CHARACTERISATION OF VIBRATION RESPONSE SPECTRA OF ROAD TRANSPORT VEHICLES

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Abstract
Planning for the transportation of large volumes of fragile products by road requires accurate characterisation of the vibrations generated by the interactions between vehicles and the road surface. This characterisation allows packaging engineers to ensure that the products are adequately, but not excessively, protected. In a packaging context, this is especially important given the environmental issues associated with consumption and disposal of single-use packaging materials. Given these issues, the impetus for optimising packaging designs is clear. Packaging optimisation relies on the accurate characterisation of the vehicle’s vibration response using both Power Density Spectra (PDS) and statistical distributions of vibration levels. In the transport packaging industry, protective packaging systems are validated by exposing them to physically simulated vibration spectra in the laboratory. Since vertical vibrations are known to be the dominant cause of product damage (compared to other modes of vibration), the process is often reduced to a simple quarter-car heave response. In reality, the vehicle’s PDS varies considerably depending on its dynamic parameters. However, due to a lack of available information, vibration response spectra are often reduced to over-simplistic PDS which fail to differentiate between the various vehicle types and payload combinations. This paper aims to address this short-coming.

Keywords: Vehicle vibrations, product damage, protective packaging, vibration spectrum, Power Spectral Density.

1. Introduction

As the problems associated with packaging and product waste during distribution are becoming more prominent across the globe, the drive for packaging systems to be designed using environmentally responsible materials and optimised such that the least amount of material is used without compromising product integrity is increasing. A delicate compromise between the costs (both financial and environmental) related to excessive packaging and those associated with product damage needs to be achieved. For this to occur, the prediction of damage rates for various packaging scenarios must be accurate and, to date, this can only be achieved if laboratory simulations of distribution environments - particularly vibrations during transport – are sufficiently accurate. It is widely acknowledged that the vibrations generated by heavy goods vehicles as they travel, at speed, over uneven roads are the most common source of damage to products during the distribution phase. Today, a number of standards designed to assist packaging engineers with implementing suitable laboratory testing regimes to simulate vibrations related to road transport exist. Although they have developed over a number of years, their provenance and formulation are not, however, always sound and many such laboratory test protocols remain reliant on flawed assumptions and limited understanding of frequency and statistical analysis techniques (Rouillard et al., 2021). In the main, road vehicle vibrations are predominantly random due to the random nature of road profiles (Rouillard, 2007). Their analysis is, therefore, most commonly undertaken in the frequency domain where vibrations for particular scenarios - vehicle (suspension) type, payload and route (road type - roughness) - are represented by average Power Density Spectra (PDS). Other potential sources of vibrations (such as drivetrain harmonics) and shocks (caused by road surface aberrations such as pot holes) are generally ignored. This article includes a historical perspective and critical review on the nature of the PDS used in many test protocols in common use. It then presents a thorough analysis of a large number of heavy goods vehicle's vibration spectra computed from data measured by the authors as well as those published in the literature. This analysis highlights the complexities associated with interpreting PDS as well as the challenges associated with combining PDS from various scenarios (vehicle type, suspension type and payload).

2. Literature Review

Methods for testing the ability of products to withstand vibrations during transport using physical simulation under controlled conditions in the laboratory have been in place for at least four decades (Ostrem and Godshall, 1979; ASTM D4728, 1987). These methods have and continue to be useful for validating the effectiveness of packaging systems for transport without having to resort to expensive, uncontrollable and statistically unrepeatable field trials. All such test protocols use the PDS to define various tests and these are usually defined as 'breakpoints' or Power Spectral Density – frequency coordinates. When used to synthesize vibrations, the PDS inherently yields random vibrations that conform to the Gaussian distribution with a constant RMS (where the RMS is defined by the square-root of the integral of the PDS). The shape of the PDS is primarily a function of the vehicle's dynamic parameters (suspension characteristics) including payload, whereas speed and road roughness are overwhelmingly responsible for the level of the vibrations and, if linear or near linear behaviour is assumed, typically have little influence on the shape of the PDS.

There exists a clear evolutionary path when it comes the characterisation and simulation of transport vibrations which has improved with technological advancements in measurement and recording devices. Rouillard et al. (2021) provide a detailed review of the evolution of the PDS used for transport simulation testing. Ostrem and Godshall (1979) were one of the first to publish information of the PDS of road vehicle response vibrations and proposed the use of a 'vibration envelope curve' generated by joining the spectral peaks. The result appears to indiscriminately encompass all frequencies present in the measured data as illustrated in Figures 1 and 2Figure . Rouillard (2007) described this approach as highly flawed (but possibly deemed necessary at the time) as simplifying the spectrum in this way effectively spreads the vibratory energy across a wide frequency range thus reducing the concentration of vibratory energy in specific frequency bands where resonances of the vehicle and/or product may exist. When used to evaluate the vibration resistance of packaged products which typically exhibit multiple resonances, the approach cannot be justified.

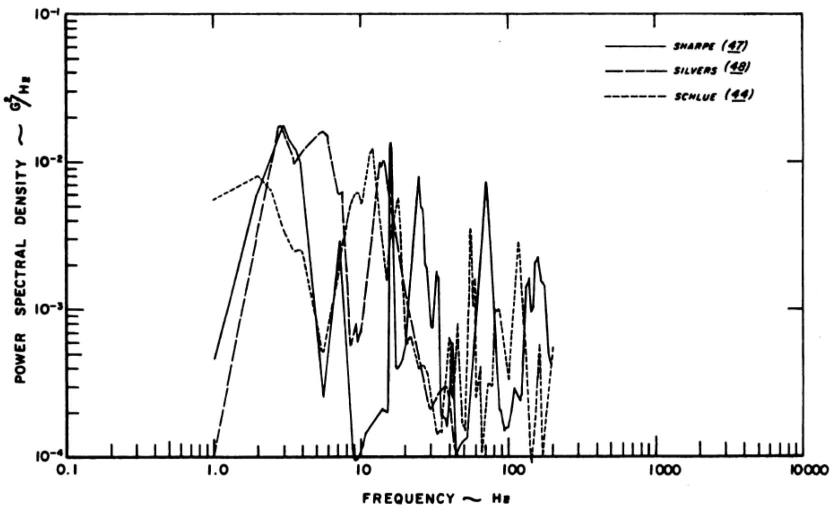

Figure 1. Typical truck frequency spectra (PDS) for three scenarios as published by Ostrem and Godshall (1979).

Despite evident limitations, simplistically joining spectral peaks to create spectral breakpoints (to include potentially a wide variety of vehicles) has repeatedly been employed by standard organisations without questioning its validity and appropriateness. The many test spectra published in test standards - illustrated in Figure 3 –continue to be used around the globe. The majority of the numerous studies that have been published on the topic clearly show that measured PDS vary significantly from what standards organisations recommend (Rouillard et al., 2021). Only a few, isolated examples involving specific vehicle types indicate some similarities to the standard spectra (Rouillard et al., 2021). Despite numerous studies into vibration PDS from road transport vehicles, there has, to date, been no formal attempt to compare the shape of the PDS and relate it to the various types of vehicles and payload conditions. Rouillard et al. (2021) published preliminary results showing PDS obtained for a selected range of RVV (road vehicle vibrations) heave data covering a broad range of routes and vehicle types (Figure 4). However, no formal comparison was undertaken.

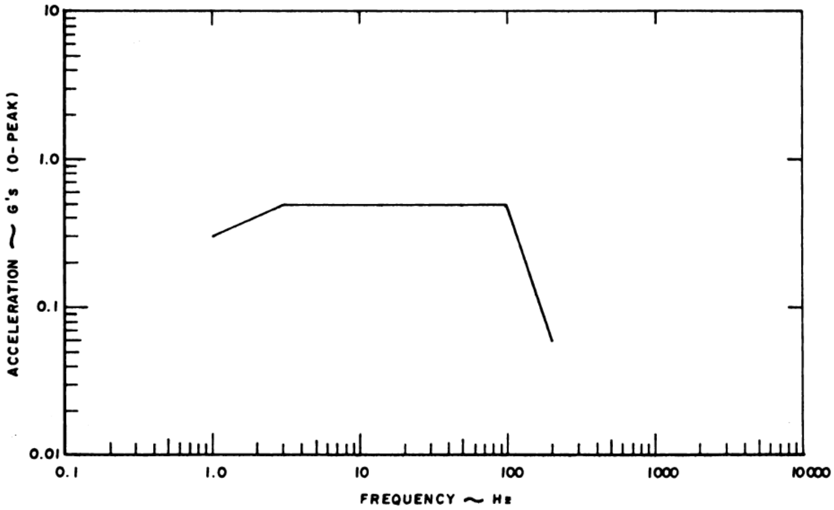

Figure 2. Truck frequency spectra (amplitude) envelope as proposed by Ostrem and Godshall (1979). Note the significant differences to the measured spectra shown in Figure 1.

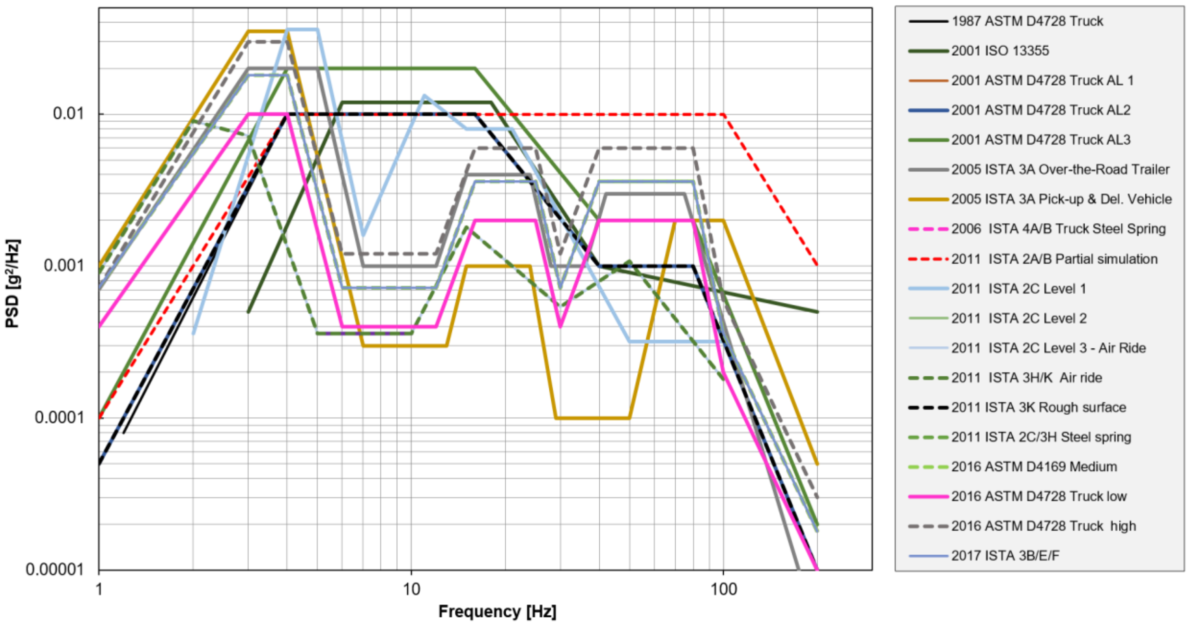

Figure 3. A comprehensive range of Truck PDS including from the original release of ASTM standard D4728 in 1987[1].

The need to undertake a broad survey of road vehicle vibrations for a variety of representative vehicle types and payloads to establish variations of spectral shape and any relationship with vehicle types and payload is clear. Some preliminary results related to this are presented in

[1] ASTM: American Society for Testing and Materials; ISTA: International Safe Transit Association; ISO: International Standards Organisation.

Figure 4; although, these represent the PDS of raw RVV heave data with no special steps taken to isolate any transients or harmonics or to categorize based on vehicle/route type.

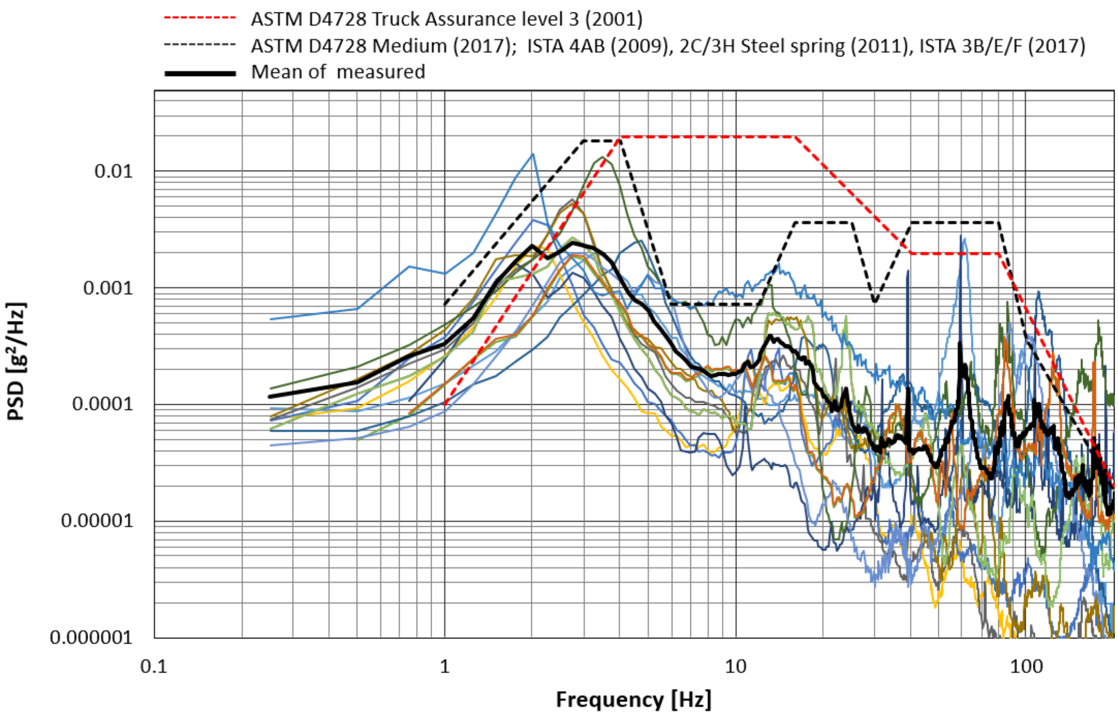

Figure 4. PDS from a variety of measured RVV along with the overall mean and two commonly-used generic spectra.

3. Aim

The aim of this papers is to propose alternative vibration response spectra to be used as representative PDS for laboratory simulation of road transport vibrations. These representative PDS are to be based on the broadest possible set of publically-available data in order to achieve a more accurate and realistic representation of the vibrations that are generated by heavy goods vehicles.

4. Methodology

The approach taken in this paper was to explore the inherent nature of the vertical (heave) vibration response spectra (PDS) measured from an as large as possible set of publicly-available information. The PDS were analysed to establish if they could be grouped in accordance with important vehicle characteristics namely suspension type and payload with the aim of producing a number of archetypal PDS that could be used to represent common vehicle configurations and their corresponding vibration response PDS. This was achieved by matching (curve-fitting) the vibration response PDS with a linear quarter-car model and extracting the important modal parameters. This required the assumption of a spectral model for the road elevation profile for all cases. In this paper, the well-established road elevation profile PDS as described in ISO standard 8608 was used[2]:

$$G_x\left(n\right)=G_x\left(n_o\right)\cdot\left(\frac{n}{n_o}\right)^{-w} \tag{1}$$

, where $G_x(n)$ is the elevation PSD (m^3/rad), n is the spatial frequency (rad/m), n_o is the reference spatial frequency (1 cycle/m) and w is the spectral exponent (usually set at two according to the standard). Arguments for different values for the spectral exponent, w, are numerous and continue to be explored by many in the field. In this paper, the influence of setting w at values other than two was explored and is described in the results section. The magnitude Frequency Response Function (FRF or transmissibility), $T(f)$, of a quarter-vehicle (that is a single-wheeled vehicle) as a function of frequency in Hz travelling at a constant speed, v (m/s), on a road with a an elevation PDS, $G_x(n_o)$, can be estimated from the vertical acceleration response PDS, $R_{\ddot{x}}(f)$, as follows:

$$T(f)=\sqrt{\frac{R_{\ddot{x}}(f)}{G_{\ddot{x}}(f)}}=\sqrt{\frac{R_{\ddot{x}}(f)}{G_x(n_0)}}\frac{f^{\left(\frac{w}{2}-2\right)}}{(2\pi)^2\left(n_0\right)^{w/2}v^{(w-1)/2}} \tag{2}$$

, where $G_{\ddot{x}}(f)$ is the acceleration PDS of the pavement surface profile obtainable by differentiating $G_x(f)$. However, $G_x(f)$ can only be known by converting $G_x(n_o)$ to the temporal domain by combining it with the vehicle velocity, v, which, in this case, is not known. However, due to the fractal nature of $G_x(n_o)$ (Ohmiya, 1991), its conversion to the temporal domain – which effectively shifts the PDS to the right with increasing values of v – results in an equivalent rise in the PSD values as illustrated in Figure 5. This feature offers a work-around by introducing a magnification factor, m, that determines the effective overall road roughness $\hat{G}_x(f_o)$ such that the transmissibility function, $T(f)$, approaches unity at zero Hz. In this case, the (acceleration transmissibility), $T(f)$, can be reduced to:

$$T(f)=\sqrt{\frac{R_{\ddot{x}}(f)}{\hat{G}_{\ddot{x}}(f)}} \qquad for \quad T(f\rightarrow 0)\rightarrow 1 \tag{3}$$

In this paper, measured vibration response PDS were combined with the road acceleration PDS to produce an estimate of the quarter-car magnitude FRF. This FRF was then modelled (fitted using regression) by the theoretical sprung-mass FRF of a 2 degree-of-freedom system described in the Laplace domain as shown in equation 4.

$$\left|FRF\right|=T\left(s\right)=\left|\frac{\omega_{n,u}{}^2\left(2\zeta_s\omega_{n,s}s+\omega_{n,s}{}^2\right)}{\left(s^2+2\rho\zeta_s\omega_{n,s}s+\omega_{n,u}{}^2+\rho\omega_{n,s}{}^2\right)\left(s^2+2\zeta_s\omega_{n,s}s+\omega_{n,s}{}^2\right)-\rho\left(2\zeta_s\omega_{n,s}s+\omega_{n,s}{}^2\right)^2}\right| \tag{4}$$

Where $\omega_{n,s}$ and $\omega_{n,u}$ are the uncoupled sprung and unsprung mass natural frequencies in rad/s respectively; ζ_s is the uncoupled sprung mass damping ratio (tyre damping is negligible and

[2] Interestingly, this spectral shape is known as a Brown(ian) spectrum or Brown noise as it is similar to the spectrum that describes Brownian motion also sometimes known as random walk or drunkard's walk. Brownian motion describes the motion of small particles suspended in a fluid due to bombardment by molecules conforming to a Maxwellian velocity distribution. It was named after R. Brown who discovered it in 1828. It was Einstein, who, in 1905, used kinetic theory to explain the phenomenon known as Brownian motion.

ignored); ρ is the sprung to unsprung mass ratio; ω is the excitation frequency in rad/s and $s = \pm i\omega$ is the Laplace operator. This approach requires that the assumed road elevation PDS (equation 1) is valid. Although generally assumed to be two, the value of the spectral exponent w may, in practice, vary (generally between 2.0 and 2.4) from road to road (Kropáč & Múčka, 2009). In this study, the influence of w on the results was evaluated by combining the acceleration response PDS with road elevation spectra using values of w between 2.0 and 2.4 evaluating the goodness-of-fit between the measured and predicted response PDS.

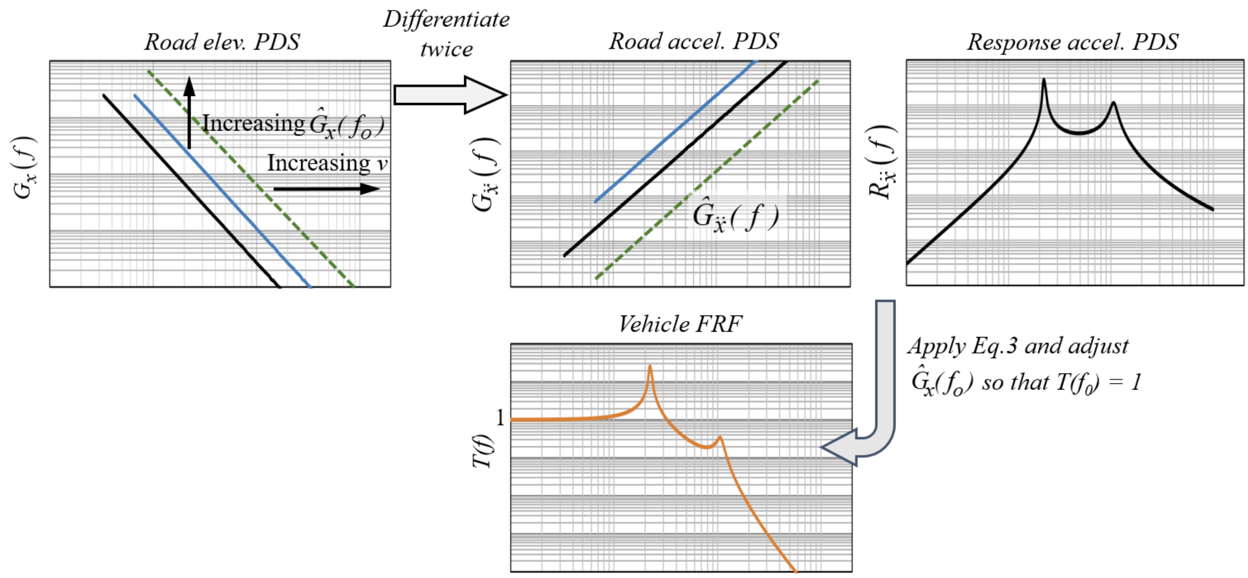

Figure 5. Effect of vehicle speed on the road elevation spectrum (top) and resulting quarter-car transmissibility by iteratively adjusting the road roughness

5. Results

The analysis herein is based on some 129 PDS computed from a variety of vehicle and road surface configurations. 56 PDS were computed using vibration data measured by the authors over the past 25 years and 73 extracted from published data (earliest reliable information published in 1992) where clear information of the vehicle type, suspension type, payload and road type(s) was available. A comprehensive summary of the completed data set including sources is freely available through this link.

5.1 Data classification

The data was classified in two broad sets: 1) Steel leaf suspension and 2) Air ride suspension. Attempts at further classifying the data based on payload (as a fraction of capacity) proved too difficult due to challenges in obtaining reliable payload and capacity information for all cases. Instead, frequency spectra were classified on the value of the first (sprung mass) natural frequency which is a function of payload and stiffness. Purpose-designed code was implemented to extract the natural frequency of each acceleration response PDS for the two sets namely leaf spring and air ride and compute their statistical distributions. The bin width of the statistical distributions had to be carefully selected to yield sensible results which are shown in Figure 6. These clearly reveal an interesting phenomenon in that the natural frequencies tend to occur in groups which can be summarised in Table 1.

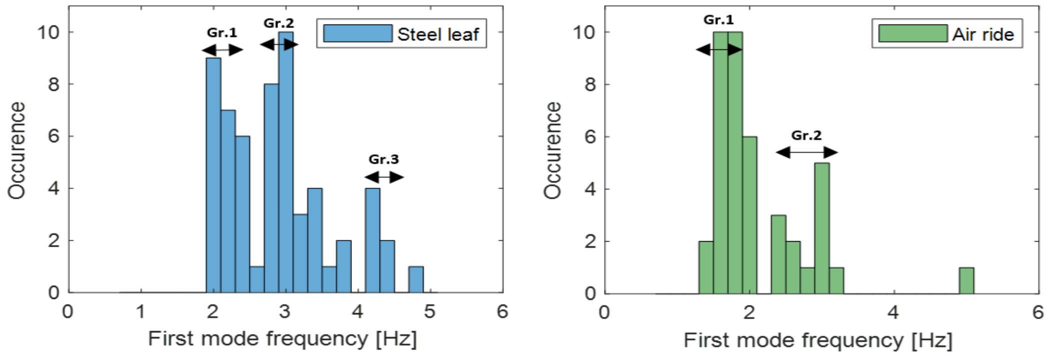

Figure 6. Distribution of first mode (sprung mass) natural frequency for vehicle with steel leaf and air ride suspensions.

Table 1. Sprung mass natural frequency grouping.

	Sprung mass natural frequency grouping [Hz]		
Suspension Type	**First**	**Second**	**Third**
Steel leaf	1.9 – 2.5	2.7 – 3.1	4.1 – 4.5
Air ride	1.5 – 2.1	2.3 – 3.1	-

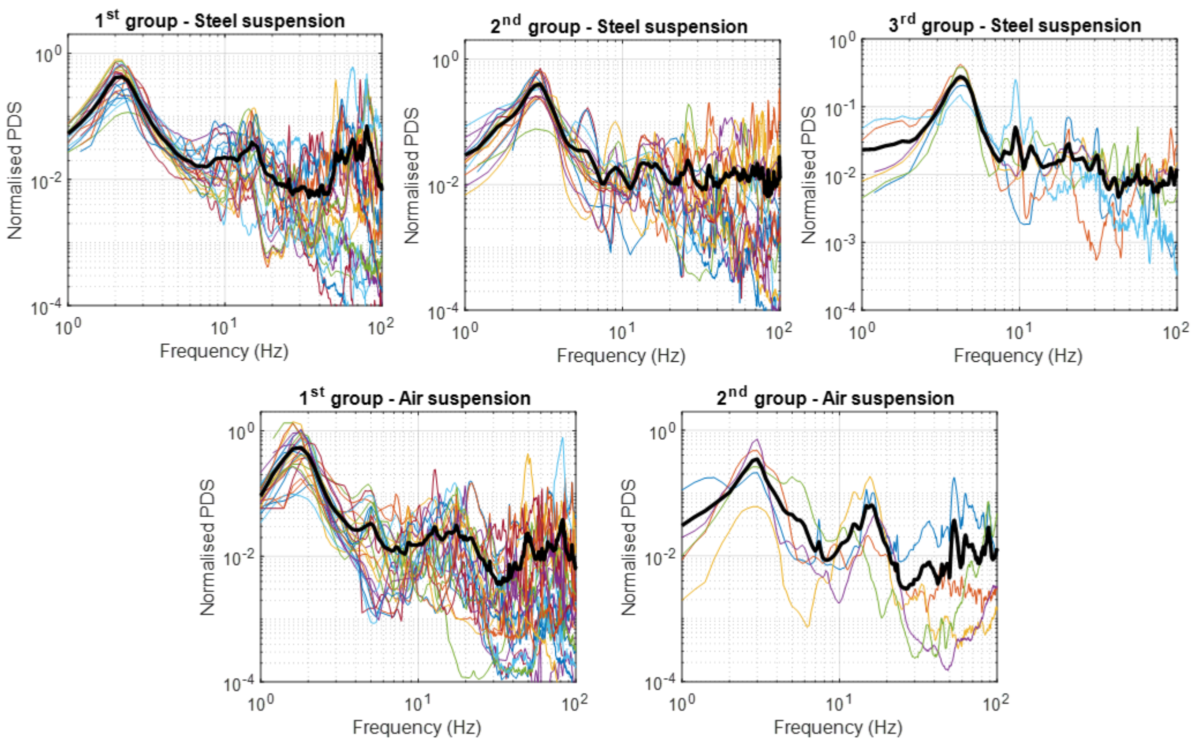

Figure 7. Individual PDS for each of the five groupings along with their mean (black).

5.2 Representative spectra

The aim here is to establish a single archetypal of overall PDS to represent each of the five categories (groups) listed in Table 1.

Although the frequency of the first resonant mode for each category are well grouped, the second mode (axle hop) is not so well-defined with its frequency depending on the vehicle's mass and stiffness ratios. It must be noted here that only the shape of the PDS is of interest in the context of this paper. If linearity is assumed, as the vibration level increases, the PDS values increase proportionally across the entire frequency bandwidth. In fact, the overall root-mean-square (RMS) vibration equates to the square-root of the area under the PDS. Therefore, all PDS were normalized to produce unity RMS across the rigid-body mode frequencies, namely 1 – 30 Hz. Vibratory energy above 30 Hz is considered to represent structural and drivetrain vibrations and is, here, ignored for the purpose of modelling rigid-body vibrations. Application of a simple average normalized PDS is not suitable as it drowns-out the second mode as shown in Figure 7 making it challenging when trying to use the mode as a manifestation of a realistic quarter-car vehicle model. Therefore, using the mean as a representative PDS for the five groups is inappropriate.

5.3 Selective elimination to produce representative PDS

In order to retain the important features of the vibration response PDSs (namely the prominent peaks corresponding to the sprung mass and unsprung mass (axle hop) resonance, a two-stage selection process was applied. First, those PDS determined to be clearly abnormal (such as those with excessively large peaks outside the typical first and second mode frequency bands) were discarded. The mean of the remaining PDS was then computed and individual PDS with large deviations from the mean across the low (rigid-body) frequency bands were progressively eliminated with the mean re-computed after each elimination resulting in the representative PDS for the group. These PDS are shown in Figure 8 for each of the five groupings. In must be noted that there are no well-established methods for sampling and determining representative specimens from small data sets with high variability and extreme outliers. This will be the subject of further investigation.

5.4 Quarter-car modelling of representative PDS

Each of the five representative PDS were combined with the generic road elevation spectrum and fitted to the quarter-car theoretical model (eq. 4) to produce estimates of the magnitude FRF for each case (Figure 9). These five typical scenarios effectively epitomise five typical vehicle configurations - suspension and sprung mass (including payload) – that can be said to be representative of transport vehicles in general namely heavily, moderately and lightly laden in terms of vehicle payload capacity. This offer far more realistic response PDS that can be used to simulate (rigid-body) road vehicle vibrations in the laboratory to verify the vibration fragility of products as well as validate and optimise packaging systems. The other important observation to be made here is that none of the representative PDS (nor any of the 130 individual PDS) resemble any of the generic spectra used by the various test standards shown in Figure 4. These latter are over-simplistic and generally fail to feature the significant resonances that occur in reality. Consequently, they tend to produce more broad-banded random vibrations that do not account for the concentration of vibratory energy around the resonances. An interesting, and potentially practically useful outcome of this research is the ability to estimate the sprung mass of a vehicle from vibration response measurements based on a single set of measurement where the sprung mass is known. Once this 'base mass ratio' is established using the approach described above, the unsprung mass of the vehicle can be evaluated and any future payload can be estimated from vibration response measurements. This approach needs, in the future, to be verified through controlled experiments.

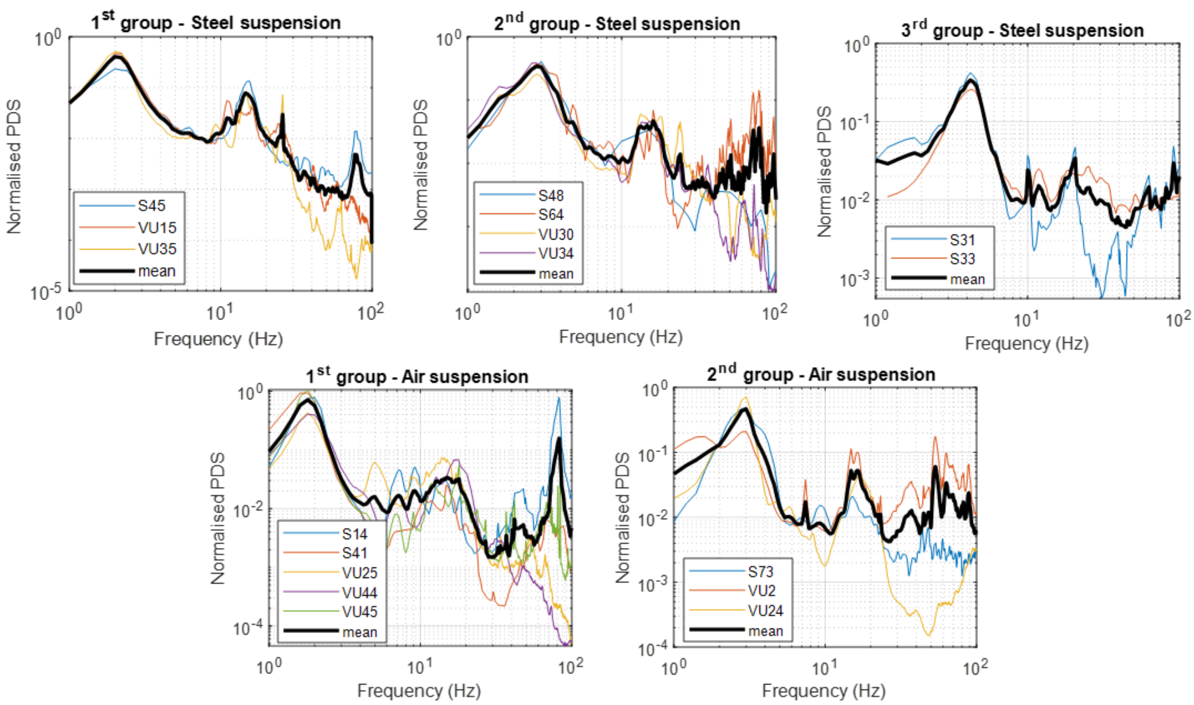

Figure 8. Individual PDS after selective elimination for each of the five groupings along with their mean.

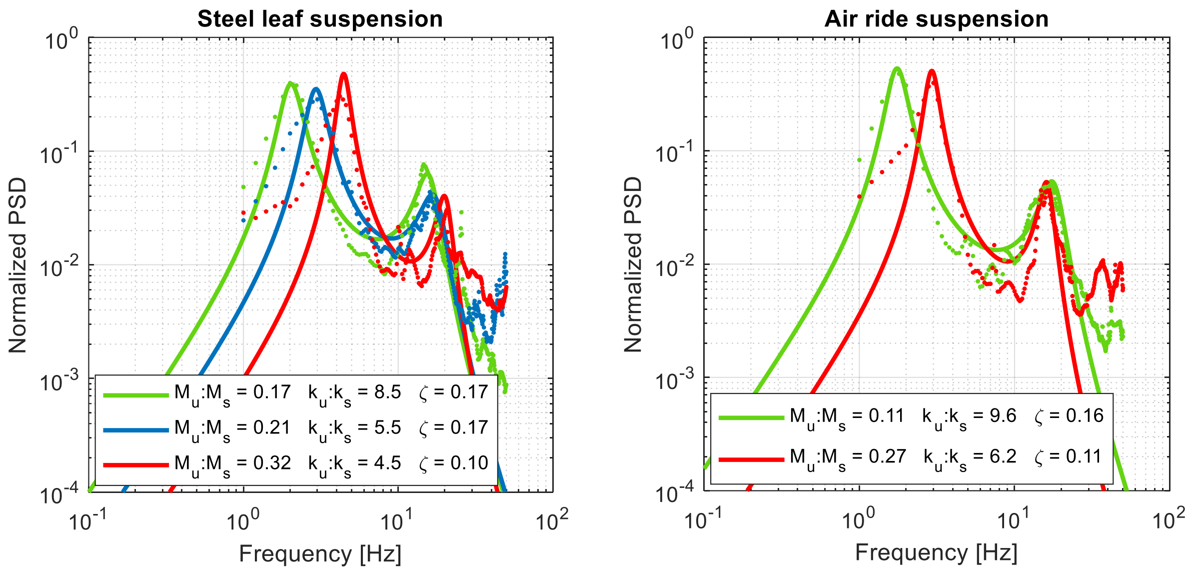

Figure 9. Measured and modelled response PDS for the five representative payload scenarios: Green: Heavy; Blue: Moderate; Red: Light. Where ζ is the equivalent SDoF damping ratio.

An interesting outcome of this work are the significant differences in stiffness ratios yielded here when compared to (albeit largely older) published values as shown in Table 2. These newly-estimated stiffness ratios vary between 4.5 and 9.6 (for *w* = 2) compared to those from published works which vary between 0.77 and 3.9. These differences cannot, at this stage, be easily explained except to conjecture that tyre and suspension technology (stiffness) have changed significantly over the years. These differences need to be investigated further by undertaking a series of deflection measurements which can be achieved during the unloading process for various typical vehicles. It must emphasized that only the shape of the PDS is of interest in the context of this paper.

Table 2. Quarter-car parameters (for *w* = 2.0 to 2.4.) along with published values.

Model	M_u:M_s	K_u:K_s	ζ	ζ_s
UMTRI Factbook 1/4 Truck (de Pont, 1994)	0.150	1.80	0.069	2.858
Quarter Truck (Prem, 1987)	0.190	0.92	0.011	0.039
Quarter Truck (Heath, 1987)	0.149	1.75	0.029	0.063
Rear (Todd & Kulakowski, 1989)	0.129	0.77	0.005	0.019
Tuned 1/4 Truck (de Pont, 1994)	0.150	1.20	0.047	0.134
Steel leaf (Cebon, 1999)	0.094	1.95	0.057	0.112
Air (Cebon, 1999)	0.094	3.90	0.143	0.212
Steel leaf heavy (This paper)	0.17 - 0.18	8.5 - 12.3	0.17 - 0.18	0.21 - 0.22
Steel leaf moderate (This paper)	0.21 - 0.27	5.5 - 7.5	0.17 - 0.17	0.21 - 0.23
Steel leaf light (This paper)	0.32 - 0.30	4.5 - 5.6	0.10 - 0.13	0.14 - 0.17
Air heavy (This paper)	0.11 - 0.19	9.6 - 14.3	0.16 - 0.16	0.14 - 0.17
Air light (This paper)	0.27 - 0.33	6.2 - 8.4	0.11 - 0.13	0.15 - 0.16

6. Conclusion

An extensive collection of vibration response data measured from a broad range of heavy goods road vehicles was analysed to produce five typical response PDS (three for steel leaf suspension and two for air ride suspension) that represent what can be considered as representative response PDS for road transport. It is recommended that these five representative spectra be used to evaluate and optimise the vibration resistance of products during road transport. These will yield more realistic vibrations than those promoted by standards organisations and should have a positive impact on the optimisation of packaging designs and a corresponding reduction in packaging waste. Additional findings from the research are i) potential for a technique to estimate the sprung mass (or payload) of road vehicles from vibration response measurements only and ii) a clear difference in the tyre : suspension stiffness ratios for the representative vehicles studied when compared to existing published data. Both these findings need to be investigated further. In addition, further work on developing and validating techniques for selecting and combining sample spectra to obtain representative PDS for selected groupings needs to be undertaken.

7. Acknowledgment

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8. References

- ASTM D4728–87. (1987). “Standard Test Method for Random Vibration Testing of Shipping Containers”.
- Cebon, D. (1999), “Handbook of vehicle-road interaction”, CRC Press.
- De Pont, J. (1994), “Road profile characterisation”, Transit New Zealand.
- Fancher, P.S. (1986), “A factbook of the mechanical properties of the components for single-unit and articulated heavy trucks”, UMTRI.
- Gillespie, T.D. and Karamihas, S.M. (2000), “Simplified models for truck dynamic response to road inputs”, International Journal of Heavy Vehicle Systems, 7(1), pp.52-63.
- Heath, A. (1987), “Vehicle parameters affecting dynamic road loading”, Proc. ARRB Symposium on Heavy Vehicle Suspension Characteristics, Canberra, Australia.
- Kropáč, O., & Múčka, P. (2009), “Effects of longitudinal road waviness on vehicle vibration response”, Vehicle system dynamics, 47(2), 135-153.
- Ohmiya, K. (1991). Fractal dimensions of terrain profiles. Journal of Terramechanics, 28(2-3), 155-165.
- Ostrem F, Godshall WD. (1979), “An assessment of the common carrier shipping environment”, Dept.of Agriculture, Forest Service, Forest Products Laboratory.
- Prem, H. (1987), “Road roughness influence on suspension performance”, Proc. ARRB Symposium on Heavy Vehicle Suspension Characteristics, Canberra, Australia.
- Rouillard, V. (2007), “On the synthesis of non-Gaussian road vehicle vibrations”, Doctoral dissertation, Monash University.
- Rouillard V, Lamb M and Gomez Tabanera A. (2019), “Developing Laboratory Test Protocols for Multi-axial Road Transport Vehicles Vibrations”, ISTA China Packaging Symposium. Shenzhen, China.
- Rouillard, V., Lamb, M. J., Lepine, J., Long, M., & Ainalis, D. (2021), “The case for reviewing laboratory-based road transport simulations for packaging optimisation”, PTS, 34(6), 339-351.
- Todd, K.B. and Kulakowski, B.T. (1989), “Simple computer models for predicting ride quality and pavement loading for heavy trucks”, Transportation research record, 1215, pp.137-150.