Standard Test Procedures For The Lateral Stability Of Heavy Vehicle Combinations

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ABSTRACT

In October 1992, Subcommittee 9 for Vehicle Dynamics and Road Holding Ability (SC9) of Technical Committee 22 for Road Vehicles (TC22) of the International Standardization Organisation (ISO) began an effort to develop standard test procedures for evaluating dynamic performance of heavy vehicles. The first result of this work, carried out by Working Group 6 for Dynamics of Heavy Vehicles (WG6)¹, is a draft proposal for lateral stability test procedures for heavy commercial vehicles and articulated buses $[1]^2$. The draft document describes standard test procedures for measuring rearward amplification, dynamic offtracking, yaw damping including mode shape information and zero damping speed. Test procedures based on pseudo random steering, single lane change, and pulse steering inputs are included in the proposed standard.

This paper reviews the proposed test procedures and the rationales on which it is based. The paper reports findings of simulation and test programs conducted by UMTRI and Volvo in support of the development of the standard. The results of these studies reveal the sensitivities of measurement results to a variety of test methodology issues.

INTRODUCTION

The lateral stability of heavy vehicle combinations is a most important part of active vehicle safety. There is consequently a need to be able to objectively test and verify the stability characteristics of these vehicles.

In order to meet the ever-increasing demand for transportation of goods, the vehicle weights are increasing and new types of vehicle combinations are being developed. This development emphasizes the need for standardized and internationally recognized test procedures. There is also a growing focusing on traffic safety in many countries and an interest in such established methods.

Therefore, in October 1992, SC9 of TC22 of the International Standardization Organization began an effort to develop standard test procedures for evaluating the dynamic performance of heavy vehicles. This work was assigned to WG6 and the first result is a draft proposal for lateral stability test prodedures for heavy commercial vehicles and articulated buses.

STABILITY CRITERIA

Stability in this context refers to oscillatory stability. Four different criteria are defined. Not all of them are suitable in all cases. Their applicability depends on the type of vehicle combination that is tested.

REARWARD AMPLIFICATION

Rearward amplification is the relationship between the maximum movements of the first and the last vehicle units during some kind of manoeuvre. It is usually given in terms of lateral acceleration or yaw velocity gain.

This criterion expresses the increased risk for a rollover and a swing-out of the last unit compared to what the driver is experiencing.

OFFIRACKING

Offtracking is the lateral deviation between the path of the front axle of the first vehicle unit and the path of the most severely offtracking axle of the last vehicle unit during a manoeuvre. It gives a measure of the additional space that is required for the last unit in that manoeuvre.

YAW DAMPING AND MODE SHAPE

The yaw damping parameter measured by this method is the relative damping of the lightest damped mode of the vehicle combination during free oscillations.

Mode shape expresses how the different vehicle units move in relation to each other for this mode.

A low level of yaw damping is not desirable in any mode of response of the vehicle combination. Modes

¹ John Aurell is the convenor of Working Group 6, and Chris Winkler is one of the experts of the group.

² Numbers in brackets indicate bibliographic references given at the end of this paper.

whose shape is characterized by an out-of-phase yaw response of succesive units may be particularly troublesome.

ZERO DAMPING SPEED

The zero damping speed is the speed at which the damping of free oscillatory movements of the vehicle combination equals zero. If the vehicle speed exceeds the zero damping speed the vehicle combination becomes unstable, ie the oscillations continue with an increasing amplitude without any steer input.

TEST PROCEDURES

PSEUDO-RANDOM STEER INPUT

The rearward amplification and its dependency on the frequency is determined with this method. It assumes that the vehicle combination can be considered as a linear system. The analysis is made in the frequency domain.

The test is carried out at a constant speed with continuous steer inputs made to the steering-wheel up to a certain steering-wheel amplitude. This limit is chosen so that the vehicle exhibits a linear behaviour. In order to ensure adequate high-frequency content, the steering input must be energetic. Both the frequency and the amplitude of the steering input may be varied randomly. A spectral analysis of the steer input is used to check that the frequency content of the input is adequate.

Rearward amplification is defined as the gain of the frequency response functions between the signals of the last and the first vehicle units. In processing the data, both the gain and the coherence functions are established. If the vehicle is operated in a range where it exhibits linear behaviour and no extraneous noise is present in the signals, the coherence is close to unity. The measurement is accepted if the coherence is at least 0.95 in the relevant frequency range. The maximum rearward amplification is the maximum gain.

SINGLE LANE CHANGE

Rearward amplification and also the offtracking may be determined with this method. The test is conducted at a constant speed for at least three frequencies. The single lane change may be carried out in two optional ways, with a single sine-wave steering input or with a single sine-wave lateral acceleration input.

Single sine-wave steering input One full period sinusoidal steering-wheel input is applied. In order to get accurate and reproducible results a stering machine is preferable.

Single sine-wave lateral acceleration input In this case the vehicle is steered in such a way that the front axle of the first vehicle unit follows a marked test course, which corresponds to one full period sinusoidal input of lateral acceleration.

In both cases rearward amplification is defined as the relationship between the maximum peak values of the signals of the last and the first vehicle units. Offtracking is measured from markings of the paths of the front axle and the most severly offtracking axle.

PULSE STEER INPUT

The purpose of this test procedure is to determine the damping and mode shape of the least damped yaw response mode of the vehicle combination. By determining the damping for a number of vehicle speeds, the zero damping speed, if any, may be estimated. If all modes of yaw motion are well damped, this method is not applicable.

When the vehicle is driven in a straight line at a constant speed a short steering pulse is applied to the steering wheel. After the steering impulse is completed the steering-wheel is held fixed in the straight ahead position in order to allow the vehicle to oscillate freely.

The damping (D) is obtained from the logarithmic decrement of the amplitudes of the articulation angle of subsequent oscillations (A_i) , equations (1) and (2).

$$r = \frac{1}{n-2} \left(\frac{A_1 + A_2}{A_2 + A_3} + \dots + \frac{A_{n-2} + A_{n-1}}{A_{n-1} + A_n} \right)$$
(1)

$$D = \frac{ln(r)}{\sqrt{\pi^2 + (ln(r))^2}}$$
(2)

The mode shape is calculated from the relationship between the yaw velocities of the vehicle units.

The zero damping speed is determined by linear curve fitting of the plotted values of damping and speed.

DISCUSSION

In a broad sense, the Draft Proposal considered in this paper deals with two basic performance regimes, namely the forced response and the free response of articulated vehicles. That is, of the four general areas of performance measures considered, two, rearward amplification and dynamic offtracking, examine forced response properties and the other two, zero-speed damping, and yaw damping and mode shape, involve free response. The following discussion is subdivided according to these two categories.

FORCED RESPONSE

Fundamental differences between the pseudo random and single lane change methods The two test procedures for the measurement of rearward amplification described above do not produce the same results. The pseudo random procedure is intended to yield a true representation of the system gain in the frequency domain. The single lane change procedure, however, provides only the composite gain of the system as results from a distributed frequency content of the specific lane change performed in the test. Dynamic offtracking is something of a secondary measure in that it can be seen as a symptomatic result of rearward amplification and the geometry of the test vehicle. (Geometry of the vehicle, of course, also influences rearward amplification, itself). The remainder of this discussion, therefore, concentrates on rearward amplification.

For this discussion, rearward amplification (be it concerned with either lateral acceleration or yaw velocity) is defined as the transfer function, or system gain, between an input motion of the first vehicle unit and the output motion of the last vehicle unit. That is:

$$RA(f) = M_{\rm L} / M_{\rm F}$$
(3)

where:

 M_L is the motion of the last vehicle unit, M_F is the motion of the first vehicle unit, RA(f) is the system gain (rearward amplification) for the motion and is a function of frequency, f.

Determining rearward amplification by experiment involves implementing equation (3). That is, an experiment is arranged in which some motion of the first vehicle unit causes a forced response of the last vehicle unit. Both motions are measured and are "divided" in order to yield a representation of the system gain, known in this case as rearward amplification.

As is generally the case for dynamic systems, the gain of articulated vehicles is known to be a function of frequency. That is, the magnitude of amplification differs as the frequency content of the forcing function (the motion of the first vehicle unit) differs. Thus, to fully evaluate the RA(f) function experimentally, the motion M_F must have a frequency content spanning the range of interest, and the analysis represented by (M_L / M_F) must be accomplished in the frequency domain.

The pseudo random test procedure described herein is meant to meet these criterion and yield a full description of rearward amplification as a function of frequency. This test procedure requires a sustained steering input with significant power throughout the frequency range of interest. During the test, the relevant motions of the first and last units are measured and the full representation of the system gain is obtained by proper analysis of this data in the frequency domain.

Conversely, when M_F has limited frequency content and the analysis is done in the time domain, as is the case for the single lane change test procedure, the resulting measure of amplification is not general, but is specific for that test manoeuvre and its frequency content. Some evaluation of the influence of frequency on rearward amplification can be obtained by performing multiple tests spanning a range of lane change periods. However, as will be explained, this does not yield a true representation of the system gain when the individual test results are evaluated in the time domain.

A single lane change manoeuvre contains distributed frequency content. As is well known, the motion (lateral acceleration or yaw velocity) time history of a perfectly executed single lane change takes the form of a single sine-wave which is preceded and followed by quiescent periods. While the sine-wave itself could be said to be pure, the pre and post periods of quiescence, and the associated sharp corners of transition into and out of the wave form, introduce a distribution of frequencies. Thus, the total content of the manoeuvre spans a range of frequencies with the power of this distribution centred, but not purely concentrated, at a frequency close to the frequency associated with the single wave. This is illustrated in figure 1.

As a consequence of the distributed frequency content of the input, the time-domain analysis of the single lane change manoeuvre does not yield a pure measure of system gain at the centre-frequency of the lane change. Rather, it yields a weighted-average of the gain in the vicinity of the centre frequency of the manoeuvre. The weighting function is determined by the distribution of frequency content in the tractor motion time history. This is illustrated conceptually in figure 2.

Both a brief computer simulation study^1 and limited vehicle testing have been conducted to confirm this interpretation of the difference between the pseudo random and single lane change testing methods.

The computer study involved a US-style doubletrailer combination consisting of a tractor and two 8meter trailers. Via simulation, rearward amplifications (tractor to second trailer) were measured both in the frequency domain and in the time domain, using the pseudo random and the single lane change methods, respectively.

The results of the simulation study appear in figure 3. As expected, the single-lane change method, produces rearward amplification measurements which, when displayed as a function of the centre-frequency of the manoeuvre, appear to "smooth" the actual system gain in the frequency domain. Again, this is due to the fact that the single lane change manoeuvre has a frequency content distributed about the centre-frequency, rather than a pure content at that frequency.

Vehicle tests were performed with a vehicle combination, consisting of a three-axle tractor, a threeaxle semitrailer and a three-axle centre axle trailer, with a GCW of 64 t. Both path-following lane changes and

The simulation programs used for all of the simulation work discussed herein were developed using AUTOSIMTM.[2] All are constrained to constant forward velocity but are relatively complete in treatment of yaw, roll, and lateral motion. They each include non-linear tire and suspension spring models.

pseudo random tests were carried out. All tests were conducted at a speed of 80 km/h. The rearward amplification of lateral acceleration and yaw velocity from these two test methods are shown in figures 4 and 5, and it appears that these results confirm the simulation results.

As a final point of interest in this section of the discussion, we note that some researchers have reported that the double lane change manoeuvre seems to elicit more dramatic rearward amplification response than does the single lane change manoeuvre . (See in particular, the work of the Swedish Road and Traffic Research Institute as reviewed in reference [3].) Analyses in the frequency domain show that the frequency content of the idealized wave form of the double lane change manoeuvre is more closely concentrated about the centre frequency than it is for a single lane change manoeuvre. See reference [4]. Thus, the experimental observation would seem to be readily explained by the concepts discussed here.

Comparison between the two types of single lane change In order to investigate the difference between the path-following lane change where the lateral acceleration is prescribed and the lane change with the steering input prescribed, simulations were conducted with a vehicle model of a combination, consisting of a three-axle truck, a two-axle dolly and a three-axle semitrailer, with a GCW of 62 t. The model has yaw and lateral translational degrees of freedom. The test speed was 80 km/h. The rearward amplifications of lateral acceleration and vaw velocity as a function of the frequency of the one period sinusoidal are shown in figures 6 and 7. It appears from the results that the lateral acceleration gains for this vehicle are very similar. For the yaw velocity gains however the difference is approximately 10 %. These results are not general. The results may be different for different vehicle combinations. Obviously the results from the two different lane change manoeuvres should not be expected to be identical. The difference between them is however smaller than the difference between lane change and pseudo random steering input.

Sensitivities of rearward amplification measurement During the development of the standard test procedures, the sensitivity of measured rearward amplification to factors other than the general test method were also examined. Some of the influences investigated were those due to manoeuvre severity, instrument type and location, and data reduction method. Results appear graphically in figures 8through 13. All of these results derive from simulation studies of the same US-style double, and all are tractor-to-second-trailer amplifications. Brief comments on the results follow.

Figures 8 and 9 show the influence of manoeuvre severity on rearward amplification of lateral acceleration and yaw velocity, respectively. These results were obtained from single lane change tests. Each show a moderate sensitivity of the result to the amplitude of input. The influence derives largely from the nonlinearity associated with saturation of the lateral capability of the rear most tyres of the vehicle. As the level of excitation at the tractor increases moderately, severe traction demands may develop at the tyres of the last trailer due to amplification of response motion. As these tyres begin to saturate, lateral acceleration of the last trailer also tends to be limited. Thus, the measured rearward amplification of lateral acceleration declines for more aggressive manoeuvres. Conversely, saturation of the lateral traction of the tyres of the last trailer tends to further exaggerate the yaw motions of this unit. Thus, this measure of rearward amplification increases for the more severe manoeuvres.

The column graph of figure 10 illustrates the potential influence of accelerometer mounting and location on measured rearward amplification. These data derive from single lane change tests. The simulated vehicle was "equipped" with four accelerometers in each unit. Two were frame-fixed; that is, their sensitive axis rolled with the vehicle. The other two were not allowed to roll, but always measured in the horizontal direction as if mounted on a gyroscopically stabilized platform. One of each type was located in the sprung mass at its centre of gravity. The others were located in the sprung mass at the same longitudinal position, but at the height of the roll axis. As expected, the data show that the signal of the frame-fixed accelerometer is significantly influenced by a gravitational component, especially in the trailer which suffers greater rolling due to generally higher accelerations. Further, the roll motion itself significantly contributes to measured accelerations, depending on the mounting height of the accelerometer.

Figure 11 illustrates three methods of determining rearward amplification from the response time histories of a single lane change manoeuvre . In real testing, it is not possible for the driver (or steering machine) to produce a perfectly simsoidal wave form at the tractor. Further, the trailer response is usually asymmetric with the second half of the wave having greater absolute magnitude and longer duration than the first.

Since trailer rollover is a primary concern, maximum absolute trailer lateral acceleration has usually been used to characterize trailer response, even though the behavior is asymmetric. However, researchers have used several methods to characterize the tractor input in an effort to yield more consistent measurement of amplification.[5,6,7]

In the three data reduction methods illustrated here, tractor motion is characterized by (1) the maximum absolute value from the trace, (2) the average of the absolute values of the positive and negative peaks, and (3) an "equivalent peak" value derived from the RMS value of the tractor signal.

Figures 12 and 13 show results obtained for the three data reduction methods applied to lateral acceleration and yaw velocity behavior of the simulated US-style double. The graphs show a significant spread in results

obtained by the three methods. Particularly with respect to yaw velocity, however, the average-peak and RMSpeak methods do tend to agree fairly well.

FREE RESPONSE

The pulse input test method described above is intended to elicit a free response of the articulated vehicle under study, in order that certain of its damping and mode shape characteristic may be observed.

Articulated vehicles have several natural modes of yaw response. In a yaw plane analysis, the number of natural modes is equal to the number of yaw articulation joints plus one. The modes of yaw plane motion of an articulated vehicle vary in damping characteristics and mode shape. Typically the shapes derive from combinations in which successive units yaw more-orless in phase with one another or more-or-less out of phase. The in-phase modes tend to be well damped while the out-of-phase modes may not be. The equations of motion of articulated vehicles with full trailers typically decouple at the pintle hitch joint so that the natural modes of the full trailer and its towing unit (truck or tractor semitrailer) are independent. (For examples, see references [2,6,7]).

In theory, the pulse steering input contains all frequencies and is intended to generate responses of all modes of yaw motion. Thus, early on in the pulse input test, the vehicle response should contain all modes of yaw motion. However, motions of the well damped modes die out quickly leaving only the less damped, oscillatory modes. In practice then, the procedure tends to be most useful for characterizing the least-damped mode of the vehicle (or, possibly, least damped modes if the vehicle decouples).

Tests with the previously mentioned vehicle combination, consisting of a three-axle tractor, a threeaxle semitrailer and a three-axle centre axle trailer, were performed in order to determine the zero damping speed. The damping was calulated from the signal of the articulation angle between the trailers at a number of vehicle speeds, starting at the lowest one. These dampings were plotted versus the speed according to figure 14. The zero damping speed was then estimated by linear regression.

In developing the specifications for this method, it was noted by members of the working group that, in experimental work, the strength of the oscillatory response of some vehicles seemed to be sensitive to the longitudinal slope of the test course. A brief simulation study was undertaken to examine the influence of slope on yaw damping of articulated vehicles.¹

The primary vehicle for this study was a 21-tonne, three-axle truck towing an 18-tonne, two-axle centre axle trailer. This vehicle was subject to pulse steer runs on slopes ranging from -3 percent (downhill) to +3 percent (uphill).

Figure 15 presents time histories of the lateral acceleration response of the trailer of this vehicle in

simulated pulse steer tests on surfaces of plus and minus 3 percent, respectively. A substantial difference in response is readily apparent with the motion of the vehicle being obviously more damped when traveling uphill rather than downhill.

The vehicle which generated the data of figure 15 was arranged to be relatively lightly damped by locating the coupling 3-meters after of the truck drive suspension and by making the hitch load only 5 percent of trailer weight. Similar simulated test runs were also made with two other, more heavily damped vehicle. One was another version of the truck and centre axle trailer with half the hitch overhang and twice the hitch load. The third vehicle was the US-style double.

Figure 16 is a plot of the measured yaw damping ratio as a function of the longitudinal slope of the test surface, for all three of these vehicles. Although the level of damping for the three vehicles is quite different, all show a similar sensitivity of damping to the longitudinal slope of the test surface. The sensitivity appears to be strongest when the level of damping is lowest. The sensitivities range from 2.2 percent damping per percent slope for the lightly damped truck trailer to 1.2 percent damping per percent slope for the well-damped US double. (Note that the US-style double is very well damped in its yaw response. Distinguishing the difference in damping shown in figure 16 by actual test would not be practical for this vehicle, but is possible using "noiseless" simulation.)

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¹ The simulation used for this task allowed constant velocity operation on a sloped plane. The constraint force needed to maintain constant velocity of the entire vehicle was applied at the center of gravity of the tractor. Thus, the trailing units were towed or restrained by longitudinal forces at the couplings. Aerodynamic and rolling resistances were not included.

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Figure 4. Lateral acceleration gain of a tractor, semitrailer and centre axle trailer combination measured by the pseudo random and path-following lane change methods



Figure 2. Experiments using a series of single lane changes yield a "smoothed" version of the system gain.



Figure 3. Rearward amplification of a simulated USstyle double as measured by the pseudo random and single lane change methods



Figure 5. Yaw velocity gain of a tractor, semitrailer and centre axle trailer combination measured by the pseudo random and path-following lane change methods



Figure 6. Lateral acceleration gain for two different lane change manoeuvres



Figure 7. Yaw velocity gain for two different lane change manoeuvres



Figure 8. The influence of manoeuvre severity on rearward amplification of lateral acceleration in a single lane change manoeuvre



Figure 9. The influence of manoeuvre severity on rearward amplification of yaw velocity in a single lane change manoeuvre



Figure 10. The influence of accelerometer type and mounting height on rearward amplification of lateral acceleration in a single lane change manoeuvre











Figure 13. The influence of data reduction method on rearward amplification of yaw velocity in a single lane change manoeuvre











Figure 16. Yaw damping ratio of three vehicles as a function of the longitudinal slope of the test surface

Path Compliance In Lane-Change Tests Designed To Evaluate Rearward Amplification

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ABSTRACT

An ISO working group is currently developing a draft proposal for a standard to evaluate the rearward amplification of heavy combination vehicles. The work is being conducted in the wake of recent efforts that established an SAE Recommended Practice on the same subject. The SAE RP identifies a physical lane-change manoeuvre designed to apply a sinusoidal lateral acceleration to a tractor that closely follows the path, and it then defines the manner in which measurements of tractor and trailer lateral accelerations can be used to estimate rearward amplification. The draft ISO proposal is expected to mirror this SAE manoeuvre very closely.

Both the SAE RP and the draft ISO proposal identify an allowable path tolerance whereby an individual test is deemed successful if the tractor steer-axle follows the prescribed path within ± 150 mm. This paper uses a variety of physical test results and computer simulations to demonstrate the degree to which the magnitude of this tolerance affects the uncertainty of the test results, and it shows that consideration should be given to using a path tolerance that is a function of the desired path frequency and amplitude.

INTRODUCTION

The evaluation of heavy commercial vehicle dynamic performance has been the focus of various research efforts for more than thirty years (eg., Jindra 1965, Hazemoto 1973, and Nordstrom and Strandberg 1978). The establishment of fairly detailed computer simulation models in the late 1970's and early 1980's (Mallikarjunarao and Fancher 1978, and Winkler et al, 1981) and subsequent work in applying and interpreting the results of simulations and physical tests have led to the development of a variety of vehicle performance measures designed to characterize the relative safety of various heavy vehicle configurations. Among these is a "rearward amplification" measure that evaluates the degree to which a combination vehicle exhibits large, lightly damped, trailer yaw motions in response to rapid steering input.

Rearward amplification is usually described as a ratio of the lateral acceleration of the rearmost trailer to that of the tractor. The precise definition varies slightly in different applications, and the measure is occasionally extended to consider the rearward amplification of yaw rate as well as lateral acceleration. Despite these variations, it is always used as an indicator of the degree to which rear vehicle units respond to lateral tractor excitations.

In 1982 Ervin and MacAdam reported the results of computer simulations that were used to examine the rearward amplification of ten vehicle configurations, and they showed the extent to which rearward amplification is highly dependent on the frequency of the steering input. Rearward amplification has remained an important indicator of vehicle performance through a variety of subsequent evaluation programs (e.g., Ervin and Guy, 1986), and through a number of proposals for future regulatory scenarios that would require specific levels of performance for vehicles of certain types (e.g., Fancher 1989, El-Gindy and Woodrooffe 1991).

Physical measurements of rearward amplification been made by a broad spectrum of researchers over the years, and in 1992 Winkler et al. outlined a specific test procedure that coalesced a considerable degree of the experience that had been gained in this practice. In their report they recommended that the test vehicle follow a single desired path designed to apply, to a point that precisely follows the path at the correct speed, a sinusoidal lateral acceleration with a peak value of 0.15 g (where 1 g = 9.81 m/s²) and a frequency of 0.4 Hz.

In spite of the well-known frequency-dependence of rearward amplification, the report recommended that the physical tests be conducted at a single frequency for two principal reasons. The first of these was that the main purpose of the test was to obtain an estimate of the peak rearward amplification of a particular vehicle, and a broad

variety of simulation and field measurements had shown that the peak rearward amplification of most heavy combination vehicles occurs around 0.4 Hz. The second compelling factor was the attractive degree of simplification that could be obtained by conducting tests at a single path frequency rather than over a range of frequencies.

Despite this, however, the authors of the report considered the test procedure to be sufficiently general that they encouraged its extension to other frequencies and amplitudes where circumstances warranted. In 1993, the Society of Automotive Engineers published J2179, a Recommended Practice for evaluating rearward amplification that was closely based on the procedure in the 1992 report.

Since that time, an ISO Working Group (ISO/TC22/ SC9/WG6) has met on a number of occasions to develop recommendations for a broader set of lateral stability test procedures for heavy commercial vehicle combinations and articulated buses. A variety of tests are covered in the current draft ISO proposal, including one for a single-lanechange manoeuvre to evaluate rearward amplification in a manner very similar to that defined in J2179. The draft ISO proposal differs, however, in that it recommends that the test be repeated at a number of different path frequencies to allow the peak rearward amplification to be identified from the test results.

Both the SAE RP and the draft ISO proposal recognize that real vehicles cannot follow an ideal lane-change path perfectly, and hence both include provisions for determining whether the path actually followed by a vehicle is sufficiently close to the desired path to be considered acceptable. This paper examines the extent to which test results that use the path tolerances of these two procedures provide an acceptable estimate of on-path performance, and it suggests an alternative tolerance definition that is particularly applicable to the multi-frequency tests currently contained in the draft ISO proposal.

THE SINGLE-LANE-CHANGE TEST

The methodologies defined in J2179 and in the draft ISO proposal require that the steer-axle of the vehicle follow a path that, if followed precisely, would impart a singleperiod sinusoidal lateral acceleration to the point following the path. The ideal path has straight entry and exit regions, and a central lane-change section in which the lateral coordinate y of a point along the path is defined as a function of the longitudinal coordinate x, the peak lateral acceleration a_y , the vehicle speed v, and the frequency f in Hz:

$$y - \frac{a_y}{(2\pi f)^2} \left[2\pi f \frac{x}{v} - \sin\left(2\pi f \frac{x}{v}\right) \right]$$
(1)

The x and y coordinates of the SAE lane-change section are shown as a solid line in Figure 1. The maximum lateral



draft ISO proposal.

displacement y_{max} in a single lane-change manoeuvre depends on the frequency and amplitude of the manoeuvre, and is defined as

$$y_{\rm max} - \frac{a_y}{2\pi f^2} \tag{2}$$

The dashed lines in the figure identify the ± 150 mm tolerance that is defined in J2179 and in the draft ISO proposal as the maximum allowable deviation from the desired path.

The tests are conducted at highway speeds, and the peak lateral accelerations are selected to ensure that adequate levels of vehicle response are obtained without approaching the vehicle rollover limits or tire friction limits too closely. Tests in which the steer axle of the test vehicle fails to remain within the specified tolerance are rejected. The current procedures require a minimum of five acceptable tests along a given path to provide a statistical measure of certainty about the magnitude of the rearward amplification. While the SAE practice confines the test to a single path with a frequency of 0.4 Hz, the draft ISO proposal requires the process to be repeated at frequency intervals of 0.1 Hz or less a sufficient number of times to establish the relationship between frequency and rearward amplification in a manner that identifies the peak rearward amplification.

Rearward amplification, RWA₁₅₀, in the draft ISO proposal is defined as

$$RWA_{ISO} = \frac{\max(|a_{y_trailer_peak_1}|, |a_{y_trailer_peak_2}|)}{\left(\frac{|a_{y_tractor_peak_1}| + |a_{y_tractor_peak_2}|}{2}\right)}$$
(3)

where $a_{y_tractor_peak_1}$ and $a_{y_tractor_peak_2}$ are the two peak lateral

accelerations of the tractor CG, and where $a_{y_trailer_peak_l}$ and $a_{y_trailer_peak_2}$ are the corresponding values for the CG of the rear-most vehicle unit. In practice, the lateral acceleration of the tractor CG is closely approximated by that of the tractor steer axle — a quantity that is much easier to measure because of the limited amount of axle roll that occurs — and this, together with the use of a root-mean-square evaluation of the steer-axle lateral acceleration for improved consistency, is the basis of the definition in the SAE Recommended Practice.

$$RWA_{SAE} - \frac{\max(|a_{y_trailer_peak_1}|, |a_{y_trailer_peak_2}|)}{\sqrt{2} a_{y_steer_axle_rms}}$$
(4)

A third, slightly different definition used later in this paper, combines portions of the SAE and ISO approaches in defining RWA_{st_avg} as the ratio of the same maximum trailer acceleration used in Eq. (1) and (2) to the average of the two peak steer-axle accelerations.

PATH TOLERANCE

THE FIXED 150-MM PATH TOLERANCE

The ± 150 mm path tolerance shown in Figure 1 introduces a degree of uncertainty about the magnitude and frequency of the actual path taken by the steer axle of a vehicle that successfully traverses the course. While actual paths can be expected to have a very wide range of potential characteristics, it is useful to consider a simplified situation in which the actual steer-axle path remains sinusoidal, but where the frequency and magnitude are different from those

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used to define the ideal path. In this case, the envelope BCDE in Figure 2 defines the limits of the various frequency/magnitude combinations that can be used to produce paths that lie within the specified tolerances for a particular ideal path.

The envelope in Figure 2 has been established for an ideal path designed for a vehicle speed of 24.4 m/s (88 km/h), a peak lateral acceleration of 0.15 g, and a frequency of 0.4 Hz (point A in the figure). The inset graphs show the path entries that are associated with various frequency/magnitude combinations at the edges of the envelope, along with dotted lines showing the +150 mm tolerance from the ideal path. For example, the path defined with a frequency of 0.22 Hz and a magnitude of 0.053 g (point E on the BCDE envelope) is shown in both the left and the lowest inset graphs as path E, a path that begins early on the outside tolerance boundary, skims past the inside boundary, and then passes through the centre point of the lane-change on its way to an exit that is a mirror image of the entry. Among these completely sinusoidal paths, path E represents the one with the lowest frequency and amplitude that can be used to negotiate the 0.4-Hz, 0.15-g lane-change in an acceptable manner. Similarly, point C identifies the frequency/magnitude combination for the path with the highest possible frequency.

In practical terms, the paths traced by vehicles conducting a physical test are not restricted to the singlefrequency-and-amplitude content of the paths considered in Figure 2. A single acceptable path traced by a typical



Figure 2. An envelope defining the sinusoidal path frequency/magnitude combinations that can be used to successfully negotiate the J2179 lane-change manoeuvre.

vehicle can be expected to contain a variety of frequency and magnitude components, some of which will range beyond the boundaries of the envelope in Figure 2. In addition, a real path will not necessarily be symmetric about the centre point of the lane change. Nevertheless, the envelope in Figure 2 provides an indication of the general nature of the frequency/amplitude variations that can be expected in paths that will be acceptable for the specified test.

Figure 2 shows that a vehicle could successfully negotiate the J2179 path at frequencies that are substantially different from the nominal path frequency of 0.4 Hz — at frequencies as low as 0.22 Hz and as high as 0.85 Hz, and that the peak lateral accelerations could range correspondingly from the nominal 0.15-g value down to 0.053 g and up to 0.55 g.

Rearward amplification is not strongly related to variations in the magnitude of the peak lateral acceleration, and thus this variation is not of significant concern. As noted earlier, however, there is often a strong relationship between rearward amplification and path or steering input frequency, and the substantial latitude that Figure 2 suggests is available for variations in path frequency evidently makes it important to consider the extent to which these significant levels of uncertainty in frequency typically introduce correspondingly significant errors in estimating the magnitude of the rearward amplification.

Before doing this, however, it is useful to look at the manner in which variations in the level of path tolerance affect the size of the BCDE envelope.

VARIABLE PATH TOLERANCE

The size of the BCDE envelope in Figure 2 is a function of the allowable tolerance, and of the frequency, amplitude, and vehicle speed used to define the desired path. Assuming that the vehicle speed in a physical test can be reasonably well controlled, Figure 3b shows how the overall envelope size varies for desired path frequencies of 0.3, 0.4, and 0.5 Hz. The increase in envelope size that accompanies the increase in frequency occurs as a result of two factors working in unison. The first is that the idealpath maximum lateral displacement y_{ma} becomes smaller with increasing frequency (Eq. (1)). The second is that the fixed path tolerance of ± 150 mm becomes a progressively larger proportion of y_{max} as y_{max} becomes smaller, thus allowing the relative significance of the path deviations to increase.

Figures 3a and 3c show similar variations in envelope sizes for a tighter and a looser path tolerance, respectively, with the obvious result that envelope size increases with increasing path tolerance. The data in these figures show that a more consistent level of uncertainty about the frequency and amplitude of the actual paths taken by test vehicles would be achieved if the required path tolerance



varied inversely with the square of the ideal-path frequency. A similar process can be used to show that the path tolerance should also vary directly with the peak lateral acceleration of the ideal path.

If it assumed that the tolerance of ± 150 mm is appropriate to a 0.15-g, 0.4-Hz path, then the tolerance ϵ , in millimetres, for a nominal path with a peak lateral acceleration a_y expressed in units of g, and a frequency f expressed in Hz, would be defined as

$$150\left(\frac{a_y}{0.15}\right)\left(\frac{0.4}{f}\right)^2$$
 (5)

With a variable path tolerance of this nature, the amplitude/ frequency variations for 0.15-g lane-change manoeuvres at 0.3, 0.4, and 0.5 Hz would be represented by the outer envelope of Figure 3a, the middle envelope of Figure 3b, and the inner envelope of Figure 3c, respectively.

e -

A variable path tolerance as defined by Eq. (5) would make it easier to perform acceptable tests at frequencies below 0.4 Hz, but it also implies the use of considerably stricter tolerances at higher frequencies. At a nominal path frequency of 1.0 Hz, for example, which the draft ISO proposal suggests is the highest frequency of interest for heavy commercial vehicles, the allowable tolerance in a 0.15-g manoeuvre would be reduced to just ± 24 mm. However, this tight tolerance appears to be more reasonable than the fixed ± 150 mm tolerance when considered in the context of the 234-mm maximum lateral displacement that will occur in this particular manoeuvre over a longitudinal lane-change distance of more than 24 m.

MEASURED PATH VARIATIONS

The extent to which the actual paths traced by the steeraxle of a vehicle configuration vary from the ideal paths can be illustrated by examining the results of a number of actual lane-change tests. The results presented below are from tests that were conducted as part of a study to evaluate the accuracy and reliability of a new computer simulation model designed to simulate the lateral and yaw stability of logtruck configurations (Preston-Thomas, 1994).

In conducting the tests, seven runs were made through a nominal 0.15-g, 0.4-Hz lane-change manoeuvre with an instrumented tractor/pole-trailer. Of the seven runs, the results of two are not included here due to technical difficulties encountered in recording the offtracking data. The remaining five all resulted in paths that are considered acceptable according to the ± 150 -mm tolerance of J2179. The steer-axle paths from these five tests are shown as thin, solid lines in Figure 4. The figure shows a relative lack of path consistency, and a marked tendency to negotiate the lane change with actual path frequencies that are somewhat lower than the ideal-path frequency.

Figures 5a and 5b show the corresponding lateral accelerations of the steer axle and the log-payload centre of gravity, along with the steer-axle lateral acceleration that would have occurred if the steer-axle had followed the ideal path precisely. The lack of path consistency is evident in a corresponding lack of consistency among the steer-axle lateral accelerations, although it is apparent that the measured peak lateral accelerations of the steer axle are generally lower than the nominal 0.15-g peak of the ideal path, and that the frequency of the actual path tends to be lower than the nominal path frequency in the exit region of







Figures 5a,b. Measured steer-axle and log-CG lateral accelerations for the vehicle/path combinations of Figure 4.

the manoeuvre.

The lack of path consistency is attributed largely to the difficulty encountered by the driver in achieving smooth and accurate variations in steering input at low levels of lateral acceleration, and contributes to the scatter observed among the five resulting rearward amplification values shown in Table 1.

Figures 6, 7a, and 7b show a set of experimental results for the same vehicle negotiating a nominal 0.25-g, 0.4-Hz lane-change manoeuvre that was devised to achieve higher levels of lateral acceleration and improved estimates of rearward amplification. For this purpose, the path tolerance was left at the fixed value of ± 150 mm recommended in J2179. Of the nine attempts made to negotiate the lanechange manoeuvre, three have been rejected for a lack of compliance with the ± 150 -mm tolerance, and a further one has been rejected for other reasons. The results shown in the Figures and in Table 1 for the remaining five runs demonstrate that the attempt to reduce the rearward amplification estimation errors was successful.

Among the five runs that were used to negotiate the nominal 0.25-g lane-change, none achieved a peak lateral acceleration of as much as 0.25 g, and all show the same tendency to exhibit lower path frequencies in the exit region of the manoeuvre. A rough estimate places the exit-region frequency at approximately 0.2 Hz, a value that is half the nominal path frequency.

Both the 0.15-g and 0.25-g results demonstrate a strong tendency for the actual steer-axle paths to be characterized by frequencies that are close to or lower than the nominal path frequency of 0.4 Hz. The extent to which this affects the accuracy of the estimate of peak rearward amplification must be examined by comparing the result to the rearward-

Nominal Peak Lateral Accel., g	Test	$RWA_{\pi_a, \alpha, q}$, based on:	
		Physical Measurements	Simulation Along the Same Path
0.15	46	1.058	1.363
	47	0.911	1.234
	49	1.188	1.105
	50	0.932	1.253
	51	1.024	1.180
	Mean	1.023	1.227
	CI-L ¹	0.917	1.136
	CI-U ²	1.128	1.318
0.25	58	1.040	1.183
	59	1.034	1.245
	60	1.076	1.186
	62	1.146	1.259
	64	1.067	1.187
	Mean	1.073	1.212
	CI-L ¹	1.030	1.177
	CI-U ²	1.116	1.247

¹Lower limit of 90% confidence interval.

² Upper *

Table 1. Rearward amplification values derived frommeasured lateral accelerations and simulationsalong five 0.15-g paths and five 0.25-g paths.

amplification frequency response of the tractor/pole-trailer — a relationship that was not measured in the tests, and that is thus best obtained through the use of computer simulation.

COMPARISON WITH SIMULATION

Logging trucks such as the tractor/pole-trailer used in the tests outlined above have design features that are







Figures 7a,b. Measured steer-axle and log-CG lateral accelerations for the vehicle/path combinations of Figure 6.



rigures 8a,0. Comparisons between measured and simulated tractor/pole-trailer lateral accelerations for one of the five experimental paths associated with the 0.25-g, 0.4-Hz lane change manoeuvre.

virtually unknown among conventional articulated highway combinations. Principal among these are an additional yaw articulation point and a sliding reach that connects the pole trailer to the back of the vehicle unit ahead of it. The unique geometry of these configurations has been taken into account in the development of a new comprehensive Log-Truck Yaw/Roll model (Tong et. al, 1995) patterned after a Yaw/Roll model developed for conventional articulated vehicles at the University of Michigan Transportation Research Institute.

In both models it is assumed that the forward velocity of the lead unit remains constant, and each of the vehicle units is characterized as a rigid body with five degrees of freedom. The axles are treated as beam axles free to roll and bounce, and the non-linear cornering force and aligning torque characteristics of the tires are represented by tabular input. Both simulation models include closed-loop pathfollower models that generate steering input based on a path perceived with a representative degree of path preview and driver lag.

While the path-follower models are capable of following the nominal lane-change path or one of the measured paths

ARTICULATED VEHICLE BEHAVIOR

to an accuracy of approximately ± 40 mm, a higher degree of accuracy was desired for the purpose of examining the path-dependent variability of the results. To achieve this, a two-iteration procedure was developed in which the vehicle was first asked to track the desired path, and was subsequently required to track a revised path that was offset by the tracking error from the first iteration. This procedure resulted in a final tracking error in the order of ± 4 mm, and ensures that the results presented below are not contaminated by significant offtracking errors introduced in the simulation process.

Figures 8a and 8b show a comparison between the measured lateral accelerations for one of the runs through the nominal 0.25-g lane-change manoeuvre and they show the lateral accelerations predicted by the Log-Truck Yaw/Roll model for the same truck following the same path. The agreement between the predicted and the measured steer-axle lateral accelerations is very good up to the 3.5-s point where no further path information was available. The measured steer-axle acceleration includes some highfrequency components that are not contained in the simulation results due to the relatively limited levels of accuracy and bandwidth associated with the measured path data. The measured and predicted log-CG accelerations in Figure 8b show even better agreement, primarily because the mechanical system filtered out the high-frequency responses of the lead unit.

Despite the apparently very good agreement, the simulation results provide an RWAst ave value of 1.175 compared to a value of 1.084 obtained from the measured data. The RWA_{st avg} values for the five experimental paths of Figure 6 are shown in Table 1. While the confidence intervals for both the measured and simulated RWAs ave values are reasonably narrow, it is clear that there is a systematic tendency for the simulation process to slightly over-estimate the rearward amplification relative to the physical measurement process. A very similar, although slightly more pronounced, tendency was observed when the rms-based RWA_{SAE} measures were used in place of the RWAss and thus it was concluded that RWAss and measures would be used exclusively for the comparisons presented here. Apart from the tendency to slightly overestimate RWA values, the comparisons in Figure 8 and Table 1 show that the simulation model is fairly effective at estimating the yaw performance of the log truck configuration used in the physical tests.

IDEAL-PATH FREQUENCY RESPONSE

The Log-Truck Yaw/Roll Model was used to establish the rearward-amplification frequency response of the tractor/pole-trailer by conducting more of the two-iteration simulations described above along a number of ideal singlelane-change paths with frequencies ranging from 0.18 to 0.5 Hz. The paths were defined for a speed of 24.4 m/s and a peak lateral acceleration of 0.25 g. The resulting discrete rearward amplification values are shown as "X"



Figure 9. Ideal-path and experimental-path frequency response performance of a tractor/pole-trailer conducting a 0.25-g lane change with a fixed path tolerance of ± 150 mm.

symbols in Figure 9, and the continuous ideal-path relationship is approximated by the wide solid line.

A second set of rearward amplification values corresponding to the RWA_{ISO} definition is shown in the figure with "+" symbols and a wide dashed line. The close relationship between the two curves at low frequencies is evident. The divergence at frequencies above 0.25 Hz occurs as the tractor yaw rate becomes significant enough to make the steer-axle lateral accelerations higher than the corresponding tractor-CG accelerations.

EXPERIMENTAL-PATH FREQUENCY RESPONSE

PATH TRANSFORMATION

The five simulation-based, 0.25-g, 0.4-Hz rearward amplification values from Table 1 are shown as individual symbols at the nominal path frequency of 0.4 Hz in Figure 9. Simulations along the individual paths resulted in a mean RWA_{sr_arg} value of 1.21 — a value slightly higher than the ideal-path value of 1.18.

The five physical lane-change tests used to obtain these RWA values were conducted with a single nominal lanechange frequency of 0.4 Hz. No additional physical lanechange experiments were conducted at nominal frequencies other than 0.4 Hz, and thus there are no physical measurements available here to define the rearward amplification of the tractor/pole-trailer at nominal path frequencies other than 0.4 Hz.

It is possible, however, to transform the five experimental paths in such a way that the response of the vehicle to experimental paths at other nominal frequencies can be simulated. A path transformation that can be used for this purpose is one in which an experimental path is "stretched out" in the longitudinal direction and where the maximum lateral displacement y_{max} is increased (in accordance with Eq. (2)) to simulate an experimental path with a lower nominal path frequency. Adjustments in the converse direction can be made to simulate an experimental path at a higher frequency. In both cases, the lateral deviation of the experimental path from the ideal path should remain constant. Figure 10 shows an example of the transformations of a single 0.4 Hz path to frequencies of 0.3 Hz and 0.5 Hz.

In mathematical terms, the y-coordinate $y_{path}(x_l, f_l)$ at an x-coordinate x_l along the path associated with the new nominal path frequency f_l is defined as

$$y_{path}(x_1, f_1) - y_{ideal}(x_1, f_1) + \{y_{path}(x_0, f_0) - y_{ideal}(x_0, f_0)\}$$
(6)

where $y_{ideal}(x_1, f_1)$ is the ideal-path y-coordinate defined by Eq. (1) for the new frequency, where $y_{path}(x_0, f_0)$ is the measured y-coordinate of an experimental path for the nominal frequency f_0 of 0.4 Hz at an x-coordinate x_0 , and where $y_{ideal}(x_0, f_0)$ is the ideal-path y-coordinate defined by Eq. (1) for the nominal frequency f_0 . The coordinate x_1 is scaled to the new nominal frequency by the equation

$$x_1 - x_0 \left(\frac{f_1}{f_0}\right) \tag{7}$$

FIXED-TOLERANCE FREQUENCY RESPONSE

Transformed experimental paths were defined at a variety of frequencies between 0.18 and 0.5 Hz in the manner described above, and additional two-iteration simulations were conducted to evaluate the response of the tractor/pole-trailer to experimental paths over this frequency range. The resulting RWA_{sr_avg} values are shown in Figure 9.



Figure 10. One of the five experimental 0.25-g, 0.4-Hz experimental paths transformed to nominal path frequencies of 0.3 and 0.5 Hz. (Ideal paths and path tolerances are included for clarity.)

The five experimental paths produced RWA values that are very tightly grouped at low frequencies, but that spread out to allow an undesirable level of experimental error at frequencies above 0.4 Hz. This characteristic is a direct result of the fixed path tolerance that allows relatively large path deviations at higher frequencies. In this particular case, the results are such that at 0.2 Hz the 90% confidence interval extends from 1.191 to 1.201 — a range of less than 1% that is much tighter than necessary, whereas at 0.5 Hz the corresponding interval extends from 0.99 to 1.19 — a range of approximately 20%.

In addition to the substantial uncertainty at higher frequencies, there is an apparent tendency for the experimental-path RWA values to over-estimate the actual frequency response at frequencies above 0.35 Hz. This over-estimation is the result of a frequency shift that will be more readily visible when the results of the variable-tolerance simulations are examined.

If the results of these experimental-path simulations were used to define the peak rearward amplification of the vehicle in the manner defined in the current draft ISO proposal, the result would be an estimated value of 1.28 at a frequency of 0.25 Hz — a value that actually compares very well with the ideal-path RWA of 1.30 at 0.28 Hz.

VARIABLE-TOLERANCE FREQUENCY RESPONSE

The sinusoidal-path envelope analysis presented earlier in this paper suggested that a variable path tolerance of the form presented in Eq. (5) might be more suitable for inclusion in the draft ISO proposal than a fixed path tolerance. The path transformation and simulation processes presented above make it relatively easy to examine the potential impact of this change.

The same five experimental paths measured during a 0.25-g, 0.4-Hz lane change were transformed using the same process described above, except that the path tolerance ϵ was varied according to the relationship

$$\epsilon - 150 \left(\frac{a_y}{0.25}\right) \left(\frac{0.4}{f}\right)^2$$
 (8)

This variation differs from that presented in Eq. (5) to the extent that the base path tolerance of ± 150 mm is associated with a 0.25-g, 0.4-Hz path rather than the 0.15-g, 0.4-Hz path defined in J2179. This variation is consistent with the earlier selection of the 0.25-g path over the 0.15-g path as a basis for determining the rearward amplification of the tractor/pole-trailer.

The results of simulations along the transformed variable-tolerance paths are presented in Figure 11. It is evident that the introduction of variable path tolerances has had two main effects. The first is that the experimental error is now relatively uniform throughout the frequency range, and was offset by the benefit of being able to conduct the low-frequency lane-change manoeuvres with path



Figure 11. Ideal-path and experimental-path frequency response performance of a tractor/pole-trailer conducting a 0.25-g lane change with a variable path tolerance defined by Eq. (8).

tolerances that were substantially higher than the fixed 150-mm value.

The second effect visible in Figure 11 is that the experimental-path RWA values appear to be uniformly shifted to the right of, and slightly below, the ideal-path RWA curve. The shift is a direct result of the tendency for the experimental paths to have actual lane-change frequencies that are lower than the nominal lane-change frequency (i.e., that are characterized by frequency/magnitude combinations in the region between points A and E in Figure 2). The result is that the RWA values that are associated with a nominal lane-change frequency of 0.25 Hz, for example, are plotted at 0.25 Hz in Figure 11, but are really the results of lane changes where the predominant frequency is about 20% lower — around 0.2 Hz in this particular case.

While the consistency of the frequency shift is at least partly attributable to the incestuous manner in which the experimental paths were generated at frequencies other than 0.4 Hz, it seems reasonable to conclude that the truly experimental paths that followed the nominal 0.25-g, 0.4-Hz lane-change provided rearward amplification values for a frequency that was 10 to 20% lower than the nominal path frequency. It also seems reasonable to conclude that a similar tendency might exist even in situations where the paths at all of the other frequencies were truly experimental too.

The peak rearward amplification defined from the results of the variable-tolerance experimental-path simulations is approximately 1.24 at a frequency of 0.32 Hz — a value that is not quite as good an estimate of the ideal-path value of 1.30 at 0.28 Hz as was achieved with the fixed tolerance approach, but that still represents a very reasonable estimate considering the benefit of being able to

use higher path tolerances at lane-change frequencies below the baseline value of 0.4 Hz. Furthermore, the variabletolerance approach readily lends itself to uniform reductions in experimental-path RWA error, simply by changing the constants in Eq. (5) or (8).

CONCLUSIONS

The fixed-path tolerance that is currently included in SAE J2179 and in a draft ISO proposal for a set of lateral stability test procedures allows vehicles to negotiate the lane change with path frequencies and peak lateral accelerations that vary substantially from the nominal path values. A simple sinusoidal path analysis has demonstrated that this tolerance has the potential to allow a vehicle to negotiate the SAE lane-change at actual frequencies that range from approximately 60% to 200% of the nominal lane-change frequency of 0.4 Hz, with corresponding variations in peak lateral acceleration from approximately 40% to 300%.

Test results presented for a tractor/pole-trailer negotiating a 0.25-g, 0.4-Hz path with a fixed tolerance of ± 150 mm showed actual variations in path frequency in the order of 10% to 20%. The nature and magnitudes of these variations need to considered when this single-lane-change method is used to estimate rearward amplification.

The use of a fixed path tolerance for paths with frequencies or peak lateral accelerations that vary from the baseline J2179 values of 0.4 Hz and 0.15 g results in substantial variations in the experimental error that is attributable to accepted path variations. A fixed tolerance of ± 150 mm provides extremely low experimental errors at low frequencies and high lateral accelerations, but this is achieved at the expense of having to stay within the specified tolerance over a path that is much longer and that has a substantially higher lateral offset than the baseline J2179 path. Conversely, the fixed tolerance results in unacceptably high experimental errors at frequencies above the baseline value of 0.4 Hz.

A variable path tolerance has been proposed in which the magnitude of the tolerance varies directly with the nominal peak lateral acceleration and inversely with the square of the nominal path frequency. This variable tolerance has the effect of making the path-based experimental errors relatively independent of frequency and magnitude. While it is certainly desirable to maintain lower experimental errors at low frequencies wherever possible, the variable path tolerance provides a recognition of what should be required rather than what might be desirable.

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