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ANALYSIS OF ACCELERATIONS AND STRAINS MEASURED ON A TIE-DOWN SYSTEM OF A HEAVY NUCLEAR TRANSPORT PACKAGE DURING A ROUTINE RAIL JOURNEY

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ABSTRACT
The design and development of nuclear packages is critical for the safe transportation of new fuel and irradiated waste. The renaissance of the nuclear industry in recent years has increased motivation for the development of optimised transport and storage solutions. The design of mechanisms to safely constrain nuclear packages, commonly referred to as tie-down systems, has become more challenging as package masses have increased.

This paper focuses on characterising the loading environment that a tie-down system is subjected to using signals processing techniques on previously measured acceleration and strain time histories. The measurements were taken on a tie-down system for a nuclear package, weighing 99.7 tonnes, during a routine rail journey. Similar previous studies on tie-downs have omitted frequency analysis of the measured signals on tie-down systems. A frequency analysis has been used to determine the nature of the loading experienced by a tie-down system and also the extent of vibration transmission into the package. A means for obtaining a suitable filter cut-off frequency is also presented by comparing frequency spectra from different measurement points.

To extract quasi-static accelerations from the raw data, several digital filters have been designed to study their effects on the resulting signals. By comparing the low pass and band pass filtered time histories some insightful trends in the accelerations peaks have been found. To demonstrate what constitutes a good or bad filter design, sensitivity studies have been conducted to show how the distributions of peaks and their statistics are altered significantly with poorer filter choices.

INTRODUCTION

The transportation and storage of nuclear waste is of great importance to sustaining the generation of electricity by nuclear power. Engineers are continuously designing new, heavy, nuclear packages used to transport and store nuclear material. The design of a tie-down system used to restrain a package to its conveyance during transport is an integral part of package design (**Figure 1**). It is recognised by several authors that there is a paucity of experimental data for the design substantiation of tie-down systems [1-4].

Cummings *et al*. [4] presented a method for obtaining real time measured data from a tie-down system suitable for design purposes. A 99.7 tonne package and its tie-down system were transported by rail from Sellafield to Barrow, in the UK. Two hours of data were measured at a sampling frequency of 1200Hz. During data collection, an anti alias, 100Hz low pass, Butterworth filter was used prior to digitisation of the samples. A total of 24 acceleration channels from 8 triaxial accelerometers and 36 channels of strain from 12 strain gauge rosettes were collected.

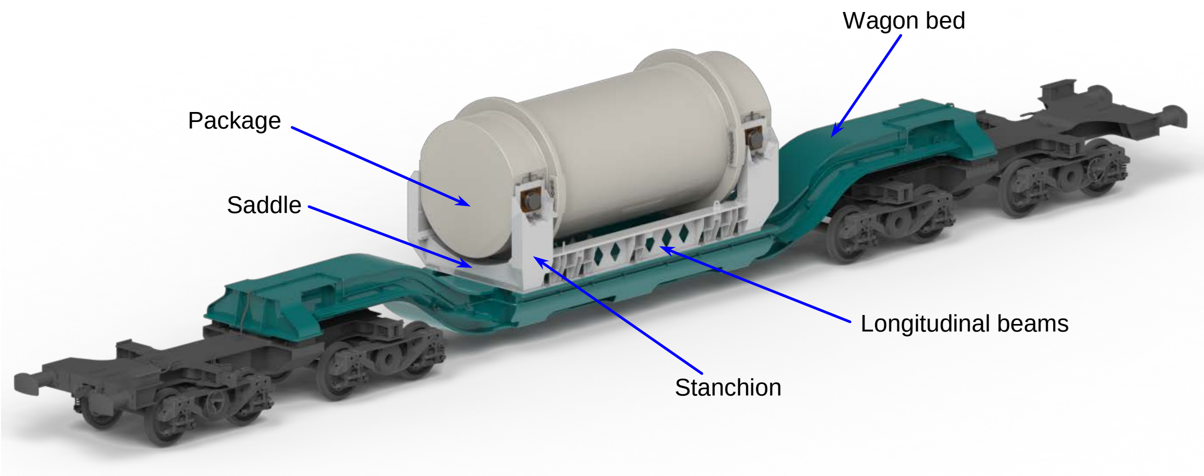


Figure 1 – CAD Model of Rail Wagon and Package

The maximum values collected on each channel show that the highest accelerations were measured nearest the track and the lowest accelerations nearest the package [4]. This paper presents a thorough analysis of the measured time history records that enhances current understanding of the behaviour of tie-down systems during transit by rail specifically for large mass packages (approximately 100 tonnes).

CHARACTERISING THE LOADING ENVIRONMENT

A strain time history has been dissected and certain key features are highlighted (**Figure 2**). Strains have been selected in preference to accelerations because the various types of loads are more easily detectable by visual observation of strain time histories. The time history consists of a number of sections where the measurements reduced to the noise floor of the instrumentation. These sections are called signal dropouts [5]. In the figure several signal dropouts exist and are highlighted in green, these sections all correspond to time periods where the vehicle came to rest.

The strain signal commences at 0 seconds and 0$\mu m/m$ but after 2 hours has drifted to ~25$\mu m/m$. The apparent drift is most likely due to temperature effects and is more pronounced in the strain signal than in any of the acceleration signals. Temperature compensated strain gauges were used so the drift was attributed to real temperature variations e.g. small amounts of thermal expansion on one side of the tie-down system exposed to solar insolation. The drift was considered reasonable due to the low overall values of strain and relatively high signal to noise ratio.

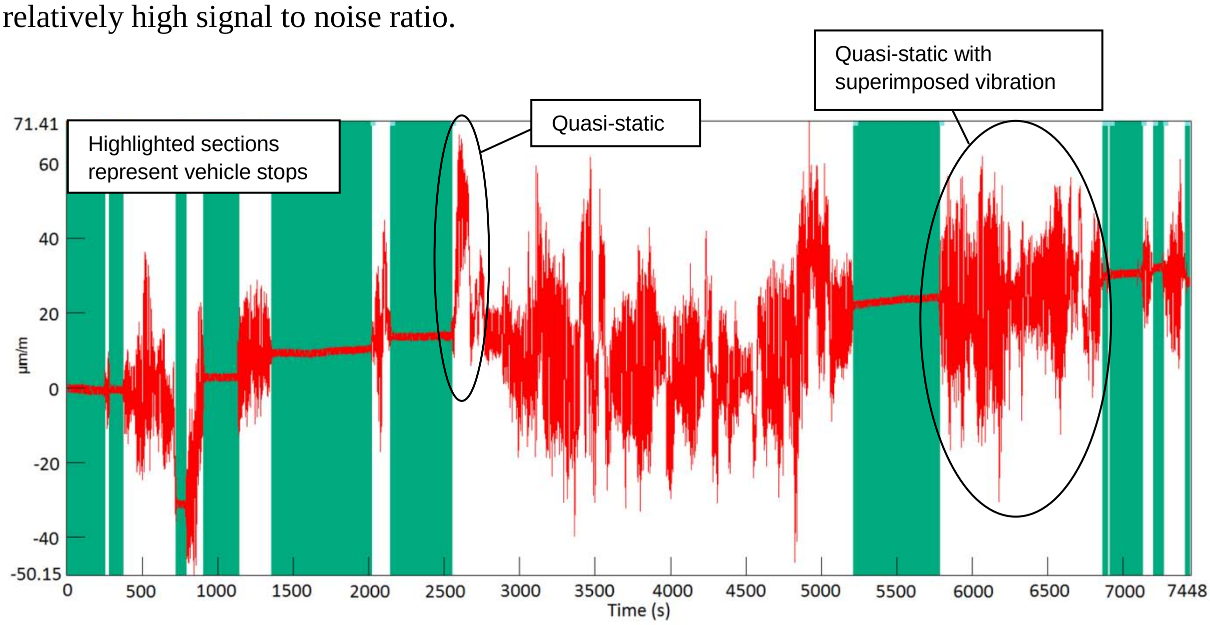


Figure 2 – Typical Strain Time History Characteristics

The structural loading imparted to tie-down systems can be categorised into three main types:

1. Quasi-static
2. Shock
3. Vibration

These categories of loading are evident with the exception of shock loading which, if present, was not defined sufficiently in the signal to be highlighted (**Figure 2**). The source of the loading and its effects on tie-down systems is a critical consideration for their safe structural design.

<u>Quasi Static Loading</u>

Quasi-static loads are generally slowly applied and therefore tend to appear in the lower frequency range i.e. <30Hz [6]. The IAEA Advisory Material [7] does not provide a specific definition for quasi-static loading, however it does offer guidance on filtering time histories to obtain quasi-static loading. It suggests that based on experience 10–20Hz is a suitable cut-off frequency for a package of 100 tonnes. For the purposes of experimental and structural analysis two formal definitions are also provided:-

1. Structural response is time dependent if loading is time dependent. However if loading is cyclic and of frequency less than roughly one-quarter of the structure's natural frequency of vibration, dynamic response is scarcely larger than static response [8].
2. For frequencies considerably below the first resonance or slowly varying time histories the response will be purely quasi-static and reasonable results can be obtained from a static analysis [9].

The definitions suggest that a filter cut-off frequency should be based upon prior knowledge of a tie-down systems first natural frequency. For large mass packages (where the ratio of package mass to conveyance mass is >1) this is not a straightforward calculation. A tie-down system is part of a chain of dynamically coupled systems including the package and rail wagon. This means that isolating the tie-down system and package to perform a modal analysis using Finite Element Analysis (FEA), or a modal test, would not produce correct results. Therefore to obtain an accurate natural frequency estimate a more complex test of the complete system (i.e. vehicle and payload) is required [10]. Multi body dynamics tools may also provide good estimates but parameter identification and validation of these techniques is challenging. To circumvent these problems a practical method of determining a suitable filter cut-off frequency has been devised.

Shock Loading

The nature of shock loading is a short transient burst of energy that occurs rapidly and involves a much larger frequency range. It is a transient response that is initially low, rises to its maximum and then decays. Shock loading will typically excite many natural frequencies of a structure. The resulting structural response consists of a weighted combination of the mode shapes, causing a significantly different response than that due to a quasi-static load [11,12]. Examples of shock loads in tie-down system operations are longitudinal coupling of rail wagons or hump shunting operations. These are considered as normal conditions of transport in the advisory material [7] and can be approximated using explicit FEA [13–16].

Vibration Loading

Vibration can be considered as the residual loading, if quasi-static loads and shock loads are removed from the signal. Vibration is categorised into two types; deterministic and stochastic. Deterministic vibratory loads are generally created by rotating machinery such as piston engines, pumps and turbines. This kind of loading can be measured and fully quantified by test; the measurements can be reproduced exactly in a subsequent test. The loading on a tie-down system during a rail journey cannot be reproduced exactly each time it is measured because it falls into the second class of stochastic or random vibration.

Random vibration can only be quantified using probabilistic methods, therefore a repeat test will produce the same statistical measures such as the root mean square (RMS) value of a signal. Depending on the tie-down system's modal characteristics and the level of energy contained in the input loads random vibration can be treated for design in three different ways:-

1. If the highest frequency content of the loading is less than a quarter of the fundamental natural frequency of the tie-down system or is slowly occurring then the loading can be treated as quasi-static.
2. If the vibration is of sufficient level and close to the fundamental natural frequency of the tie-down system then resonance effects should be accounted for in structural integrity calculations.
3. If the level of vibration loading is insufficient to affect the tie-down system or the lowest frequency of the loading is much higher than the fundamental natural frequency of the tie-down system then the influence of random vibrations can be safely neglected from structural integrity calculations.

To extract the quasi-static content from the signals various digital filter designs have been explored.

DIGITAL FILTERING

The advisory material [7] states that digital filtering of measured acceleration time histories is necessary to define quasi-static loads and to demonstrate design compliance. Filters have been applied for two main purposes in this paper:

1. **Frequency Analysis** – For spectral analysis a filter which minimises pass band ripple and has a high roll-off rate was required to refine the anti-alias filter used during measurement.
2. **Structural Analysis** – For peak analysis the filter applied should minimise time domain ringing (to avoid distorting peaks) whilst providing a roll-off rate sufficient to extract the quasi-static content.

Filter Requirements

The design of a filter to extract low frequency content for structural analysis involves careful consideration of both its frequency and time domain characteristics. In the frequency domain the filter behaviour in the pass band, transition band and stop band can be crucial. The filter roll-off rate dictates at what frequency the minimum stop band attenuation is achieved. Since the preservation of peaks in the pass band is of utmost importance to the structural engineer, pass band ripple should be minimised or eliminated during filter design. In addition filtering causes a phase distortion which has been corrected by the use of a forward-backward filtering algorithm.

Butterworth and Chebyshev Type 1 filters continue to attenuate in the stop band i.e. beyond the cut-off frequency signal attenuation continues indefinitely. Elliptic and Chebyshev Type 2 filters behave differently as they enable control over the permissible stop band attenuation. A minimum acceptable stop band attenuation was set to 1% of the original signal amplitude (-40dB). This was considered sensible to avoid degradation of the filtered signal.

As an example the frequency at which the minimum stop band attenuation of 1st, 2nd, 4th and 8th order, low pass, Butterworth filters with a cut-off frequency of 17.5Hz is shown in **Table 1**. Higher order filters provide better roll-off rates and therefore achieve the desired stop band characteristics at lower frequencies. The transition band is the frequency range between the filter cut-off frequency and the frequencies listed in **Table 1**. The higher the roll-off rate the narrower this band becomes and the more accurately the filter removes content above the cut-off frequency.

Table 1 – Frequencies at which Various Butterworth Filters Attenuate to -40dB

Filter Order	Frequency [Hz]
1	517
2	164
4	55
8	31

Higher roll-off rates are achieved at the expense of poorer filter performance in the time domain. Increasing filter orders, decreases stability and the filter exhibits overshoot and ringing in the time domain, which can degrade its performance. This is the area of most uncertainty in this filter design; firstly because the exact filter cut-off frequency is not known and secondly it is difficult to quantify the level of error that can be attributed to a filter that leaks in the transition band and a filter that overshoots and rings in the time domain.

Sensitivity studies of these filter characteristics have been carried out. Three types of Infinite Impulse Response (IIR) filters were considered for the structural analysis; the Butterworth, Elliptic and Chebyshev Type 1 filters. A 4^{th} order Butterworth filter was chosen initially due to the compromise between overshoot and ringing in the time domain and roll-off rate and stop band attenuation in the frequency domain. For comparison the Bode magnitude and step response plots of the various filters used in the sensitivity study are provided (**Figures 3–4**). For the frequency analysis an 8^{th} order Butterworth filter was used as it has suitable characteristics for refining the roll-off rate and stop band attenuation of the anti-alias filter.

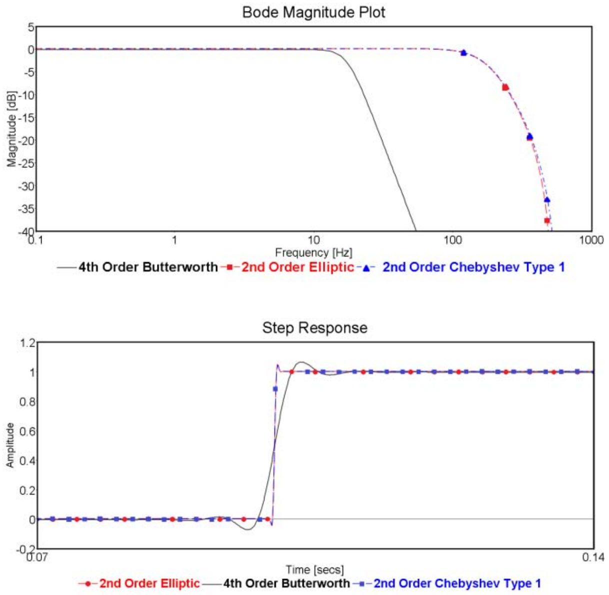


Figure 3 – Filters Designed to Minimise Time Domain Ringing

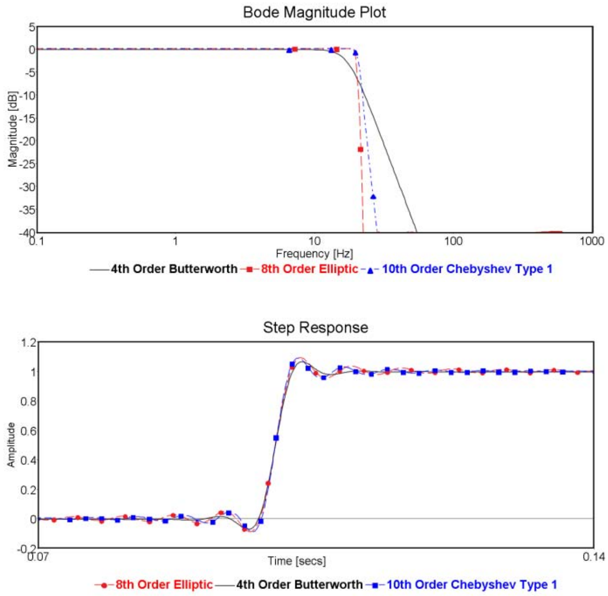


Figure 4 – Filters Designed to Maximise Roll-off Rate

FREQUENCY ANALYSIS

<u>Power Spectral Density</u>

The Power Spectral Density (PSD) enables the study of random time histories in the frequency domain. It is used to show which frequency band(s) of a signal contain the most energy and also highlights resonant frequencies as peaks. Here the PSD is used predominantly to understand the signal content and as a guide for selecting a filter cut-off frequency. The measured time histories have been transformed to PSDs using the following calculation method [17]:-

1. Each time history has been low pass filtered at 100Hz, to refine the anti-alias filter using an 8^{th} order Butterworth filter. This filter is maximally flat in the pass band (no frequency domain ringing), provides good roll-off characteristics and good stop band attenuation [18–20].
2. The time histories were then subdivided into segments of 4096 points (n = 2182 segments) which produced a $\Delta f \sim 0.3$Hz ($T_f \sim 3.41$secs). The selection of this segment size was considered optimal to present results but was occasionally adjusted to ensure the conclusions of the frequency analysis were reasonable.
3. To minimise the effects of leakage, each segment was passed through a Hanning window function and overlapped to minimise random error.
4. The final PSDs were calculated using a standard Fast Fourier Transform method on each segment and a linear average calculated to improve their statistical properties [21–23].

Strain PSDs

The strain PSDs did not contain any spectral information of significance above 40Hz, so a frequency range of 0–40Hz was plotted. The strain PSDs were all very similar so here an example is provided from each leg of the strain gauge rosettes from two of the welded joints (**Figure 5**). Each channel exhibited similar spectral signatures; the energy is distributed in three distinct frequency bands, 0–4Hz, 4–16Hz, 19–29Hz.

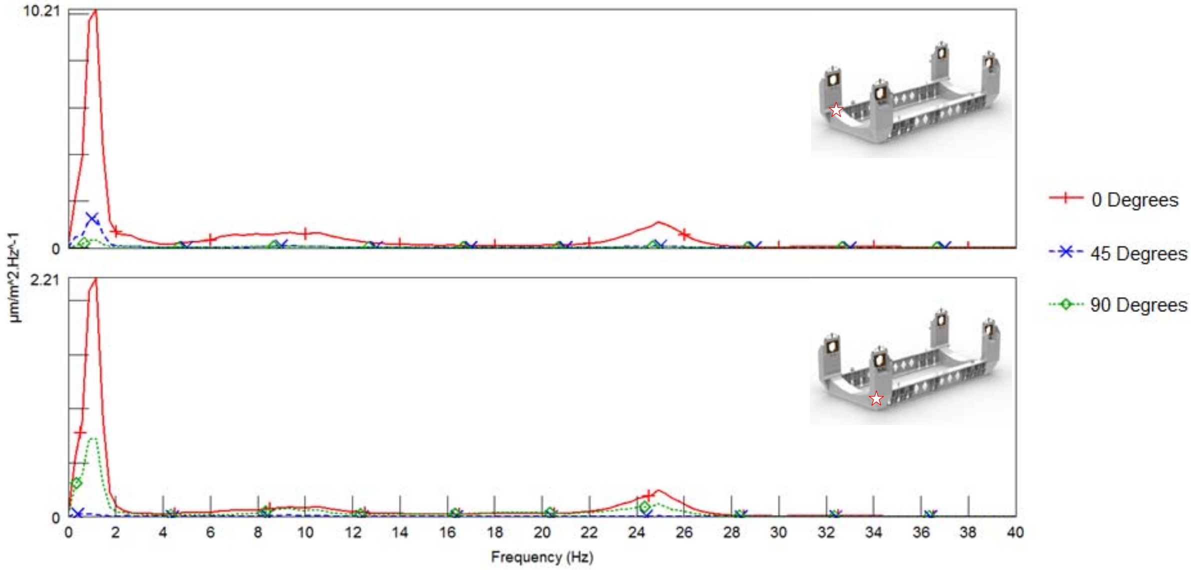


Figure 5 – Example Strain PSDs

Acceleration PSDs

The acceleration PSDs from the tie-down system were found to have similar spectral content. However a comparison between these and the wagon bed and bogie showed differences in the PSDs. There were also differences between the lateral, vertical and longitudinal channels at each location.

The acceleration levels from the wagon bed were higher than that measured on the tie-down system. The highest overall accelerations were measured at the bogie which has a broadband spectrum that is dominated by vertical vibration energy. The difference between the overall magnitude and area under the PSD curves at the bogie and wagon bed shows how much of the vibration energy is attenuated by the rail vehicle suspension system which acts as a mechanical filter (**Figure 6**).

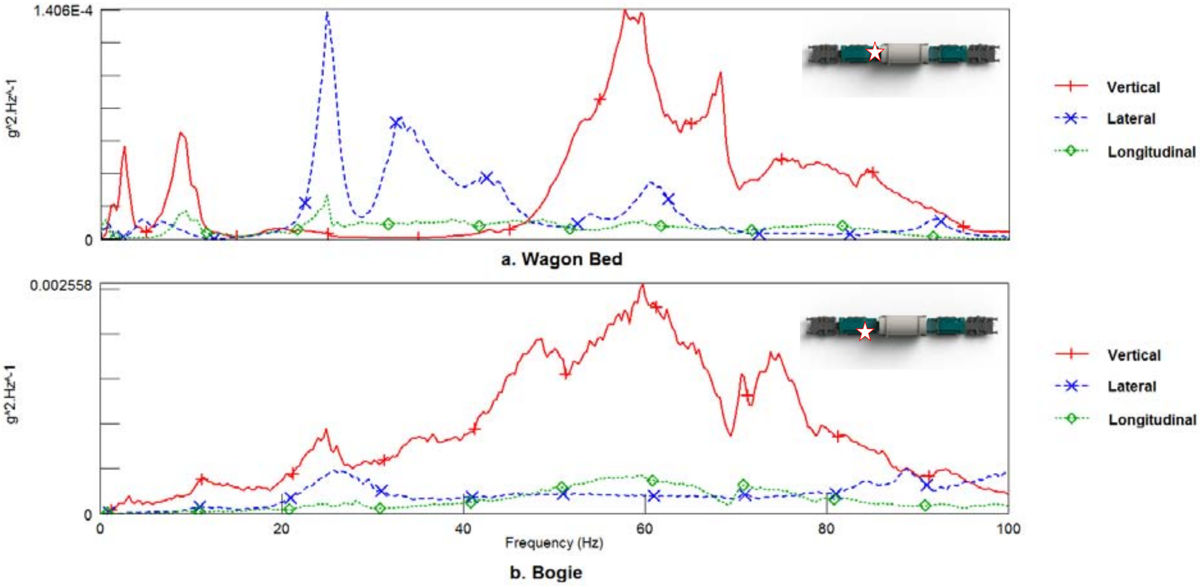


Figure 6 – Acceleration PSDs

The equivalence or similarity of the acceleration PSDs at many of the locations on the tie-down allowed for data reduction. The lid end accelerations have been omitted, concentrating on the slightly higher base end data. Only one of the stanchions is considered since both the base end stanchions spectra were identical.

Accelerometers from the wagon bed, the stanchion and the saddle, have been selected for comparison (**Figure 7**). The wagon bed accelerometer has been included in the selection since this provides the best location to determine what relative motion occurs between the base of the tie-down system and its stanchions.

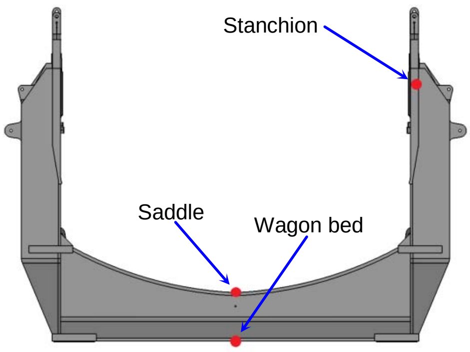


Figure 7 – Accelerometers Selected for Detailed Analysis

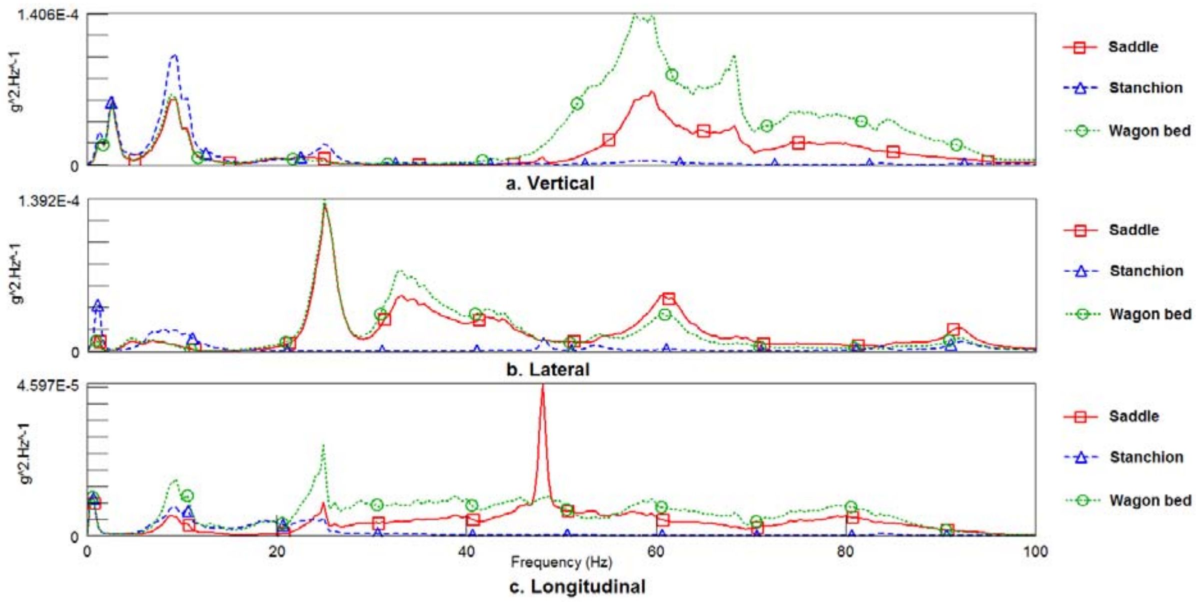


Figure 8 – PSDs of Acceleration Time History

<u>**Vertical Acceleration PSDs at the Wagon Bed and Tie-Down System**</u>

The vertical acceleration PSDs for the three accelerometers are shown (**Figure 8a**). At frequencies below 40Hz three peaks are present between 0–4Hz, 4–16Hz and 19–29Hz, these frequency bands match those in the strain PSDs. At frequencies below 40Hz the energy is marginally higher at the stanchion than at the wagon bed or saddle. Above 40Hz the vibration intensity is much higher at the wagon bed and saddle than it is at the stanchion.

Lateral Acceleration PSDs at the Wagon Bed and Tie-Down System

The lateral acceleration PSDs for the three accelerometers are shown (**Figure 8b**). Below 30Hz peaks occur at the same frequency bands, 0–4Hz, 4–16Hz and 19–29Hz. There are, however, some subtle changes in the vibration signatures.

Below 20Hz the stanchion vibration intensity is marginally higher than the wagon bed and saddle. Above 20Hz the energy levels at the stanchion are significantly reduced whereas at the wagon bed and saddle they increase. There is also a significant peak at 25Hz at the wagon bed and saddle which is not present in the stanchion PSD.

Longitudinal Acceleration PSDs at the Wagon Bed and Tie-Down System

The longitudinal acceleration PSDs for the three accelerometers are shown (**Figure 8c**). Their overall vibration level is much lower than in the vertical and lateral directions. Three small peaks are evident at 9.5Hz, 25Hz and 48Hz. The energy level is very low at the stanchion across the whole frequency range with marginally higher levels of vibration existing at the wagon bed and saddle.

Estimating a Filter Cut-Off Frequency from Displacement PSDs

A method for obtaining a suitable filter cut-off frequency has been devised by comparing PSDs at the stanchion and wagon bed. The acceleration PSDs were integrated twice in the frequency domain to produce displacement PSDs. As the purpose of the filter is to obtain quasi static loads for structural design, displacements PSDs were considered to be more closely related to structural stress and strain than accelerations. Since the standard approach used in tie-down system design is to apply the loading to the centre of mass of the package, it is postulated that the cut-off frequency can be determined as the frequency at which the PSDs become lower at the stanchion, than those at the wagon bed. As a first approximation, three potential cut-off frequencies have been identified from the PSDs (**Figures 9–11**) (**Table 2**). These are similar to the filter cut-off frequency recommended in the advisory material [7].

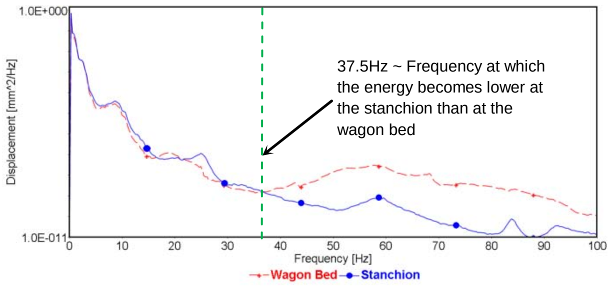


Figure 9 – Vertical Displacement PSD from Wagon Bed and Stanchion

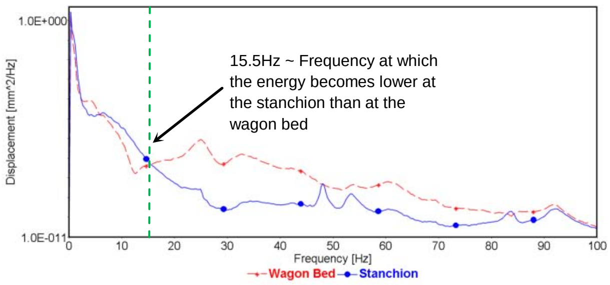


Figure 10 – Lateral Displacement PSD from Wagon Bed and Stanchion

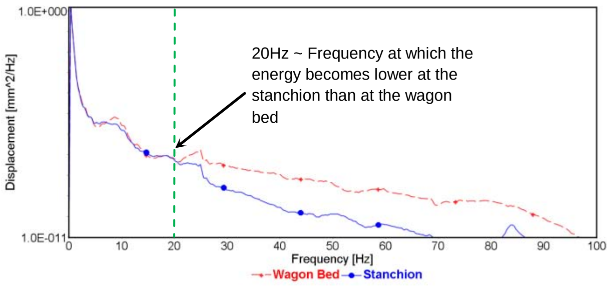


Figure 11 – Longitudinal Displacement PSD from Wagon Bed and Stanchion

Table 2 – Possible Filter Cut-Off Frequencies

	Longitudinal	Lateral	Vertical
Frequency [Hz]	20	15.5	37.5

TRENDS OBSERVED IN THE STATISTICS OF ACCELERATION EXTREMA

Using the filter cut-off frequencies from **Table 2** and selecting a 4th order, forward-backward, Butterworth filter the wagon bed and stanchion time history records have been filtered and the resulting signals have been compared. Three comparative filtering studies have been conducted on the signals by:

1. Low pass filtering with a cut-off frequency of 100Hz
2. Low pass filtering with the cut-off frequencies from **Table 2**
3. Band pass filtering, where the lower cut-off frequencies have been taken from **Table 2** and the upper cut-off frequency set to 100Hz

Some statistics from the resulting time histories are presented (**Tables 3–5**).

Table 3 – Statistics of Acceleration Signals Low Pass Filtered at 100Hz

	Wagon Bed [g]			Stanchion [g]		
	RMS	Max	Min	RMS	Max	Min
Longitudinal	0.03	0.31	-0.43	0.02	0.12	-0.32
Vertical	0.07	0.78	-0.87	0.04	0.26	-0.32
Lateral	0.05	0.57	-0.56	0.03	0.22	-0.18

Table 4 – Statistics of Acceleration Signals Low Pass Filtered at Cut-off Frequencies from Table 2

	Wagon Bed [g]			Stanchion [g]		
	RMS	Max	Min	RMS	Max	Min
Longitudinal	0.02	0.12	-0.12	0.01	0.09	-0.11
Vertical	0.03	0.22	-0.19	0.04	0.23	-0.31
Lateral	0.02	0.11	-0.15	0.02	0.16	-0.16

Table 5 – Statistics of Acceleration Signals Band Pass Filtered, Lower Cut-off Frequencies from Table 2 and Upper Cut-off Frequency of 100Hz

	Wagon Bed [g]			Stanchion [g]		
	RMS	Max	Min	RMS	Max	Min
Longitudinal	0.03	0.33	-0.37	0.01	0.17	-0.25
Vertical	0.06	0.78	-0.81	0.01	0.09	-0.10
Lateral	0.04	0.54	-0.56	0.01	0.14	-0.16

On the wagon bed the largest peak acceleration of -0.87g (highlighted with a red circle) was measured in the vertical direction (**Table 3**). The signal is shown in a close up of this peak and the low pass and band pass filtered signals are also shown for comparison (**Figure 12**). It is evident that the peak is a high frequency oscillation. To understand how this peak transmits through to the package, a similar figure has been created for the stanchion time history during the corresponding time period (**Figure 13**). The overall level of acceleration is lower at the stanchions than the wagon bed (**Figures 12a and 13a**) and the low frequency content is similar at the wagon bed and stanchions (**Figures 12b and 13b**). The peaks in **Figures 12 and 13** are summarised in **Table 6**.

Table 6 – Transmission of Peak Vertical Acceleration from Wagon Bed into Package

	Low Pass 100Hz	Low Pass 37Hz	Band Pass 37.5Hz–100Hz
Wagon Bed [g]	-0.87	-0.15	-0.79
Stanchion [g]	-0.16	-0.14	-0.02

The high frequency peaks measured at the wagon bed have been attenuated by an order of magnitude at the stanchions, from -0.79g to -0.02g (**Figures 12c and 13c**). These results are emphasised in overlaid time history plots of the peak at both the stanchion and wagon bed (**Figures 14 a, b**). **Figure 14a** is low pass filtered at 37.5Hz and **Figure 14b** is band pass filtered between 37.5Hz–100Hz. It is evident that the signals are in-phase and hence at low frequency a state of near rigid body motion exists.

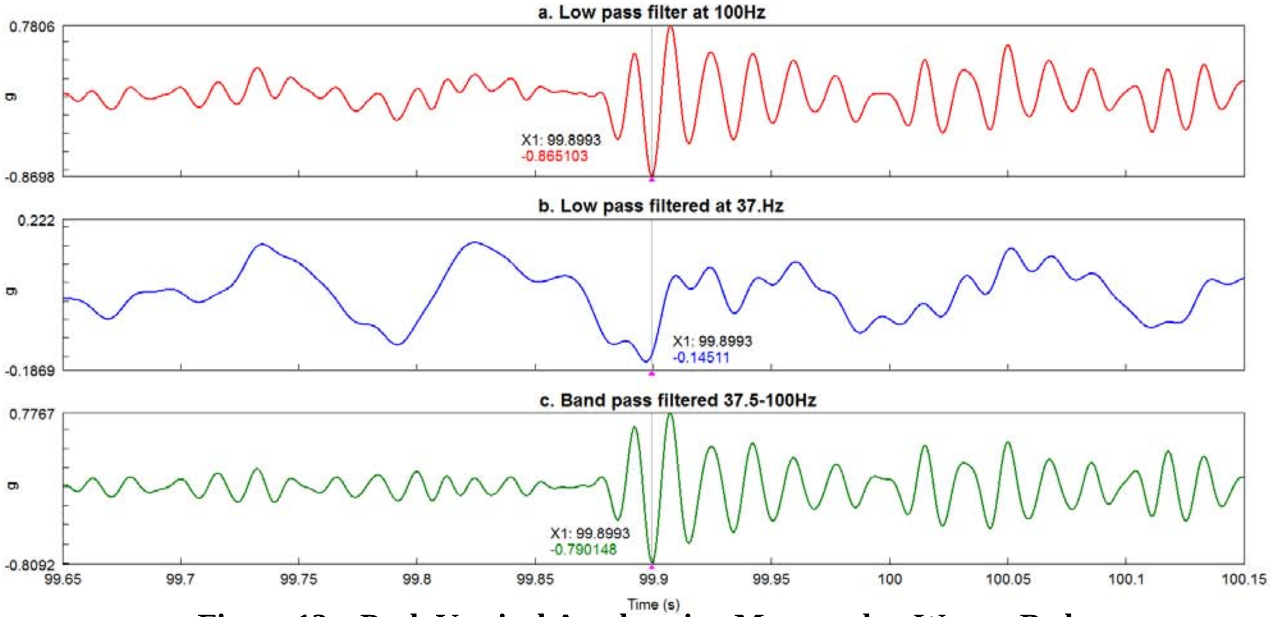


Figure 12 – Peak Vertical Acceleration Measured at Wagon Bed

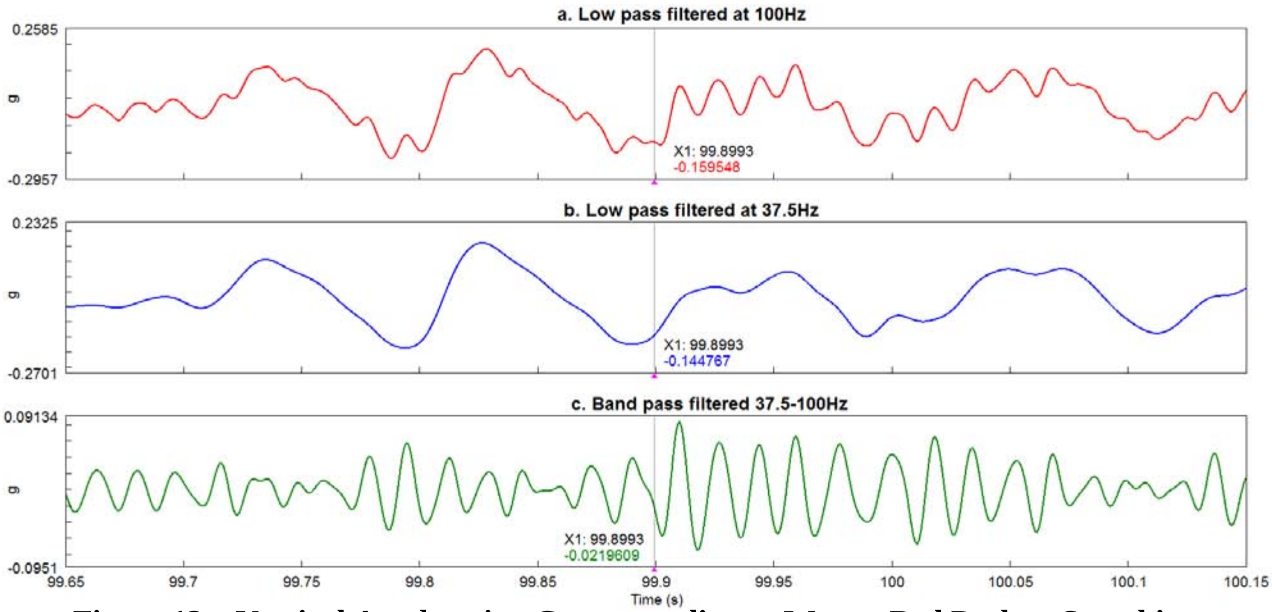
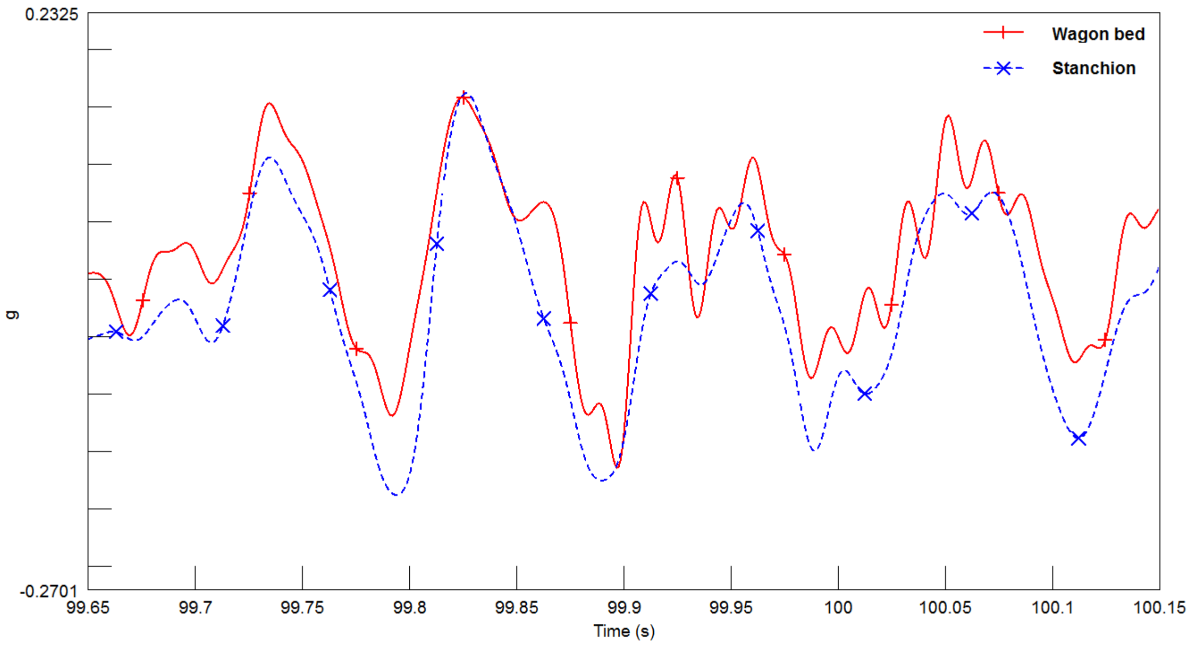
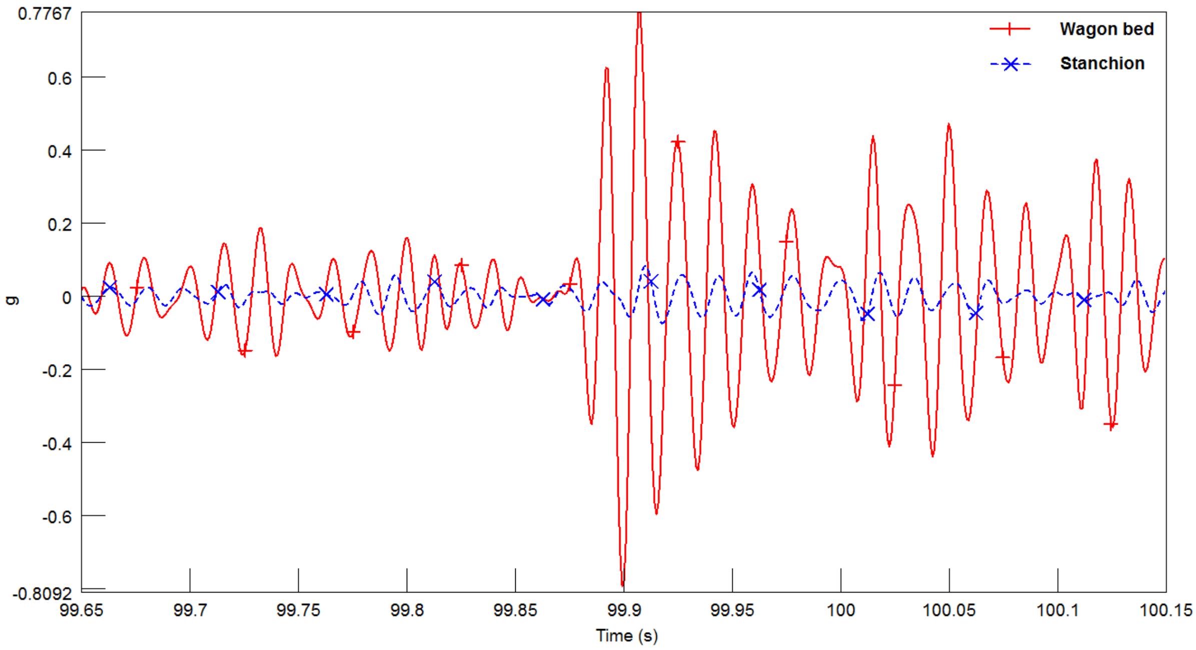


Figure 13 – Vertical Acceleration Corresponding to Wagon Bed Peak at Stanchion



a. Low pass filtered at 37.5Hz



b. Bandpass filtered 37.5 - 100Hz

Figure 14 – Filtered Peak Vertical Acceleration Measured at Stanchion and Wagon Bed

SENSITIVITY OF ACCELERATION EXTREMA DUE TO FILTER DESIGN

For this part of the study the triaxial accelerations measured by accelerometer N, at the base end stanchion have been used. This accelerometer was chosen because it was closest to the centre of mass of the package.

To assess the sensitivity of the acceleration extrema the filter cut-off frequency was set to 17.5Hz. The signals were first filtered with a 4th order, forward-backward, low-pass Butterworth filter (**Figures 3–4**). Four other forward-backward, low pass filters were also used to compare the effects of filter roll-off rates and time domain ringing on the extrema. To minimise time domain ringing a 2nd order Chebyshev Type 1 and a 2nd order Elliptic filter were selected (**Figure 3**). To maximise roll-off rate a 10th order Chebyshev Type 1 and an 8th order Elliptic filter were selected (**Figure 4**). The allowable passband ripple of all the Chebyshev and Elliptic filters was set to -0.0001dB which tends towards a flat passband response at the expense of roll-off rate. The poorer roll-off rate is particularly prominent in the lower order Chebyshev and Elliptic filters. The Elliptic filters stopband attenuation was set to a minimum of -40dB.

The results of the two studies using Chebyshev Type 1 and Elliptic filters are provided for comparison with those from the 4th order Butterworth filter (**Tables 7–9**). The statistics in **Tables 7–9** provide some clues about the distributions that the data sets produce. They also show some discrepancy between the lower order filters designed to minimise time domain ringing and the higher order filters designed to maximise roll-off rate.

Table 7 – Statistics of Filtered Lateral Accelerations

Filter Designed to Maximise Roll-Off Rate				
Filter Type	Mean [g]	Std. Dev	Min [g]	Max [g]
4th Order Butterworth	0.0065	0.0213	-0.1590	0.1590
8th Order Elliptic	0.0065	0.0214	-0.1595	0.1612
10th Order Chebyshev Type 1	0.0064	0.0214	-0.1604	0.1609
Filter Designed to Minimise Time Domain Ringing				
Filter Type	Mean [g]	Std. Dev	Min [g]	Max [g]
4th Order Butterworth	0.0065	0.0213	-0.1590	0.1590
2nd Order Elliptic	0.0065	0.0283	-0.2692	0.2713
2nd Order Chebyshev Type 1	0.0065	0.0289	-0.2817	0.2840

Table 8 – Statistics of Filtered Vertical Accelerations

Filter Type	Mean [g]	Std. Dev	Min [g]	Max [g]
Filter Designed to Maximise Roll-Off Rate				
4^{th} Order Butterworth	-0.0195	0.0259	-0.2434	0.2086
8^{th} Order Elliptic	-0.0195	0.0262	-0.2540	0.2189
10^{th} Order Chebyshev Type 1	-0.0194	0.0264	-0.2605	0.2190
Filter Designed to Minimise Time Domain Ringing				
Filter Type	Mean [g]	Std. Dev	Min [g]	Max [g]
4^{th} Order Butterworth	-0.0195	0.0259	-0.2434	0.2086
2^{nd} Order Elliptic	-0.0195	0.0295	-0.3239	0.2702
2^{nd} Order Chebyshev Type 1	-0.0195	0.0295	-0.3242	0.2715

Table 9 – Statistics of Filtered Longitudinal Accelerations

Filter Type	Mean [g]	Std. Dev	Min [g]	Max [g]
Filter Designed to Maximise Roll-Off Rate				
4^{th} Order Butterworth	-0.0010	0.0106	-0.0932	0.0804
8^{th} Order Elliptic	-0.0010	0.0111	-0.0951	0.0816
10^{th} Order Chebyshev Type 1	-0.0010	0.0114	-0.1051	0.0825
Filter Designed to Minimise Time Domain Ringing				
Filter Type	Mean [g]	Std. Dev	Min [g]	Max [g]
4^{th} Order Butterworth	-0.0010	0.0106	-0.0932	0.0804
2^{nd} Order Elliptic	-0.0010	0.0132	-0.4091	0.2130
2^{nd} Order Chebyshev Type 1	-0.0010	0.0132	-0.4134	0.2247

To understand the likelihood of seeing larger peaks, histograms were constructed from the various filtered signals by carrying out a level crossing analysis. A level crossing analysis is used to count the number of occasions a signal exceeds a given level [24]. By setting intervals and counting the number of crossings within each interval a histogram of the results is obtained. The level crossing histogram is often a precursor for probabilistic analysis on extreme values [22–25].

An example of all the lateral acceleration histograms is shown in **Figure 15**. The abscissa of the histograms is set to the range -0.165 to 0.165 and the ordinate shows the number of crossings. It is evident that the shape and size of the histograms due to the higher order filtered signals are all similar but those due to the lower order filtered signals are significantly different. In particular the lower order filters produced a much larger number of crossings. This is because these filters allow far more of the higher frequency content of the signal through the transition band.

The statistical properties of the higher order filtered signals were very similar and comparable to the results from the 4^{th} order Butterworth filter. The lower order filters produced vastly different histograms due to the leaky nature of the filters in the frequency domain (**Figure 3**). This demonstrates the importance of a high roll-off rate and shows that some overshoot and ringing in the time domain is permissible.

In all cases the distributions indicate clearly that peak accelerations occur rarely and the larger the peaks the less likely they are to occur. This is because the tails of the distributions are exponentially decreasing therefore the likelihood of larger accelerations gets smaller as the peaks get larger.

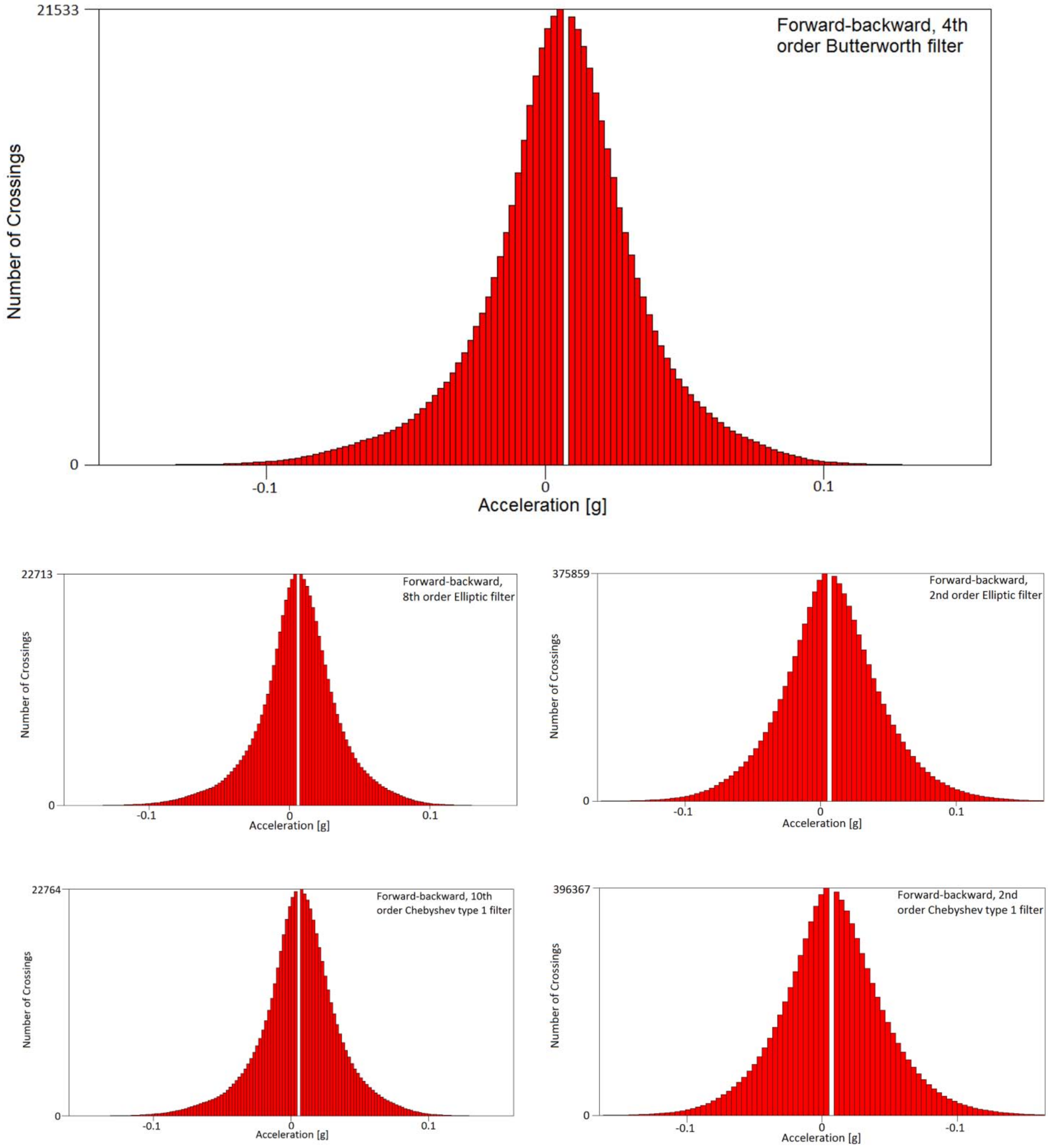


Figure 15 – Level Crossing Histograms of Lateral Accelerations with Various Filters Applied

DISCUSSION AND CONCLUSIONS

Frequency Analysis

The high frequency oscillations (>25Hz) of the wagon bed are attenuated at the stanchions by the large package mass which doesn't have time to respond to the motion due to inertia. The energy at the wagon bed in the lateral and vertical acceleration PSDs appears to be the cause of the peak at 25Hz in the strain PSDs. If a tie-down design was produced using reduced accelerations factors compared with current guidance material, this could influence fatigue life due to the larger number of cycles which occur at high frequencies. It should be noted that no fatigue damage was calculated from any of the measured strain channels for this tie-down system.

In general the loading expected to affect tie-down system design is low frequency i.e. <25Hz. Two main frequency ranges of interest were identified, between 0–4Hz and 4–16Hz. In the range 0–4Hz the strain peak was quite pronounced and corresponded with both lateral and vertical accelerations. In the range 4–16Hz there was no distinct peak in the strain PSD just marginally higher spectral content. The acceleration PSDs differed; the vertical PSD exhibited a distinct peak whilst the lateral PSD displayed a band of increased energy, similar to the strains.

Filter Design

Throughout the study different filters and their characteristics have been assessed to ensure the robustness of the analysis. When designing a filter to obtain quasi-static accelerations where the main concern is preserving the acceleration extrema, the results showed that the roll-off rate of the filter was the most influential characteristic. For this reason when applying higher order, forward-backward filters the resulting signals all possessed similar statistical properties but for 2nd order filters the statistical properties differed.

An estimate of the filter cut-off frequency was based on the postulate that the frequency at which the energy levels at the stanchion fall below those at the wagon bed is the most suitable to use as a cut-off frequency. This is logical since current design practice of tie-down systems is to apply loads at the centre of mass of the package. The results also suggested that the cut-off frequencies were, in general, close to those suggested in the advisory material [7]. As this method is not directly based on the natural frequencies of the tie- down system and package it is not necessarily the most accurate way of separating quasi-static content from the signals.

Peak Analysis

Comparing the quasi-static acceleration factors quoted in the advisory material to the results of this study from the stanchion accelerometer highlights two main differences. The measured accelerations are quoted as approximate as their actual value depends on which filter is used:

1. The longitudinal acceleration factor of 1g for dedicated movements with special rail wagons is an order of magnitude higher than those measured (≈0.1g).
2. The lateral acceleration factor of 0.5g is also considerably larger than the measured accelerations (≈0.16g).

The vertical acceleration factor of 1±0.3g, where 1g is assumed to be the force of gravity, appears to agree with the measured accelerations (downwards ≈0.26g and upwards ≈0.22g). A summary of this comparison is shown in **Table 10.**

Table 10 – Comparison of Measured vs Advisory Acceleration Factors

	TS-G-1 (Table IV.2) [g]	Measured [g]
Lateral	0.5	0.16
Longitudinal	1.0	0.1
Vertical	1.0±0.3	0.22 (U) 0.26 (D)

The vertical quasi-static accelerations at the wagon bed were similar to those at the stanchion and the signals were in-phase. However the acceleration peaks at the wagon bed at higher frequencies were much larger.

The level crossing histograms are approximately bell-shaped but an attempt to achieve a good fit to several statistical distributions failed. From their shape the histograms exhibit an exponentially decaying process and therefore the likelihood of higher acceleration peaks is small.

It is acknowledged that the data set examined is limited due to the length of journey and relatively low vehicle speed, however this is representative of a real routine journey by rail in the UK. The results indicate that there are very large margins of safety between current design parameters and the actual strains and accelerations measured during this test. In conclusion the current acceleration factors, for routine conditions of transport, used for the design of tie-down systems for heavy packages are adequate and appear to be conservative.

FURTHER WORK

If the parameters for tie-down system design for heavy packages were lowered based on the results presented here, then an investigation into fatigue loading on tie-down systems would become necessary. Filtered accelerations may not be appropriate as the fatigue life would depend not only on the quasi-static loading but also on the residual vibratory and shock loading. To optimise package and tie-down system design, which has many benefits to the future of the nuclear transport industry, the expected fatigue life of auxiliary equipment needs to be fully understood. This is particularly true in the UK were rail gauge constraints place limitations on the rail wagon design which severely restrict the size of tie-down systems.

A better method to separate quasi-static loading would also be beneficial. The underlying process of the quasi-static loading is likely to be attributed to rail curvature and undulation and vehicle speed and manoeuvres. The residual shock and vibratory processes that occur during routine conditions of transport would depend more on the vehicle suspension, the wheel-track interface and track irregularities. The track irregularities could be considered as superimposed on the curvature and undulating profile of the rail, which is largely responsible for the quasi-static response. It may be possible to extract improved acceleration data for tie-down system design by careful modelling of these characteristics of the rail environment. This would also assist in understanding the variation in accelerations that arise when nuclear packages of different masses and geometric configurations are transported by rail, thereby allowing for the selection of design parameters to suit a particular tie down system and package configuration.

Using modern computer modelling a parametric study of package mass, vehicle running speed and tie down system stiffness, in conjunction with a measurement programme for validation, would allow for a more scientific basis for revising existing design criteria.

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