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A Road Profile Based Truck Ride Index (TRI)

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Abstract

Trucks are a major user of the road network and contributor to road wear, and yet most empirically derived road user cost relationships are based on traditional car-response based pavement roughness measures such as the International Roughness Index (IRI). The IRI is widely accepted as a reliable low-cost measure of relative comfort for occupants of passenger cars, providing significant input into the estimation of road user costs for all vehicles and for use in pavement management systems. There are a number of anecdotal instances of truck drivers complaining of poorer ride quality than is indicated by traditional roughness data. To better represent truck response in asset management systems, a road profile based Truck Ride Index (TRI) has been developed that is similar in concept to the IRI but tuned to the ride response characteristics of trucks. The TRI comprises a "quarter-truck" model, a combined seat and driver model, and a human tolerance to vertical vibration frequency-weighting curve. The TRI was tested on a range of road profiles and it was found that for any particular value of the IRI the TRI could cover a wide range of values. The TRI was found to replicate the key dynamic characteristics of truck ride, and the parameter set proposed for the TRI produces a waveband filter that is both centred on the dominant truck ride frequency and covers the frequency range of interest. At the critical frequency the gain factor for the TRI is less than 2.0, which is consistent with what appears to be required to achieve acceptable ride in trucks used on high-speed sealed-road conditions in Australia.

1. INTRODUCTION

Trucks are a major user of the road network and the major contributor to road wear, yet most empirically derived road user cost relationships are based on road roughness measures defined for passenger vehicles.

Traditional pavement roughness measures, such as the International Roughness Index (IRI) and NAASRA Roughness (in Australia), are widely accepted as a reliable low-cost measure of relative comfort for occupants of passenger cars. The IRI and NAASRA roughness also provide significant input into the estimation of road user costs for all vehicles, and are used in pavement management systems to determine maintenance intervention intervals and treatments.

There are a number of anecdotal instances of truck drivers complaining of poorer ride quality than is indicated by traditional roughness data, and there is evidence that truck drivers are exposed to higher levels of ride vibration than drivers of modern cars. Potential detrimental effects of poor ride for trucks include premature driver fatigue and increased risk to other road users, increased vehicle operating costs, damage to goods, and longer journey times as a result of lower average speeds.

The research underlying the material presented in this paper has identified and developed a road profile index that is tuned to truck ride and which can be applied to previously collected network-level profile data.

2. THE ELEMENTS OF RIDE

2.1 Perception of Vibration

The perception and evaluation of vehicle ride deals principally with understanding of the relationship between exposure to whole-body vibration and comfort. Many factors contribute to determine the feeling of well being in an individual and vibration is just one of these factors. In the context of this research it has been assumed than vibration is the main source of discomfort to the truck driver; discomfort due to factors other than vibration, such as noise and heat, have not been considered.

A large number of studies have been conducted over the years dealing with whole-body vibration, discomfort and ride in both ground and air transportation vehicles. Early studies were concerned primarily with the effects of varying vibration along a single axis, the more recent have investigated complex conditions, including multiple-frequency vibration, multiple-axis vibration and shock, and the interaction between noise and vibration (Leatherwood, et. al., 1980, for example).

In the context of the research described in this paper, the vibration imposed on the seated driver in the cabin of a truck travelling along a road is the prime measure of interest.

2.2 Whole-Body Vibration Tolerance

The evaluation of ride must deal with human tolerance to whole-body vibration. Gillespie (1985) presents a review of the vibration limits for human comfort and certain common features can be seen in the results of recent studies. Fig. 1, for example, shows lines of constant discomfort as determined in various research for a range of individuals subjected to vertical vibration over a broad range of frequencies. Because of the different interpretations of discomfort the nominal level of one curve is not directly comparable with that of another. Overall, however, the majority show the sensitivity of the human-body to vertical vibration has a maximum in the frequency range of 4 to 8Hz, and the curves illustrate that the human body is not equally sensitive to vibration at different frequencies.

Whole-body vibration tolerance curves are a useful way to determine how the vibration is likely to be perceived by the driver, and a range of curves presented in various formats can be found in scientific literature discussed in more detail in Section 3.3.

2.3 Truck Response to Road Profile

The prime source of truck vibration that determines ride comes from road surface unevenness. A number of other possible sources of excitation exist, that can be difficult to quantify but significant in their effect, which also contribute to the overall vibration levels in the truck cabin. These include wheel out-of-round and imbalance, tyre non-uniformities, and engine and drive train vibration. These have not been considered in this paper.

The nature of the vibration caused by road roughness will depend on the dynamic response characteristics of the truck measured between the areas where the tyres make contact with the road and the areas of contact with the driver, such as through the cabin floor, seat and steering wheel. Generally, however, only vertical vibration that transmit through the seat to the driver are considered in ride studies. To develop a truck ride index the dynamic response characteristics and vibration isolation properties of the truck and other elements in the vibration path to the driver, such as the seat, must be determined.

Fig. 2 shows how the road input to a truck is modified by a truck's ride response function to produce the acceleration response spectrum. The acceleration response spectrum describes the driver's vibration environment to highlight the dominant vibration frequencies. Further, the area under the curve provides a measure of the average vibration level.

The dominant peak in the vehicle response shown in Fig. 2 centred near 1Hz is associated with the bounce natural frequency of oscillation; a vertical oscillatory motion of the truck sprung mass on its suspension and tyres.

Various models for predicting the ride response of a typical truck have been reviewed and the most promising for a truck ride index has been identified.

2.4 Truck Ride and the IRI

A simple comparison of the IRI response function with a typical truck-ride response function is shown in Figs 3(a) and 3(b), both as a function of temporal frequency, expressed in cycles/s or Hz, and spatial frequency (or wavenumber) expressed in cycles/m. Spatial frequency is related to temporal frequency through the following expression involving travel speed.

[Eq. 1] $$f = V\upsilon$$

where:

f = temporal frequency (cycles/s)
V = speed (m/s)
υ = spatial frequency (cycles/m)

While the IRI response function describes the relationship between the motion of the unsprung-mass measured relative to the sprung-mass, the truck ride response function describes the relationship between the motion of the sprung mass relative to the road input.

Figs 3(a) and 3(b) reveal there are fundamental differences between the IRI and an index that may be indicative of truck ride. Trucks, like other highway vehicles, share the vibration isolation properties common to a sprung mass supported by a suspension system at each axle. The sprung mass supported by the suspension and tyres establishes the first resonant frequency seen as the peak near 1Hz. This peak in the response function is common to truck ride and the IRI. The second resonance at approximately 10Hz corresponds to wheel hop, seen in Fig. 3(a) as the bump near to 10Hz, which has a much larger influence on the IRI than on truck ride. The IRI measures axle-to-body relative movement whereas the truck driver is largely isolated from vertical vibration of the axle through the truck's suspension and the seat vibration isolation system. This can be seen in Figs 3(a) and 3(b) as the sharp fall off and increasing attenuation of road inputs in the truck ride response function for frequencies greater than 1Hz.

Fig. 3(b) shows the same information as 3(a), but in terms of wavenumber, which translates simply to distances along the road surface. Hence, the IRI is sensitive to road roughness in the wavelength range from 30m to 1.0m (a wavenumber range of 0.03 to 1.0 cycles/m), whereas truck ride would appear to be sensitive to wavelengths greater than about 3m (wavenumber less than 0.33 cycles/m).

3. VEHICLE AND HUMAN VIBRATION MODELS

An extensive literature review of the major transport and vehicle research databases was conducted (Prem, Ramsay and McLean, 1998). A range of vehicle models were identified that would be able to simulate the vibration levels experienced by the driver in the truck cabin and several standards relating to human tolerance to whole-body vibration were

examined. The review separately considered the following areas that are directly relevant to truck ride:

a) Vehicle models,
b) Cabin suspension models,
c) Seat suspension models,
d) Human biodynamic response models, and
e) Vibration limits for human comfort.

3.1 Vehicle and Cabin Suspension Models

Several types of vehicle models were identified, both linear and non-linear, having a wide range of complexity. The simplest of models were quarter-vehicle models, emulating either a single- or half-axle of a vehicle's suspension. More complicated models provide a detailed picture of entire vehicle dynamics and take account of the interactions between connected units (prime-mover and semi trailer, for example) and axles. Some include complex non-linear features of multi-leaf steel spring and airbag suspensions and the different bump and rebound characteristics of hydraulic dampers. Few vehicle models were detailed enough to consider cabin or seat suspensions, since the majority of them concentrated on the dynamic loads imposed on pavements by heavy vehicles.

3.1.1 Quarter-Vehicle Models

Fig. 4 shows a typical linear quarter-truck model. The majority of quarter-truck models that were identified do not include tyre damping. Considerable simplification can be performed using linear models, but often at the expense of model accuracy. The quarter-vehicle parameters for the models reviewed by Prem, Ramsay and McLean (1998) are presented in Table 1. The sprung mass suspension stiffness is significantly lower for the car-based IRI and the front-suspension based models by Amirouche, Palkovics and Woodrooffe (1995), and Todd and Kulakowski (1989). These result in lower sprung mass bounce frequencies.

The sprung mass responses of the models are shown in Fig. 5. Considerable variation is present in the location and magnitudes of the sprung mass resonant peaks. However, for all models, frequencies above 10Hz are strongly attenuated.

3.1.2 Half- and Full-Vehicle Models

The simple quarter-vehicle models do not account for many of the complex interactions that occur between the components of a full vehicle.

Half vehicle models consider one-half of a (four wheeled) vehicle. By combining two quarter-vehicle models that share the same sprung mass, a model is produced having four degrees of freedom. With one wheel following the left road profile, and the other following the right, the result is a roll-plane half-vehicle model. The roll-plane model will add effects due to differences in the left and right wheelpath road profiles. Cab-over-engine trucks,

having the driver positioned high above the steer axle, are particularly susceptible to this, however, other research has shown that vertical vibration generally are more dominant than the lateral vibration induced by differences in the wheelpath profiles (Gillespie, 1985).

Pitch-plane half-vehicle models have the rear unsprung mass following the same profile as the front unsprung mass, but with a time delay introduced that depends on vehicle speed and axle spacing, or wheelbase. Once again, drivers of cab-over trucks experience fore-aft vibration due to pitching of the vehicle.

More complicated models have been created that examine the pitch-plane behaviour of entire prime-mover semi-trailer systems. The most simple of these have three unsprung masses representing the steer, drive and trailer axle groups and two sprung masses representing the prime mover and semi-trailer. With the large variety of heavy vehicles in existence, selection of one wheelbase for a pitch-plane half-vehicle model may not accurately model the response of another. For example, a short-wheelbase cab-over prime mover will have a different response to a long wheelbase coach.

Full-vehicle models can be used to examine both pitch and roll of a vehicle. Typically, for a four-wheeled vehicle, rigid unsprung masses that are free to bounce and roll are used for the axles, and sprung masses that are free to bounce, roll and pitch are used.

Considerable work has been carried out by the University of Michigan Transportation Research Institute (UMTRI) and a number of full-vehicle models capable of predicting ride vibration have been produced (Gillespie and MacAdam, 1982; Gillespie and Sayers, 1987). Cebon (1985) presented results of a road loading simulation of a three-axle rigid full-vehicle model. Results from this closely resembled those of the real vehicle, but obviously would not be appropriate for a different configuration vehicle.

The complexity of half- and full-vehicle models and the extensive parameter data set required to run these models makes them unattractive for predicting truck ride from large amounts of road profile data collected at the network level.

3.2 Seat and Human Body Response

The vehicle model and its suspension elements provide sufficient information to define the vibration response characteristics of the cabin-floor to road excitations. In the study of ride, however, the primary response variable of interest is the vibration experienced by the seated driver.

There are a number of areas of contact between the vehicle and the human body. These include the seat-to-buttocks interface, backrest to back, armrests, floor-to-feet and steering wheel to hands, for example. Only vertical vibration transmitted to the human body through the seat-to-buttocks interface have been considered in this paper because of the lack of information in the literature on empirically derived human-body transmissibility models that consider other vibration paths. Gillespie (1985) reports that ride studies have

shown that visual and hand/foot vibration is also important to the perception of ride in trucks, though these are difficult to quantify.

3.2.1 Measured Seat-Response Characteristics

Seats used in modern trucks range in complexity from simple fixed units to air-suspended models with various adjustments and automatic compensation for driver weight. Seats differ in their dynamic characteristics and vibration isolation properties. Figs 6(a) to 6(d) reproduced from various sources show the transmissibility characteristics of a number of different commercial seats across a range of applications. The transmissibility characteristics of a seat will depend on a number of factors including cushion and suspension stiffness and damping, as well as the mass and build of the seated driver.

The seat transmissibility characteristics given in Figs 6(a) to 6(d) all show that cabin floor vibration will be attenuated to varying degree for frequencies greater than 10Hz. Vibration at the base of the seat shown in Fig. 6(a), for example, will be amplified within the frequency range 2 to 5Hz by as much as 2.5 times between 3 and 4Hz. The aircraft seats shown in Fig. 6(b) on the other hand do not attenuate the vibration as well, but the amplification of vibration in the frequency region 2 to 8Hz is significantly lower and does not exceed 1.4. For the air-suspension truck seat shown in Fig. 6(c) the vibration isolation performance is seen to be far better than the other examples cited but only when the seat suspension is properly adjusted (mid-ride position).

The 4 to 8Hz frequency region is particularly important because the human body has maximum sensitivity (minimum tolerance) to vertical vibration within this frequency range. The sensitivity is well recognised as the vertical resonances of the human abdominal mass (Gillespie, 1985; Bruel and Kjaer, 1988), and it is commonly suggested that the first major resonance of the human body occurs at about 5Hz, corresponding to the transmissibility of vertical vibration to the head (Griffin, 1990).

3.2.2 Human Body and Seat Models

A model that can reproduce the dynamic response characteristics of the seated driver in a realistic driver-vehicle vibration environment is an important element of truck ride prediction.

The human body can be considered a mechanical system and various biodynamic models have been developed to simulate its response to vibration. Fig. 7 shows a greatly simplified mechanical model of the body, where the various limbs/organs have been represented by simple mass, spring and damper units. Fig. 7 is intended to draw the reader's attention to the fact that when subjected to vibration, the human body will resonate over a range of frequencies and not just at a single frequency. Asymmetry and dependence of vibration response on the direction of application and the magnitude of the excitation will further complicate this behaviour.

Recent work by Boileau et al (1997) has produced a combined seat and driver model for ride vibration prediction based on a non-linear suspension seat model and a four-degree of freedom human-body model taken from previous work by Boileau (1995). Whilst the complexity and non-linear features of the seat model would add unnecessary detail to a simple truck ride model, the driver model established under sinusoidal and random vibration excitations would appear to be representative.

This model can be further simplified if low frequency ride vibration is of principle interest. Inspection of the model will reveal that the low frequency dynamics will be largely controlled by the seat suspension spring stiffness, K_s, damping, C_s, and the total supported mass, m_t. The total supported mass comprises the combined mass of the head and neck, chest and upper torso, lower torso, thighs and pelvis, and the mass of the seat.

The simplified driver-seat model is presented in Fig. 8(a) and its transmissibility function is shown in Fig. 8(b). There is good agreement between the response of the simplified model with the lower of the 5 measured curves shown in Fig. 6(d).

3.3 Tolerance to Whole-Body Vibration

A number of standards have been developed that seek to define reasonable and reliable procedures for quantifying the severity of complex vibration and shock environments. The most widely used, ISO 2631-1:1985 (International Standards Organisation, 1985), was recently replaced by ISO 2631-1:1997 (International Standards Organisation, 1997), which is closely aligned with BS 6841 (British Standards Institution, 1987). The dependency on exposure duration of the various effects on people had been assumed in ISO 2631-1:1985 to be the same for different effects (health, working proficiency and comfort) but the concept was not supported by research results in the laboratory.

BS 6841 (and ISO 2631-1:1997) covers the 0.5 to 80Hz frequency range. The frequency weighting function from BS 6841 for vertical vibration is shown in Fig. 9 together with the equivalent ISO 2631-1:1985 weighting over the frequency range of interest. The BS 6841 weighting function for vertical vibration are comparable to the fatigue decreased proficiency weighting in ISO 2631-1:1985, which were used to quantify vertical vibration with respect to its effect on performance (fatigue decreased proficiency). However, the British Standard (and ISO 2631-1:1997) employ a slightly different weighting for frequencies between 0.5 and 2Hz, increasing the importance of vibration frequencies below 1Hz and above 8Hz, as shown in Fig. 9. In general the frequency weighting indicate greatest sensitivity to acceleration between 4 and 8 Hz.

Researchers at the NASA Langley Research Centre in the USA developed a ride quality model for evaluating air and land transportation vehicles. The model incorporates the interactions between multi-axis vibration and interior noise (Leatherwood et al, 1980). The constant comfort lines for vertical vibration from the work of Leatherwood et al (1980) are shown in Fig. 10. The curves show that the sensitivity as a function of frequency is dependent on the acceleration level. At low levels of vibration the sensitivity curves

suggest comfort is independent of frequency, and low vibration levels are equally objectionable regardless of their frequency content. As vibration levels increase, however, the curves generally match those published by others.

Leatherwood et al's (1980) model is not as widely recognised as ISO 2631 or BS 6841. Further, for the present application the interior noise component of the model is not appropriate for a vehicle ride model that only considers road profile as the input.

In summary, the vibration acceleration frequency weighting defined in BS 6841 (or ISO 2631-1:1997) would be suitable for use in a truck ride index formulation.

4. TRUCK RIDE INDEX (TRI)

4.1 Practical Considerations

A requirement of the Truck Ride Index (TRI) is that it must be fully transportable, reproducible and measurable with a range of equipment in a similar fashion to the IRI established by the World Bank in 1986. Ideally it should also be no more complicated than the IRI in both form and ease of implementation.

The review has found that the prediction of truck ride from road profile measurements requires modelling of certain heavy vehicle dynamic responses. It is necessary to choose a representative vehicle model that can predict the vertical acceleration response of a truck's sprung-mass (at the drivers' location) to road profile inputs. A number of comprehensive and detailed models have been found in the literature, however, these are considered unnecessarily complicated, and a simple quarter-truck model based on a set of parameters that correspond to the front suspension properties of a typical truck was considered adequate. It was also considered necessary to model the seat and driver combination to account for the biodynamic response of the human-body to vibration, and dynamics of seat suspension system. Further, frequency weighting is required to account for the sensitivity of the human body to various vibration frequencies, particularly those occurring in the critical 4 to 8 Hz frequency range. Therefore, a human vibration tolerance curve such as BS 6841, for example, was selected and the overall weighted vibration response reported.

The process of estimating truck ride from road profile is summarised in Fig. 11.

4.2 The TRI and its Characteristics

The proposed truck ride model is shown in Fig. 12. The parameters that define the TRI are given in Table 2, and the truck ride response transfer function corresponding to this data set and the model described in Fig. 12 is shown in Fig. 13(a) and 13(b) as a function of temporal and spatial frequency, respectively.

Fig. 13(a) shows that the TRI response is centred on a frequency between 2 and 3 Hz. Fig. 13(b) shows that for a travel speed of 100 km/h the waveband impacting on truck ride is centred on a wavenumber of 0.085 cycles/m, which corresponds to a wavelength of about 11.2m, and there is some contribution from longer wavelengths. The overlap between the IRI and the TRI indicates some roughness features will be sensed by both indices. This may account for the ability of the IRI to predict truck ride on some pavements. However, the IRI would clearly pick up roughness features that excite wheel-hop vibration that would have very little impact on the TRI.

4.3 Guide to Effects of Vibration Magnitudes

The TRI is intended to provide a uniform and convenient method of indicating the subjective severity of the vibration environment for a driver seated in a truck that is travelling along a road. To help asset managers and others to determine the range of the TRI for a road network, and in the longer term establish acceptable road roughness levels for truck ride, a suitable TRI scale is required. In establishing such a scale it is important to recognise that the limiting acceptable values for the TRI will depend on many factors that may vary with each application, and that acceptable conditions in one environment may be unacceptable in others. The following values, taken from BS 6841, provide very approximate indications of the likely driver reactions to various magnitudes of frequency-weighted root-mean-square (rms) acceleration:

less than 0.315 m/s^2	not uncomfortable
0.315 to 0.63 m/s^2	a little uncomfortable
0.5 to 1.0 m/s^2	fairly uncomfortable
0.8 to 1.6 m/s^2	uncomfortable
1.25 to 2.5 m/s^2	very uncomfortable
greater than 2.0 m/s^2	extremely uncomfortable

Determination of threshold values for TRI will most likely need to consider a wide range of factors. These factors may include, for example, the transport environment (urban, rural and remote) and prevailing traffic speed. The type of transport operation (long/short haul) will influence the duration a driver is exposed to a vibration environment, and long haul applications may require smooth roads over long distances in order to maintain low vibration levels over long periods of time.

5. DEMONSTRATION OF TRI

Analysis and testing of the TRI was performed against a range of road profiles and separately using data from a study of the dynamics of heavy vehicles in which ride measurements were taken.

5.1 General Characteristics

Road authorities participating in the study identified approximately 550km of road profile that covered a wide range of unevenness conditions and ride vibration levels for trucks and other road users. TRI and IRI were calculated and the relationship between them is shown in Fig. 14. The limiting values for various levels of truck ride vibration are indicated using the descriptions given in Section 4.3 of this paper - these descriptions provide very approximate indications of likely driver reactions to ride vibration levels.

The following conclusions are drawn from these results:

1) Overall there is broad agreement between the two profile indices (due to the overlap in the TRI and IRI response functions shown in Fig. 13(b));

2) For any particular value of IRI the TRI covers a range of values. For example, sections of pavement that have an IRI of say 3.0m/km can have TRI values in the range 0.2 to 2.5m/s^2, and the likely reactions of truck drivers could range from "not uncomfortable" to "extremely uncomfortable". This finding supports the anecdotal instances of truck drivers complaining of poorer ride quality than indicated by traditional roughness data;

3) For a particular value of TRI the IRI covers a range of values, and pavements that have TRIs as low as 0.3m/s^2, and may be regarded as "not uncomfortable" in a truck, can have IRIs in the range 1.0 to 5.0m/km. An IRI of 1.0m/km is normally associated with new construction, whereas a pavement with an IRI of 5.0m/km is either close to, or has reached, the end of its useful life and would require some form of remediation (Austroads, 1992).

Road profiles that were supplemented with subjective ratings of ride provided by the road authority were investigated. Both the TRI and the IRI were generally found to agree with the subjective ratings, except for some profiles that indicated either the subjective rating was lower than suggested by the profile index, ie ride was rated as unacceptable when the profile index suggested it should have been acceptable, or vice versa. This was most likely due to the very imprecise method of subjectively rating pavement rideability and truck ride.

5.2 FORS Truck Study

In a study commissioned by the Australian Federal Office of Road Safety (FORS) into alleged problems with the dynamic behaviour of trucks and their effects on vehicle dynamics and drivers, on-road instrumented testing and data analysis was carried out on seven prime-mover and semi trailer combinations (Sweatman and McFarlane, 1999). Two of the test vehicles were benchmark vehicles provided by truck manufacturers. Measurements taken in the FORS study were specifically directed towards evaluating ride quality and handling. The seven vehicles tested had the following general characteristics:

- Long wheelbase highway prime movers (typically with a wheelbase over 4.5m)
- Mechanical multi-leaf steel spring suspension on the steer axle
- Air suspension on the drive axle group

- Higher powered engines (typically with 450hp or greater)
- Hauling semi-trailers over long distances at highway speeds (approx. 100 km/h).

Table 3 shows some specific details of the prime movers tested. All prime movers were tested with the same semi-trailer, which was fitted with wide single tyres and a mechanical multi-leaf suspension. The test course covered approximately 330km of high-speed sealed-road conditions.

5.2.1 Truck Ride Vibration

Vibration measurements were taken at a number of locations for assessment of ride; those relevant to this study are acceleration in the vertical direction taken on the seat, which are shown in Fig. 15(a).

Fig. 15(a) reveals the dominant vertical vibration covers the frequency range 2 to 10Hz. Within this frequency range there are two peaks, a large peak centred on approximately 2Hz that covers the frequency range 2 to 4Hz, and a second smaller peak centred on about 8Hz. The first and larger of the two peaks would correspond to vibration at the body bounce frequency. The frequency of the second peak coincides with a wheel rotation frequency of 9 Hz at the prevailing travel speed of approximately 100km/h, and is most likely transmission of vibration from a combination of wheel out-of-round, imbalance and variations in tyre radial stiffness.

5.2.2 Confirmation of the TRI

A generic road acceleration input spectrum was determined for the class and condition of roads used in the study, and the ride response transfer functions, shown in Fig. 15(b), were calculated for each of the heavy vehicles tested. The ensemble of transfer functions were averaged to produce an overall response function, also shown in Fig. 15(b), that was taken to represent the average response characteristics of the type of heavy vehicle investigated in the study.

Fig. 15(b) shows there is wide variation in the characteristics of the test vehicles, with vibration in the 2 to 4Hz frequency range amplified by as much as 3 times by some of the vehicles. It is important to note that vibration complaints were received in the study for three of the test vehicles, which showed higher seat vibration than the benchmark vehicles (Sweatman and McFarlane, 1999). This would suggest gain factors for the TRI transfer function should not exceed about 2.0.

The average transfer function for the test vehicles shown in Fig. 15(b) is in effect a "TRI" for the specific class and type of vehicle used in the FORS study. The average ride response is compared in Fig. 16 to the (generic) TRI proposed in the present study intended for general use across the heavy vehicle fleet. A TRI based on the data from the FORS study, denoted as TRI_FORS, which is the TRI model shown in Fig. 12 but tuned to the FORS test vehicles average transfer function shown in Fig. 15(b), is also shown in Fig. 16. TRI_FORS has been developed and included in this paper only to provide further support

for the proposed topology of the TRI model, which, although simple, is shown can predict the ride response of a relatively complex system reasonably well.

6. SUMMARY AND CONCLUSIONS

A Truck Ride Index (TRI) has been developed that comprises a "quarter-truck" model, a combined seat and driver model, and a human tolerance to vibration frequency weighting curve.

The TRI was tested on a range of road profiles and there was broad agreement between the TRI and the IRI. However, it was found that for any particular value of the IRI the TRI could cover a wide range of values, which supports the anecdotal instances of truck drivers complaining of poorer ride quality than indicated by traditional roughness data. Furthermore, for a particular value of TRI the IRI could cover a range of values, and pavements that have a TRI as low as $0.3m/s^2$, and the ride may be regarded as "not uncomfortable" in a truck, can have IRIs in the range 1.0 to 5.0m/km. An IRI of 1.0m/km is normally associated with new construction, whereas a pavement with an IRI of 5.0m/km is either close to, or has reached, the end of its useful life and would require some form of remediation.

On most sites TRI and IRI values were generally consistent with the perceptions of ride and descriptions used by road authority staff. However, on some sites the IRI values suggested a somewhat less than acceptable pavement condition (and less than acceptable ride), for at least some road users, than indicated by the subjective ratings. Also, there were a few sites where ride was rated as unacceptable for passenger cars but the IRI values indicated that it should be acceptable.

The FORS data confirmed that the TRI adequately replicates the key dynamic characteristics of truck ride for one class of vehicle. The parameter set proposed for the TRI, which produces a waveband filter that is centred on the dominant truck ride frequency and appears to adequately cover the frequency range of interest, was also confirmed. Further, at the critical frequency the gain factor for the TRI is less than 2.0, which is consistent with what appears is required to achieve acceptable ride in trucks used on high-speed sealed-road conditions in Australia.

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TABLES AND FIGURES

TABLE 1 - Parameters for Various Linear Quarter-Vehicle Models

Model	μ (m_u/M_s) (-)	K_1 (k_t/M_s) (s^{-2})	K_2 (K_s/M_s) (s^{-2})	C_s (C_s/M_s) (s^{-1})
IRI	0.150	653	63.3	6
UMTRI Profile Index	0.036	5120	390	17
Prem (1987)	0.190	413	381	1.6
Heath (1987)	0.149	473	270	2
UMTRI Factbook (Fancher et al, 1986)	0.150	237	427	88
de Pont (1994)	0.150	500	600	6
Amirouche, Palkovics and Woodrooffe (1995)	0.125	875	125	3.8
Cebon (1993)	0.124	393	225	4.5
Todd and Kulakowski (1989) - Front	0.111	322	80.9	1.1
Todd and Kulakowski (1989) - Rear	0.130	218	284	0.7

TABLE 2 – Specification of the Truck Ride Index (TRI)

Description of Parameter	Variable	Value*	Units
Effective mass of seat and driver	m_{sd}	0.067	-
Seat suspension stiffness	k_s	8.26	s^{-2}
Seat suspension damping	c_s	0.7	s^{-1}
Sprung mass	M_s	1	-
Suspension stiffness	K_2	250	s^{-2}
Suspension damping	C_s	30	s^{-1}
Unsprung mass	μ	0.150	-
Tyre stiffness	K_1	400	s^{-2}
Tyre enveloping (base length)	b	300	mm
Human tolerance to vibration (BS 6841, W_b frequency weighting)	-	-	-
Travel Speed			
rural/remote	V	100	km/h
urban		60	

NOTE: Values have been normalised by the sprung mass, M_s.

TABLE 3 – Characteristics of Prime Movers used in the FORS Study

ID	Vehicle	Tare Weight (t)	Wheelbase (m)
F1	Kenworth T900	10.6	5.94
F3	Kenworth C501T	9.7	5.94
F4	Mack CH Fleetliner	8.4	5.40
F6	Ford LT 9513	9.5	4.95
F26	Mack CH Elite	8.7	5.40
BM1	Mack CH Fleetliner	8.5	5.30
BM2	Kenworth T900	8.7	5.80

Notes: IDs prefaced with "BM" denotes a benchmark vehicle

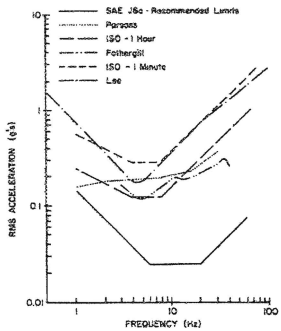

Fig. 1 Human tolerance limits for vertical vibration (from Gillespie, 1985).

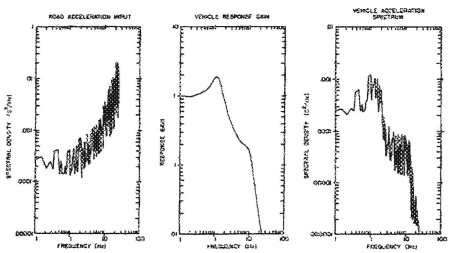

Fig. 2 Vertical accelerations produced by road excitations acting through the vehicle's response properties (from Gillespie, 1985)

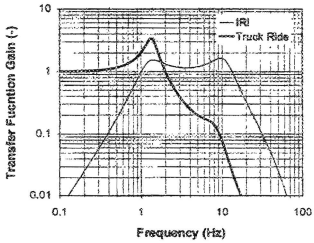

Fig. 3(a) Comparison of the IRI and the response of the sprung mass of a simple truck to road-profile response model (temporal frequency).

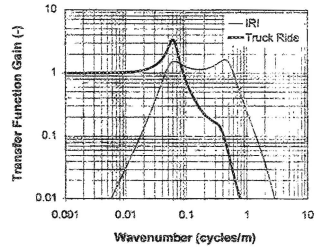

Fig. 3(b) IRI and truck ride responses as a function of spatial frequency (travel speed of 80 km/h).

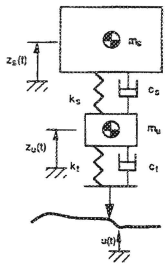

Fig. 4 Typical two degree-of-freedom 'quarter-truck' model (from Cebon 1993).

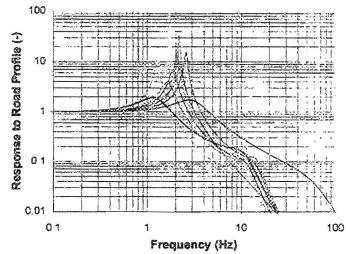

Fig. 5 Sprung-mass responses to road input of several quarter-vehicle models shown in Table 1.

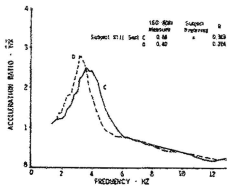

Fig. 6(a) Vertical transmissibility of two fixed seats (from Gillespie, 1985).

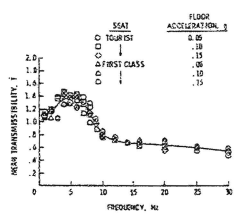

Fig. 6(b) Transmissibility of tourist and first-class aircraft seats (from Leatherwood et. al., 1980).

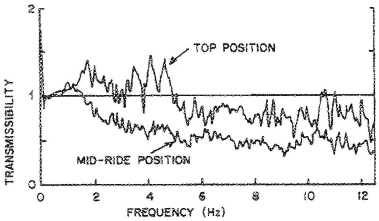

Fig. 6(c) Transmissibility of an air suspension truck seat when adjusted properly, and when adjusted incorrectly (taken from Gillespie, 1985).

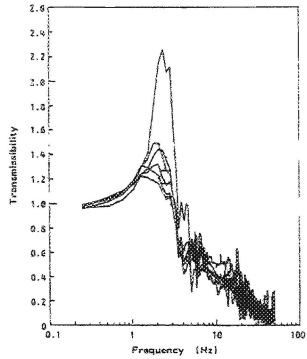

Fig. 6(d) Vertical transmissibility of six truck suspension seats (reproduced from Griffin, 1990).

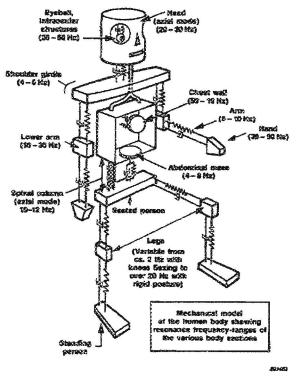

Fig. 7 Simplified mechanical model of the human body showing a range of resonant frequencies (reproduced from Bruel & Kjaer, 1988).

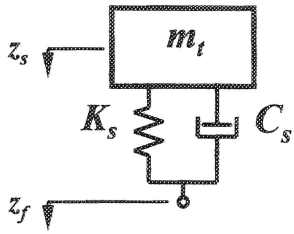

Fig. 8(a) Simplified driver and seat model.

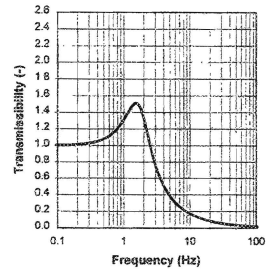

Fig. 8(b) Transmissibility of the simplified driver and seat model.

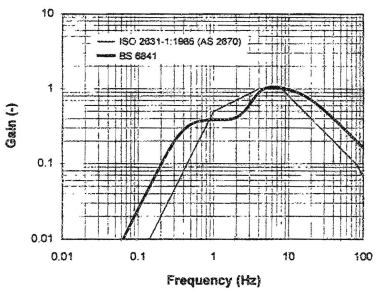

Fig. 9 Comparison between BS 6841 and ISO 2631-1:1985 frequency weighting for vertical vibration.

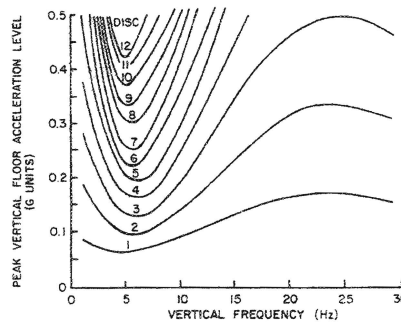

Fig. 10 NASA discomfort curves for vibration in transport vehicles (reproduced from Leatherwood, Dempsey and Clevenson, 1980).

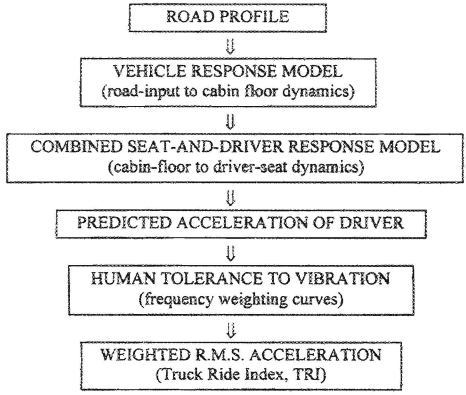

Fig. 11 The process of estimating truck ride from road profile.

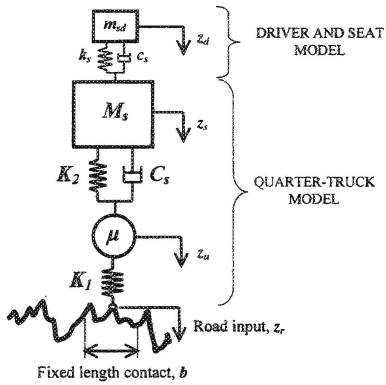

Fig. 12 Proposed dynamic model for truck ride prediction (parameter data set shown is given in Table 2).

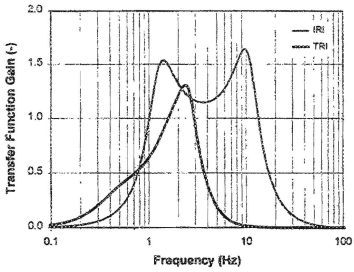

Fig. 13(a) Truck ride response - includes the seat and driver model shown in Fig. 17(a) and the BS 6841 frequency weighting shown in Fig. 20 – compared to the IRI.

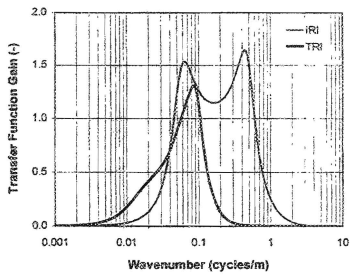

Fig. 13(b) Comparison of the IRI and the TRI in the spatial frequency domain. (TRI computed for a 100km/h travel speed.)

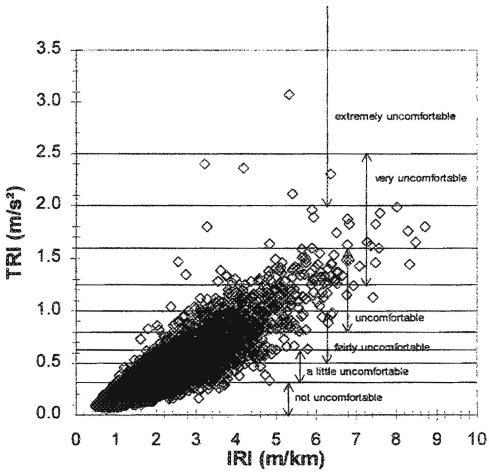

Fig. 14 Relationship between the TRI and the IRI.

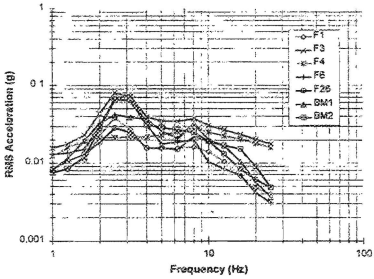

Fig. 15(a) Vertical acceleration measured on the seat of seven test vehicles (from Sweatman and McFarlane, 1999).

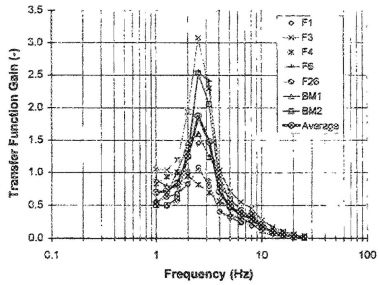

Fig. 15(b) Transfer function gain estimates for FORS study test vehicles.

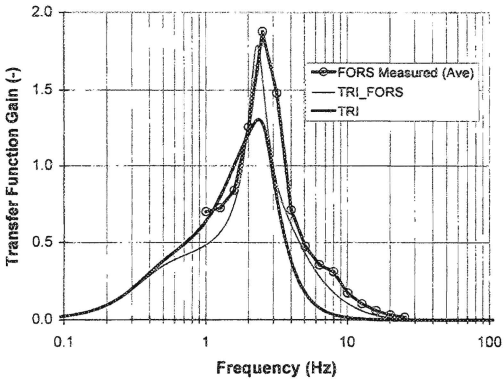

Fig. 16 TRI compared to FORS study data and TRI_FORS.