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Technical Report

THE USE OF SHOCK ISOLATION MOUNTS IN SMALL HIGH-SPEED CRAFT TO PROTECT EQUIPMENT FROM WAVE SLAM EFFECTS

by

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NSWCCD-80-TR-2017/022 The Use of Shock Isolation Mounts in Small High-Speed Craft to Protect Equipment from Wave Slam Effects

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This report summarizes an investigation of the use of shock isolation mounts in small high-speed craft to protect equipment from wave slam effects. Mount design methods are discussed and calculations are presented that illustrate the significant challenges encountered when attempting to design effective shock isolation systems. An alternative method for minimizing the risk of equipment failure during high-speed operations in rough seas is discussed.

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SYMBOLS, ABBREVIATIONS, AND ACRONYMS

π .... approximately 3.1415926
ξ .... damping ratio
% .... percent
ω .... circular frequency
A_{MAX} .... maximum or peak acceleration
ANSI .... American National Standards Institute
ASRS .... acceleration shock response spectrum
c .... damping coefficient
cg .... center of gravity
DSRS .... relative displacement shock response spectrum
f .... natural frequency
g .... acceleration due to gravity (32.2 ft/sec^2, 9.81 m/sec^2)
Hz .... Hertz (cycles per second)
lbs .... pounds
LCG .... longitudinal center of gravity
k .... stiffness
k_{eff} .... effective stiffness
Kilo 1000
m .... mass
ms or msec .... millisecond
MR .... mitigation ratio
MR_{peaks} .... mitigation ratio based on peak acceleration ratio
MR_{SRS} .... mitigation ratio based on SRS ratio
R .... ratio of shock pulse duration to natural frequency
SDOF .... single degree of freedom
SEC .... seconds
SRS .... shock response spectrum
S&V .... shock and vibration

t .... time
T .... wave impact shock pulse duration
τ_{SYS} or tau .... system natural period
V.. velocity change
VSRS ....Pseudo-velocity shock response spectrum
x, y, z...coordinate axes
X(t)...motion of SDOF system mass
Y(t).. base input motion of SDOF system
Z(t) ....relative displacement of SDOF system
Z_{MAX}.. maximum relative displacement
% ....percent
Δ .... maximum relative displacement
ϖ .... circular frequency

ADMINISTRATIVE INFORMATION

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Summary

This report summarizes an investigation of the use of shock isolation mounts in small high-speed craft to protect equipment from wave slam effects. Mount design methods are discussed and calculations are presented that illustrate the significant challenges encountered when attempting to design effective shock isolation systems. An alternative method for minimizing the risk of equipment failure during high-speed operations in rough seas is discussed.

Introduction

Background

Shock isolation mounts are often installed in small high-speed craft with the intention of protecting sensitive electronics equipment from the damaging effects of severe wave impacts. The mounts can be installed directly under equipment or under a larger cabinet that houses many equipment items, as shown in Figure 1. Wire rope mounts are shown in the photograph on the bottom left and right sides and center of the cabinet, and on top of the cabinet wire rope shock mounts are installed to provide lateral stability.

Figure 1. Shock Mounted Equipment Rack

The use of shock isolation mounts is one of two approaches used to reduce the risk of equipment malfunction or failure due to wave impacts. The other approach, referred to as equipment hardening, is to demonstrate in a laboratory test that the equipment is sufficiently rugged to withstand a shock pulse (or pulses) whose effects are similar to the effects of shocks caused by wave impacts. This is usually achieved using a laboratory shock test machine or drop test apparatus [1 - 4]. This report deals solely with the use of shock isolation mounts as a means of protecting equipment.

Objectives

The objectives of this report are to present shock mount design procedures, to present wave impact acceleration data that characterizes the wave slam shock environment, and to present numerous shock mount design calculations that illustrate the difficulties encountered when attempting to find an acceptable shock mount solution for protecting equipment installations in small high-speed craft.

Scope

The computational results, conclusions, and recommendations presented in this report apply only to the effects of wave impact shock pulses on small high-speed planing craft, mono-hulled or multi-hulled, manned or unmanned, operating at high speeds in rough seas. The results apply to craft with displacements in the range of 14,000 pounds to 170,000 pounds and lengths on the order of 36 ft (11 m) to 85 ft (29.5 m). These ranges correspond to the planing craft whose rough water seakeeping trials provided wave slam acceleration data used to characterize the shape, peak amplitude, duration, and jerk of wave slam shock pulses.

The primary focus of this report is on electrical and electronic communication, navigation, control, computer, and sensor systems. It does not address propulsion machinery or shock isolation seats. Propulsion machinery and power generation components are typically more massive, inherently rugged, and located aft of the longitudinal center of gravity (LCG) where rigid body accelerations caused by wave impacts are smaller. Therefore they do not typically require shock isolation from wave impact effects. The spring-damper assemblies of typical passive shock isolation seats are uniquely different from the small shock isolation mounts typically employed for equipment. The uniqueness has to do primarily with the large strokes and large dampers that require large volumes that is not practical for most equipment installations.

Shock Definition

The word shock in this report implies mechanical shock, as opposed to chemical, electrical, or thermal shock. Mechanical shock is often defined as a non-periodic excitation (e.g., a motion of the foundation or an applied force) of a mechanical system that is characterized by suddenness and severity and usually causes significant relative displacements [2]. It will be shown that shock mount motions caused by acceleration pulses recorded during wave impacts in small high-speed craft fit this definition. Therefore, the terms shock-pulse and wave slam are used synonymously to describe a severe wave-impact pulse.

Shock Mount Design

Process

There are numerous engineering handbooks, texts, and papers that address the effects of shock on equipment and shock isolation design [1 - 11]. There are also engineering consulting firms that specialize in dynamic environments that will design or select appropriate shock mounts based on client needs.

The basic process includes identifying size, weight, shape, and fundamental modes of vibration of the equipment to be protected, describing the characteristics of the shock environment (including pulse direction, shape, amplitude, rate of load application, and pulse duration), and determining excursion space allocations and mount placement options where the equipment is to be located. Table 1 lists an example of a generic process for selecting appropriate shock mounts.

Table 1. Generic Shock Mount Selection Process

Step	Description	Comment
1	Determine weight and CG of equipment	This may involve multiple components in a cabinet
2	Identify mount locations based on surrounding structure, including stabilizing mounts for tall cabinets	If possible place mounts in symmetric locations to avoid x, y, z axis motion coupling and non-uniform mount loads
3	Calculate static load per mount	
4	Define the shock environment	Direction, shape, amplitude, duration, jerk
5	Identify desired shock reduction factor	Typically based on equipment survivability level[1] plus an assumed safety margin
6	Determine allowable excursion space	Include a safety margin
7	Compute mount natural frequencies that satisfy the desired shock reduction factor	Use different values of mount damping to bound a solution
8	Iterate steps 6 and 7	To identify the mount natural frequency that satisfies shock reduction factor and allowable excursion
9	Compute the mount stiffness	Assume linear elastic
10	Compute stiffness per mount	Includes parallel mounts and mounts in series
11	Select appropriate mount from vendor offerings	Static load, capacity, size, stiffness, damping, excursion
Note 1: Survivability levels can be replaced by equipment fragility information, but few items are subjected to fragility testing and fragility is more difficult to determine because of the many possible failure modes		

Isolation Challenges

Shock is controlled by large deflection (i.e., mount excursion) [12], but a large deflection shock mount that protects equipment from short duration pulses (e.g., a blast load) will not be effective for long duration wave impact pulses [2, 12, 13]. Shock isolation mounts must be designed (or selected) for the specific shock environment. In small high-speed craft the biggest challenges are related to having sufficient mount excursion space in tight spaces and being able to determine typical wave impact pulse durations. When pulse duration is known it is then important to be able to quantify the severity of the shock environment, including heave, surge, and pitch motions (sway and roll in some cases may also be a factor). These factors and the importance of knowing characteristics of the multiple degree of freedom shock environment will be discussed later in the report.

An equipment item and its many shock mounts are typically referred to as a mechanical system. Multiple mounts underneath the item and perhaps overhead mounts for stability are treated as parallel or series mounts as shown in Figure 2. In each coordinate axis a single effective stiffness (i.e., spring) can be computed for series and parallel mounts, assuming all mounts are symmetric and x, y, z motions and rotations are uncoupled [2]. This greatly simplifies the computational process and leads to equation (1) for computing the mount system natural frequency (f). The symbol (m) represents the total mass of the isolated system.

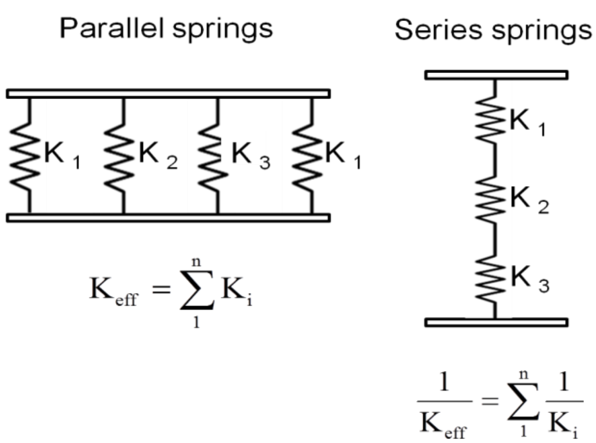

Figure 2. Parallel and Series Mount Configurations

$$f = \frac{1}{2\pi}\sqrt{\frac{K_{eff}}{m}} \qquad \text{Equation (1)}$$

In an x, y, z three-coordinate frame of reference, there are six degrees of freedom, including three translations in x, y, z directions plus rotations around each axis. If the center of gravity of the equipment installation is not symmetric in x, y, and z axes relative to the mount locations, the response motion translations and rotations of the equipment will be coupled. In a coupled system a shock force applied in the x direction will result in responses in the x and y or z directions plus rotations about the z or y axes. If coupling occurs, it is a more complicated response environment, and multiple degree of freedom computational models are required to design effective mount systems.

If the center of gravity of the equipment installation is symmetric in x, y, and z axes relative to the mount locations, the response motion translations and rotations of the equipment can be uncoupled. In an uncoupled system, a shock input in one direction (translation or rotation) results only in a response in that direction. In other words, the shock input and the shock response occur in only one degree of freedom. The one degree-of-freedom condition sufficiently simplifies the dynamic environment such that a relatively simple mathematical model can be used to investigate how key parameters influence effective shock isolation.

The Simple Theoretical Model

The study of the effects of dynamic motions on systems begins with the simple single-degree-of-freedom (SDOF) spring-mass-damper model shown in Figure 3. Its simplicity enables the opportunity to understand the fundamental cause and effect relationships between an input absolute displacement y(t) and the absolute response x(t) of a spring-mass-damper system. The relative displacement across the spring-damper is z(t).

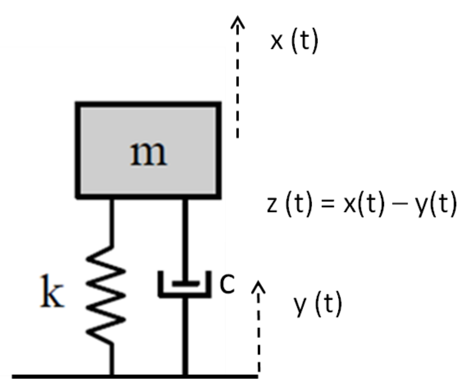

Figure 3. The Single-Degree-of-Freedom Spring-Mass-Damper Model

When a free-body diagram of the forces acting on the mass (m) is constructed for a given base input displacement y(t), it can be shown that a summation of forces acting on the mass results in the second-order partial differential equation (2). A single dot over a parameter indicates the first derivative (velocity) and two dots indicate the second derivative (acceleration). This equation applies for any input motion, including shock pulses or vibration oscillations, but isolation for vibrations is drastically different from isolation for shock pulses. Appendix A presents common misunderstandings between shock isolation and vibration isolation.

$$m\ddot{z} + 2\xi\omega\dot{z} + \omega^2 z = -\ddot{y} \qquad \text{Equation (2)}$$

where the undamped system frequency in Hertz (Hz) is

$$f = \frac{\omega}{2\pi} = \frac{1}{2\pi}\sqrt{\frac{k_{eff}}{m}}$$ Equation (3)

and the damping ratio is

$$\xi = \frac{c}{2m\omega}$$ Equation (4)

The equation for the damped natural frequency can be found in reference [12].When damping is added to the system it can be shown that the natural frequencies are reduced on the order of 4 percent for damping ratios up to 30% and mount natural frequencies from 1 Hz to 4 Hz. Throughout the report the undamped system frequency will be shown in plots of computational results.

The solution of equation (2) is beyond the scope of this report, but many examples of shock input and response calculations will be presented. All calculations were performed using UERDTools, a general purpose software package that includes a finite difference algorithm that solves equation (2) [14]. The calculations are based on the following assumptions [2]. The foundation below the spring-damper is sufficiently rigid such that the motion x(t) does not change the input motion y(t). The input vertical translation results only in vertical response translation of the mass (m). The spring stiffness (k) is linear-elastic with z(t). The damping coefficient (c) is linear-elastic with relative velocity.

Example Shock Pulse Calculation

Figure 4 shows calculated responses of a vertically oriented SDOF passive spring-damper shock isolation system for two different shock inputs. The SDOF system natural frequency is 2 Hertz (Hz) and the damping ratio is 20 percent.

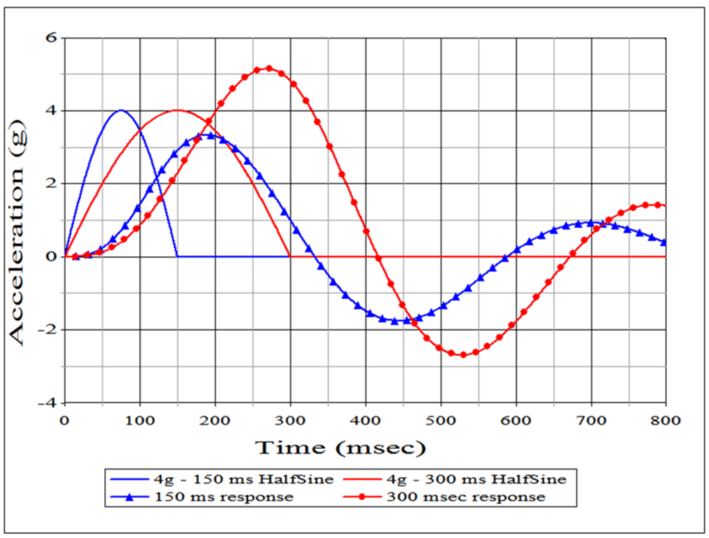

Figure 4. SDOF Model 2 Hz – 20 % Damped Responses

One shock input (blue solid line) is represented by a half-sine pulse with peak amplitude 4 g and pulse duration of 150 milliseconds (msec). The calculated time-history response to this input (shown by the blue line with circle symbols) reaches a maximum acceleration of 3.33 g. This is an example of shock mitigation. The peak acceleration has been reduced.

The second shock input in Figure 4 is also a half-sine pulse with the same 4 g peak amplitude, but it has a 300-msec pulse duration. The SDOF peak response for this input is 5.14 g. This is an example of peak acceleration amplification (i.e., also called dynamic amplification) where the response peak acceleration is greater than the input peak acceleration. This is the first indication that if shock mount properties represented by the spring-mass-damper assembly are chosen incorrectly, the system may amplify the peak acceleration input rather than reduce it (i.e., mitigate the input).

The peak acceleration parameter is an intuitive parameter for comparing shock severity. Most engineering handbooks and texts that deal with shock use the ratio of input and response peak accelerations to construct a shock transmissibility curve. However, peak acceleration is not the only important parameter to consider when comparing shock severity. More will be discussed on the other important parameters later in the report.

Shock Transmissibility

Multiple solutions of equation (2) are used to create the shock transmissibility curves shown in Figure 5 for half-sine shock inputs [2]. A value on the ordinate transmissibility scale less than 1.0 indicates peak acceleration reduction (i.e., mitigation). A value greater than 1.0 indicates dynamic amplification of the input peak acceleration.

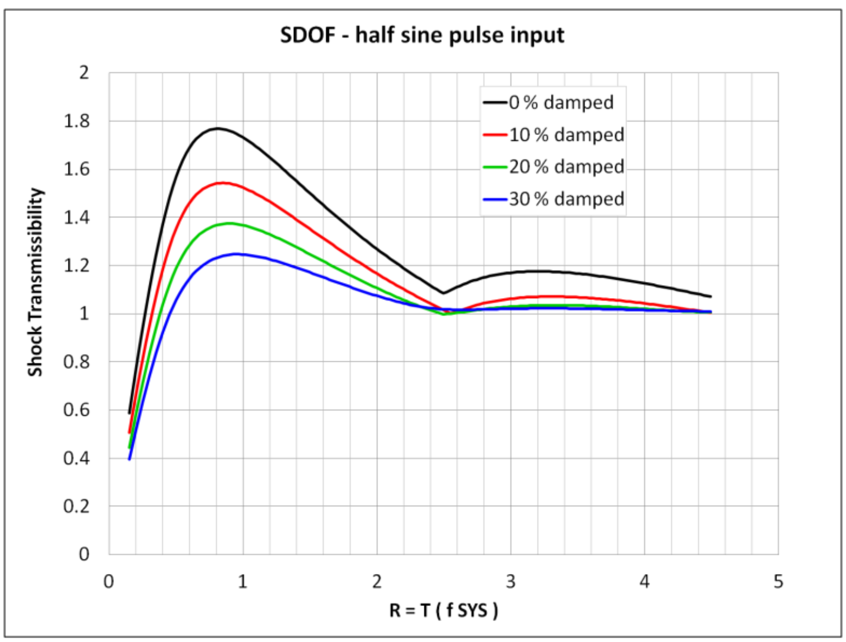

Figure 5. Shock Transmissibility Curve for Half-Sine Pulse Input

In Figure 5 the transmissibility scale (y axis) is the ratio of the response peak acceleration divided by the input peak acceleration. This is also referred to as the peak acceleration mitigation ratio (MR_{peaks}). The x axis is the non-dimensional parameter (R) obtained by multiplying the half-sine pulse duration (T) by the natural frequency (f) of the oscillator. Equation (5) shows that the value R is also identical to the ratio of the half-sine pulse duration (T) and the natural period (τ_{sys}) of the oscillator[1].

$$R = T\,(f_{sys}) = \frac{T}{\tau_{sys}} \qquad \text{Equation (5)}$$

Dynamic Amplification Due to Pulse-Period Mismatch

Dynamic amplification occurs when the mitigation ratio (MR) is greater than 1.0. Figure 5 shows that the theoretical upper limit for amplification is on the order of 1.77 for no damping. Conversely, shock reduction (i.e., shock mitigation) occurs when MR is less than 1.0. For the example calculations shown in Figure 4, the pulse-period ratio (R) from Figure 5 is 0.15 / 0.5 = 0.3 for the 150-msec shock pulse, and 0.3 / 0.5 = 0.6 for the 300-msec shock pulse. The 2 Hz 20% damped shock mount mitigates the 150-msec pulse (MR = 0.83), but it amplifies the 300-msec pulse (MR=1.28). Dynamic amplification is like a sling-shot effect where the response amplitude over-shoots the input amplitude because of the long duration of the input shock pulse relative to the natural period of the mount system. This is illustrated by the red curves in Figure 4. When transmissibility is greater than 1.0, dynamic amplification occurs because of pulse-period mismatch (i.e., the pulse duration is too large for the mount system natural period to achieve mitigation). Appendix B presents example acceleration data from high-speed craft trials that illustrate dynamic amplification caused by pulse-period mismatch (i.e., the sling-shot effect with no bottoming out).

Shock Mitigation

The first step in effective shock mitigation is to achieve a MR value (i.e., shock transmissibility) less than 1.0. This occurs when R values are less than 0.31 for 10% damping, 0.38 for 20% damping, and 0.45 for 30% damping. Using equation (5), this means for 10% damping the shock pulse duration must be less than 0.31 times the natural period of the oscillator. Conversely, it means the natural period of the oscillator must be larger than 3.2 times the shock pulse duration. For a 100-msec wave slam pulse this corresponds to a system natural frequency less than 3.1 Hz. Additional steps for shock mount design related to avoidance of mount bottoming are discussed later in the report.

Step-Velocity Model Assumptions

Many texts and engineering handbooks present shock mount design equations for an assumed shock pulse with a step-velocity time history, where the shock velocity input jumps

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[1] Some texts define the R ratio as 2T divided by system natural period for comparison with vibration transmissibility curves. See Appendix A.

from zero to a maximum value (V) instantaneously. This is referred to as a step-velocity input. The solution for this assumption, derived from conservation of energy equations, results in easy to use algebraic equations for mount deflection and maximum acceleration transmitted above the mounts, respectively [2]. The equations and sample calculations are presented in Appendix C. These simple equations are appropriate for very short duration shocks, like blast loads, but they should not be used for shock mount design for long duration wave impact pulses. The assumption that the velocity change for the half-sine pulse is equal to the velocity change for the step-velocity pulse is not appropriate for wave slam shock pulses.

Wave Slam Shock Pulse Characterization

Shock Pulse Shape

At any location on a craft the direction of the shock pulse during a wave slam can be aligned with coordinate axes X (surge acceleration, positive forward), Y (sway acceleration, positive to port), and Z (heave acceleration, positive up). The shape of the rigid body vertical acceleration when impact forces dominate can be simplified for analytical study as a half-sine pulse [15]. Figure 6 illustrates the half-sine representation of the rigid body vertical acceleration pulse for a wave impact where the largest amplitude is A_{MAX} and the pulse duration is T. The red curve is the unfiltered vertical acceleration recorded near the LCG of a craft. The black curve is the estimated rigid body acceleration obtained using a 20 Hz low pass filter[2]. While the sequence of wave encounters in terms of wave height and time between impacts varies randomly, the vertical response of the craft to a single wave impact appears to be repeatable in shape with amplitudes that vary primarily with speed, craft weight, wave period, and wave height [16].

Shock Pulse Duration

Figure 7 is a plot of shock pulse duration versus vertical peak acceleration recorded for wave impacts at different locations in 16 different craft during head-sea trials in rough water. The peak accelerations are the rigid body peak acceleration estimated using a 10 Hz or 20 Hz low-pass filter. All wave impacts with peaks greater than 3 g were analyzed. Lower amplitude pulses were surveyed for trends. The open circles correspond to shock pulses recorded at the LCG on six craft that weighed from 14,000 pounds to 18,000 pounds [17]. The open squares in the plot correspond to LCG data for six craft that weighed from 22,000 pounds to 38,000 pounds, and the open triangles were recorded at the LCG on a craft that displaced 105,000 pounds. The solid symbols are for the largest peak bow accelerations recorded during a trial on seven craft within the same weight ranges at bow, LCG, and stern locations.

The data indicates that the shortest impact durations regardless of impact severity are on the order of 100 msec, and the longest durations decrease as peak acceleration increases. The

[2] The half-sine approximation is not exact. Many wave impact pulses are skewed slightly to the left, but the half-sine is an appropriate approximation for illustrating the theoretical affects of varying peak acceleration and pulse duration on craft system responses. The exact pulse shape will affect the calculated impact velocity, but this parameter for long-duration pulses can be shown to be important only for correlation with laboratory drop test heights to simulate a wave impact acceleration pulse.

variation in the impact duration for a given peak acceleration is caused by several variables, including craft weight, speed, wave height, impact angle, deadrise, and where the craft impacted the wave (i.e., on the leading flank, crest, or following flank).

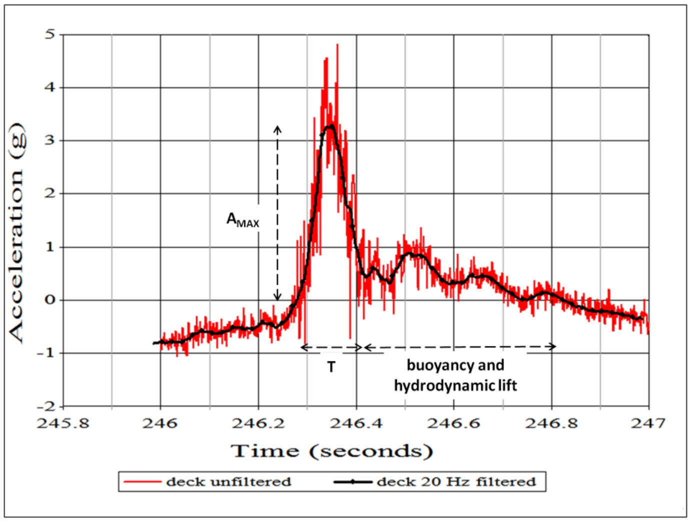

Figure 6. Half-Sine Pulse Approximation[3]

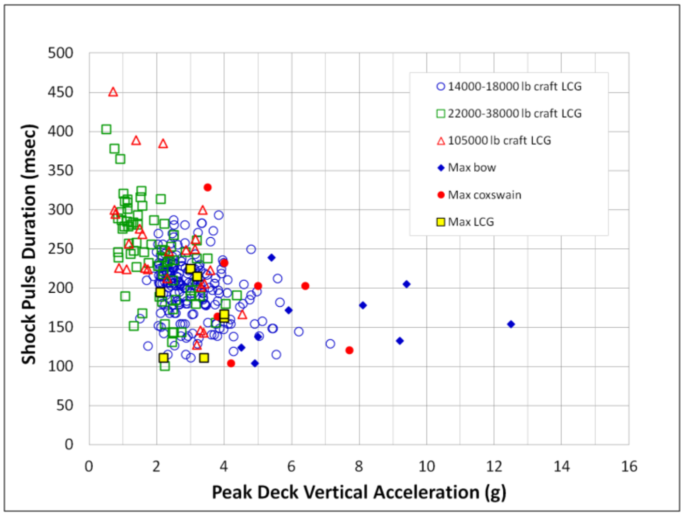

Figure 7. Wave Impact Pulse Duration Trends

[3] The half-sine pulse approximation applies only to that portion of the time history when impact forces dominate over buoyancy and hydrodynamic forces (which dominate after the impact period).

Mitigation Ratio Calculations

Mitigation Ratio vs. Mount Natural Frequency

The shock transmissibility plot shown in Figure 5 was used to create the curves shown in Figure 8 for 10%, 20%, and 30% damping for a 100-msec half-sine shock pulse[4]. Damping values up to 60% were added merely to show the range of the calculations if those damping values could be achieved. The transmissibility axis is relabeled mitigation ratio. For each curve the abscissa R value in Figure 5 was multiplied by 0.10 seconds to yield mount natural frequency in Hz. The curves cross MR = 1.0 at 3.1 Hz for 10% damping, 3.8 Hz for 20% damping, 4.5 Hz for 30%, 5.4 Hz for 40%, 6.9 Hz for 50% and 9.6 Hz for 60% damping. These are the frequencies below which the shock mounts must be designed (or selected) in order to mitigate the 100-msec half-sine shock inputs. It is important to note that the MR value is independent of the peak acceleration input of the half-sine pulse. The MR is only a function of shock pulse duration (T), mount-natural frequency (f), and damping.

Similar calculations were performed for 70 msec[5], 200 msec, and 300 msec shock pulse durations. The results are shown in Figure 9 and tabulated in Table 2. For each pulse duration value, there are five curves corresponding to 10% to 50% damping in increments of 10. The upper curve corresponds to 50% damping, and the lower curve is for 10% damping. The most striking observation shown in Figure 9 is the very low mount frequencies required to mitigate 200 msec (red curves) and 300 msec pulses (blue curves), on the order of 1.5 Hz to 3.5 Hz for 200 msec, and 1.0 Hz to 1.3 Hz for 300 msec. As pulse duration decreases to 100 msec, the frequency range increases to 3 Hz to 7 Hz (green curves). This rise in frequency as pulse duration decreases is illustrated further for a 70 msec half-sine pulse (orange curves), where the frequencies vary from 4.5 Hz to 10 Hz. The curves illustrate the significant importance of the shock pulse duration in shock mount design.

Another observation from Figure 9 and Table 2 is that mount frequencies equal to or greater than 10 Hz are not capable of mitigating typical wave slam shock pulses (i.e., 100 – 400 msec pulses). This explains why shock mounts originally designed for mine blast protection in the 10 Hz to 25 Hz range are not appropriate for wave slam protection. This includes many types of wire rope mounts.

The next step in the shock mount design process is to investigate the relative displacement that must occur for the different frequency values tabulated in Table 2. The relative displacement

[4] A survey of commercially available equipment mounts found typical characteristics of 5 Hz to 25 Hz and 5% to 25% damping for elastomeric and wire rope mounts and up to 33% with friction damping.

[5] A 70-msec wave slam shock pulse has not yet been recorded during seakeeping trials. It is included here to bound the solution space and to anticipate shorter pulse durations for craft that are smaller than those described in the scope of this report.

calculation is important because it determines the excursion space required to prevent mount bottoming.

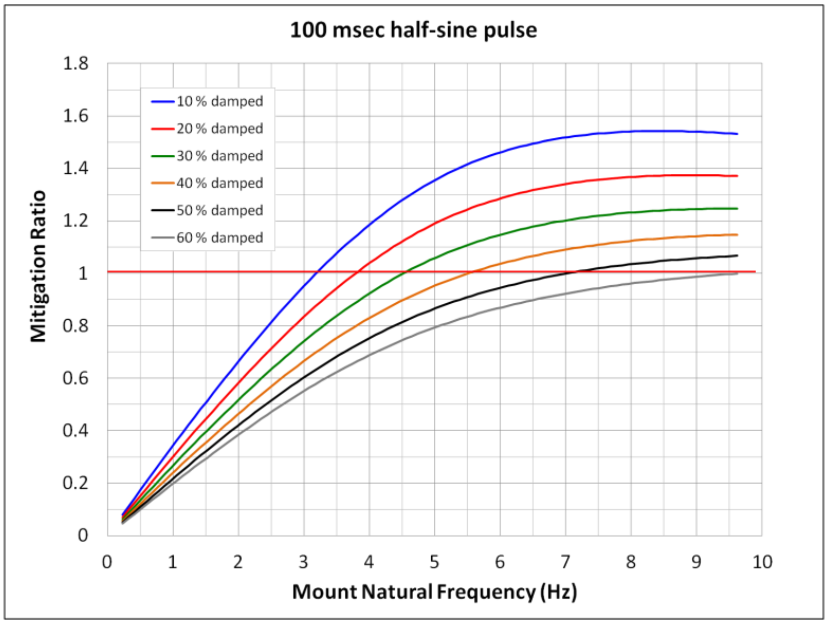

Figure 8. Mitigation Ratios for 100-msec Shock Pulse

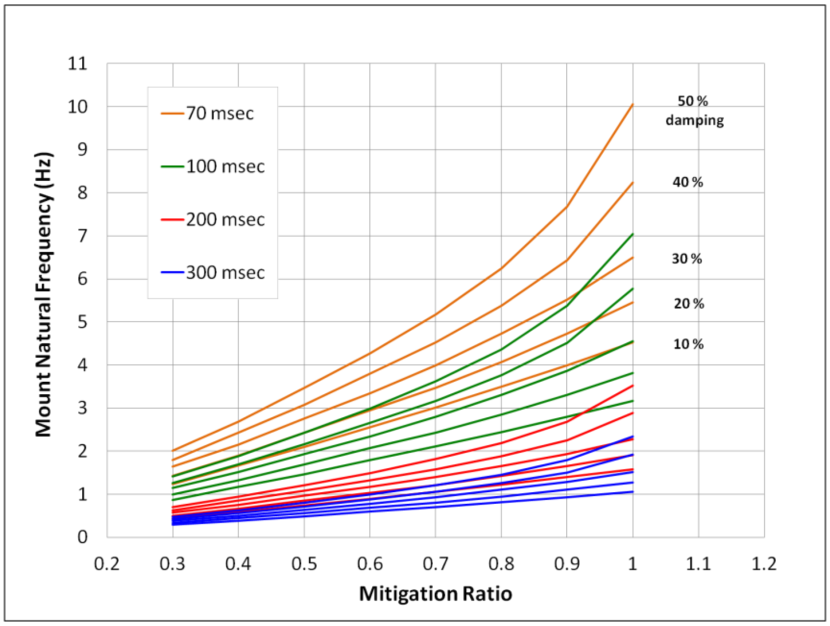

Figure 9. Natural Frequency vs. Mitigation Ratio

Table 2. Mount Frequency and Mitigation Ratio

Shock Pulse Duration	Mitigation Ratio	Frequencies (Hz) needed to achieve MR					
		10% damped	20% damped	30% damped	40% damped	50% damped	60% damped
70 msec	0.3	1.24	1.43	1.64	1.80	2.01	2.16
	0.4	1.67	1.90	2.16	2.43	2.68	2.97
	0.5	2.10	2.43	2.76	3.08	3.47	3.81
	0.6	2.56	2.96	3.34	3.80	4.27	5.23
	0.7	3.01	3.47	4.00	4.53	5.18	5.84
	0.8	3.50	4.07	4.73	5.38	6.24	7.28
	0.9	4.00	4.73	5.53	6.44	7.68	9.40
	1	4.53	5.45	6.50	8.24	10.05	13.87
100 msec	0.3	0.87	1	1.15	1.26	1.41	1.51
	0.4	1.17	1.33	1.51	1.7	1.88	2.08
	0.5	1.47	1.7	1.93	2.16	2.43	2.67
	0.6	1.79	2.07	2.34	2.66	2.99	3.66
	0.7	2.11	2.43	2.8	3.17	3.63	4.09
	0.8	2.45	2.85	3.31	3.77	4.37	5.1
	0.9	2.8	3.31	3.87	4.51	5.38	6.58
	1	3.17	3.82	4.55	5.77	7.04	9.71
200 msec	0.3	0.435	0.5	0.575	0.63	0.705	0.755
	0.4	0.585	0.665	0.755	0.85	0.94	1.04
	0.5	0.735	0.85	0.965	1.08	1.215	1.335
	0.6	0.895	1.035	1.17	1.33	1.495	1.83
	0.7	1.055	1.215	1.4	1.585	1.815	2.045
	0.8	1.225	1.425	1.655	1.885	2.185	2.55
	0.9	1.4	1.655	1.935	2.255	2.69	3.29
	1	1.585	1.91	2.275	2.885	3.52	4.855
300 msec	0.3	0.29	0.33	0.38	0.42	0.47	0.50
	0.4	0.39	0.44	0.50	0.57	0.63	0.69
	0.5	0.49	0.57	0.64	0.72	0.81	0.89
	0.6	0.60	0.69	0.78	0.89	1.00	1.22
	0.7	0.70	0.81	0.93	1.06	1.21	1.36
	0.8	0.82	0.95	1.10	1.26	1.46	1.70
	0.9	0.93	1.10	1.29	1.50	1.79	2.19
	1	1.06	1.27	1.52	1.92	2.35	3.24

Mount Relative Displacement Calculations

Computational Methods

The SDOF model for a 3 Hz mount with 30% damping was used to create the shock input acceleration and SDOF response acceleration curves shown in Figure 10. The half-sine inputs had pulse durations that varied from 70 msec to 300 msec. The predicted results show that the 3 Hz 50% damped mount can mitigate the 70 msec (MR = 0.54) and 100 msec half-sine pulses, (MR = 0.74), but it amplifies the 200-msec (MR = 1.14) and 300-msec (MR = 1.24) pulses. These estimated MR values are based on ratios of the peak accelerations above and below the mounts. The time histories were then double integrated to yield absolute displacements of the base input to the SDOF model and the absolute displacement of the mass. These absolute displacements were then subtracted, as shown in Figure 3, to yield the predicted relative displacement z (t). Figure 11 shows the predicted relative displacements for all five of the 4 g half-sine input pulses shown in Figure 10. The negative values indicate mount compression. The maximum relative displacement for each response was then tabulated.

In Figure 11 the relative displacements for the mitigated input pulses in Figure 10 are 4.1 inches for the 70 msec pulse (MR = 0.54) and 5.7 inches for the 100 msec pulse (MR = 0.74).

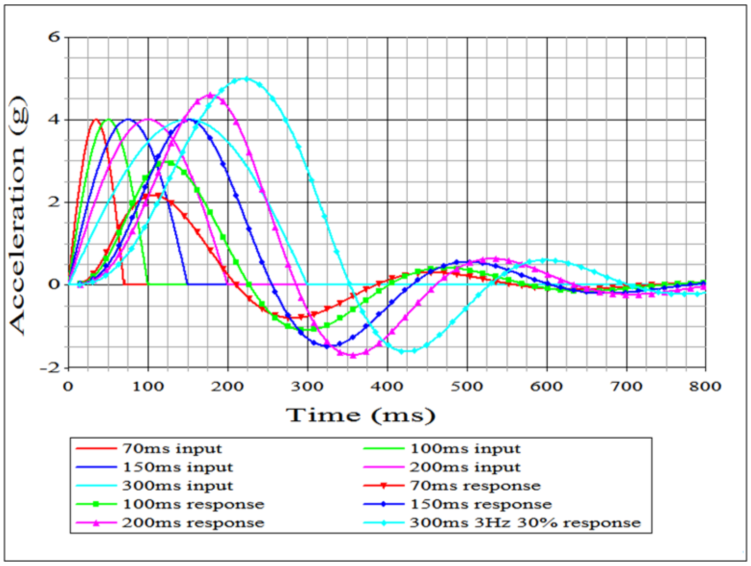

Figure 10. Input Shock Pulses and Predicted SDOF Responses

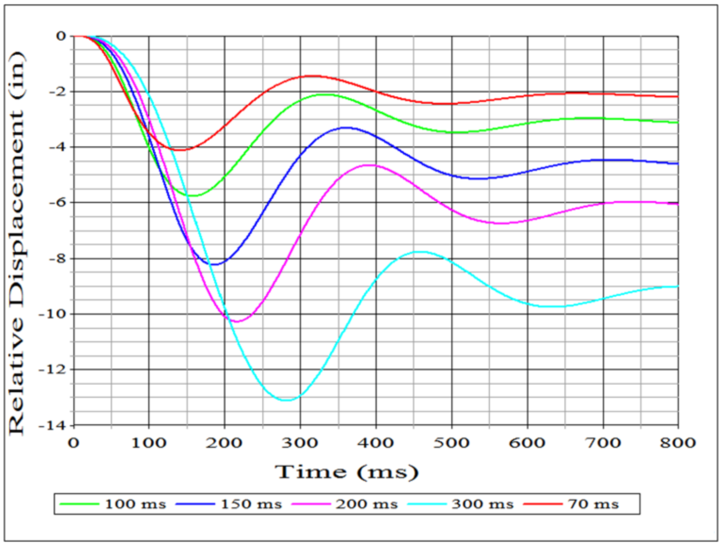

Figure 11. Predicted Relative Displacement for 3 Hz 30% Damped Mount

The acceleration curves shown in Figure 10 were then used to compute a mitigation ratio using equation (8) that accounts for above and below mount differences in pulse shape, duration, jerk, and peak acceleration.

$$MR_{SRS} = \frac{SRS_{\text{ABOVE MOUNT RESPONSE}}}{SRS_{\text{BELOW MOUNT INPUT}}} \qquad \text{Equation (8)}$$

For these calculations the SRS is the relative displacement shock response spectrum computed using 9% damping. Appendix D describes the SRS, and Appendix E explains the rationale for using the ratio of input and response SRS to compute the mitigation ratio.

Calculation Results

Half-sine pulse calculations similar to those shown in Figures 10 and 11 were completed for 2 Hz, 3 Hz, and 4 Hz mounts, for 70 msec, 100 msec, 150 msec, 200 msec, and 300 msec pulses with peak accelerations that varied from 2 g up to 8 g. The results are tabulated in Table 3. Blank spaces in the table correspond to peak accelerations and pulse durations not observed in the data base shown in Figure 7. The column in Table 3 labeled MR (i.e., mitigation ratio) shows SRS mitigation ratios computed using equation (8).

Table 3. Predicted Shock Mount Maximum Relative Displacements

Mounts	Pulse Duration	MR	Mount Relative Displacement (inches)						
	msec		2 g	3 g	4 g	5 g	6 g	7 g	8 g
2 Hz 20% damped	70	0.41	2.96	4.45	5.93	7.41	8.89	10.37	11.85
	100	0.58	4.18	6.26	8.36	10.45	12.54	14.63	16.72
	150	0.83	6.08	9.12	12.16	15.2	18.25	21.29	24.33
	200	1.03	7.77	11.66	15.55	19.44	23.32		
	300	1.28	10.37	15.55	21.74	25.92			
2 Hz 30% damped	70	0.36	3.13	4.69	6.26	7.82	9.39	10.95	12.52
	100	0.51	4.42	6.64	8.84	11.07	13.28	15.5	17.71
	150	0.74	6.49	9.73	12.97	16.22	19.47	22.72	25.97
	200	0.92	8.38	12.58	16.27	20.97	25.16		
	300	1.14	11.56	17.34	23.12	28.9			
3 Hz 20% damped	70	0.6	1.94	2.92	3.87	4.85	5.81	6.81	7.78
	100	0.83	2.7	4.05	5.39	6.75	8.11	9.44	10.81
	150	1.12	3.79	5.68	7.58	9.47	11.37	13.26	15.16
	200	1.28	4.61	6.91	9.22	11.51	13.83		
	300	1.37	5.49	8.23	10.98	13.72			
3 Hz 30% damped	70	0.54	2.06	3.09	4.12	5.15	6.18	7.22	8.25
	100	0.74	2.88	4.32	5.77	7.21	8.65	10.08	11.53
	150	0.99	4.11	6.17	8.23	10.29	12.35	14.41	16.47
	200	1.14	5.13	7.7	10.27	12.84	15.41		
	300	1.24	6.56	9.84	13.12	16.4			
4 Hz 30% damped	70	0.69	1.52	2.28	3.04	3.8	4.55	5.32	6.09
	100	0.92	2.09	3.14	4.19	5.24	6.29	7.34	8.39
	150	1.14	2.89	4.33	5.78	7.22	8.67	10.11	11.56
	200	1.23	3.46	5.2	6.93	8.67	10.4		
	300	1.22	4.23	6.35	8.47	10.59			
4 Hz 40% damped	70	0.62	1.62	2.44	3.25	4.06	4.87	5.69	6.5
	100	0.83	2.26	3.39	4.52	5.66	6.79	7.92	9.06
	150	1.03	3.2	4.8	6.41	8.01	9.61	11.22	12.82
	200	1.12	3.98	5.98	7.97	9.97	11.96		
	300	1.14	5.25	7.89	10.53	13.16			

Relative displacement calculations for shock mounts with a natural frequency of 1 Hz were only performed for 2 g and 6 g half-sine pulses (with 70 msec to 300 msec durations) to illustrate the impractical mount relative displacements (i.e., excursions). Figure 12 shows that 1 Hz yields mitigation ratios less than 0.58 for durations less than 200 msec, but the relative displacements required to achieve these results are not practical for high-speed craft.

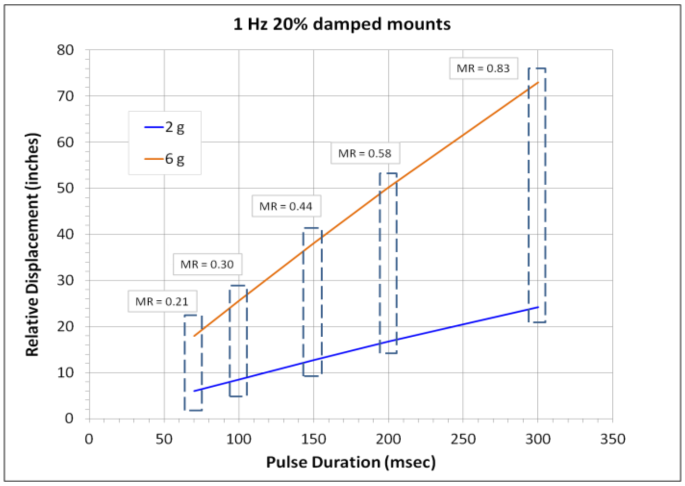

Figure 12. Impractical Displacement for 1 Hz Shock Mounts

The mitigation ratio column in Table 3 shows that the 2 Hz to 4 Hz mounts included in the analysis cannot mitigate 300 msec pulses. Only the 2 Hz 30%-damped mounts mitigates 200 msec pulses (MR = 0.92). Appendix F presents plots of maximum relative displacements and mitigation ratios listed in Table 3.

Table 4 lists only the mounts with MR values less than 1.0. The results show that the 3 Hz and 4 Hz mounts can only mitigate 100 msec pulses, and 2 Hz and 3 Hz mounts can only mitigate 100 msec and 150 msec pulses.

The color code in Table 4 corresponds to mount size (i.e., excursion space) that can accommodate the predicted relative displacements[6]. For example the green color is for 6-inch mounts. In other words, the predicted relative displacements in green cells are all less than 6 inches. Yellow is for 8-inch mounts that can accommodate the predicted displacements between 6 inches and 8 inches (as well as the 6-inch mount displacements). Orange is for 10-inch mounts, and blue is for 12-inch mounts. The 12-inch mount was assumed to be the upper bound practical mount for discussion purposes. Relative displacements greater than 12 inches are not color shaded in the figure. Table 4 shows the very narrow range of practical mount designs (i.e., frequency and damping) that can mitigate only the shorter duration pulses (i.e., 100 – 150 msec).

[6] These values do not include a design margin for excursion space.

Table 4. Mount Relative Displacements for MR Less Than 1.0

Mounts	Pulse Duration msec	MR	Mount Relative Displacement (inches)						
			2 g	3 g	4 g	5 g	6 g	7 g	8 g
2 Hz 20% damped	100	0.58	4.18	6.26	8.36	10.45	12.54	14.63	16.72
	150	0.83	6.08	9.12	12.16	15.2	18.25	21.29	24.33
2 Hz 30% damped	100	0.51	4.42	6.64	8.84	11.07	13.28	15.5	17.71
	150	0.74	6.49	9.73	12.97	16.22	19.47	22.72	25.97
	200	0.92	8.38	12.58	16.27	20.97	25.16	n	n
3 Hz 20% damped	100	0.83	2.7	4.05	5.39	6.75	8.11	9.44	10.81
3 Hz 30% damped	100	0.74	2.88	4.32	5.77	7.21	8.65	10.08	11.53
4 Hz 30% damped	100	0.92	2.09	3.14	4.19	5.24	6.29	7.34	8.39
4 Hz 40% damped	100	0.83	2.26	3.39	4.52	5.66	6.79	7.92	9.06

Color Code (inches)				
	6	8	10	12

The computational results of Table 3 and the color code shown in Table 4 are combined to create Figure 13. It shows the original shock pulse duration data base from Figure 7 with color coded symbols to show mount design options. The plot contains 324 data points (i.e., individual wave impacts). The 243 red-x symbols (i.e., 75 percent of the database) correspond to all pulse durations that cannot be mitigated (i.e., MR > 1.0) by 2 Hz to 4 Hz mounts with 20% to 40% damping.

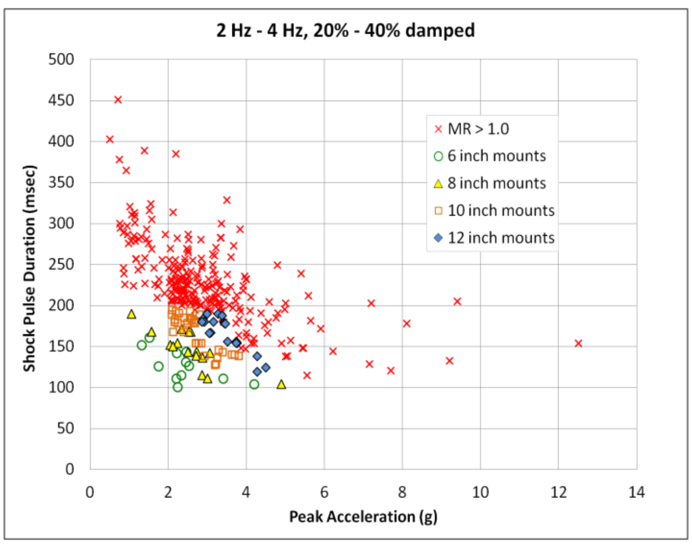

Figure 13. Shock Mount Design Solutions

Seventeen data points (5.2%) represented by the green circle symbol can be mitigated using 6-inch mounts. Thirty-three data points (10.1%) can be mitigated by yellow triangle 8-inch mounts (Note: Thirty three includes the 6-inch mount data points). Sixty-six data points (20.3%) can be mitigated by orange square 10-inch mounts (i.e., includes the 6-inch and 8-inch mount data points), and eighty-one data points (25%) can be mitigated by blue diamond 12-inch mounts.

Observations

Low Mount Frequencies

Table 2 shows that the typical pulse durations in the 100-msec to 300-msec range require very low frequency mounts to achieve mitigation ratios less than 1.0. Even with unrealistically high damping of 50% and 60% the results indicate that mounts with frequencies higher than 5 Hz cannot provide protection for typical wave slam pulses. This precludes the use of typical 10 Hz to 20 Hz wire rope mounts for wave slam protection in high-speed craft.

Mounts with natural frequencies less than 2 Hz can provide mitigation ratios on the order of 0.6 or less depending upon pulse duration, but this frequency range is probably only practical for shock isolation seats with large damping or seismic protection of equipment in earthquake design for buildings [18, 19]. The very large relative displacements for mounts with natural frequencies this low are not practical for equipment in small high-speed craft.

Small Solution Space

The database presented in this report does not include all wave impacts recorded during rough water trials, so it is not a complete data set. All wave impacts for 16 different craft with peaks greater than 3 g were analyzed and lower amplitude pulses were surveyed for trends. The plot is useful for identifying the design solution space for the more severe impacts. Figure 13 showed that only 25 percent of the database presented in this report can be mitigated with 3 Hz - 20% or 30% damped mounts with 12-inch excursion space. In addition 4 Hz – 30% or 40% damped mounts with 10-inch excursion space can mitigate only 20.3% of the impacts. This means that 3 Hz or 4 Hz mounts with 10-inch to 12-inch excursion space cannot provide equipment protection for approximately 70% to 80% of the impacts shown in Figure 13.[7]

Large Excursion Envelopes

The standard practice for shock mount design is to add a safety margin to ensure adequate installation excursion space [12]. This is an approximate approach for including uncertainties related to multi-axis shock inputs or cross-axis-response coupling due to un-symmetric mount positioning. For example, the design mount excursion envelopes for the 10-inch relative displacement shown in Figure 13 would correspond to 11.5-inch to 15-inch excursion envelopes if safety margins of 1.25 to 1.5 were used. Likewise the calculated 12-inch relative displacement would require 15-inch to 18-inch mounts. These are impractical excursion envelopes for small

[7]The market survey suggests current commercial off-the-shelf mounts with 8-inch to 15-inch excursions are not available.

high-speed craft, and even if it could be achieved, only 20% to 25% of impacts shown in Figure 13 could be mitigated.

Large Peak Accelerations

It is also standard practice in shock mount design to include a safety margin on the peak shock input acceleration used to compute mount relative displacements. The calculations shown in Table 3 only go up to a peak input acceleration of 8 g, because values larger than this are typically observed only for bow locations. The 8 g limit was selected to reduce the time required to calculate a large number of predicted excursions. If a 25% margin were included for higher peak accelerations, the 8 g maximum value used herein would become 10 g or higher. This increase in peak acceleration would also increase the design excursion envelope beyond 15 to 18 inches.

Complex Shock Environment

The calculations presented herein are for ideal single-degree-of-freedom shock inputs and responses with assumed shock mount placement that only results in uncoupled response motions. These assumptions provide tractable computational steps that yield results useful as a baseline for understanding how shock mounts respond to long-duration wave impact shock pulses. In small high-speed craft the wave impact environment is not well modeled as a single-degree-of-freedom. Craft pitching may add significant fore-aft shock impulses in addition to the vertical shock pulse. Multiple axis shock impulses and cross-axis coupling due to lack of mount symmetry will likely occur. This means the predicted excursion envelopes listed in Table 4 are lower-bound estimates. More accurate engineering estimates can only be obtained using two or three degree-of-freedom system models [9, 10]. These more complex mathematical modeling approaches require user expertise and unique software that adds time and cost to mount design investigations.

Repeated Wave Impact Shocks

Figure 4 shows predicted acceleration responses of a 2 Hz isolation system with 20% damping for single wave impacts with 150-msec and 300-msec shock pulse durations. The predicted response accelerations have damped periods for the first oscillation of response on the order of 600 msec to 700 msec. Successive wave impacts may occur on the order of every 500 msec to 1000 seconds (depending upon the wave height environment, craft length, and craft speed). In this circumstance the response motion of the 2 Hz 20% damped isolation system may not damp out before the next wave impact. If this occurs a very complex mathematical model with multiple wave impact pulses would be required to achieve an effective shock mount design. This would also require user expertise and unique software that adds time and cost to mount design investigations.

Alternative Risk Mitigation

The shock isolation theory and the results of the calculations for SDOF systems presented herein suggest that effective equipment wave impact shock isolation will be very hard to achieve in small high-speed craft. The complex environment precludes the effective use of simple SDOF computational approaches, and the more complex (i.e., two or three dimension solutions) and expensive computational techniques will likely result in large excursion allowances in all axes and shock absorber sizes that are impractical for the majority of equipment installations.

The market survey conducted for this report provides a sampling of available mount characteristics. Unique mounts not included in the survey may already be available that could provide tractable isolation solutions for craft. Shock isolation could therefore be pursued for very sensitive electronics or expensive hardware, but caution is advised because effective solutions will likely only be achieved by experienced shock isolation designers who pursue unique isolation strategies (e.g., seismic-mass dampers, pneumatic or hydraulic isolation systems).

An alternative approach to mitigating the risk of equipment malfunction or failure due to wave impact shock is equipment hardening (i.e., laboratory shock testing prior to installation in a craft). Laboratory test requirements for equipment hardening are presented in Appendix G. Subjecting an equipment item to the testing described in the appendix reduces the risk of failure for all the impact peak accelerations and pulse durations shown in Figure 13.

Shock Isolation Seats

The primary focus of this report has been on the protection of electronics equipment, but the shock isolation theory also applies to shock isolation seats. Seat solutions are achievable as long as pulse-period mismatch does not occur. The seats can incorporate large stroke mechanisms (i.e., which allows large relative displacements to avoid seat bottoming) with low natural frequencies and high damping from large shock absorbers (to avoid pulse-period mismatch). Recent seakeeping trial results have demonstrated that passive seats can provide protection, but the results of other trials also show they can provide little or no shock protection and even amplify wave impact shock inputs if not properly designed [20 – 24]. These findings have led to the development of laboratory drop tests for shock isolation seats to demonstrate mitigation effectiveness prior to installation in a craft [25 – 26]. Seats properly designed to avoid pulse-period mismatch and seat bottom impacts can provide wave impact protection.

Conclusions and Recommendations

The relatively long duration of wave impact shock pulses in small high-speed craft is the reason they are difficult to mitigate. These long durations, on the order of 100 msec to 450 msec, can result in pulse-period mismatch (i.e., shock transmissibility greater than 1.0) which causes dynamic amplification rather than shock isolation. The key design parameters for avoiding dynamic amplification are the mount-natural frequency (f), mount damping, and the shock pulse duration (T). Based on single-degree-of-freedom (SDOF) design calculations presented herein, mount natural frequencies must be less than approximately 5 Hz to mitigate 100 msec to 300 msec pulses. Mount frequencies equal to or greater than 10 Hz are therefore not capable of mitigating typical wave slam shock pulses. This explains why shock mounts originally designed for blast protection in the 10 Hz to 25 Hz range are not appropriate for wave slam protection in small high-speed craft.

Low frequencies less than 5 Hz require large excursion allowances (i.e., large relative displacements) and high system damping, both of which typically lead to very difficult isolation solutions. The large excursions will likely be impractical for many electronics equipment installations. For example, calculations show that 3 Hz to 4 Hz mount systems with 20% to 40% damping and 10-inch to 12-inch excursion space can provide equipment protection for shock pulses with durations less than 200 msec and peak accelerations less than 5 g. This resulted in only 20% to 30% of the 324 individual wave impacts analyzed during this investigation to be mitigated. Pulse durations greater than 5 g are predicted to result in 10-inch to 12-inch mount bottom impacts.

The calculations presented herein are for ideal SDOF vertical shock inputs. They illustrate mount design challenges for craft. They did not include design margins or take into account the complex multi-degree of freedom environment which can lead to even larger excursions. In the more complex environment more accurate engineering estimates of shock isolation can only be obtained using more sophisticated two or three degree-of-freedom system models.

The market survey conducted for this report provided a sampling of available mount characteristics. Unique mounts not included in the survey may already be available that could provide tractable isolation solutions for craft, but caution is advised because effective solutions will likely only be achieved by experienced shock isolation designers who pursue unique isolation strategies (including seismic-mass dampers, pneumatic or hydraulic isolation systems) using multiple degree of freedom software tools.

The implication of these findings is that the risk of failure for the majority of equipment installations should be reduced by some means other than the use of shock isolation mounts. Equipment hardening is an alternative approach. This approach subjects equipment to laboratory shock testing to demonstrate ruggedness prior to installation in a craft. Example laboratory test requirements for equipment are presented in Appendix G.

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APPENDIX A. Common Misunderstandings

Acceleration Data Low-Pass Filtering

The most important parameters to characterize when investigating shock loads are shock pulse shape, shock pulse duration, shock amplitude, and jerk. These parameters can only be quantified by applying response mode decomposition to unfiltered acceleration data [A1]. This involves the use of low-pass filtering to separate rigid body shock pulses from local vibrations in the vicinity of the accelerometer. Anecdotal evidence suggests that ineffective shock isolation attempts in past high-speed craft applications has been due to the use of unfiltered acceleration data. The use of unfiltered data focuses the designer's attention (or data analyst's attention) on the very short duration of local vibration oscillations (e.g., nominal 25 to 50 msec or less) rather than rigid body shock pulse durations (e.g., 100 msec and more). It has only been recently that the consistent standardized use of low-pass filtered acceleration data has identified the true long duration character of wave impact shock pulses. This may explain why shock isolation mounts originally designed for blast loads were installed in craft for wave slam protection, only to learn later that they actually amplify wave impact shock pulses due to pulse-period mismatch.

Mass Effect

Another common misunderstanding involves the concept of mass participation in dynamic environments. Almost all accelerometer installations in craft record local vibrations as well as the absolute heave, surge, and sway motions (i.e., rigid body motions). Consider a single vertical accelerometer installed on the deck at the center of a deck plate. The recorded acceleration time history will include motions with acceleration components attributed to both local deck-plate vibrations and rigid body heave. If a 200-pound mass had been installed next to the accelerometer the vibration content would have been mitigated significantly, but the amplitude and duration of the rigid body heave component will not be changed. This is because the 200-pound mass is much smaller than the mass of the craft at the cross-section where the accelerometer is positioned. Vibrations in craft do not transmit wave impact shock pulses to equipment or people. Rigid body motions (e.g., rapid change in heave) transmit shock load.

Isolation Mount Differences

Shock is controlled by large relative displacement [A2]. This means the mounts must be able to experience large relative displacements during a shock event. Figure A1 shows wire rope mounts installed in a high-speed craft in an attempt to protect the equipment installation from the effects of wave impact shock. The shape of the mount is intended to allow large excursions on the order of several inches in order to mitigate shock inputs, but the natural frequency of this type of installation causes dynamic amplification because of pulse-period mismatch.

Figure A2 shows examples of typical rubber vibration mounts. The hard rubber mount on the left can be used in sets of four or more for effective vibration isolation of small electrical equipment. The two hard-rubber mounts on the right are different types of engine vibration mounts that have been used in high-speed craft. They are relatively stiff rubber mounts because the intended vibration environments involve very small relative displacements. Hard rubber vibration mounts do not provide wave impact shock mitigation.

Figure A1. Example Shock and Vibration Isolation Mounts in a Craft

Figure A2. Example Hard Rubber Vibration Mounts

Shock and Vibration Calculations

The following paragraphs explain common misunderstandings related to when the words shock and vibration (S&V) can be used together in the same sentence. As will be shown, this is not always the case. Figure A3 shows the shock transmissibility curves for a wave impact half-sine shock pulse (on the left) and vibration transmissibility curves for sine wave vibration excitations (on the right) acting on single-degree-of freedom system (SDOF). The color coding indicates different amounts of assumed damping in the SDOF isolator. These curves are typically shown in engineering handbooks. In the shock transmissibility curves the region of effective shock mitigation is on the left side of the plot where the mitigation ratio (MR) values are less than 1.0. In the vibration transmissibility curves effective vibration mitigation (i.e., MR < 1.0) is on the right side of the curves. The reason for MR < 1.0 on the left side for shock and right side for vibration is because the ordinates axis for one is the inverse of the other. Note also the definition of time (T) for the excitation is different in the two plots. For shock the value of T is the duration of the half-sine shock pulse. For vibration the value of T is the natural period of the vibration excitation. This can be confusing and lead to misunderstandings about when an isolation system can be effective for shock inputs or for vibration inputs.

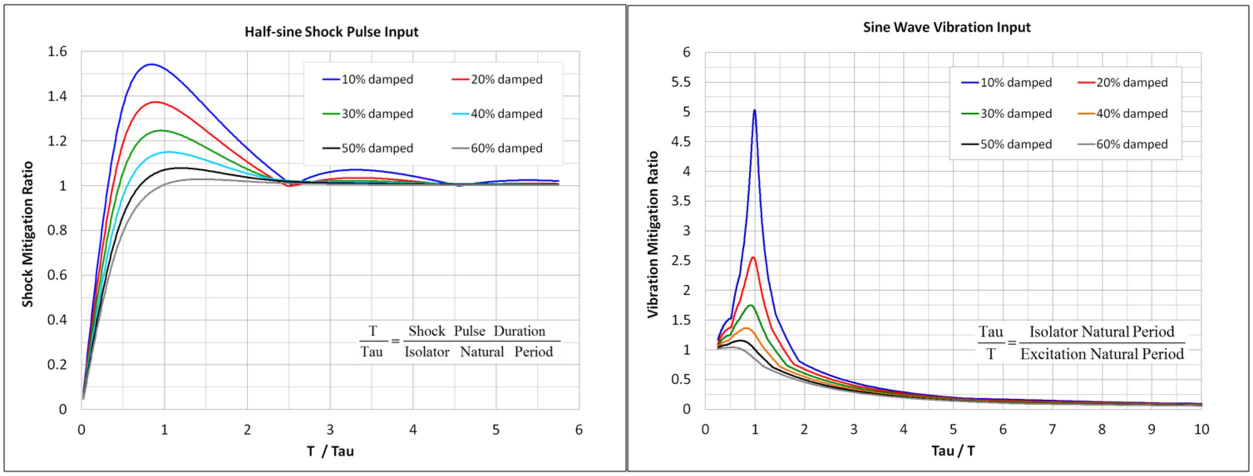

Figure A3. Shock Transmissibility Curves and Vibration Transmissibility Curves

Figure A4 shows the same vibration transmissibility curves as Figure A3, but the ordinate for the shock transmissibility curves has been inverted (i.e., the isolator natural period divided by the shock pulse duration). The inverted time ratio for the shock curves moves the shock mitigation region (i.e., MR < 1) to the right in the figure. The blue curves are vibration transmissibility for 10% to 40% damping. The red curves are half-sine pulse shock transmissibility for 10% to 40% damping. The upper line for both sets of curves corresponds to 10% damping. As damping increases the curves move downward. These curves were used to compute values listed in Table A1.

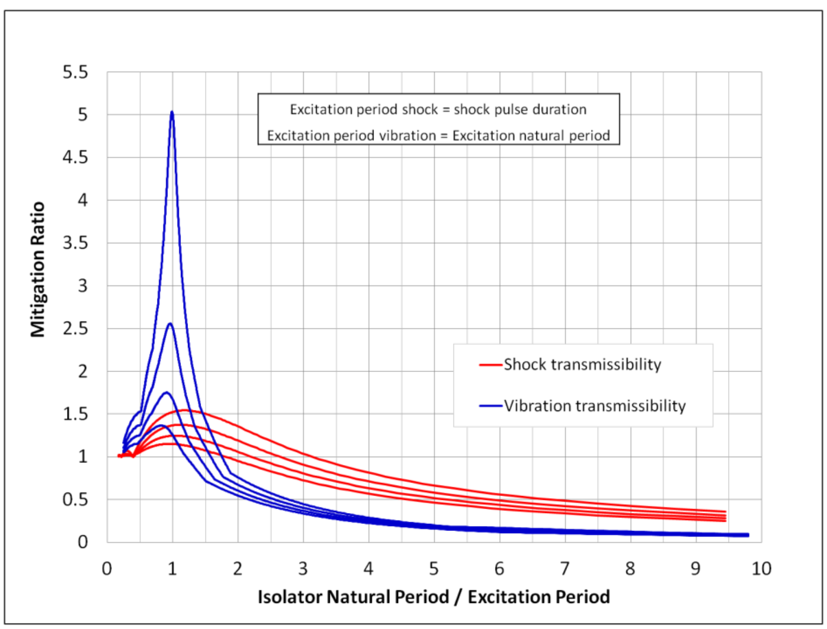

Figure A4. Transmissibility Curves for Shock and Vibration

Table A1 lists computational results for a generic SDOF isolator that has a system natural frequency of 10 Hz, a natural period (Tau) of 0.1 seconds, and a 10% damping ratio. It is assumed that the isolator is subjected to vertical excitations acting at the base of the mount, which include two vibration inputs and five shock pulse inputs. The vibration excitations have natural frequencies of 25 Hz and 50 Hz. The natural period (T) for the vibrations is the reciprocal of the natural frequency. Green shading indicates mitigation (MR < 1) in the results section of the table and the orange shading indicates amplification (MR > 1).

The calculations indicate that the isolator will mitigate the vibration excitations with MR = 0.19 for the 25 Hz input vibration and MR = 0.57 for the 50 Hz vibration. The 10 Hz isolator is therefore an effective vibration isolator for 25 Hz to 50 Hz vibrations.

Table A1. Shock and Vibration Mitigation Ratio (MR) Calculations

Excitation	Isolator Properties			Excitation Characteristics		Calculated Values	
	Frequency Hz	Tau sec	Damping %	Frequency Hz	T sec	Tau / T	MR
Vibration Sine Wave	10 Hz	0.1	10	25	0.040	2.50	0.19
				50	0.020	5.00	0.57
Half-Sine Shock Pulse	10 Hz	0.1	10	blast pulse	0.005	20.00	0.18
					0.010	10.00	0.35
					0.050	2.00	1.35
				wave impact pulse	0.100	1.00	1.52
					0.200	0.50	1.16

For the half-sine shock pulses five different shock pulse durations (T) were assumed. Two of the shock pulses with durations of 0.005 seconds (sec) and 0.01 sec are characteristic of blast shock pulses. They are typically very high amplitude but very short duration pulses. The other three shock pulses have durations of 0.050 sec, 0.10 sec, and 0.20 sec. The 0.1 sec and 0.2 sec half-sine pulses are characteristic of wave impact shock pulses. The computed MR values show that the 10 Hz mount is an effective isolator (MR < 1) for the blast shock pulses, but it amplifies (MR > 1) the wave impact shock pulses.

These results illustrate how an isolator can be effective for both *blast shock* and vibration, but it is not an effective isolator for wave impact shock. In a blast environment it can be an effective *shock and vibration* isolator, but in a small high-speed craft it is only effective for *vibration* isolation. The misunderstanding that often occurs in small craft applications is that an isolator available in the market place as a shock and vibration isolator is assumed to be good for wave impact shock, which is not true.

Shock Pulse Duration Definition

Figure A5 shows the shock and vibration transmissibility curves, but in this case the shock excitation period is defined as 2 times the half-sine shock pulse duration ($T_{half\text{-}sine}$). Many engineering handbooks and texts use the 2T definition for half-sine pulse calculations and plots, ostensibly because two-half sine durations result in a full *period* for an input pulse. When the 2T format is used, the computed MR values in Table A1 are the same, but caution is advised. The visual appearance of the plot suggests that the mitigation performance for the half-sine shock pulse is very similar to the mitigation performance for vibrations. This is not the case as illustrated in Table A1 for different shock pulses (i.e., blast and wave impact).

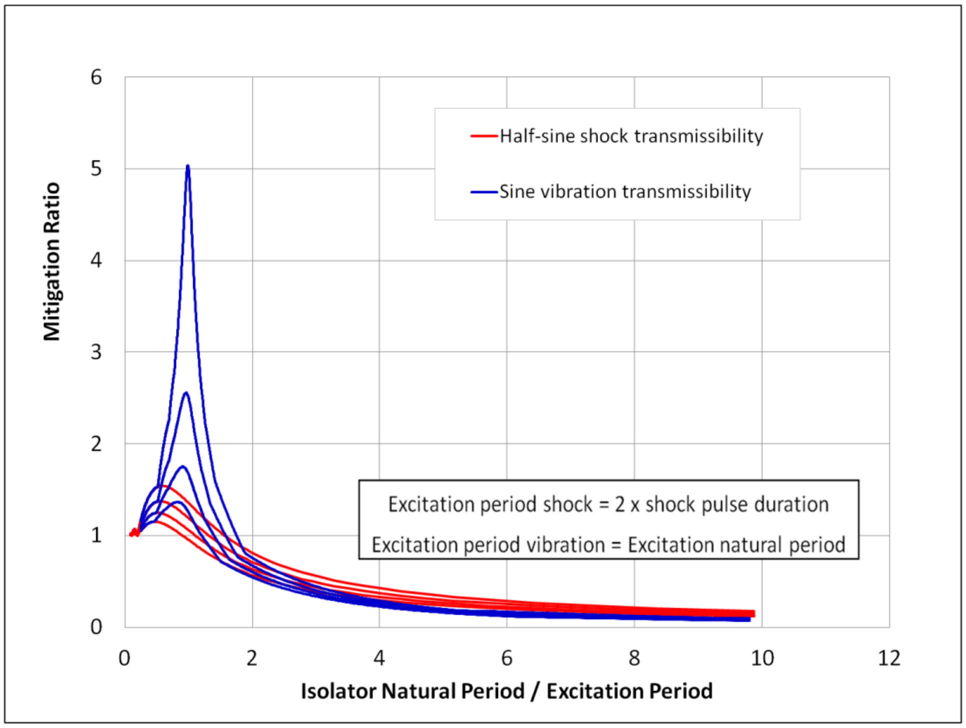

Figure A5. Shock Excitation Period Cautionary Note

S & V Language

The words *shock and vibration* (S&V) appear together in so many venues that it may seem to imply that shock isolation and vibration isolation have the same engineering solutions. In many instances this is true. Well known engineering handbooks include the words *shock and vibration* in book titles. Company brochures and advertisements refer to *shock and vibration* products or services. Online web sites show collections of isolators titled *shock and vibration* mounts. Procurement documents for individual equipment installations often use language that refers to *shock and vibration* test requirements in the same section of the document under the title *shock and vibration*.

Caution is advised because long term exposure to this *shock and vibration* language, as though the two are synonymous, can lead to confusion and misunderstandings related to effective shock isolation design for high-speed craft.

Transient vibration environments in craft induced by wave impacts are characterized by small relative displacements (e.g., unloaded mid-span deck plating less than 0.2 inches) with frequencies ranging from 20-60 Hz or higher. Propulsion system induced vibrations have smaller relative displacements (e.g., less than to 0.01 inches) and frequencies from 40-500 Hz. On the other hand, the wave impact shock environment is characterized by shock pulses acting over roughly 100 - 450 milliseconds and large absolute displacements (e.g., 6 – 24 inches or more) as the hull plunges into a wave. Wave encounter rates are less than 2 impacts per second in rough seas (i.e., 2 Hz or less). These very different dynamic environments require drastically different isolation solutions. Hard rubber vibration mounts installed under equipment installations in high-speed craft are not effective shock isolation mounts, and wire rope mounts with natural frequencies greater than 5 Hz are not effective wave impact shock isolation mounts.

Appendix A References

A-1. Riley, Michael R., Coats, Timothy W., “Acceleration Response Mode Decomposition for Quantifying Wave Impact Load in High-Speed Planing Craft”, Naval Surface Warfare Center Carderock Division Report NSWCCD-80-TR-2014/007, April 2014.

A-2. LeKuch, H., Chapter 39, *Harris’ Shock and Vibration Handbook, Sixth Edition*, Piersol, A., Paez, T., editor-in-chiefs, McGraw-Hill Companies, Inc., New York, New York, 2010.

Appendix B. Example Of Shock Mount Dynamic Amplification

An example of acceleration data that shows shock mounts that amplify wave slam shock inputs is illustrated in Figure B1. The upper data plot shows the vertical acceleration input recorded on a foundation below the shock isolation mounts (blue curve). The vertical acceleration response above the mounts at the equipment item is the red curve. The pulse durations of the blue input and red response curves are approximately the same, but the peak response accelerations above the mounts are greater than the shock input peak accelerations recorded below the mounts. The explaination for this amplification can be shown mathematically using the single degree of freedom (SDOF) model shown in the figure. The black curve in the lower right plot is the same acceleration time history below the mounts shown in the upper plot between 348 and 349 seconds (i.e., the blue curve). It was used as the shock input pulse for calculations for a SDOF model with a natural frequency of 12 Hz and 20% damping. The red curve in the lower right plot is the predicted motion above the mounts. The time-history prediction shows shock mount amplification caused by pulse-period mismatch that is similar to that observed in the data.

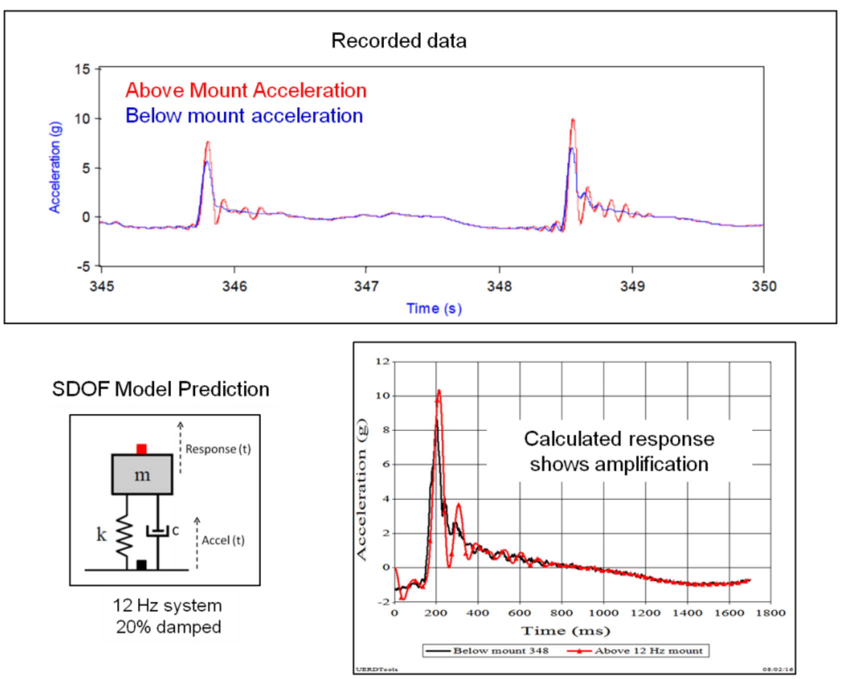

Figure B1. Recorded and Predicted Shock Mount Amplification

In Figure B1 the shock pulse duration for the wave slam recorded between 348 and 349 seconds is approximately 0.15 seconds. For a 12-Hz mount this corresponds to an R value of 1.8. It can be shown on the shock transmissibility curve that R = 1.8 for 20% damping is in the shock amplification region with a predicted MR = 1.18 (i.e., 18% amplification).

Shock mitigation of a 150-msec pulse assuming 20% mount damping requires an R value less than approximately 0.38 (i.e., the limit value for 20% damping). Selection of new mounts for this installation would therefore focus on selecting a mount effective natural frequency less than 2.5 Hz (i.e., 0.38 divided by 0.15 seconds) while ensuring that sufficient excursion space is allowed to prevent mount bottoming.

Appendix C. STEP VELOCITY SHOCK PULSE EQUATIONS

Many texts and engineering handbooks present shock mount design equations for an assumed shock pulse with a step-velocity time history, where the shock velocity input jumps from zero to a maximum value (V) instantaneously. This is referred to as a step-velocity input. The solution for this assumption, derived from conservation of energy equations, results in easy to use algebraic equations (C1) and (C2) for mount deflection and maximum acceleration transmitted above the mounts, respectively [C1].

$$\text{mount deflection } \Delta = \frac{V}{2\pi f} \qquad \text{Equation (C1)}$$

$$\text{shock transmitted } A_{MAX} = 2\pi f V \qquad \text{Equation (C2)}$$

Equation (C1) was used to compute mount relative displacements and equation (C2) was used to compute peak acceleration mitigation ratios for 2 Hz, 3 Hz, and 4 Hz mounts with 20%, 30%, and 40% damping, assuming half-sine shock inputs with 2 g to 8 g peak accelerations and 100-msec to 300-msec pulse durations. Although not a valid assumption, for these calculations it was assumed that the velocity change of the half-sine pulse was equal to the step-velocity change (i.e., in order to illustrate the problem). Tables C1 to C3 list all computational results.

Compared with solutions for a single-degree-of-freedom model and a half-sine pulse shock input, the relative deflection error using equation (C1) is within +/- 16 percent (see Table C3), but the transmitted acceleration Amax using equation (C2) was from 1.4 to 4.2 times the Amax values predicted for the half-sine pulse inputs (see Table C1 and C2 mitigation ratio values). Equations (C1) and (C2) are appropriate for very short duration shocks, like blast loads, but they should not be used for shock mount design for long duration wave impact pulses. The assumption that the velocity change for the half-sine pulse is equal to the velocity change for the step-velocity pulse is not appropriate for wave slam shock pulses.

Table C1. Half-Sine Pulse SDOF Model Results

Mounts	Pulse Duration msec	MR	Mount Relative Displacement (inches)						
			2g	3g	4g	5g	6g	7g	8g
2 Hz 20% damped	70	0.41	2.96	4.45	5.93	7.41	8.89	10.37	11.85
	100	0.58	4.18	6.26	8.36	10.45	12.54	14.63	16.72
	150	0.83	6.08	9.12	12.16	15.20	18.25	21.29	24.33
	200	1.03	7.77	11.66	15.55	19.44	23.32		
	300	1.28	10.37	15.55	21.74	25.92			
2 Hz 30% damped	70	0.36	3.13	4.69	6.26	7.82	9.39	10.95	12.52
	100	0.51	4.42	6.64	8.84	11.07	13.28	15.50	17.71
	150	0.74	6.49	9.73	12.97	16.22	19.47	22.72	25.97
	200	0.92	8.38	12.58	16.27	20.97	25.16		
	300	1.14	11.56	17.34	23.12	28.90			
3 Hz 20% damped	70	0.60	1.94	2.92	3.87	4.85	5.81	6.81	7.78
	100	0.83	2.70	4.05	5.39	6.75	8.11	9.44	10.81
	150	1.12	3.79	5.68	7.58	9.47	11.37	13.26	15.16
	200	1.28	4.61	6.91	9.22	11.51	13.83		
	300	1.37	5.49	8.23	10.98	13.72			
3 Hz 30% damped	70	0.54	2.06	3.09	4.12	5.15	6.18	7.22	8.25
	100	0.74	2.88	4.32	5.77	7.21	8.65	10.08	11.53
	150	0.99	4.11	6.17	8.23	10.29	12.35	14.41	16.47
	200	1.14	5.13	7.70	10.27	12.84	15.41		
	300	1.24	6.56	9.84	13.12	16.40			
4 Hz 30% damped	70	0.69	1.52	2.28	3.04	3.80	4.55	5.32	6.09
	100	0.92	2.09	3.14	4.19	5.24	6.29	7.34	8.39
	150	1.14	2.89	4.33	5.78	7.22	8.67	10.11	11.56
	200	1.23	3.46	5.20	6.93	8.67	10.40		
	300	1.22	4.23	6.35	8.47	10.59			
4 Hz 40% damped	70	0.62	1.62	2.44	3.25	4.06	4.87	5.69	6.50
	100	0.83	2.26	3.39	4.52	5.66	6.79	7.92	9.06
	150	1.03	3.20	4.80	6.41	8.01	9.61	11.22	12.82
	200	1.12	3.98	5.98	7.97	9.97	11.96		
	300	1.14	5.25	7.89	10.53	13.16			

Table C2. Step-Velocity Assumption Results

Mounts	Pulse Duration msec	Eq. C2 MR	MR % error	Eq. C1 Mount Relative Displacement (inches)						
				2 g	3 g	4 g	5 g	6g	7g	8g
2 Hz 20% damped	70	0.56	36.59	2.73	4.10	5.46	6.83	8.19	9.56	10.92
	100	0.80	37.93	3.90	5.85	7.80	9.75	11.70	13.65	15.60
	150	1.20	44.58	5.85	8.78	11.70	14.63	17.55	20.48	23.40
	200	1.60	55.34	7.80	11.70	15.60	19.50	23.40		
	300	2.40	87.50	11.70	17.55	23.40	29.25			
2 Hz 30% damped	70	0.56	55.56	2.73	4.10	5.46	6.83	8.19	9.56	10.92
	100	0.80	56.86	3.90	5.85	7.80	9.75	11.70	13.65	15.60
	150	1.20	62.16	5.85	8.78	11.70	14.63	17.55	20.48	23.40
	200	1.60	73.91	7.80	11.70	15.60	19.50	23.40		
	300	2.40	110.53	11.70	17.55	23.40	29.25			
3 Hz 20% damped	70	0.84	40.00	1.82	2.73	3.64	4.55	5.46	6.37	7.28
	100	1.20	44.58	2.60	3.90	5.20	6.50	7.80	9.10	10.40
	150	1.80	60.71	3.90	5.85	7.80	9.75	11.70	13.65	15.60
	200	2.40	87.50	5.20	7.80	10.40	13.00	15.60		
	300	3.60	162.77	7.80	11.70	15.60	19.50			
3 Hz 30% damped	70	0.84	55.56	1.82	2.73	3.64	4.55	5.46	6.37	7.28
	100	1.20	62.16	2.60	3.90	5.20	6.50	7.80	9.10	10.40
	150	1.80	81.82	3.90	5.85	7.80	9.75	11.70	13.65	15.60
	200	2.40	110.53	5.20	7.80	10.40	13.00	15.60		
	300	3.60	190.32	7.80	11.70	15.60	19.50			
4 Hz 30% damped	70	1.12	62.32	1.37	2.05	2.73	3.41	4.10	4.78	5.46
	100	1.60	73.91	1.95	2.93	3.90	4.88	5.85	6.83	7.80
	150	2.40	110.53	2.93	4.39	5.85	7.31	8.78	10.24	11.70
	200	3.20	160.16	3.90	5.85	7.80	9.75	11.70		
	300	4.80	293.44	5.85	8.78	11.70	14.63			
4 Hz 40% damped	70	1.12	80.65	1.37	2.05	2.73	3.41	4.10	4.78	5.46
	100	1.60	92.77	1.95	2.93	3.90	4.88	5.85	6.83	7.80
	150	2.40	133.01	2.93	4.39	5.85	7.31	8.78	10.24	11.70
	200	3.20	185.71	3.90	5.85	7.80	9.75	11.70		
	300	4.80	321.05	5.85	8.78	11.70	14.63			

Table C3. Step-Velocity Relative Displacement Error

Mounts	Pulse Duration msec	Relative Displacement Percent Error						
		2 g	3 g	4 g	5 g	6g	7g	8g
2 Hz 20% damped	70	-7.77	-7.98	-7.93	-7.89	-7.87	-7.86	-7.85
	100	-6.70	-6.55	-6.70	-6.70	-6.70	-6.70	-6.70
	150	-3.78	-3.78	-3.78	-3.78	-3.84	-3.83	-3.82
	200	0.39	0.34	0.32	0.31	0.34		
	300	12.83	12.86	7.64	12.85			
2 Hz 30% damped	70	-12.78	-12.69	-12.78	-12.72	-12.78	-12.74	-12.78
	100	-11.76	-11.90	-11.76	-11.92	-11.90	-11.94	-11.91
	150	-9.86	-9.82	-9.79	-9.83	-9.86	-9.88	-9.90
	200	-6.92	-7.00	-4.12	-7.01	-7.00		
	300	1.21	1.21	1.21	1.21			
3 Hz 20% damped	70	-6.19	-6.51	-5.94	-6.19	-6.02	-6.46	-6.43
	100	-3.70	-3.70	-3.53	-3.70	-3.82	-3.60	-3.79
	150	2.90	2.99	2.90	2.96	2.90	2.94	2.90
	200	12.80	12.88	12.80	12.95	12.80		
	300	42.08	42.16	42.08	42.13			
3 Hz 30% damped	70	-11.65	-11.65	-11.65	-11.65	-11.65	-11.77	-11.76
	100	-9.72	-9.72	-9.88	-9.85	-9.83	-9.72	-9.80
	150	-5.11	-5.19	-5.22	-5.25	-5.26	-5.27	-5.28
	200	1.36	1.30	1.27	1.25	1.23		
	300	18.90	18.90	18.90	18.90			
4 Hz 30% damped	70	-10.20	-10.20	-10.20	-10.20	-10.00	-10.20	-10.34
	100	-6.70	-6.85	-6.92	-6.97	-7.00	-7.02	-7.03
	150	1.21	1.33	1.21	1.28	1.21	1.26	1.21
	200	12.72	12.50	12.55	12.46	12.50		
	300	38.30	38.19	38.13	38.10			
4 Hz 40% damped	70	-15.74	-16.09	-16.00	-15.95	-15.91	-16.04	-16.00
	100	-13.72	-13.72	-13.72	-13.87	-13.84	-13.83	-13.91
	150	-8.59	-8.59	-8.74	-8.71	-8.69	-8.76	-8.74
	200	-2.01	-2.17	-2.13	-2.21	-2.17		
	300	11.43	11.22	11.11	11.13			

Appendix C References

C1. Harris, Cecil M., editor-in-chief, *Shock and Vibration Handbook, Fourth Edition*, McGraw-Hill Companies, Inc., New York, New York, 1995.

APPENDIX D. SHOCK RESPONSE SPECTRUM

A shock response spectrum (SRS) is a computational tool used extensively to compare the severity of different shock motions [references D1 to D7]. It is also referred to as a maximum response spectrum that can be used to analyze any dynamic event, even vibration signals [Reference D7]. It is especially useful for comparing different shock pulses that have different pulse shapes, peak amplitudes, jerk, and pulse durations.

The SRS uses a model of the single-degree-of-freedom (SDOF) system shown in Figure D1 to compute the effects of an input motion y (t) on the SDOF system. The system has a base attached to a mass (m) by a spring with stiffness k and a damper with damping coefficient c. For a prescribed time varying shock input motion y (t) at the base of the system the resulting response of the mass (m) is x (t). The relative displacement z (t) between the base and the mass is x (t) minus y (t). The equation of motion of the system given by equation (D1) is obtained by summing the inertial force of the mass and the forces within the spring and damper.

It is important to note that the model in Figure D1 is not a model of a shock isolation system. It is merely a model that will be used as a mathematical yard-stick to quantify shock severity for an input shock motion y (t). The predicted response motion x (t) is used to quantify shock severity. The input shock that has the largest amplitude response is the most severe shock input when comparing two inputs.

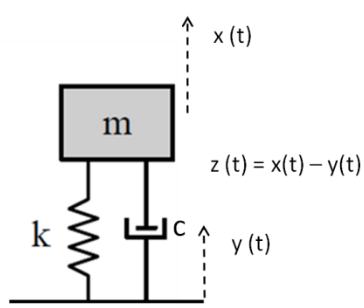

Figure D1. The Single-Degree-of-Freedom Spring-Mass-Damper Model

When a free-body diagram of the forces acting on the mass (m) is constructed for a given base input displacement y (t), it can be shown that a summation of forces acting on the mass results in the second-order partial differential equation (D1). A single dot over a parameter indicates the first derivative (velocity) and two dots indicate the second derivative (acceleration). This equation applies for any input motion, including shock pulses or vibration oscillations, but isolation for vibrations is drastically different from isolation for shock pulses. Appendix A presents common misunderstandings between shock isolation and vibration isolation.

$$m\ddot{z} + 2\xi\omega\dot{z} + \omega^2 z = -\ddot{y}$$ Equation (D1)

where the undamped system frequency in Hertz (Hz) is

$$f = \frac{\omega}{2\pi} = \frac{1}{2\pi}\sqrt{\frac{k_{eff}}{m}}$$ Equation (D2)

and the damping ratio is

$$\xi = \frac{c}{2m\omega}$$ Equation (D3)

The solution of equation (D1) provides the predicted response motion of the mass (m) caused by the base input motion either in terms of the absolute motion of the mass x (t) or the relative displacement z (t) between the base and the mass.

An SRS is the maximum response of a set of single-degree-of-freedom (SDOF), spring-mass-damper oscillators to an input motion. The input motion is applied to the base of all oscillators, and the calculated maximum response of each oscillator versus the natural frequency make up the spectrum [D7]. The relative displacement SRS is often used as a parameter to quantify shock severity when two input shock motions are being compared. It is an intuitive engineering measure of severity because the relative displacement across the spring is proportional to the strain in the spring. The shock pulse that causes the larger strain, and therefore the largest damage potential, is judged to be the more severe of the two base input shock pulses.

The plot on the left in Figure D2 shows three vertical acceleration time histories recorded at three locations on a craft (bow, helm, and LCG). The plot on the right is the computed maximum relative displacement SRS (DSRS) for each time history as a function of SDOF model natural frequency. Visual inspection of the time histories on the left indicate that the red bow shock pulse is the most severe. The DSRS curves on the right quantify the difference in severity. The key feature of the SRS approach is that it quantifies shock severity based on its effect on a set of SDOF models with different natural frequencies.

The SRS can also be plotted using other SDOF response parameters as shown in Figure D3. In this figure the spectra compare the severity of a 3g – 100-ms half-sine pulse to the severity of a 2 g – 150-ms half-sine pulse. The upper left plot shows the two input pulses in the time domain; the other three plots show maximum responses in the SRS frequency domain (i.e., as a function of oscillator natural frequency). The upper right plot shows how the absolute peak acceleration response of the mass varies with system natural frequency. They are called the absolute acceleration shock response spectra (ASRS).

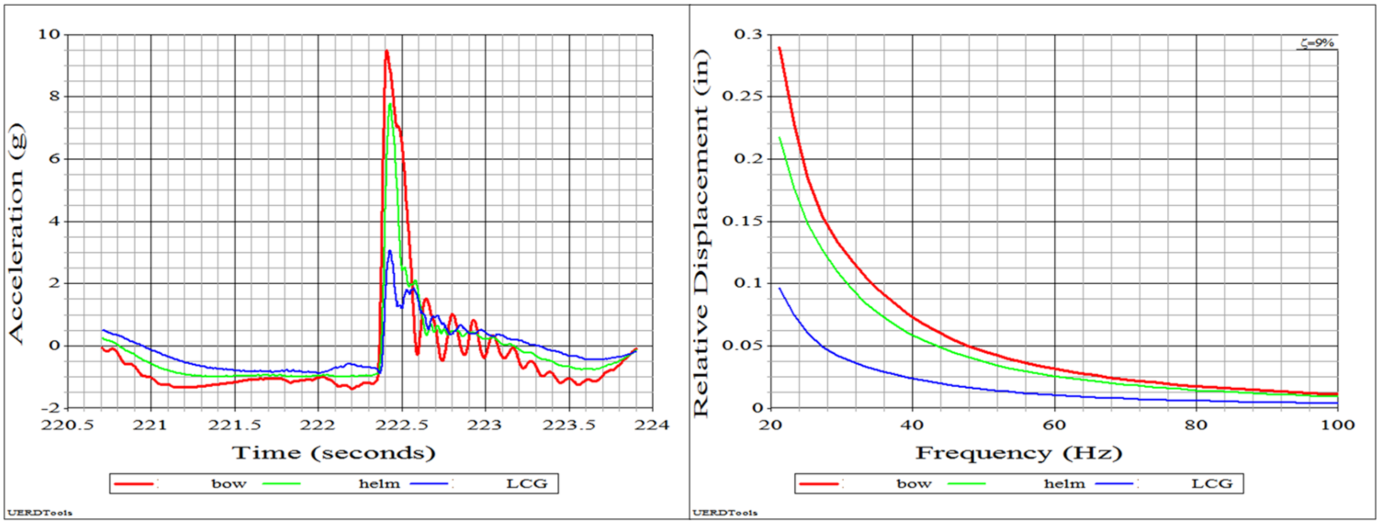

Figure D2. Three Wave Slam Shocks and Relative Displacement SRS[8]

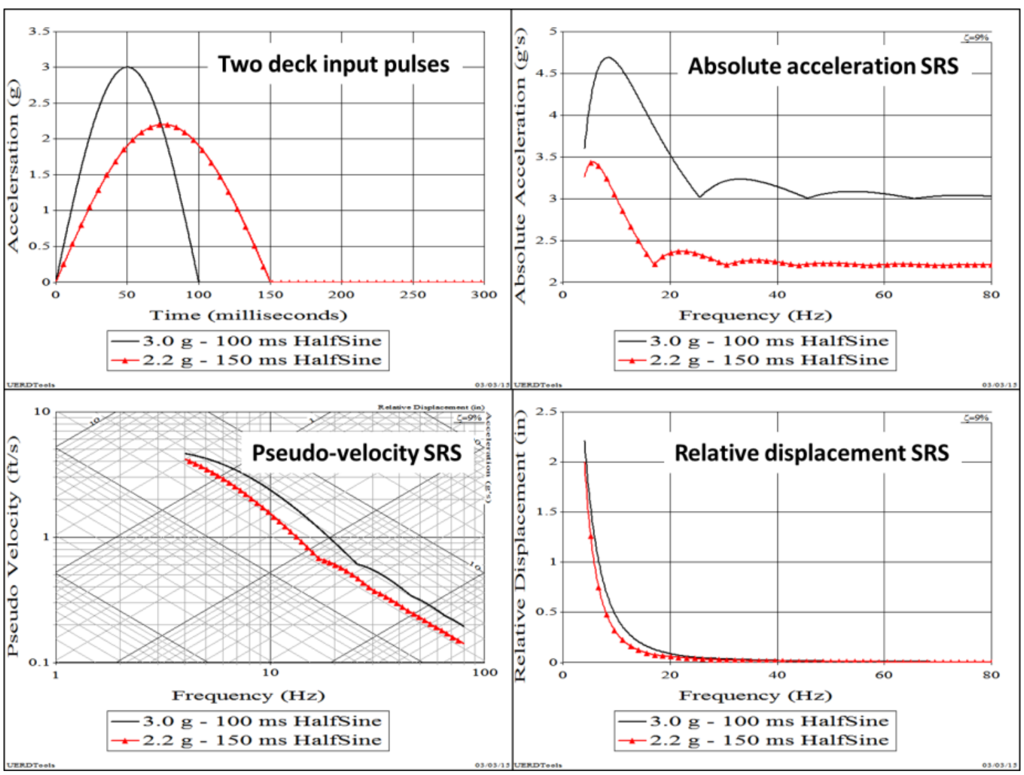

Figure D3. Different Types of Shock Response Spectra

[8] All data plots and SRS shown were created using UERDTools [D8].

The lower right plot in Figure D3 is the relative displacement SRS for each input pulse, and the lower left plot is the pseudo- velocity SRS (VSRS) for each pulse. Logarithmic scales are used on all four axes of the VSRS. The horizontal lines are the pseudo-velocity scale. Vertical lines are the system natural frequency scale. Lines sloping downward to the left show the predicted maximum relative displacement scales. Lines sloping downward to the right show the predicted maximum response accelerations. The log-log VSRS is a useful format because it provides a measure of the shock severity in units of maximum displacement, velocity, and acceleration. The acceleration scale is referred to as the pseudo-acceleration (A_{MAX}) for damped systems and the velocity scale is referred to as the pseudo-velocity when the maximum values are calculated using equations (D5) and (D6), which applies for lightly damped or zero damped systems [D1]. Z_{MAX} is the maximum relative displacement of the oscillator spring.

$$A_{MAX} = \left(2\pi f\right)^2 Z_{MAX} \qquad \text{Equation (D5)}$$

$$V_{MAX} = \left(2\omega f\right) Z_{MAX} \qquad \text{Equation (D6)}$$

Appendix D References

D1. Harris, Cecil M., editor-in-chief, *Shock and Vibration Handbook, Fourth Edition*, McGraw-Hill Companies, Inc., New York, New York, 1995.

D2. ANSI/ASA S2.62-2009, *Shock Test Requirements for Equipment in a Rugged Shock Environment*, American National Standards Institute and Acoustical Society of America, Melville, N.Y., 2009.

D3. Department of Defense Test Method Standard, *Environmental Engineering Considerations and Laboratory Tests*, Military Standard, MIL-S-810G, change 1, Method 516.7, Shock, 15 April 2014.

D4. STANAG 4559, *Testing of Surface Ship Equipment on Shock Testing Machines*, NATO Standardized Agreement, 13 May 2008.

D5. Alexander, J. Edward, *The Shock Response Spectrum – A Primer*, Society of Experimental Mechanics Inc., Proceedings of the IMAC XXVII, Orlando, Florida, USA, 9 – 12 February 2009.

D6. Gaberson, Howard A., *Shock Severity Estimation*, Sound and Vibration Magazine, Volume 46 Number 1, January 2012.

D7. ISO-18431-4: 2007, *Mechanical vibration and shock – Signal processing – Part 4: Shock response spectrum analysis*, International Organization for Standardization, Geneva, Switzerland, 2007.

D8. Mantz, Paul A., Costanzo, Fredrick A., *An Overview of UERDTools Capabilities: A Multi-Purpose Data Analysis Package*, Proceedings of the IMAC-XXVII Conference and Exposition on Structural Dynamics, Society of Experimental Mechanics, Inc., 9-12 February 2009, Orlando, Florida, 2009.

Appendix E. Shock Mitigation Ratio

Mitigation Ratio Using SRS

The universal approach to quantifying shock transmissibility is by dividing the severity of the shock response pulse above the mounts by the severity of the base input shock pulse [E1]. In this report the term shock mitigation ratio is the same as shock transmissibility. Many texts define the mitigation ratio (or transmissibility) as the ratio of the peak response acceleration above the mounts divided by the peak acceleration of the shock input. This approach is satisfactory as long as the shock input pulse and the shock response pulse above the mounts have similar shape, jerk, and pulse duration.

When pulse shapes, jerk, and duration are not similar the preferred method of quantifying shock mitigation is to use the shock response spectra ratio given by equation E1. This is because the SRS ratio inherently accounts for differences in the key shock parameters, including shape, duration, and jerk, as well as differences in peak acceleration [E2].

$$\text{Mitigation Ratio} = \frac{\text{SRS}_{\text{Response}}}{\text{SRS}_{\text{Input}}} \qquad \text{Equation (E1)}$$

If the ratio is greater than 1.0, the shock pulse for the response is more severe than the shock pulse for the base input. If the ratio is less than 1.0, the shock pulse for the response is less severe than the shock pulse for the base input. As an example, Figure E1 shows relative displacement SRS (DSRS) for two hypothetical half-sine pulses, 7 g – 100 msec base input acceleration and 5 g – 210 msec above-mount response acceleration. The question is how much less severe or more severe is the above-mount response pulse compared to the base input pulse?

Figure E2 was constructed to answer this question by dividing the DSRS for the 5 g – 210 msec pulse by the DSRS for the 7 g – 100 msec pulse. A damping ratio of 22 percent was assumed for the calculations. It shows that over a broad frequency range the 5 g – 210 msec shock pulse is less severe than the 7 g – 100 msec pulse (i.e., the ratio is less than 1.0). For natural frequencies greater than approximately 30 Hz the mitigation ratio is approximately 0.70 (i.e., the 5-g pulse is 30 percent less severe than the 7-g pulse). Between 4 Hz and 30 Hz the mitigation ratio varies from 0.55 to 0.7 (i.e., 30 percent to 45 percent less severe).

The mitigation ratio based on relative displacement shock response spectra (DSRS) is a convenient relative measure of shock input severity because (1) it takes into account the effects of acceleration magnitude, pulse duration, and the rate of acceleration application (i.e., jerk), and (2) because of its relationship to compressive strain or stress in the SDOF mathematical model [E3]. The concept of stress as a measure of shock severity is not new. The early NASA studies concluded that magnitude (i.e., peak acceleration) alone does not define shock severity, nor does acceleration cause damage in a system. Stress (or strain), a result of acceleration, causes damage [E2].

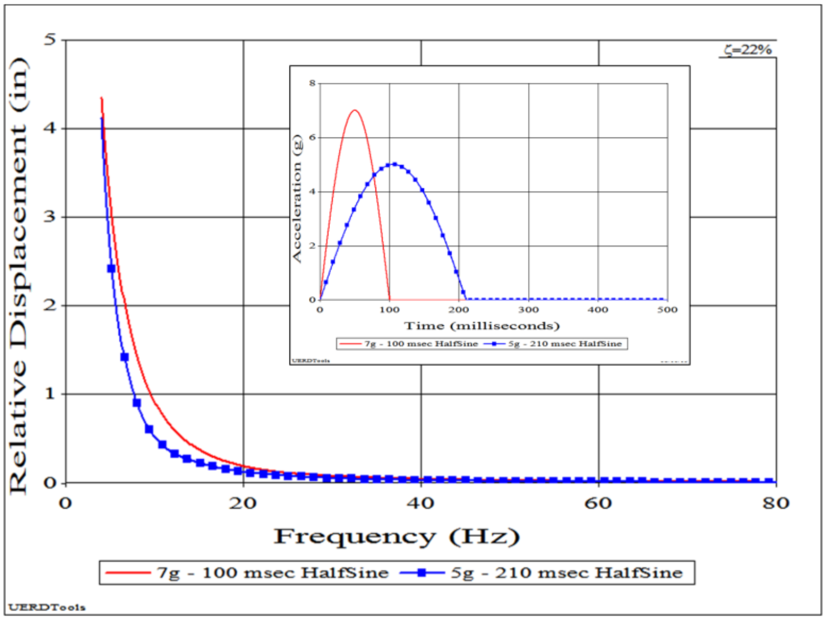

Figure E1. Comparison of Hypothetical DSRS

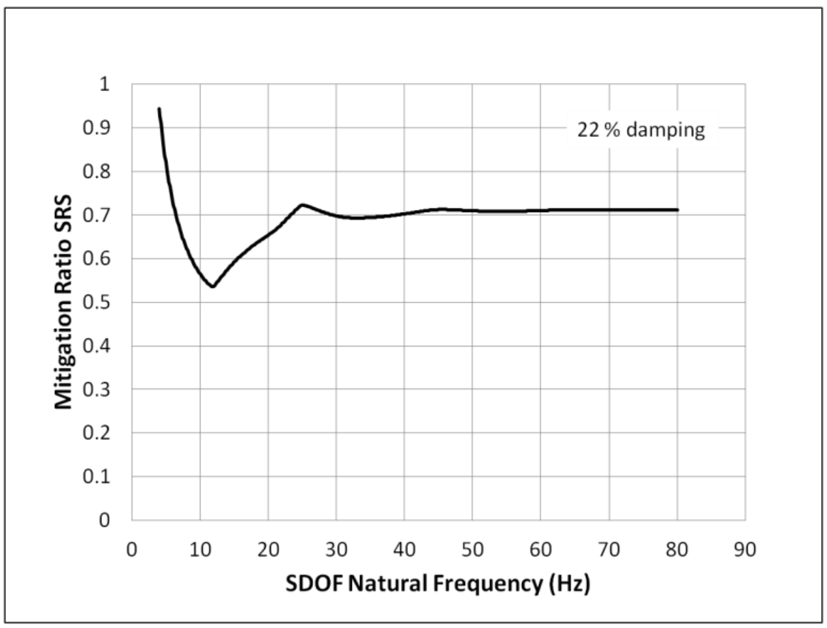

Figure E2. Mitigation Ratio for 5 g and 7 g Half-sine Pulses

MR Frequency of Interest

Selection of the frequency value of interest (i.e., on the natural frequency abscissa in Figure E2) and the SRS damping ratio for the mitigation ratio calculation is based on the assumption that there is no intent to specifically model the item being subjected to the shock. The mathematical model of the SDOF system in this application is simply a mathematical ruler for relative comparisons of shock intensity. But the ruler can be made more relevant for the investigation by considering the frequency (or frequencies) and damping characteristics of interest. In Figure E2 the frequency scale is related to the natural mode(s) of vibration of the item being investigated. If relatively stiff items (e.g., electronics components) are being subjected to shock then the frequencies of interest may be 50 Hz to 70 Hz or more. Figure E2 shows that in this frequency range of interest the MR values level off at a plateau level and remain constant as natural frequency increases. In the main body of this report MR values at 100 Hz and 9% damping are tabulated and compared. These values were selected because electronics components typically have fundament response modes in terms of hundreds of Hertz, with lower bound estimates in the 30 Hz to 40 Hz range, and light internal damping (e.g., 3% to 9% vice 20% to 40%).

The intent is not to model the item being subjected to the shock, but rather to select a relevant frequency that renders the mathematical ruler (i.e., the SRS mitigation ratio) more meaningful for the application.

It can be shown that MR_{SRS} is identical to MR_{Peaks} for 9% damping and natural frequencies of interest greater than roughly 50 Hz.

Appendix E References

E1. Harris, Cecil M., editor-in-chief, *Shock and Vibration Handbook*, Fourth Edition, McGraw-Hill Companies, Inc., New York, New York, 1995.

E2. Eiband, Martin A., *Human Tolerance to Rapidly Applied Accelerations: A Summary of the Literature*, Lewis Research Center, National Aeronautics and Space Administration, Memorandum 5-19-59E, Cleveland, Ohio, June 1959.

E3. Riley, M. R., Coats, T.W., *Development of a Method for Computing Wave Impact Equivalent Static Accelerations for Use in Planing Craft Hull Design*, The Society of Naval Architects and Marine Engineers, The Third Chesapeake Powerboat Symposium, Annapolis, Maryland, USA, June 2012

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Appendix F. Plots of Maximum Relative Displacement and Mitigation Ratio

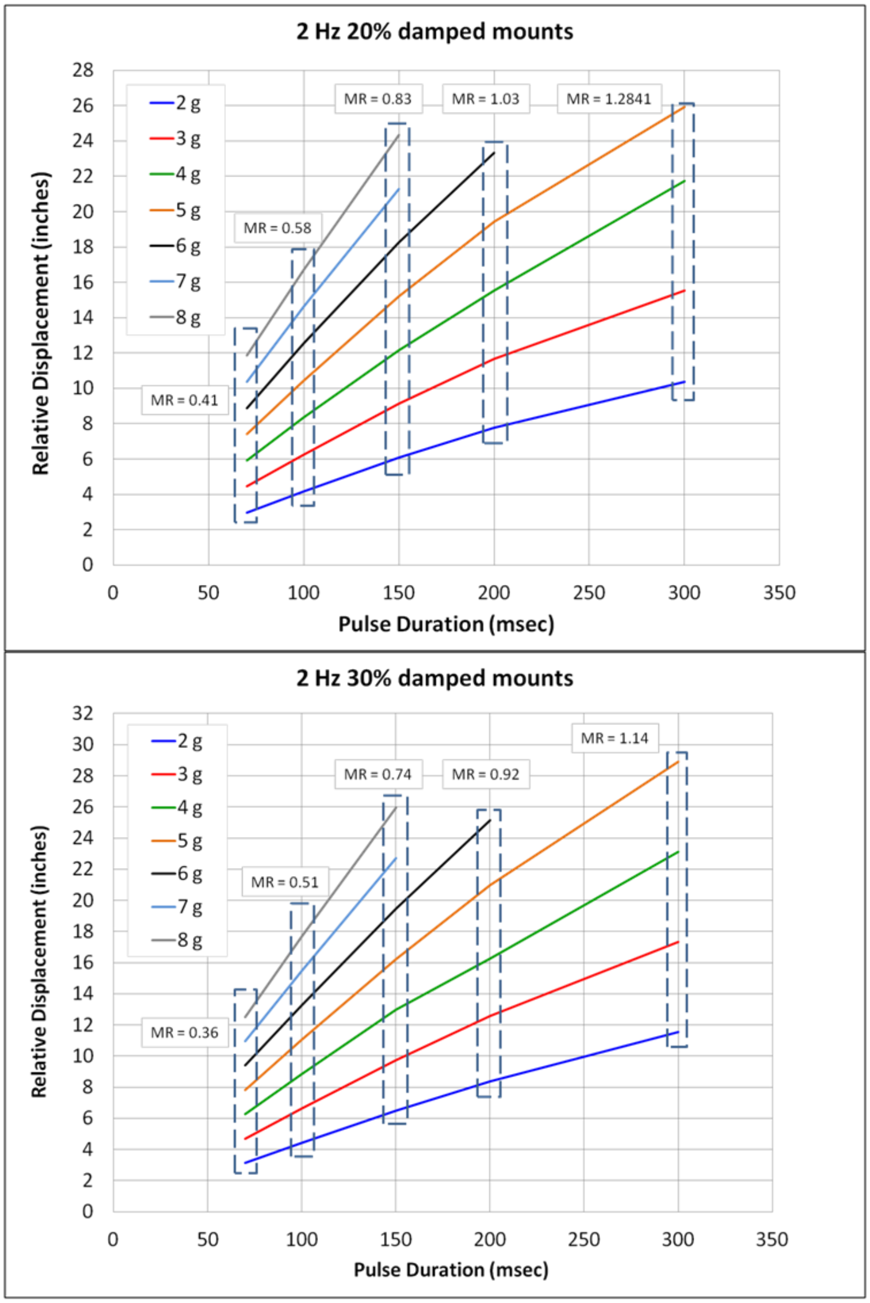


Figure F1. 2Hz Mount System Computational Results

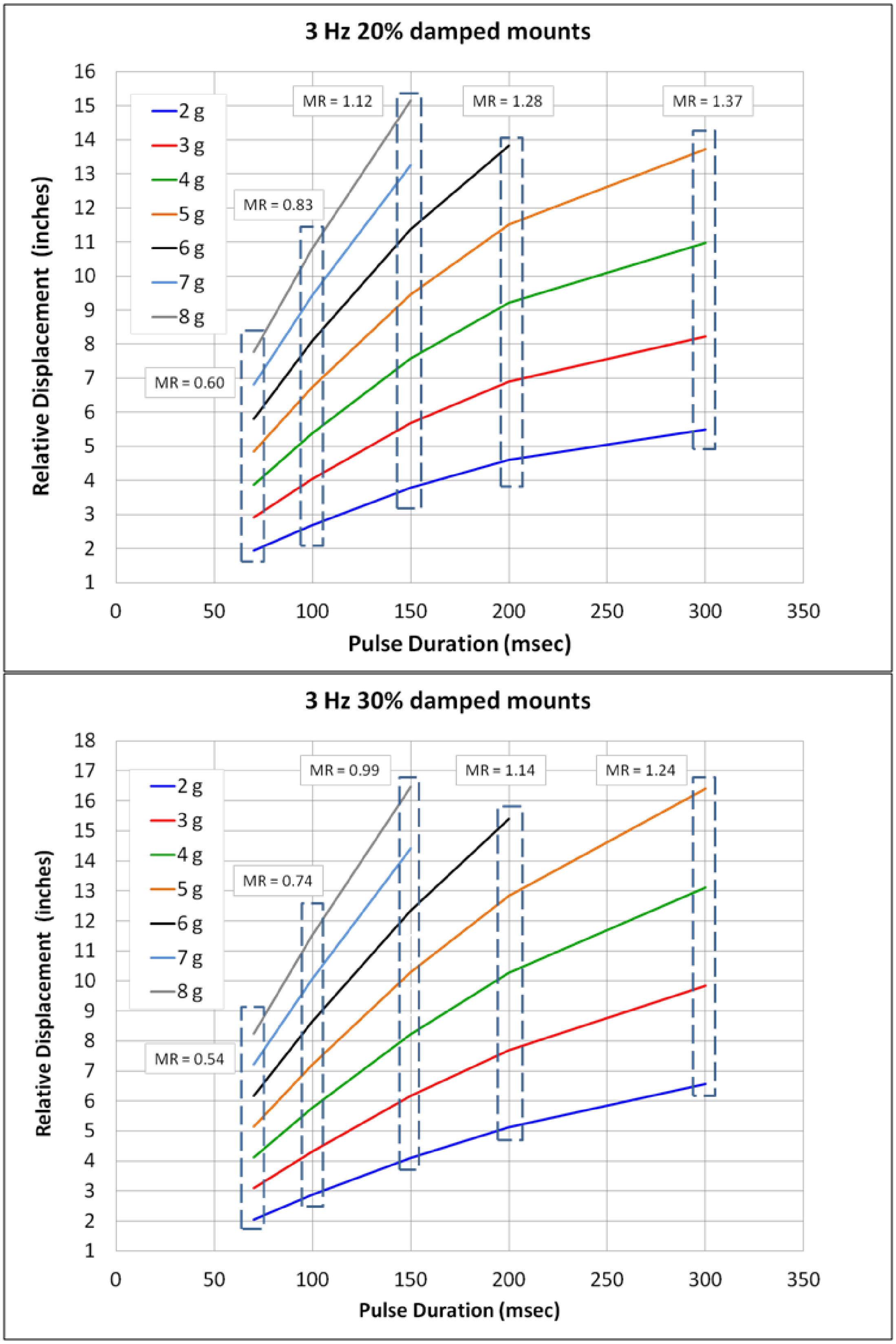
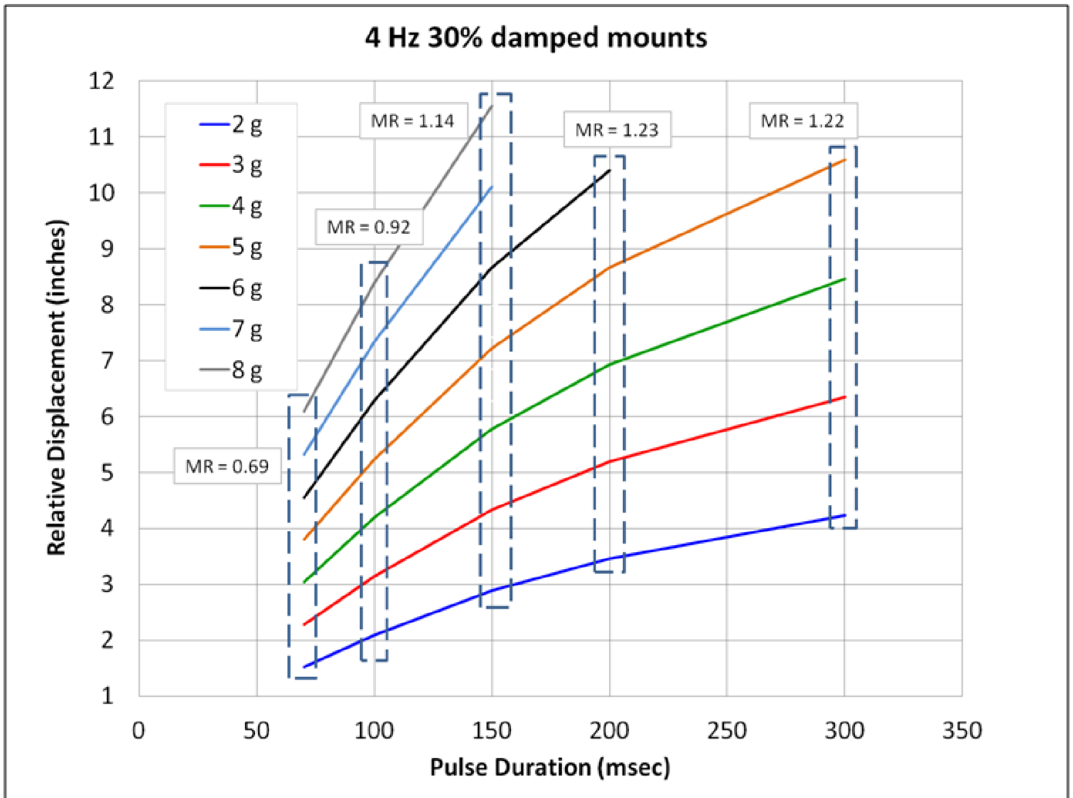


Figure F2. 3 Hz Mount System Computational Results



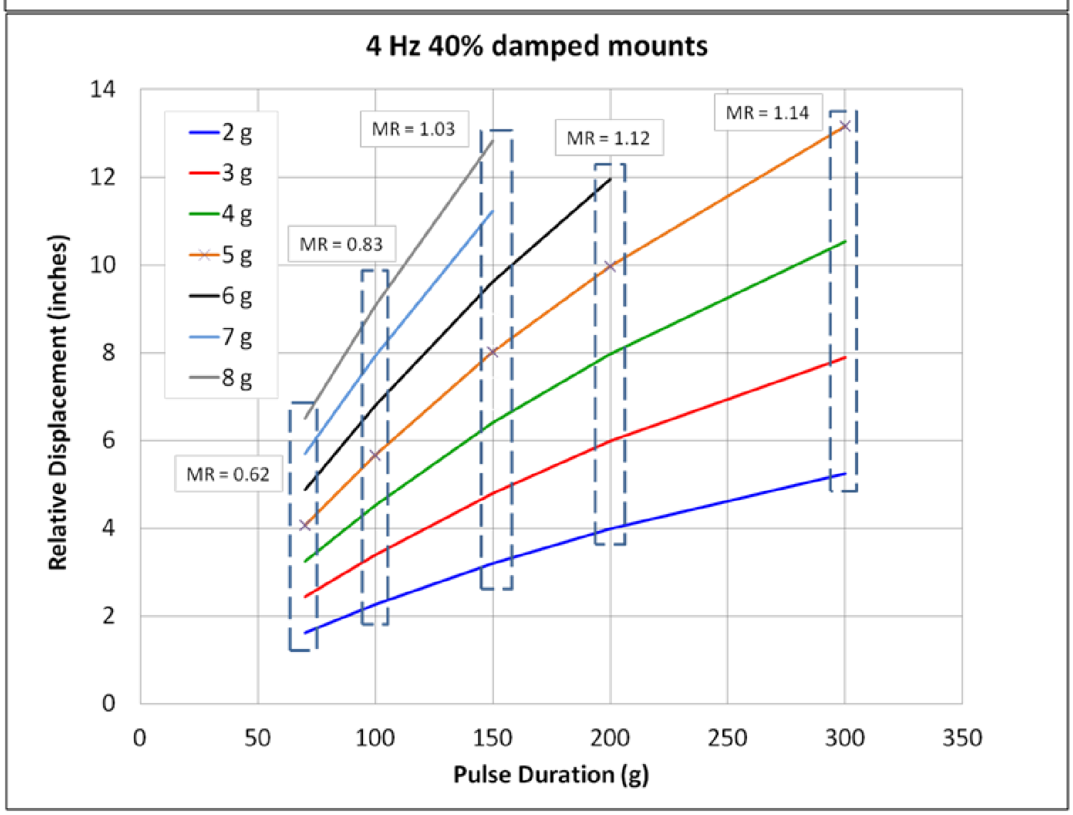


Figure F3. 4 Hz Mount System Computational Results

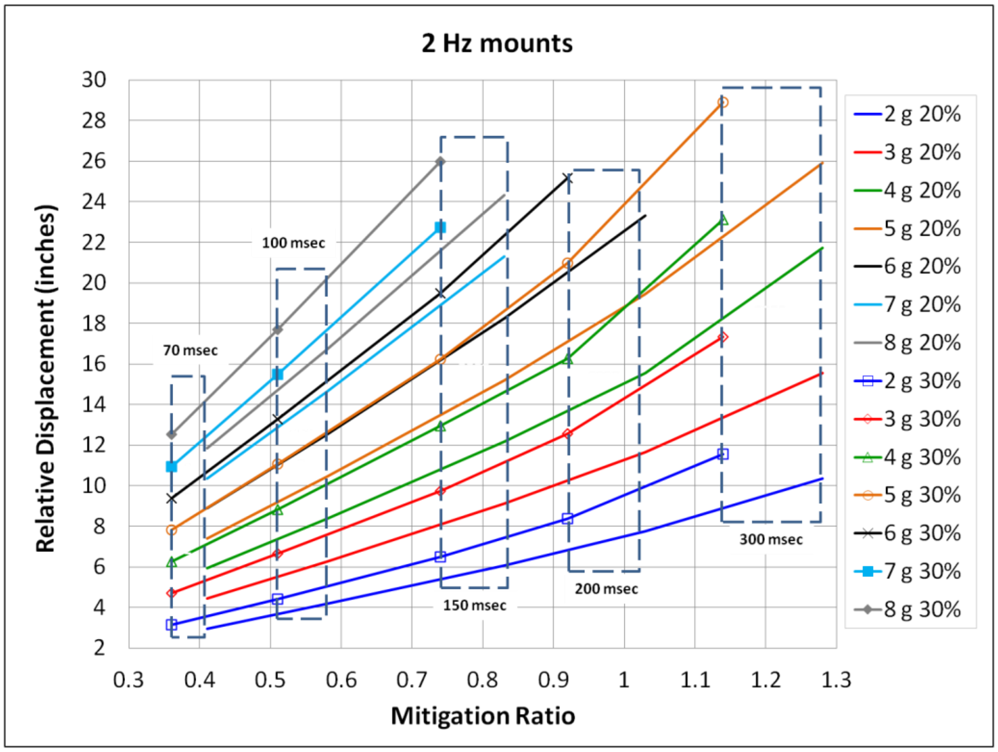


Figure F4. 2 Hz Mount Results for 20% and 30% Damping Ratios

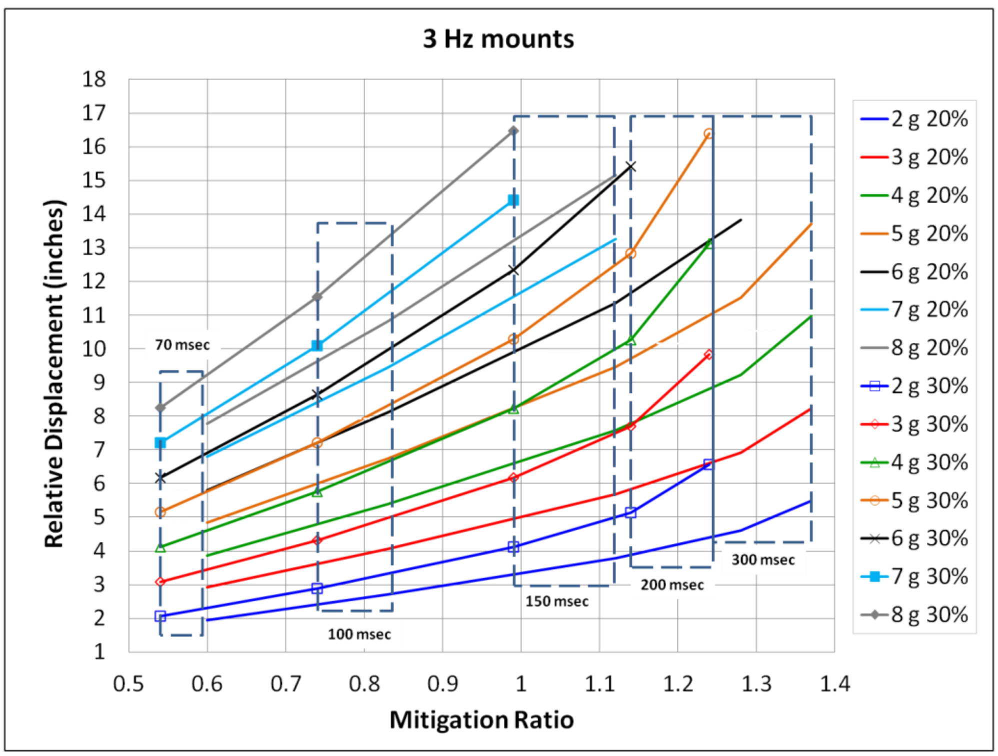


Figure F5. 3 Hz Mount Results for 20% and 30% Damping Ratios

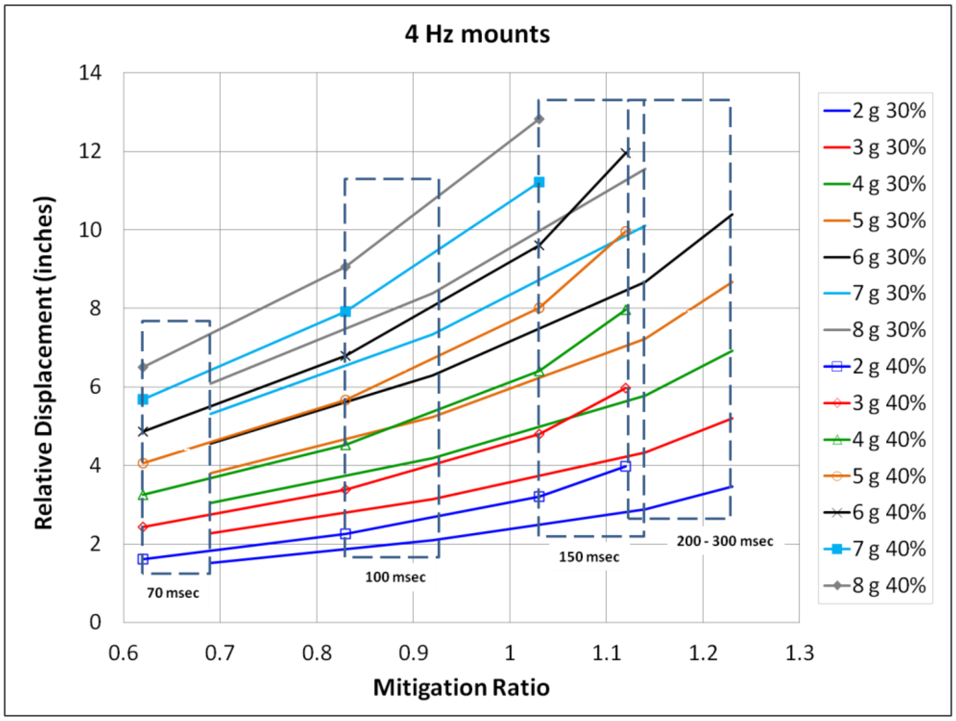


Figure F6. 4 Hz Mount Results for 30% and 40% Damping Ratios

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APPENDIX G. EQUIPMENT SHOCK HARDENING

Laboratory Shock Tests

When equipment is installed without shock isolation mounts in a dynamic environment it is referred to as hard-mounted equipment. When hard-mounted equipment successfully survives a laboratory shock test before installation in a craft it is referred to as hardened equipment for the intended environment (i.e., hardened to withstand shock). Figure G1 shows an example of a laboratory test machine capable of generating appropriate shock pulses. In this photograph a small test item is seen installed on a test fixture on top of the test machine. The payload capacity on this machine varies from 650 lbs to 1100 lbs (for 20 g – 23 msec and 15 g – 23 msec half-sine pulses, respectively).

Figure G1. Example Laboratory Shock and Vibration Machine[9]

[9] Ling Dynamic Systems Vibration System, Model V894/440T S/N 89101; Unholtz-Dickie Vibration Controller Model VWINN II S/N 00275921, photograph courtesy of SPAWAR Atlantic, Charleston, S.C.

Equipment Damage Mechanisms

There are numerous damage mechanisms (i.e., damage modes) that can lead to equipment malfunction or failure, including failure of attachment bolts, screws, enclosures, or internal structures due to material overstresses, broken lead wires, cracked solder joints, delaminated printed circuit boards, and electrical shorts. Failures can also occur due to broken or disconnected plugs, sockets, circuit cards, or circuit card subcomponents. In high speed craft these damage modes can be excited by a single severe wave slam, which can lead to any of the modes of failure, or damage modes can be excited by hundreds of lower severity wave impacts that can lead to solder joint failures or dislodged friction fittings (e.g., circuit cards or plugs) over time. Laboratory test methods should therefore simulate the effects of both failure modes: a single severe impact and repeated low severity impacts.

Laboratory Shock Test Requirements

The following shock machine test requirements are consistent with Procedure I - Functional Shock cited in Military Standard, MIL-STD-810G, Change 1, Method 516.7, Shock when implemented using laboratory shock test machines. Procedure I tests equipment in its functional modes to assess physical integrity, continuity, and functionality when exposed to the effects of operational shock loads. Alternative testing methods such as ANSI Standard S2.62-2009 may also be used. The half-sine pulse amplitudes and durations are different than actual wave impact shock pulses, but shock response spectra comparisons were used to show the test machine pulses have greater damage potential (i.e., includes a margin) than wave impact pulses to mitigate the risk of failure[10].

Standard Case

These standardized test requirements are applicable for all craft. Equipment may be installed at any location on any craft in any orientation for all planned craft speeds and operating sea states after successful completion of these tests. It is recommended that two types of shock tests be required to minimize the risk of electrical and electronics equipment malfunction or failure in high-speed craft. The first test is a single severe shock test in each axis repeated 3 times.[11] The second test is one with 800[12] lower severity shock pulses spaced at 1-second intervals repeated in each axis. Example language for hard mounted equipment requirements is presented in the following paragraphs.

Single Severe Shock Test

The test item shall maintain its physical integrity, continuity, and functionality during and following a laboratory shock machine test that subjects it to a single half-sine acceleration pulse of 20g – 23 msec in each of its three axes (positive and negative axis directions) in accordance with MIL-STD-810G w/change 1, section 516.7, Procedure 1, Functional Shock. Each test shall

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[10] Margins documented in a separate NSWCCD technical report to be published soon.
[11] Three repeated shock tests per axis is recommended by MIL-STD-810G.
[12] The 800 number was selected to simulate a 15 to 20 minute seakeeping trial. Experience suggests that new equipment that can withstand its first exposure to low severity trials will not fail in this mode during subsequent runs.

be repeated 3 times. Operational testing and visual inspection shall be conducted after each test to verify physical integrity, continuity, and functionality.

Repeated Low Severity Shock Test

The test item shall maintain its physical integrity, continuity, and functionality during and following exposure to 5.0 g - 23 msec half-sine pulses, 800 pulses at 1.0 second intervals in each of its three axes (or as specified) in accordance with MIL-STD-810G w/change 1, section 516.7, Procedure 1. Operational testing and visual inspection shall be conducted after the test to verify physical integrity, continuity, and functionality.

Known Orientation and Location Case

Except for equipment mounted on a mast, arch, or cabin top, equipment that is installed only in a vertical (Z) up orientation may be subjected to a single severe half-sine pulse of 10g – 23 msec in its X (surge) and Y (sway) axes, and 20 g – 23 msec in its positive vertical (Z) axis each test repeated 3 times.

Craft rigid-body pitching in rough seas results in severe response motions on the mast (or arch, or cabin-top) in the X (surge) direction that can be equal in amplitude to the bow vertical acceleration depending upon moment arm relationships. Therefore equipment installed in the vertical up orientation on a mast, arch, or cabin top structure should be tested in surge (X), sway (Y), and heave (Z) directions using the standard 20 g - 23msec pulse.

Limited Application Case

Table G1 lists test severity options for acquisition flexibility for unique procurements (e.g., high value or fragile components) where general installation in any craft at any location is not anticipated. Instead of the standardized 20 g single-severe test for the vertical (Z) axis, a 10 g or 15 g vertical peak acceleration may be used for the 23 msec half-sine shock pulse depending upon the craft and location.

Table G1. Limited Application Requirements by Craft Size

Craft Size		**Location**		
Length (ft)	**Weight (Kilo-lbs)**	**LCG**	**Coxswain**	**Bow**
65 - 85	**105 - 160**	**10g**	**15g**	**20g**
40 - 70	**35 - 70**	**10 g**	**15 g**	**15 g**
35 - 40	**14 - 25**	**15 g**	**15 g**	**20 g**

Isolated Equipment

The peak acceleration amplitudes for the 23-msec half-sine shock pulses listed in Table G1 are applicable only for hard-mounted equipment. Equipment installed on vibration mounts or with internal vibration mounts shall be shock tested with vibration mounts installed. A vibration mount for HSC is considered any mount with less than 2-inches of excursion space. The peak acceleration amplitudes for the 23-msec half-sine shock pulses listed in Table G1 and G2 are not applicable for testing shock isolation seats or equipment installed on shock isolation mounts.

Summary

Table G2 summarizes the single severe shock tests (i.e., 3 times in each axis) for standardized and limited applicability cases. The additional test with 5 g – 23 msec half-sine pulses delivered 800 times at 1-second intervals is applicable for both standard and limited applicability cases.

Table G2. Standard and Limited Application Single Severe Test Requirements

Test Requirement	Scope	Equipment Test Axis	Half-sine Shock Pulse
Standard	All craft, all orientations, all sea states, all locations including masts and arches	X	20 g - 23 msec
		Y	20 g - 23 msec
		Z	20 g - 23 msec
Limited Applications	Equipment installed only vertical up (except masts/arches)	X	10 g - 23 msec
		Y	10 g - 23 msec
		Z	20 g - 23 msec
	Unique acquisitions, high value, craft specific, location specific	See Table G1	

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