured at the centre of the steer axle, is required to not deviate by more than ± 150 mm from a precisely prescribed path over a distance of 61m. The paths followed by the B-double models are shown in Fig. 10(a) together with the path specified in SAE J2179 (Society of Automotive Engineers, 1993). All three models are within the specified path tolerance, as revealed in second plot shown in Fig10(a), showing the lateral error in position is within a range of ± 15 mm for Yaw/Roll and ADAMS, and within ± 30 mm for the model created in AUTOSIM. This level of precision in path following performance would be challenging for a driver to consistently achieve in practice.

The influence of the slightly different steer paths is compared in Figs 10(b) in the yaw rates and lateral accelerations for the three models. The yaw rates show the responses are similar; the most significant differences occurring on the peaks; of the order of 10 to 15% on the major peaks and much larger on some of the smaller peaks. Also evident is some high frequency in the Yaw/Roll and AUTOSIM responses due to activity of their respective steer controllers; seen more clearly in the lateral acceleration trace which is the second plot in Fig. 10(b).

b) Truck/Trailer

The two variables used to compare the B-double responses in the lane change for the truck/trailer are shown in Fig. 11. The first plot in Fig. 11 shows that all three models meet the path requirements. There is less error in the steer controllers for the ADAMS and Yaw/Roll models in this task than AUTOSIM.

It was also observed that the errors in lateral position are larger for the truck/trailer models than for the B-double models, suggesting the prescribed path task for this vehicle is more challenging for the steer controllers. Furthermore, it was also noted that the steer inputs for the truck/trailer were different to those for the B-double; the truck/trailer controller is required to be more active for the duration of the run, there are a larger number of path error reversals, and the nature of the error is more erratic indicating a dynamically more active vehicle. This is seen very clearly in Fig. 12, which compares the steer angle inputs from the steer controller to the ADAMS truck/trailer and B-double models in the same manoeuvre. The number and size of the steering reversals is much greater for the truck/trailer, indicating there is a corresponding increase in "workload" imposed on the steer controller.

A driver in a real-world lane change task matching this one may be exposed to a similar increase in workload. The contrasting examples of the B-double and the truck/trailer illustrated suggests that the number and size of the steering reversals is likely to be correlated with configuration and class of vehicle and may provide a useful vehicle-specific measure of driver workload or some other vehicle-performance related issue. Steering reversals have been used in driver behaviour studies as a measure of driver workload and performance, including driver fatigue (McLean and Hoffmann, 1975).

The increase in steering activity leads to a more dynamic response, as evidenced in the yaw rate plot in Fig. 11. The agreement in the outputs from the models is clearly affected by the now larger differences in the magnitude and sequencing of steer input between the controllers.

These results emphasise the central role of the steer controller in closed-loop prescribed path tasks, suggesting the path tolerance specification for the SAE lane change may be too large for computer-based modelling. This also suggests that the current path tolerance may also be too large for physical testing of heavy vehicles leading to significant variability between runs (discussed later in the paper).

4.2.2 Low-Speed Turn

The vehicle factors influencing responses in a low-speed turn are different to those in an evasive manoeuvre such as the lane change. In a low-speed turn the vehicle mechanical properties such as mass, suspension and tyre characteristics play only a very minor role because the (inertia) forces acting on the masses and on the suspensions due to motion are no longer large enough to be significant. In a low-speed turn wheelbases and the location of couplings play a significant role in determining offtracking performance.

For the above reasons, the task of the steer controller in a near zero-speed path following task is different to the lane change. As such, the steer controller – although still required to follow a path – is now dealing with a system possessing essentially no dynamics, and its characteristics would be required to be adjusted or adapted accordingly.

a) B-double

The paths of the steer, drive and trailer axles from the B-double low-speed turn simulations are compared in Fig. 13 and it to be of the order of a few percent.

b) Truck/Trailer

The same variables are compared for the truck/trailer in Fig. 14 showing a similar level of agreement between the models.

4.3 Performance Measures

4.3.1 Comparisons

Direct comparison of the traces from the simulations presented in the previous Sections of this paper provides a very detailed view of how well the responses match across a large number of variables in four separate and distinct simulations. However, in order to help the end-user to determine whether or not to consider the performance of a vehicle as acceptable, the outputs from the simulations are usually condensed into one or several numeric values corresponding to one or a number of different performance measures. Several performance measures from Prem et al (2001a) are normally calculated from the outputs for the same or similar manoeuvres described in this paper, and the B-double and truck/trailer models have been compared in this way in Tables 1 and 2 using the safety-related performance measures that are listed.

Multiple entries for rearward amplification and load transfer ratio appear in Tables 1 and 2. These entries correspond to calculations based on the outputs from the SAE lane change and pulse steer simulations. Rearward amplification and load transfer ratio can be calculated not just from the SAE lane change but also for any manoeuvre involving an abrupt steer input, though the results and threshold levels of acceptability may not be directly comparable with those from the SAE lane change.

The values in Tables 1 and 2 indicate percentage differences as low as 1.9% and generally well below about 7% for the majority of the measures from the simulations, except for the truck/trailer SAE lane change simulations, which shows a maximum difference between the models in excess of 30% on two of the measures. This is a direct result of the greater deviation from the desired path taken by one of the models and suggests that to achieve acceptable levels of agreement, equal to or better than say 10%, a path tolerance smaller than ± 150 mm will be required.

For the B-double simulations a deviation of less than ± 30 mm led to differences between models in the SAE lane change of 6.1%, 3.3% and 7.3%, respectively, for the high-speed transient offtracking, rearward amplification and load transfer ratio performance measures.

Using the above as a guide, it is recommended that for computer-based models proper execution of the SAE lane change requires that the front axle of the vehicle does not deviate by more than ± 30 mm from the desired path defined by the test course.

This conclusion is equally applicable to physical testing and suggests the current tolerance on the path for proper execution of the test manoeuvre may lead to significant variability in test results. It is recognised that a path tolerance of ± 150 mm may be difficult to achieve in practice, and a tolerance one-fifth this value will be more difficult to achieve consistently, if at all. A statistical approach may need to be considered.

4.3.2 Concluding Comments

The pulse and step steer simulations provide a direct measure of vehicle responses to precisely defined steer inputs in the absence of a steer controller. Tables 1 and 2 show that there was closer agreement between the performance measures for the pulse steer simulations than for the SAE lane change.

Given the above, the pulse and step steer tests could be considered as viable alternatives to the SAE lane change, having the benefit of avoiding introducing additional differences from the steer controllers that are necessary to perform the prescribed path lane change. The tests would provide very precise and repeatable steer inputs to the vehicle, and excite them over a range of frequencies, rather than a single frequency (0.4Hz) that may coincide with a lightly damped resonant response in a particular vehicle.

Test methods do exist for evaluating the road holding ability of passenger cars that are based on pulse, step, random and sinusoidal steer inputs (International Standards Organisation, 1988), and similar tests involving pseudo-random and pulse steer inputs recently have been proposed for heavy vehicles (International Standards Organisation, 2000).

5. SOURCES OF VARIATION AND SOME PRACTICAL CONSIDERATIONS

5.1 Sources of Variation

5.1.1 Input Datasets

One of the key inputs to any modelling work is the dataset describing the mechanical properties of each component. It is not a trivial task to collate, interpret and make use of the input data describing each and every part in the model. In some cases these data will not be available and will either need to be estimated or physically measured.

If the datasets are either incomplete or the input data are inaccurate, the predictions from the models will either be inaccurate or in the worst possible case the predictions will be meaningless. The level of accuracy is an important issue and some parameters will have a much greater influence on the accuracy of the models than others. The key parameters should be identified either from previous similar studies or through parametric studies, and every attempt made to obtain accurate input data for all of the important components.

5.1.2 Component Models

The component models are another source of variation that can lead to differences between the outputs from simulations. In the study of vehicle dynamics, the tyres and the suspensions are key components having a significant influence on vehicle responses. It is therefore essential to include all of the necessary properties in the component models.

Piecewise linear interpretations versus continuous functions for tyres, for example, as illustrated in this paper, can lead to differences in the estimation of tyre forces – though this can be addressed by simply using more data and better defining the component model characteristics. However, inappropriate simplifications in models, such as the use of linear characteristics for tyres and suspensions in manoeuvres where the components will be operating in the non-linear regions will lead to errors. Further, using the component models outside of the range of the measured data on which they are based will also lead to inaccurate outputs from simulations.

5.1.3 Vehicle Models

The number of components that are used to define the model, and the way the various component models are connected to each other, referred to as the topology, will define the overall vehicle model. The level of detail in the model and number of separate component models required will be dictated by the problem at hand; highly detailed models do not necessarily produce more accurate results, and a high level of detail may not be necessary. Further, if a high level of detail has been included it will require more input data and the computer simulation will take longer to run.

In the study described in this paper the topologies of the various models were not defined, each Consultant determined model detail by referring to the input dataset and the descriptions provided in the accompanying text.

5.1.4 Test Manoeuvres

If meaningful comparisons of the outputs from simulations are to be made, and vehicle performances compared, then the test manoeuvres that are simulated must be identical. Small differences in the specification of a test (size of steer input, test speed or path followed) can lead to significant differences in outputs from simulations. In the study described in this paper each Consultant used the same precisely defined test manoeuvres.

5.1.5 Modelling Packages

The modelling package (software) should not lead to any significant differences in simulation outputs nor should it introduce any errors. The software should be robust, free from coding errors and thoroughly tested, preferably in a wide range of applications or at least in the application in which it will be used.

As discussed in this paper, it was assumed that the three modelling packages used in this study would not need to be verified or validated against experimental tests because they have been verified in other studies. For the models and simulations carried out, excellent agreement in the outputs from the modelling packages was demonstrated, except in the high-speed prescribed path simulations where the path deviations led to poor agreement between models.

5.1.6 Post Processing

The end-user may require the outputs from the simulations to be condensed into one or several numerics corresponding to one or a number of different performance measures. In order to prevent errors from occurring when computing a particular performance measure, the same outputs from the simulations should be used and the same (or similar) calculations performed. This requires both a clear description of the outputs that will be used in the calculations and a clear description of either the actual calculation or the performance measure. If these considerations have not been addressed adequately there is chance of introducing errors that may range in significance from small to substantial, the latter occurring in the worst possible case.

5.2 Practical Considerations

While the modelling packages may be free from error and perform "as advertised", the predictions from the models created by a particular Service Provider will strongly depend on the Provider's ability to create a realistic and accurate model. The model will need to contain all the necessary features that will ensure accurate simulations and robust predictions, as well as conduct simulations in the standard agreed test manoeuvres to the specified tolerances.

The final test of whether simulation outputs are adequate can be determined when the predictions from the model are compared to actual measurements derived from physical testing. When computer-based modelling has been used to support a new design, final "sign-off" should only occur when the computer-based performance predictions have been confirmed in practice.

6. SUMMARY

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

Time histories of a wide range of variables from the simulations were compared as well as numeric values from a selection of performance measures. For the more stable of the two vehicle models, the B-double, the time histories from the pulse steer and step steer simulations were in close agreement between all three modelling packages. Agreement in the outputs from the simulations in all manoeuvres was generally better than 8% for the performance measures considered. These were marginally influenced by the characteristics of the steer controller in the SAE lane change though agreement was still generally better than 8%.

The truck/trailer model, representing a less stable and dynamically more active vehicle compared to the B-double, produced larger but acceptable amounts of variation in the pulse and step steer simulations and low-speed 90° turn. However, in the SAE lane change the differences between the models were much too large as a result of the greater deviations in the path followed. To achieve acceptable agreement in the SAE lane change between models, a deviation from the desired path not greater than ± 30 mm is required and is recommended.

Simulations that provide a direct measure of only the vehicle responses to precisely defined steer inputs generally lead to consistent results than simulations that require steer controllers and closed loop path following. When there

is a choice, open loop manoeuvres should be selected in preference to closed loop manoeuvres that require the use of steer controllers.

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TABLES & FIGURES

Test ¹	Performance Measure	ADAMS	Yaw/Roll	AUTOSIM	Max % Diff
PS ²	Yaw Damping Coefficient3	0.545	0.550	0.559	2.6
PS	Rearward Amplification (-)4	0.873	0.890	0.887	1.9
PS	Load Transfer Ratio (-)	0.773	0.729	0.729	6.0
SAE	High-Speed Transient Offtracking (mm)	347	337	330	5.2
SAE	Rearward Amplification (-)	1.304	1.266	1.300	3.0
SAE	Load Transfer Ratio (-)	0.380	0.369	0.354	7.3
LST	Low-Speed Offtracking (mm)	5518	5658	5513	2.6

Table 1 - Performance Summary for the B-double

Test ¹	Performance Measure	ADAMS	Yaw/Roll	AUTOSIM	Max % Diff
PS ²	Yaw Damping Coefficient3	0.238	0.270	0.263	13.4
PS	Rearward Amplification (-)4	1.269	1.163	1.230	9.1
PS	Load Transfer Ratio (-)	0.982	0.961	0.931	5.5
SAE	High-Speed Transient Offtracking (mm)	840	635	663	32.3
SAE	Rearward Amplification (-)	2.252	1.895	1.889	19.2
SAE	Load Transfer Ratio (-)	0.943	0.923	0.707	33.4
LST	Low-Speed Offtracking (mm)	2117	2174	2190	3.4

FOOTNOTES:

 PS denotes Pulse Steer, SS denotes Step Steer, SAE denotes the SAE lane change, and LST denotes the Low-Speed 90° Turn.

2) The pulse steer used in this study is more aggressive than the one specified in Prem et al (2001a).

3) Calculation based on the amplitude of the second and fourth peaks of the yaw rate response.

4) Rearward amplification for the pulse steer is based on the ratio of sprung mass lateral accelerations.



Fig. 1(a) Computer-based model of the reference B-double (RTDynamics' model shown).



Fig. 1(b) Computer-based model of the reference truck/trailer (RTDynamics' model shown).



Deflection (mm)



Fig. 2(a) Force/deflection relationship for the truck and prime-mover springs (adapted from Ervin and Guy, 1986).



Fig. 3(a) Side force characteristics for a Fig. 3(b) Aligning moment characteristics for a Michelin XZA 11R22.5 truck tyre (from Ervin and Guy, 1986).





Michelin XZA 11R22.5 truck tyre (from El-Gindy and Kenis, 1998).



Fig. 4(a) Tyre slip angles and vertical forces from the B-double pulse steer simulations.



Fig. 4(b) Tyre self-aligning moment and suspension forces from the B-double pulse steer simulations.



Fig. 4(c) Lateral acceleration and roll angle from the B-double pulse steer simulations.



Fig. 5(a) Influence of tyre model on side Fig. 5(b) Influence of tyre model on aligning



Fig. 6(a) Slip angles and yaw rate from the truck/trailer pulse steer simulations.



Fig. 6(b) Lateral acceleration and roll angle from the truck/trailer pulse steer simulations.



Fig. 7 Lateral acceleration and roll angle from the B-double step steer simulations.



Fig. 8 Lateral acceleration and roll angle from the truck/trailer step steer simulations.



Fig. 9(a) Lateral acceleration response (from Sayers and Riley, 1996).

Fig. 9(b) Roll angle response (from Sayers and Riley, 1996).

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Fig. 10(a) Steer paths and steer axle lateral position error for the B-double in the SAE lane change.



Fig. 10(b) Yaw rate and lateral acceleration for the B-double in the SAE lane change.



Fig. 11 Steer axle lateral position error and yaw rate for the truck/trailer in the SAE lane change.



Fig. 12 B-double and the truck/trailer steer angle inputs in the SAE lane change simulations (RTDynamics' ADAMS models).



Fig. 13 Trajectories and offtracking for the B-double in the low-speed turn.



Fig. 14 Trajectories and offtracking for the truck/trailer in the low-speed turn.