NSWCCD-80-TR-2018/015 The Rationale, Assumptions, and Criteria for Laboratory Test Requirements for Evaluating the Mechanical Shock Attenuation Performance of Marine Shock isolation Seats

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Technical Report

THE RATIONALE, ASSUMPTIONS, AND CRITERIA FOR LABORATORY TEST REQUIREMENTS FOR EVALUATING THE MECHANICAL SHOCK ATTENUATION PERFORMANCE OF MARINE SHOCK ISOLATION SEATS

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CONTENTS

FIGURES

TABLES

SYMBOLS, ABBREVIATIONS, AND ACRONYMS

A_{MAX} .... vertical peak-acceleration
ATD .... anthropomorphic test device
c .... damping coefficient
DRI .... Dynamic Response Index
DSRS .... relative displacement shock response spectrum
f .... frequency
ft .... feet
g .... acceleration due to gravity
Hz .... hertz
IMO .... International Maritime Organization
k .... stiffness
m .... meter
msec .... millisecond
MR .... mitigation ratio
NATO .... North Atlantic Treaty Organization
NSWCCD .... Naval Surface Warfare Center Carderock Division
R .... ratio of shock pulse duration divided by natural period
RMS .... root mean square
RMQ .... root mean quad
SDOF .... single degree of freedom
sec .... second
SRS .... shock response spectrum
T .... shock pulse duration time
τ .... natural period of vibration
ω .... circular frequency

ADMINISTRATIVE INFORMATION

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SUMMARY

This report presents the rationale for engineering assumptions and criteria used to develop standardized requirements for laboratory testing of marine shock isolation seats prior to installation in high speed craft.

INTRODUCTION

Purpose

The purpose of this report is to document the engineering rationale, assumptions, and criteria used to develop laboratory testing requirements for passive, marine, shock isolation seats. The requirements are published in NSWCCD-80-TR-2015/010 Rev A, 22 June 2018, *Laboratory Test Requirements for Shock Isolation Seats* [1]. The original version was published in May 2015 [2].

Background

In the acquisition process, the systems engineer is faced with the daunting task of specifying the needed, and sometimes desirable, characteristics and features while not overly restricting potential industry solutions. In a 'system of systems', (e.g., an entire vehicle) the human machine interfaces that impact mission effectiveness, especially the ones that deal with safety or comfort, have tremendous impact on the long term suitability of the platform for the mission.

Within the marine industry of high speed boats, over three decades worth of effort has been focused on one item in the human machine interface, the operators seat/bolster. This single component has been a hotly debated topic among leading experts, and the challenge for the systems engineer writing a boat specification has not been lessened. In fact, the systems engineer is faced with a broader understanding of what remains unknown and what should be considered, rather than what is definitively known. It stands to reason that understanding the dynamic interactions between a random sea surface and a boat moving at high speed at the interface of two fluids, and then determining the effect of wave impact shocks on a human operator is a complex problem to solve. But the issue is not to solve a complex problem. The first step in resolving the complexity can be simplified by answering a different question. How do we specify a "good" seat?

The test requirements presented in Reference 1 are not the panacea defining every attribute of what comprises a "good" seat. The requirements are however, a first step, to ensure a minimum bar can be established. In Reference 1 the first step requires that only seats that mitigate simulated wave impact shock when evaluated against the test criteria are to be considered for use in the marine platform. Thus far the seat manufacturers, who have the knowledge how to achieve, test and document this mitigation, have also demonstrated an acceptable understanding of other features (e.g. comfort, weight, space, adjustability). It is anticipated the current version of Reference 1 will not be the test criteria in use 5-10 years from now, but a beginning point must be selected to separate seats that mitigate from seats that amplify shocks due to wave impacts.

Passive shock isolation seats have typically used coil springs and dampers (i.e., shock absorbers) or leaf-spring assemblies to reduce the shock forces experienced during severe wave impacts. When the severity of the shock force transmitted through the seat assembly is less than the severity of the impact force at the base of the seat it is referred to as mechanical shock attenuation, or shock mitigation. The word mechanical refers to the excitation of the mechanical spring-damper assembly. The primary purpose for installing marine shock isolation seats is to reduce the accelerations delivered to seat occupants, but recent observations from numerous sources have reported that improperly designed seats can amplify shock inputs [3 – 14]. There was therefore a need to ensure before installation in a boat, that seats can reduce wave impact shock inputs. The test requirements in Reference 1 fill that need.

The laboratory test requirements in Reference 1 are a collection of best practices, preferences, and expectations for laboratory testing based on lessons learned from collaborative international investigations that included analyses of data from high-speed craft trials and acceleration data from seat drop tests performed at different laboratories [15 – 29]. Reference 1 does not specify the type of test method to be used, but it does illustrate recent testing that used different drop test apparatus. The inclusion of sixty-one references in the main body of this report demonstrates the engineering foundation upon which the laboratory seat test requirements were developed.

Completion of laboratory testing and successful demonstration of reduced shock inputs to seat occupants is a first step toward integration into a boat design. Many other seat design attributes related to form, fit, function, comfort, fatigue and safety must also be considered to determine which seat design is most appropriate for different high-speed boat applications.

Scope

This report explains the rationale for requirements for laboratory testing of marine shock isolation seats published in Reference 1. Reference 1 is applicable for testing shock isolation seats to be installed in high-speed planing craft with nominal lengths of 7-meter (22.9 feet) to 30-meter (98.4 feet). The test procedures are intended only for passive seats with no active sensors or mechanisms for real-time adaptation to the dynamic environment. The impact test severities are representative of a broad range of wave impact severities observed in different craft with various mission profiles. While it is possible that higher impact severities might be observed in future trials, there is no intent to test to all possible at-sea scenarios. The test requirements do not address the potential for adverse health effects (e.g., discomfort or injury).

RATIONALE FOR TEST ASSUMPTIONS AND CRITERIA

Low Cost

International collaborations were established in 2013 to develop a draft standard for laboratory testing marine shock isolation seats [26]. As a guiding principle it was agreed that test requirements should focus on simple test methods and procedures that can be implemented by seat designers, seat manufacturers, or independent testers with minimal resources while maintaining sufficient technical rigor to evaluate mechanical shock attenuation. Three factors were identified as cost drivers:

- the use of anthropomorphic test devices (ATD)
- seat occupant weight for male and female populations
- wave impact shock pulse duration

The implications of each of these related to testing cost is discussed in the following sections.

Anthropomorphic Test Devices

The use of payload ballast weights is allowed during testing to simulate the mass of a seat occupant. If available, ATDs may be used.

Anthropomorphic test devices (ATD) are routinely used in research laboratories to investigate impact effects on seat occupants for numerous modes of transport. While the use of ATDs is highly desirable, requiring their use adds cost. They are therefore not required during seat testing.

Seat Occupant Weight

Three seat payload weights were selected for testing: 5^{th} percentile female, 50^{th} percentile male and 95^{th} percentile male.

Anthropometric measurements for female and male populations typically include body weights for the 5^{th}, 50^{th}, and 95^{th} percentile weights. It would have been desirable to test seats with payloads that simulate all three male and female percentiles (i.e., six payload weights), but as a time and cost reduction measure, only three payload weights were selected for testing.

Shock Pulse Duration

The nominal shock pulse duration of 100 milliseconds (msec) with allowed tolerances was selected for testing.

Analyses of at-sea trials data shows that wave impact pulse durations from 100 msec to 450 msec vary depending upon craft weight, heading, speed, and sea state [27, 30]. It would be desirable to adopt test methodologies that cover a broad range of values (e.g., 100 msec to 350 msec or more), however, it was known that controlling long duration pulses requires reconfiguration of test apparatus between tests and it would require multiple pulse durations (e.g., 2 or 3 values) for the range of selected impact severities (e.g., rigid body peak accelerations from 3 g to 10 g). The time and cost to perform a large matrix of tests (e.g., up to 36 to 54 tests) was considered cost prohibitive. Therefore, a single nominal duration was selected for six test severities and three payload weights (i.e., 18 tests). The selected 100-msec pulse duration is characteristic of the more severe wave impacts observed during at-sea trials [30].

Test Method

It was determined that test requirements should not hinder use of innovative shock test methodologies, or require expensive test apparatus, or require specific test apparatus that may not be widely available. Any method that applies a vertical shock pulse to the shock isolation seat may be used. The determination to require a vertical pulse is explained under the heading *Shock Pulse Severity*.

Number of Tests

Each test (i.e., at different severity levels) specified in Reference 1 must be repeated three times.

Common practice in shock test standards is to conduct a specified test three times to demonstrate repeatability of outcomes [22]. Addition of this requirement resulted in a total of up to 54 tests (6 severities x 3 payloads x 3 repeats) depending upon the type of craft.

Mechanical Shock Attenuation

Reference 1 uses a seat performance metric based on mechanical shock attenuation properties of the seat rather than a biomechanical metric related to seat occupant comfort.

Exposure to single severe impacts or repeated wave impacts over time can lead to seat occupant discomfort or possible adverse health effects. A robust laboratory method to address these risks is highly desirable. However, the multitude of biomechanical effects related to all possible injury mechanisms induced by multi-axis forces and rotations for a broad range of seat occupant age, size, fitness, or pre-existing conditions cannot be simulated in simple or cost effective laboratory tests. The requirements in Reference 1 therefore do not employ biomechanical metrics or multi-axis forces or rotations.

As an alternative cost effective engineering approach, the focus of the required test methodology is to ensure that the mechanical properties[1] of a passive seat are properly tuned to avoid shock amplification[2]. Figure 1 shows classical shock and vibration transmissibility curves for linear single-degree-of-freedom (SDOF) isolation mounts where the ordinate axis is the ratio of the natural period of the isolator (τ) divided by the natural period of the excitation force[3]. The blue curves are for a sinusoidal vibration excitation. The red curves are for a half-sine shock pulse. In each set of curves the upper curve is for 10 % damping with successive lower curves for 20 %, 30%, and 40% damping. The transmissibility axis is the peak response acceleration above the isolator divided by the base input peak acceleration. The curves show that there are

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[1] Example seat mechanical design parameters include stiffness, damping, and excursion space.
[2] This is analogous to properly tuning vibration mounts to avoid resonance.
[3] The period of shock excitation for these tests is the half-sine pulse duration (T).

regions where the transmissibility is greater than one, meaning the excitation is amplified. In vibration environments this is the well-known resonance condition. For a single shock pulse excitation it is referred to as dynamic amplification[4]. The design parameter in this simple example that controls effective mechanical shock attenuation is the ratio of the isolator's natural period (τ) divided by the shock pulse duration (T). For the shock transmissibility curves (red) this means the period ratio must be greater than approximately 2 to 3 depending upon the damping characteristics of the system. If the ratio is less than 2 to 3 the transmissibility will be greater than 1 and dynamic shock amplification will occur. For shock pulses that result in dynamic amplification a ratio less than 2 to 3 is referred to as pulse-period mismatch.

The purpose of the test requirements in Reference 1 is to demonstrate that a passive shock isolation seat falls in the attenuation region (i.e., transmissibility < 1) in Figure 1 and not the amplification region (i.e., transmissibility > 1). Appendix A provides example acceleration data that illustrates examples of seat mechanical amplification and seat mechanical attenuation.

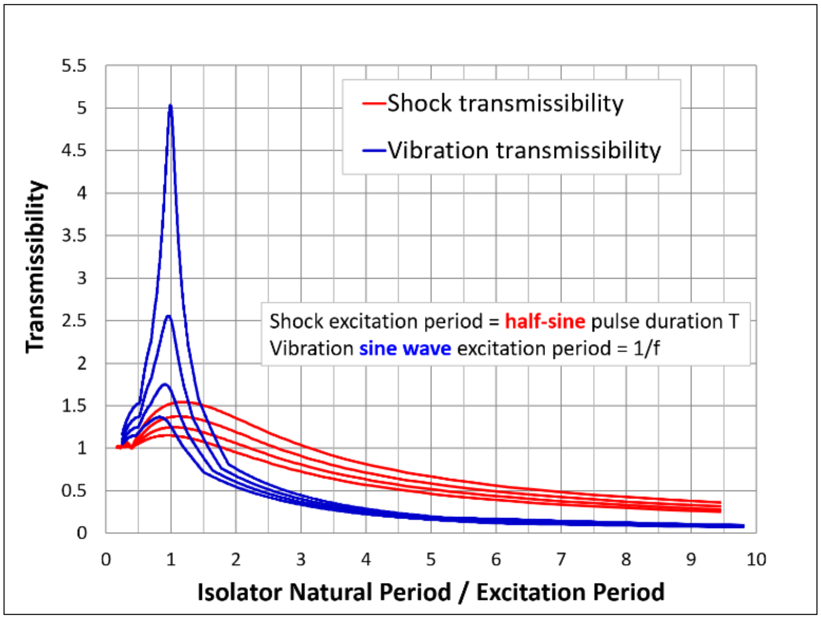

Figure 1. Transmissibility Curves for Shock and Vibration

Shock Pulse Severity

Key Parameters

Five key parameters for quantifying shock pulse severity were selected based on earlier investigations by the National Space and Aeronautics Administration (NASA).

NASA studied human tolerance to abrupt accelerations using numerous testing methods devised to simulate different impulsive loads (e.g., catapult acceleration and ejection seat accelerations) that could push the limits of human tolerance [3, 30, and 31]. They relied on the

[4] This is not seat bottom impact, which is a second cause of shock pulse amplification.

study of rigid body accelerations to evaluate aircraft dynamics or ejection seat thrust. Their conclusions were that human tolerance to rapidly applied acceleration depends primarily upon five factors:

- the direction in which the accelerating force (i.e., shock pulse) is applied
- the magnitude of the shock pulse
- the shape of the shock pulse
- the duration time of the shock pulse
- how rapidly the shock pulse is applied (i.e., jerk)

The implications of each of these related to shock mitigation seats is discussed in the following sections.

Acceleration Direction

The test method requires that a vertical shock pulse be applied at the base of a seat.

The forces applied to high-speed craft in rough seas can result in complicated seat occupant motions in all six-degrees of freedom. The largest recorded accelerations at the base of a seat are always in the vertical direction. Although it would be desirable, a multi-axis seat test requirement would add significant complexity, time, and cost. As more data becomes available, and test methods are refined, accelerations in the other directions and rotations should be considered in the future. In the interim, Reference 1 requirements only focus on attenuation of the most severe vertical shock input[5]. This is a simple first-step approach to ensure that seats can demonstrate vertical shock attenuation in the laboratory before being considered for installation in a craft.

Acceleration Magnitude

A range of rigid-body peak accelerations from 3 g to 10 g (depending upon type of craft) was selected for testing.

The amplitude of the rigid body acceleration is proportional to the shock force acting during an impact [33 - 47]. Reference 1 requires that a 20 Hertz (Hz) Butterworth low-pass filter be applied to recorded acceleration data to remove the high-frequency vibration content[6] in the signal so that only the low-frequency rigid body acceleration remains[7]. The precedent for using the 20 Hz low-pass filter in a standard for single shock impacts with water was set by International Maritime Organization (IMO) test requirements for lifeboat drop testing [37].

Collaborative comparisons of high-speed craft data concluded that the maximum acceleration (i.e., rigid body) for laboratory testing should be on the order of 8 g to 10 g [21, 25, 27, 47]. These are the peak acceleration amplitudes of the vertical shock pulses (i.e., shock inputs) to be simulated during a laboratory test. A sequence of laboratory tests with successively higher peak accelerations from 3 g up to 10 g was therefore adopted. Unless otherwise specified, Table 1 lists the peak acceleration levels to be achieved for different types of craft (referred to as craft class) depending upon planned operational envelopes (e.g., sea state and maximum speed).

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[5] Reference 1 does include an optional inclined test to evaluate the effects of an off-axis impact.

[6] The high frequency content is caused by small displacement vibrations in the vicinity of the accelerometer.

[7] The use of low-pass and high-pass filters is referred to as response mode decomposition [46]. A low-pass filter is used to estimate the rigid body acceleration in a raw unfiltered acceleration signal. A high-pass filter can be used to estimate the vibration content in the unfiltered signal.

Table 1. Craft Class and Test Severity Levels

Test Severity					Craft Class				
Threshold Level	Peak Acceleration	Nominal Impact Duration	Nominal Drop Height		Class 4			Class 3	Class 2
	g	sec	m	ft	Military 4-3	Military 4-2	Military 4-1	Search and Rescue	High Speed Commercial or Leisure
6	10.00	0.10	2.07	6.78					
5	8.00	0.10	1.32	4.34					
4	6.00	0.10	0.74	2.44					
3	5.00	0.10	0.51	1.69					
2	4.00	0.10	0.33	1.08					
1	3.00	0.10	0.18	0.61					

The threshold levels in Table 1 are based on class definitions described in Appendix B. The shaded boxes in Table 1 under each class correspond to the sequence of tests and the most severe test to be performed for the class. For example, a shock isolation seat for a Class 3 search and rescue craft would be subjected to four threshold test levels, starting at 3 g and ending at 6 g. Three tests are specified for each severity level in accordance with standard laboratory shock test methods [22]. It is important to note that the NASA conclusions also stated that "magnitude alone does not define human tolerance, nor does acceleration cause injury". "Stress, a result of acceleration causes injury. Any cogent discussion of magnitude and human tolerance is fraught with danger without appreciating the role of (pulse) duration [31]".

Shock Pulse Shape

A half-sine acceleration shock pulse was selected for seat shock testing.

The shape of the rigid body vertical acceleration observed in high-speed craft data when impact forces dominate buoyancy and hydrodynamic forces can be simplified for laboratory testing or analytical study as a half-sine pulse [34, 39, 43 - 45]. Some wave impact pulse shapes tend more toward a half-sine shape that is skewed slightly to the left, however the half-sine is well suited for laboratory testing. Figure 2 illustrates the half-sine representation of the rigid body vertical acceleration pulse (i.e., the black curve) 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.

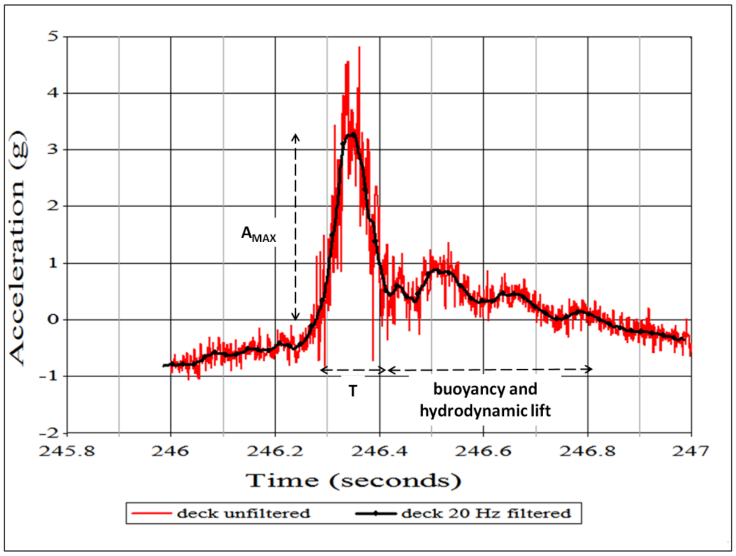

Figure 2. Half-Sine Pulse Approximation[8]

When rigid body accelerations are less than roughly 2 g the shape of the pulse may or may not compare well with a half-sine shape. These different shapes vary depending upon the hull impact location on the leading flank, crest, or following flank of a wave. The lower magnitude wave impacts less than 2 g are not of interest for laboratory testing because operator feedback indicates impacts at the low amplitudes are not uncomfortable [48].

Laboratory tests have used a drop test method with sand as the impact medium to achieve an approximate half-sine pulse shape and to control shock pulse duration [1, 50].

Acceleration Pulse Duration

The nominal 100-msec shock pulse duration was selected for testing.

In Figure 2 the shock pulse duration (T) is shown as the time period when the rapidly increasing rigid body acceleration crosses zero and then decreases to an ambient level associated with vertical forces attributed to buoyancy and hydrodynamic lift. Figure 3 is a plot of shock pulse duration (T) versus vertical rigid body peak acceleration recorded for wave impacts at different locations in sixteen different craft during head-sea trials in rough seas. The peak accelerations are the vertical rigid-body peak accelerations estimated using a 10 Hz or 20 Hz low-pass filter. The choice of the low-pass filter cutoff frequency does not significantly affect the pulse duration [34] (i.e., for pulse durations greater than 100 msec). In this figure all wave impacts recorded during a trial 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 longitudinal center of gravity (LCG) on six craft that weighed from 14,000 pounds to 18,000 pounds [30]. The open squares in the plot correspond to data recorded at the LCG 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

[8] 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).

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 deck peak acceleration increases. The 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). Appendix C provides further descriptions of pulse duration in terms of the change in vertical velocity during a wave impact.

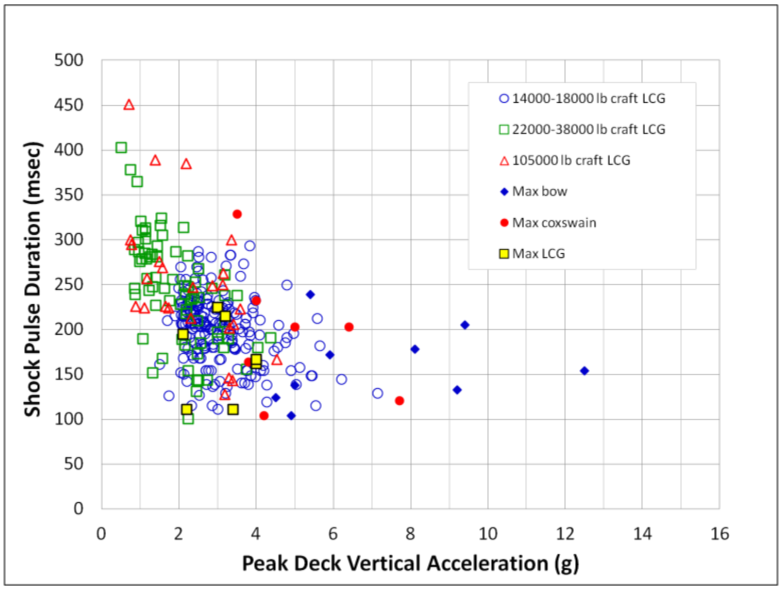

Figure 3. Wave Impact Pulse Duration Trends

Rate of Acceleration Application

The range of values for rates of acceleration application (i.e., jerk) characteristic of shock pulses recorded in craft are simulated in the laboratory by specifying the acceleration envelopes defined in the next section.

Shock Pulse Envelopes

The rigid-body (i.e., 20 Hz low-pass filtered) vertical acceleration recorded at the base of the seat is allowed to fall within tolerance criteria shown in Table 2, where (A) is the peak acceleration (20 Hz low-pass filtered using a Butterworth filter).

For each threshold severity level listed in Table 1 the peak recorded acceleration is allowed to vary within a twenty-percent envelope. The twenty percent envelope was chosen to be consistent with standardized shock test procedures [22, 50].

Table 2. Coordinates for Vertical Acceleration Tolerance Envelopes

Upper Envelope	
Time (t) in seconds	Acceleration (g)
-0.075	0
-0.02	1.2 A
0.02	1.2 A
0.06	0.6 A
0.5	0.15
0.6	0.15
Lower Envelope	
Time (t) in seconds	Acceleration (g)
-0.09 (A + 2) / (2 A)	-2
0	A
0.09 (A + 2) / (2 A)	-2

Figure 5 shows the tolerance envelopes (red dotted lines) from Table 2 constructed around the half-sine acceleration pulse (blue curve) to be achieved for a Level 4 test threshold. The larger envelope tolerances from approximately 0.05 seconds to 0.5 seconds are intended to envelope the seat base movement that may occur for some seat designs due to spring-damper oscillations after the impact. A ten-percent margin was selected as the minimum allowable shock input pulse duration (i.e., 90 msec) because seat mitigation performance can improve rapidly as shock pulse duration decreases below 100 msec (i.e., the observed minimum in Figure 3). A 20 Hz low-pass filter must be applied to the acceleration data before comparing with the allowable envelopes.

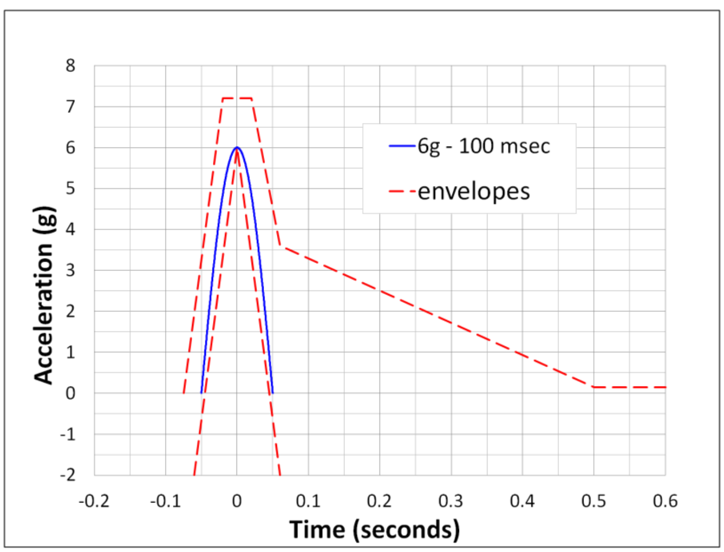

Figure 4. Example Envelopes for Level 4 Threshold Acceleration

Shock Mitigation Metric

The performance metric for mechanical shock attenuation specified by Reference 1 is the shock response spectrum mitigation ratio (MR_{SRS}). It is based on the computed response of a single degree of freedom mathematical model with a natural frequency of 8.4 Hz and 22.4 percent of critical damping.

The MR_{SRS} approach was selected because the shock response spectrum (SRS) provides an established methodology for quantifying shock pulse severity [22, 50 – 56]. As explained in Appendix D, it is used to quantify the mechanical attenuation performance of a seat by dividing the shock response spectrum of the seat cushion acceleration divided by the shock response spectrum of the deck input acceleration. The MR_{SRS} is the only shock metric currently available that is able to account for differences in deck and seat cushion pulse durations, jerk, pulse shape, and peak amplitudes. As a relative measure of impact severity it provides a consistent mathematical tool for comparing the relative severity of different shock pulses.

Recent investigations of the SRS mitigation ratio for marine seats in high-speed craft used a natural frequency of 8 Hz and 9 percent damping. The 9-percent damping value was selected because it provided a broader range of ratio values over which to evaluate seat mitigation performance [24]. The 22.4 percent damping value was subsequently selected to be consistent with Reference 37, *Testing and Evaluation of Life-Saving Appliances,* Maritime Safety Committee Resolution MSC.81(70), Life-Saving Appliances, 2003 Edition, International Maritime Organization, 2003 (see Appendix D). For single shocks this is equivalent to the ratio of the Dynamic Response Index (DRI) for the seat cushion response acceleration divided by the DRI for the deck acceleration input.

The DRI is used internationally in standards and technical literature [15, 58 – 61]. Table 1 lists numerous international documents where the DRI is specified or used as a criterion for quantifying shock load for seated occupants caused by vertical acceleration forces during single shocks. The DRI is used by the International Maritime Organization as the criterion for evaluating spinal force and seat occupant safety during ship lifeboat drop tests [37]. It is specified by the North Atlantic Treaty Organization (NATO) as the criterion for evaluating the risk of spinal injury to seat occupants in armored vehicles [58, 59]. It has been used as a shock isolation seat design criterion for individual severe wave impacts in a high speed craft [60], and it has been used to quantify the severity of different individual wave impacts recorded during high-speed craft tests [61].

Table 1. Use of DRI for Single Shocks

Document Type	Seat Occupant Application	Document Title	Organization	Reference
Test Standards and Criteria	Ship lifeboat freefall water impact	Testing and Evaluation of Life Saving Appliances, Lifeboat Drop Tests	International Maritime Organization (IMO)	IMO-MSC.81(70), 2003
	Land vehicle mine effects	Procedure for Evaluating the Protection Level of Armored Vehicles	North Atlantic Treaty Organization (NATO)	NATO-AEP-55, Volme 2 (Edition 2), 2011
	Land vehicle mine effects	Test Methodology for Protection of Vehicular Occupants Against Anti-Vehicular Landmine Effects	North Atlantic Treaty Organization (NATO) Human Factors and Medicine Panel	NATO-RTO-TR-HFM-090, 2007
Engineering Handbook / Guide	Fixed and rotary wing aircraft crash protection	Crew Systems Crash Protection Handbook	US Dept of Defense	JSSG-2010-7
	General vertical shock load applications	Effects of Shock and Vibration on Humans, Chapter 41	McGraw-Hill publication	Harris' Shock and Vibration Handbook, 2010
Technical documents	Seated anthropomorphic test device shock load response comparison	Comparative Analysis of THOR-NT-ATD and Hybrid-III ATD in Laboratory Vertical Shock Testing	US Army	ARL-TR-6648, 2013
	Aircraft ejection seat cushion study	Seat Interfaces for Aircrew Performance and Safety	US Air Force	AFRL-RH-WP-TR-2010-0083, 2010
	High speed craft shock isolated seat design study	Analysis, Optimization, and Development of a Specialized Passive Shock isolation System for High Speed Planing Boats	Tayloe Devices, Inc, and Tayco Developments, Inc.	A. Klembezyk, M. Mosher, 2003
	High speed craft shock isolated seat single impact injury parameter	The Modeling and Measurement of Humans in High Speed Planing Boats Under Repeated Vertical Impacts	University of Virginia and US Navy	C. Bass, R. Salzar, A. Ziemba, S. Lucas, R. Petersen, 2005

Computational results presented in Appendix E show that MR_{SRS} values using 80 Hz low-pass filtered acceleration data are within –-5 percent to +3 percent MR_{SRS} values using 20 Hz low-pass filtered acceleration data. The close comparisons indicate that the vibration signals in the 80 Hz low-pass deck data has little or no effect on seat cushion comparisons for an 8 Hz SDOF model. This suggests it would be satisfactory to use 1 Hz to 80 Hz bandwidth acceleration data for computing SRS mitigation ratios (MR_{SRS}). This is not the case however for determining shock pulse amplitude. Appendix E explains why a 20 Hz low-pass filter must be used when comparing the vertical peak acceleration recorded at the base of a seat with the allowable tolerance envelopes.

Conclusions and Recommendations

The rationale and criteria for shock isolation seat test requirements presented in this report are based on demonstrating in a laboratory before installation in a craft that a passive seat can mechanically attenuate in the vertical direction 100-millisecond half-sine shock pulses of varying rigid body peak acceleration amplitudes. Test severity options include six different peak acceleration levels corresponding to a craft's classification. This testing approach reduces the risk of installing a seat design in a craft that amplifies the severity of single severe wave impacts either by dynamic amplification or seat bottom impact.

The test requirements are intended only for passive seats with no active sensors or mechanisms for real-time adaptation to the dynamic environment.

The test rationale and criteria do not address seat occupant comfort or the potential for adverse health effects (e.g., extreme discomfort or injury), and they do not address the effects of repeated exposure to wave impacts over time.

It is recommended that the following three sentences be avoided because they misrepresent laboratory test goals by using non-descript words related to comfort, safety, or injury.

(1) *The laboratory test requirements demonstrate that a seat is safe (or not safe) for operational use.* This is not correct. The test requirements ensure that only safer seats are installed in a craft (i.e., safer seats that mitigate relative to seats that amplify).

(2) *The purpose of laboratory testing is to demonstrate that seat occupant injury will be avoided.* This is not correct. The test requirements do not evaluate the risk of injury. The testing reduces the risk of adverse effects by avoiding seat designs that amplify 100-msec shock pulses.

(3) *Successful testing demonstrates that seat occupants will not experience discomfort while operating in a high-speed craft in rough seas*. This is not correct. The testing minimizes the risk of discomfort by avoiding seat designs that amplify 100-msec shock pulses.

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Appendix A. Amplification of Deck Shock Inputs in High-Speed Craft

Physical Causes of Shock Amplification

Two improper design features of a seat can lead to shock amplification. First, there must be sufficient excursion space across the passive spring-damper assembly to prevent a seat bottom impact. The occurrence of seat bottom impact is primarily a function of the severity of the wave impact force. The larger the impact force the larger the required excursion space.

Second, dynamic amplification must be avoided. Dynamic amplification (i.e., without seat bottom impact) means the spring-damper assembly will amplify the shock input rather than attenuate it. Avoiding dynamic amplification requires that the ratio of the shock pulse duration (T) divided by the natural period of response (τ) of the passive spring-damper assembly (i.e., the inverse of the system natural frequency) be less than a limit value that depends upon the system damping properties. The following paragraphs explain both of these amplification phenomena.

Seat Bottom Impact

Figure A-1 shows an example of seat bottom impact in the unfiltered acceleration data recorded at the base of a shock isolated seat (black curve) and on the seat pan (red curve). The event is characterized by very large negative and positive high-frequency acceleration spikes. The impact spikes occur when the spring bottoms-out due to metal-to-metal contact. These large spikes are not associated with the force of the wave impact. They are the result of relative motion and contact between the deck and the seat pan response.

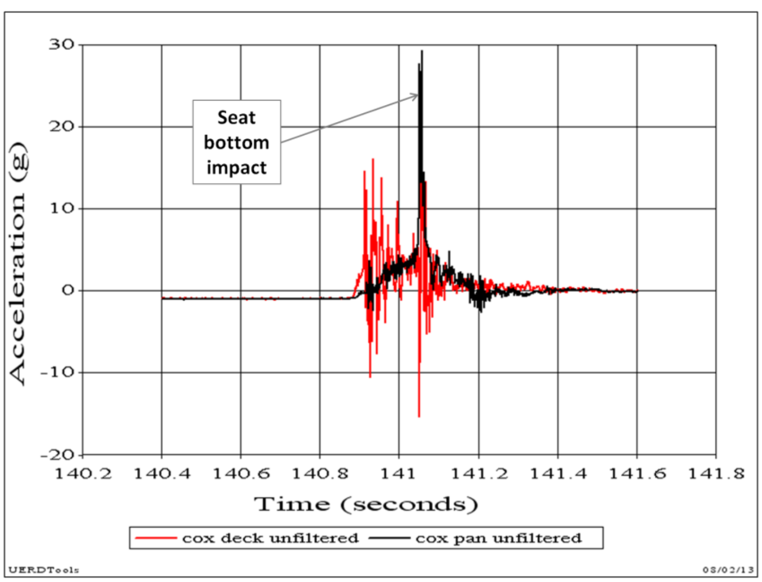

Figure A-1. Example of Seat Bottom Impact

Seat Dynamic Amplification

The second phenomenon that can lead to the seat response being more severe than the deck input is referred to as dynamic amplification. It is caused by pulse-period mismatch. An example of dynamic amplification caused by pulse-period mismatch is shown in Figure A-2. The black curve is the 20 Hz low-pass filtered acceleration on the deck at the base of a shock isolated seat. The red curve is the 20 Hz low-pass filtered seat pan acceleration. The seat pan 3.5 g peak acceleration (red) is greater than the 2.5 g deck peak acceleration with no indication of the occurrence of a seat bottom impact (i.e., peaks greater than 10 g).

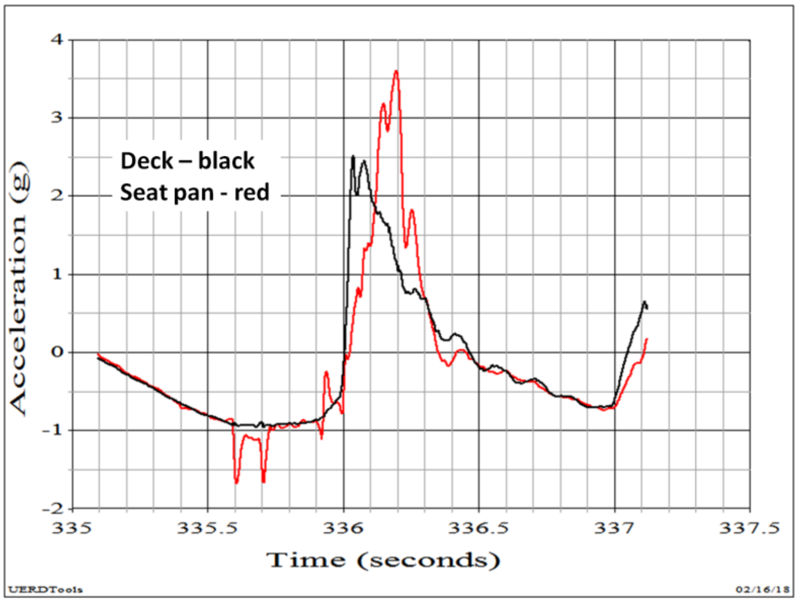

Figure A-2. Dynamic Amplification Due to Pulse-Period Mismatch

The physical relationships that cause dynamic amplification are illustrated by the shock transmissibility curve for half-sine acceleration shock pulses shown in Figure A-3 [A-1]. The curves were created using a single-degree-of-freedom (SDOF) spring-damper mathematical model and applying vertically upward acceleration shock pulses with half sine shapes. The shock transmissibility scale is the response peak acceleration divided by the input peak acceleration[9]. A value on the transmissibility scale (i.e., ordinate) 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. The different color curves correspond to different amounts of assumed damping in the SDOF model.

The X-abscissa scale in Figure A-3 is the non-dimensional parameter (R) obtained by multiplying the half-sine pulse duration (T) by the natural frequency (f) of the spring-damper

[9] The transmissibility scale is also referred to as the peak acceleration mitigation ratio (MR_{PEAKS}).

oscillator. Equation (A-1) shows that the value R is also identical to the half-sine pulse duration (T) divided by the natural period (τ_{sys}) of the oscillator[10].

$$R = T(f_{sys}) = \frac{T}{\tau_{sys}} \tag{A-1}$$

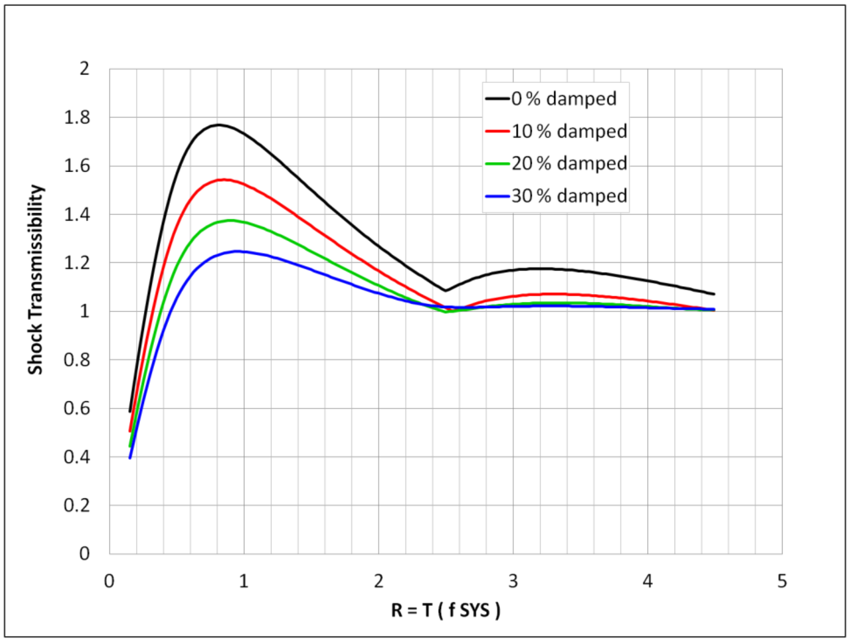

Figure A-3. Shock Transmissibility Curve for Half-Sine Pulse Input[11]

The R-limit value for 10-percent to 40-percent of critical damping varies from about 0.32 to 0.58, respectively. Values of R greater than these limits result in the seat's transmissibility greater than 1.0 (i.e., dynamic amplification).

Example Calculation

Figure A-4 shows calculated responses of a vertically oriented passive spring-damper SDOF model for two different shock inputs. The SDOF system natural frequency is 2 Hertz (Hz) and the damping ratio is 20 percent. One shock input (blue solid line) is represented by a half-sine pulse with peak amplitude 4 g and pulse duration of 150 msec. The calculated time-history response to this input (shown by the blue line with triangle symbols) reaches a maximum acceleration of 3.33 g. This is an example of shock mitigation. The peak acceleration has been reduced.

[10] Some texts define the R ratio as 2T divided by system natural period for comparison with vibration transmissibility curves for sine waves.

[11] This is similar to Figure 1 in the main body of the report, but the X-abscissa axis has been inverted (i.e., 1/R) in Figure 1.

The second shock input in Figure A-4 is also a half-sine pulse with the same 4 g peak amplitude, but it has a 300-msec pulse duration (red curve). The SDOF peak response for this input is 5.14 g (red with circles). This is an example of dynamic amplification where the response peak acceleration is greater than the input peak acceleration. This is an example of an incorrect natural frequency for a passive seat that will amplify the peak acceleration input rather than reduce it.

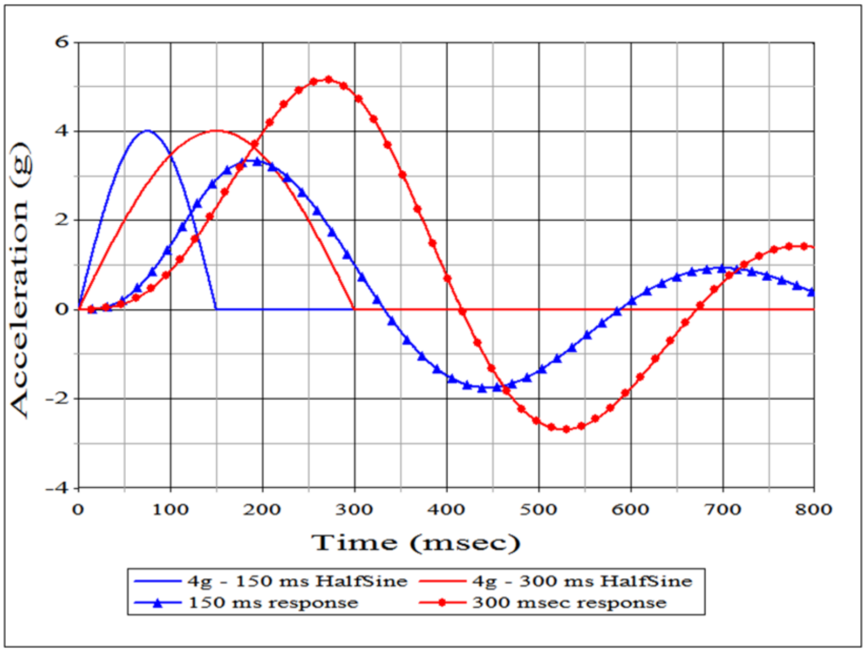

Figure A-4. SDOF Model 2 Hz – 20 % Damped Responses

Dynamic amplification can occur when the mitigation ratio (MR_{PEAKS}) is greater than 1.0. Figure A-3 shows that the theoretical upper limit for MR_{PEAKS} amplification is on the order of 1.77 for zero damping. Conversely, shock reduction (i.e., shock mitigation) occurs when MR_{PEAKS} is less than 1.0. For the example calculations shown in Figure A-4, the pulse-period ratio (R) from Figure A-3 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_{PEAKS} = 0.83), but it amplifies the 300-msec pulse (MR_{PEAKS} = 1.28). Dynamic amplification is like a resonance 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 spring-damper isolation system. This is illustrated by the red curves in Figure A-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).

The ratio of the peak acceleration on the pad divided by the peak acceleration on the deck (MR_{PEAKS}) has often been used as an estimate of the seat pad mitigation ratio. Most engineering handbooks and texts that deal with shock use the ratio of input and response peak accelerations to construct a shock transmissibility curve (e.g., Figure A-3). This approach is appropriate for shock pulses with durations much less than the natural frequencies (i.e., modes of vibration) of the system subjected to the shock pulse (e.g., shocks caused by blast). But for shocks caused by long duration wave impacts this ratio will often misrepresent the magnitude of a marine seat's attenuation performance (See Appendix E). This is because the ratio of peaks does not account for differences in jerk, pulse duration, and pulse shape between the input and the response accelerations. These three additional parameters are important when evaluating the effects of long duration shock pulses on lower frequency systems (e.g., not a rigid mass).

Appendix A Reference

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APPENDIX B. CRAFT CLASS DEFINITIONS and TEST SPECIFICATIONS

Four suggested craft classifications provide a framework for specifying test severities for shock isolation seats [B-1]: Class 1 Low Speed Commercial/Leisure, Class 2 High Speed Commercial / Leisure, Class 3 Search and Rescue, and Class 4 Military. This class rating scale was developed from trials experience on commercial, leisure, search and rescue, and military rigid hull inflatable boats with vessel lengths from 5 to 10 meters. Generic descriptions of Class 1, 2, 3, and 4 are provided below. An example test specification that could be included in craft acquisition requirements is also presented.

Class 1: Low Speed Commercial / Leisure

Class 1 describes small high speed craft carrying passengers of various ages and physical conditions, possibly including children and the elderly. Typical applications include ferry craft and sightseeing tours. Class 1 craft will typically operate at low speeds except in extremely calm conditions. Wave impacts are avoided. Class 1 vessels generally do not operate in poor weather. Class 1 corresponds to craft with operational environments not typically requiring personnel protection in the form of shock isolation seats.

Class 2: High Speed Commercial / Leisure

Class 2 describes small high speed craft similar to Class 1 vessels, where the Class 2 vessel operator may choose to operate at higher speeds, as limited by their own tolerance.

Typical applications include commercial operators offering thrill rides and marine wildlife tour boats that are capable of high speed transits. Some applications, such as maritime wind farm maintenance boats, may require operations in poor weather. Crew and passengers of Class 2 vessels are often required to meet physical fitness standards. Engines on Class 2 vessels are typically more powerful than on Class 1 vessels, and so speeds are typically higher, perhaps in excess of 20 knots when conditions allow. Wave impacts are more common on Class 2 vessels than on Class 1 vessels.

Class 3: Search and Rescue

Class 3 describes small high speed craft used for search and rescue (SAR), which often requires operations at high speed in poor weather, and in relatively high sea states. Class 3 vessel personnel are highly motivated, and well trained. They are experienced at operating in severe conditions, and are generally physically fit and healthy. Engines on Class 3 vessels often provide sufficient power to exceed 30 knots when conditions allow. Severe wave impact slamming events are typical for normal operation on Class 3 vessels.

Class 4: Military

Class 4 describes high-speed craft used for military operations. Personnel in Class 4 vessels are usually physically fit and very highly motivated. As a result of their training and experience they are more accustomed to sustained, extreme motions and wave impacts during high-speed operations.

Example Seat Specification

The class definitions identify broad applications across leisure, commercial, and military craft where there is potential for operating in successively more severe wave impact environments. It is understood that the definitions may or may not fit a specific commercial, search and rescue, or military craft. It is therefore important that craft owners, program managers, or operators develop seat test requirements that identify the maximum exposure severity for specific craft applications.

The following text provides an example of a seat test specification that could be included in acquisition requirements for a high-speed craft.

1.The craft shall have (X) permanently installed marine grade, adjustable, shock isolating seating, with arm rests, footrests and seat belts for crewmembers: (names of stations). Seats shall permit occupant use in both the seated and standing positions.

2. Shock isolation seats shall be laboratory tested in accordance with (cite reference). Testing shall be conducted three times at each severity level up to and including test severity Level 4 (Class 4-1) with three inert seat payloads equal to Y1 lbs., Y2 lbs., and Y3 lbs. (Note: if seats include table top arm workstation, the work station shall include the weight of any rated payload, installed at a representative location, during the performance of the test. (e.g. if the (name) station has an attached workstation table for controls/displays and is rated to accommodate Z lbs of controls/displays at the forward end of the workstation surface, then Z lbs of weight shall be placed on the forward end of the table during conduct of the testing.))

Appendix B Reference

B-1. Colwell, J.L., Gannon, L., Gunston, T., Langlois, R.G., Riley, M.R., Coats, T.W., *Shock Mitigation Seat Test and Evaluation*, The Royal Institute of Naval Architects, 2011-288.

APPENDIX C. WAVE IMPACT SHOCK PULSE DURATION

Sequence of Events

A vertical (heave) acceleration time history for one wave impact sequence and the velocity and absolute displacement (i.e., heave) curves obtained by integration are shown top to bottom in Figure C-1 [C-1].

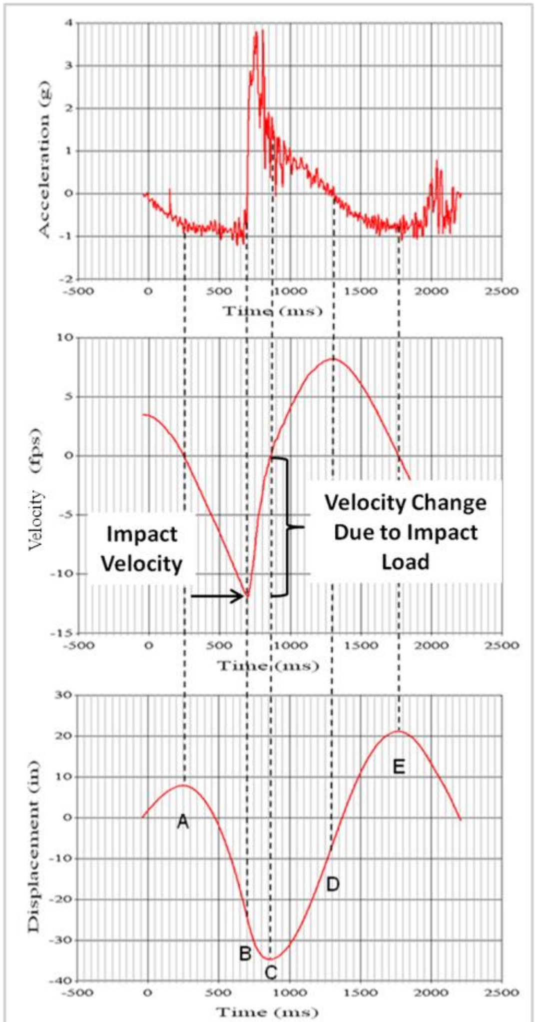

Figure C-1. Wave Impact Sequence of Events

The curves illustrate the wave impact period and non-impact periods. At time A, the -0.9 g vertical acceleration indicates a condition very close to free fall. The relatively constant -0.9 g from time A to time B and the linear decrease in velocity suggests that the craft is rotating downward with the stern in the water. The drop in height from time A to B is most likely a combination of heave and pitch. At time B, the craft impacts the incident wave, the velocity is at a minimum, the negative slope changes rapidly to a positive slope, and the force of the wave impact produces a sharp rise in acceleration. From time B to time C, the craft continues to move down in the water, the velocity approaches zero, and the acceleration decreases rapidly. At time C the downward displacement of the craft reaches a maximum, the instantaneous velocity is zero, and the impact event is complete. From time C to D forces due to buoyancy, hydrodynamic lift, and components of thrust and drag combine to produce a net positive acceleration. From time D to E, gravity overcomes the combined forces of buoyancy, hydrodynamic lift, and components of thrust and drag as another wave encounter sequence begins. The period of time in Figure C-1 from point B to point C is the wave impact period, i.e., the shock pulse duration (T). It is this period of time from B to C that is important for understanding and evaluating shock effects caused by wave impacts [C-2].

The shock pulse duration (i.e., the pulse period) is an important parameter for designing shock isolation systems for high-speed craft. When the pulse duration or range of durations is known the mechanical properties of a passive spring-damper assembly can be selected to ensure that the system is tuned to properly attenuate the shock pulses. Appendix A describes isolation system tuning in terms of the ratio of the shock pulse period (i.e., duration T) and the natural period (τ) of the isolation system.

Appendix C Reference

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C-2. Riley, Michael R., Coats, Timothy W., “The Simulation of Wave Slam Impulses to Evaluate Shock Mitigation Seats for High-Speed Planing Craft”, Naval Surface Warfare Center Carderock Division Report NSWCCD-80-TR-2013/16, May 2013.

APPENDIX D. SHOCK RESPONSE SPECTRUM

The shock response spectrum (SRS) is a computational tool used internationally to compare the severity of different shock motions [D-1 to D-11]. It is also referred to as a maximum response spectrum that can be used to analyse any dynamic event. It is especially useful for evaluating and comparing two different shock pulses (i.e., at the deck and on the seat cushion) that have different pulse shapes, peak amplitude, jerk, and pulse duration. It is used as a measure of mechanical shock isolation performance of a seat by comparing the deck input acceleration SRS with the seat pad response acceleration SRS for individual wave impacts.

The SRS uses a model of the single-degree-of-freedom (SDOF) system shown in Figure D-1 to compute the effects of an input motion X (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 X (t) at the base of the system the resulting response of the mass (m) is Y (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 D-1 is obtained by summing the inertial force of the mass and the forces within the spring and damper.

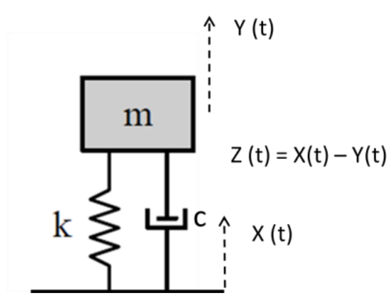

Figure D-1. Single-degree-of-freedom Mathematical Model

$$m\,\ddot{y}(t) = -k\,z(t) - c\,\dot{z}(t) \qquad \text{(D-1)}$$

Where *t* is time and:

$$\omega = \sqrt{\frac{k}{m}} \qquad \text{(D-2)}$$

The undamped natural frequency (f) in Hertz (Hz) of the SDOF system is given by equation (D-3).

$$f = \frac{\omega}{2\pi} = \left(\frac{1}{2\pi}\right)\sqrt{\frac{k}{m}} \text{ Hz} \tag{D-3}$$

The solution of equation D-1 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 Y(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 [D-7].

The relative displacement SRS is often used as a parameter to compare shock severity when two input shock motions are being compared. It is an intuitive engineering measure of severity because the relative displacement is proportional to the strain in the spring in Figure D-1. The shock pulse that causes the larger strain, and therefore the largest damage potential, is judged to be the more severe of the two shock pulses. Figure D-2 shows three vertical acceleration time histories recorded at different locations on a craft. The plot on the right is the computed maximum relative displacement SRS (DSRS) for each acceleration shock pulse. Visual inspection of the time histories on the left indicate that the red bow shock pulse is the most severe because of its higher amplitude. The DSRS curves on the right quantify the differences in severity. The key feature of the SRS approach is that it quantifies shock severity based on its effect on SDOF oscillators with varying natural frequencies. It characterizes the shock severity in the response domain, so that the effects of shock pulse shape, peak amplitude, jerk, and pulse duration can be taken into account for systems across a broad range of natural frequencies.

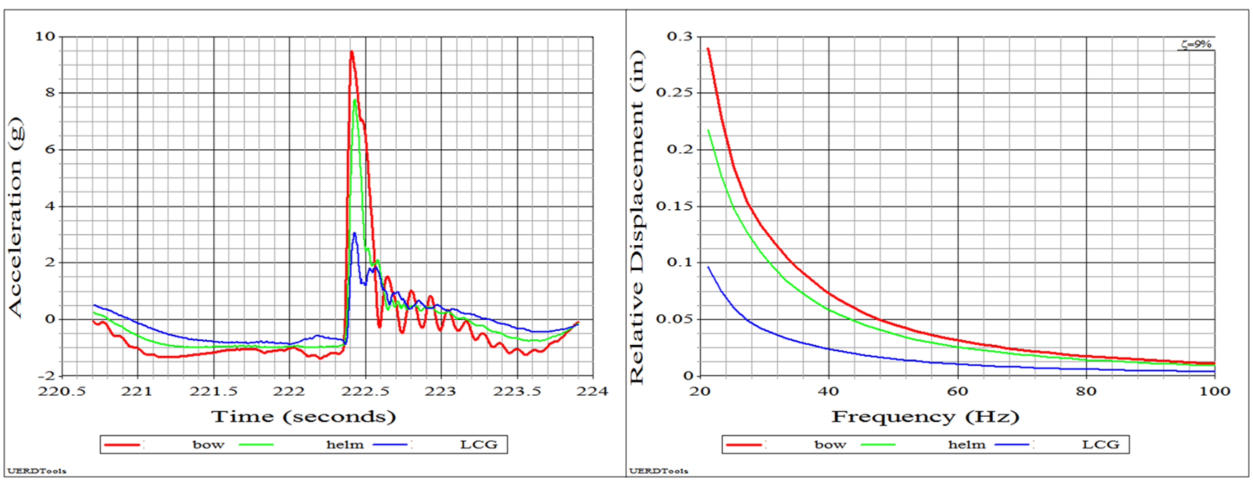

Figure D-2. Three Wave Slam Shocks and Relative Displacement SRS

Mitigation Ratio Using SRS

The universal approach to quantifying shock transmissibility is by dividing the severity of the shock response pulse above the isolation system by the severity of the base input shock pulse

[D-1]. Many engineering texts define a 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 is appropriate 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 vales, and pulse durations are not similar the preferred method of quantifying shock mitigation is to use the shock response spectra ratio given by equation D-4. This is because the SRS ratio inherently accounts for differences in the important shock parameters, including pulse shape, peak amplitude, pulse duration, and jerk.

$$\text{Mitigation Ratio} = \frac{\text{SRS}_{\text{Response}}}{\text{SRS}_{\text{Input}}} \qquad \text{(D-4)}$$

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. This is called dynamic amplification. 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 D-3 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 D-4 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 from approximately 45 Hz to 500 Hz the mitigation ratio is approximately 0.71 (i.e., the 5-g pulse is 29 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 [D-12]. 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 (i.e., strain or relative displacement), a result of acceleration, causes damage [D-13].

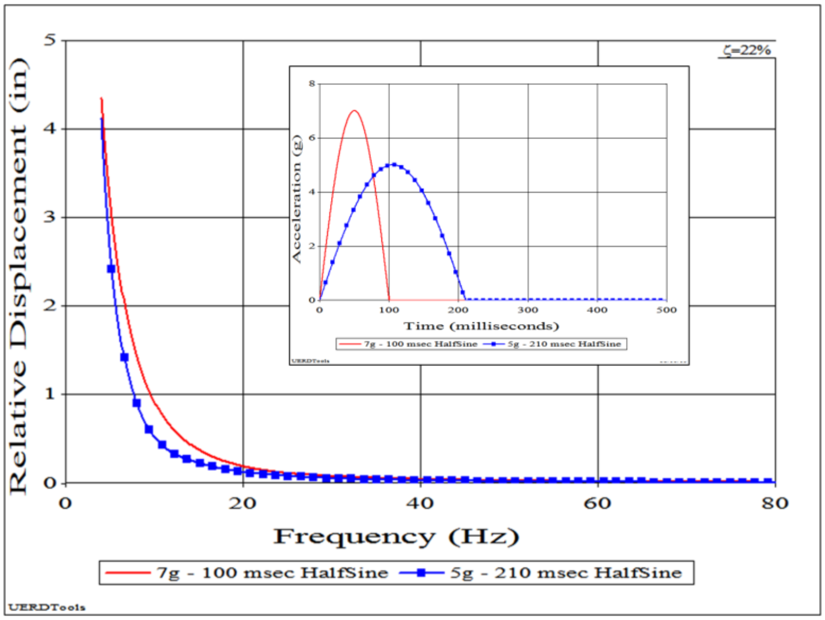

Figure D-3. Comparison of Hypothetical DSRS

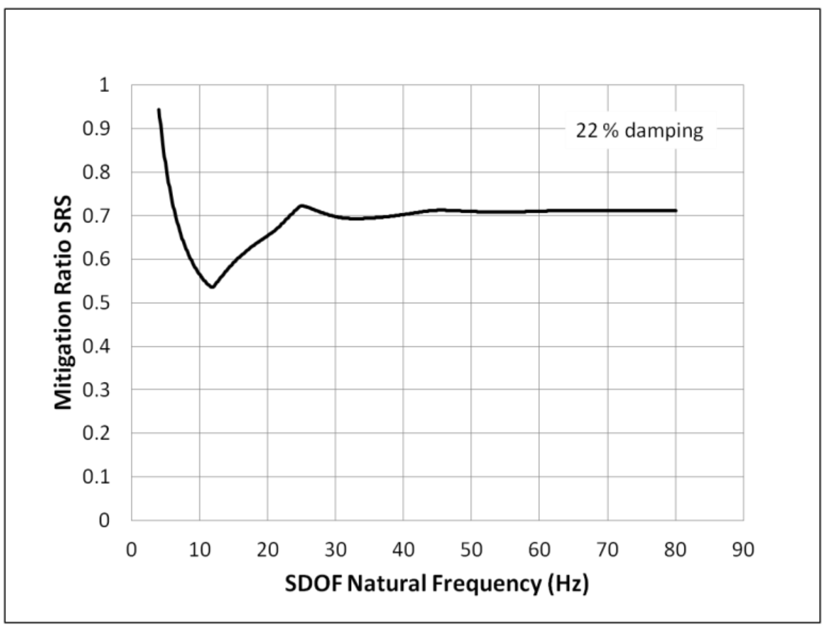

Figure D-4. Mitigation Ratio for 5 g and 7 g Half-sine Pulses

SRS Frequency of Interest

Numerous studies of the effects of a vertical shock load on a seated human have used an analogous, lumped mass model of the human body consisting of a mass, a spring, and a damper. The lumped mass model (i.e., single-degree-of-freedom model) was first studied in 1957 [D-14] and applied in 1969 [D-15] to describe the impact of jet aircraft ejection seats to the human body.

It was reported to be a simple model that was well validated for the risk of spinal injury based on ejection seat data and able to account for shock pulse duration dependency [D-16]. The model, called the Dynamic Response Index (DRI), computes the maximum relative displacement of the lumped mass model assuming a natural frequency of 8.4 Hz and 22.4% damping. The maximum relative displacement is determined by solving equation D-1. Since it is a single degree of freedom model, as shown in Figure D-1, it is identical to an SRS calculation for 8.4 Hz and 22.4% damping. These values represent the natural frequency and damping ratio of interest applicable for SRS mitigation ratio calculations.

The DRI has been used previously as a performance measure to evaluate shock isolation seat performance during drop testing [D-17]. It has also been used in numerous studies of the effects of different types of single shock pulses on seated humans. It is specified by the International Maritime Organization as the criterion for evaluating spinal force and seat occupant safety during ship lifeboat drop tests [D-18]. It is specified by the North Atlantic Treaty Organization (NATO) as the criterion for evaluating the risk of spinal injury to seat occupants in armored vehicles [D-19, D-20]. It has been used as a shock isolation seat design criterion for individual severe wave impacts in a high speed craft [D-21], and it has been used to quantify the severity of different individual wave impacts recorded during high speed craft tests [D-22].

The DRI is the only metric currently available that has been used to investigate single impact shock effects on seated humans that is able to quantify the severity of different shock pulses that have different shapes, peak amplitudes, jerk values, and pulse durations. As a relative measure of impact severity (i.e., especially when used in a mathematical ratio) the SRS mitigation ratio provides a consistent mathematical tool for determining the severity of a seat cushion acceleration pulse compared to the deck input acceleration pulse. For this application the SRS mitigation ratio is not a measure of the potential for adverse health effects. It is rather a measure of mechanical shock attenuation.

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Appendix E. Acceleration Data Bandwidth

Low-pass Filter Cutoff Frequency

Low-pass filters are used in a process referred to as response mode decomposition to separate rigid body[12] motions from higher frequency motions caused by vibrations at the gauge location [E-1][13]. In high-speed craft the rigid body heave acceleration is proportional to the shock load acting at the gauge location. Based on Fast Fourier Transform analysis of high-speed craft acceleration data (and laboratory drop test data) a 20 Hz low-pass filter is recommended for estimating the rigid body acceleration (i.e., for shock pulses with durations equal to or greater than 100 msec).

There are other methods for evaluating human exposure to whole body vibrations that call for the use of acceleration data with a bandwidth from 1 Hz to 80 Hz. This includes general requirements for evaluating human health, comfort, and perception in seated, standing, and recumbent positions [E-2], in requirements for continuous and shock-induced vibrations in buildings [E-3], and in a method for evaluating adverse health effects on the lumbar spine due to vibrations containing multiple shocks [E-4]. An 80 Hz low-pass filter can be applied before the calculations are completed if acceleration data is recorded with a higher bandwidth.

The following paragraphs explain why an 80 Hz low-pass filter should not be used to determine acceleration shock pulse amplitude (i.e., rigid body amplitude) for single shocks in laboratory testing or in evaluating the shock amplitudes of single impacts in high-speed craft data.

Figure E-1 shows vertical acceleration data recorded on the deck of a craft during a high speed head-sea run in rough seas. The location on the gauge on the deck recorded both rigid body heave motion and deck plate vibrations excited by the wave impact. In this particular record the dominant vibration frequency is 26 Hz. The black curve is the unfiltered acceleration and the red curve is the 80 Hz low-pass filtered acceleration. The unfiltered deck data is shifted 5 milliseconds (msec) to the right so that both curves could be seen.

The 80 Hz low-pass filtered curve from Figure E-1 is compared in Figure E-2 with the 20 Hz low-pass filtered acceleration. In the 20 Hz data (i.e., the black curve) a small remnant of the 26 Hz vibration signal still remains because of its close proximity to the 20 Hz cutoff frequency. The peak acceleration of the 20 Hz acceleration is a better estimate of the rigid body acceleration at the gage location than the 80 Hz data because the 80 Hz signal contains both rigid body acceleration amplitude plus the majority of the 26 Hz vibration amplitude. The vibration signal is a measure of a local shock response motion. The rigid body signal (i.e., 20 Hz low-pass filtered) at the base of a seat is an estimate of the shock severity input (i.e., input shock force).

[12] Rigid body motions are sometimes referred to as whole body motions.
[13] Vibration amplitude and frequency varies during testing based on the exact location of the gauge.

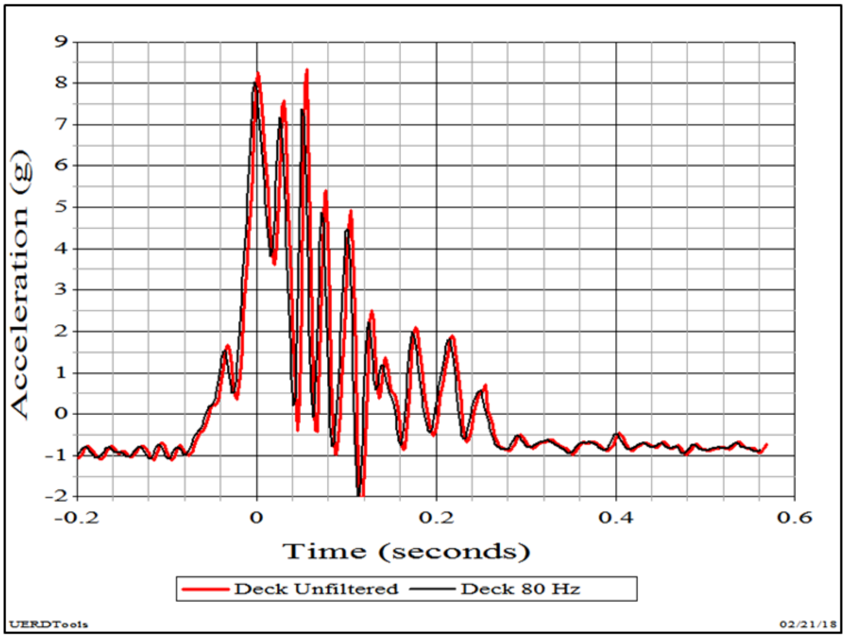

Figure E-1. Unfiltered and 80 Hz Low-pass Filtered Wave Impact Acceleration Data

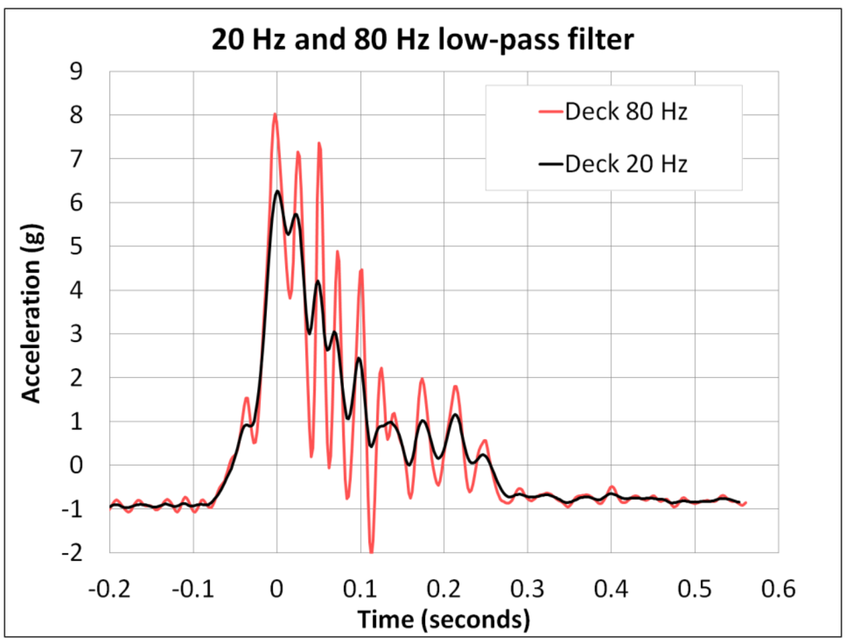

Figure E-2. 80 Hz and 20 Hz Low-pass Filtered Acceleration Data

In Reference E-1 the severity of each seat laboratory test is determined by comparing the 20 Hz low-pass filtered acceleration pulse with the allowable acceleration envelopes. For example, Figure E-3 shows the 20 Hz and 80 Hz data from Figure E-2 compared with the allowable envelopes for a level 4 threshold laboratory test (i.e., nominal 6 g peak acceleration).

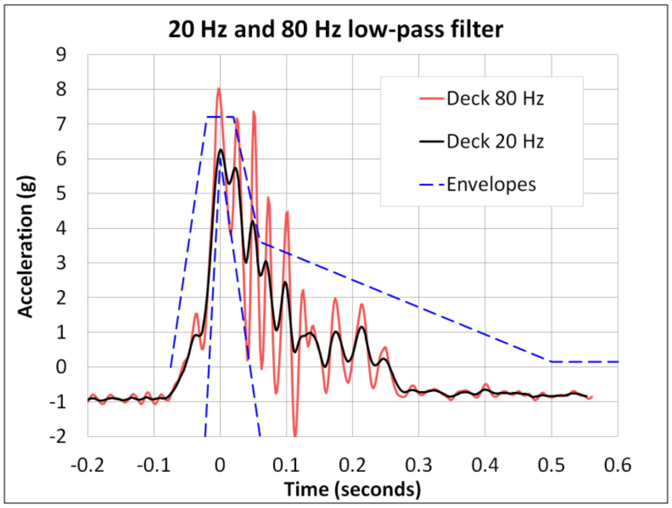

Figure E-3. Allowable Envelopes with 20 Hz and 80 Hz Low-pass Filtered Data

The data in Figure E-3 shows that the rigid body acceleration of the deck input shock pulse (i.e., 20 Hz data) falls within the allowable envelopes shown by the blue curves. This shock pulse is therefore an acceptable level 4 threshold shock pulse (i.e., nominal 6g rigid body amplitude). The peak acceleration of the 80 Hz acceleration data is 8.02 g. If it had been used to compare with allowable envelopes it would have been considered a level 5 threshold severity test (i.e., nominal 8 g peak acceleration), which is not correct.

The next example presented in Figure E-4 looks at the rigid body amplitude of all wave impacts recorded during a craft high-speed run. It shows the effect of different cutoff frequencies on the amplitude of all wave impact accelerations recorded during a 10-minute run at the base of a seat while a high-speed craft was operating in rough seas. The peak acceleration of each wave impact was extracted using the StandardG algorithm [E-5]. StandardG was run using 20 Hz, 80 Hz, and 500 Hz low-pass filters. The 500 Hz filter yields peak accelerations that are approximately equal to unfiltered peak accelerations for this run. The low-pass filtered peak accelerations were tabulated largest to smallest and then plotted in the figure as X-Y pairs. The figure shows how the vibration content in the record increases the peak accelerations above the rigid body acceleration amplitudes (i.e., estimated using 20 Hz low-pass filter). For the 500 Hz low-pass filter (i.e., approximately unfiltered) the peak amplitudes are on the order of 2 times to 3.6 times larger than the 20 Hz values. The 80 Hz data for this record is on the order of 1.42 times larger than rigid body peak accelerations up to 5 g, and 1.8 times more than the two largest rigid body peaks of 5.57 g and 6.28 g.

Unfiltered data or data filtered with an 80 Hz low-pass filter should not be used when evaluating shock pulse peak-G amplitudes or for comparing recorded shock pulses with allowable envelopes. This includes evaluating whether or not an acceleration shock pulse fits within the allowable acceleration envelopes for either laboratory test data or craft data.

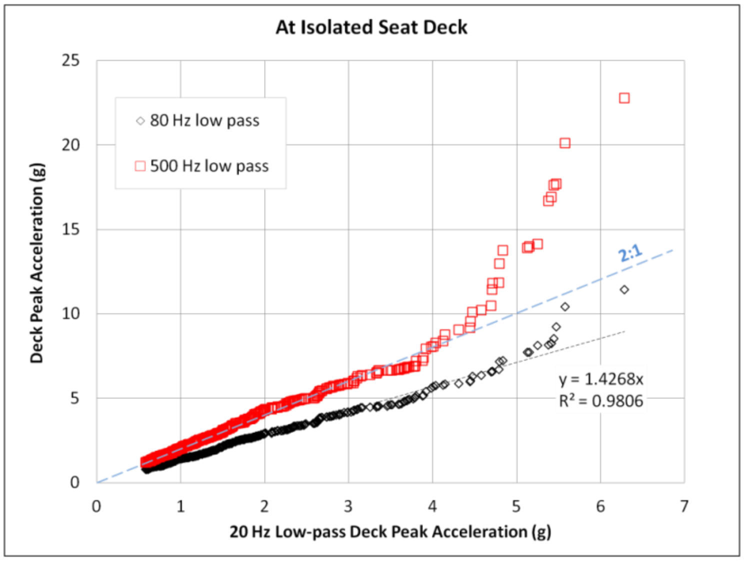

Figure E-4. Comparison of 20 Hz, 80 Hz, and 500 Hz Kaiser Low-Pass Filtered Data

Unfiltered vs. Low-pass Filtered Acceleration Data

Figure E-5 shows another example of why response mode decomposition is critically important when evaluating mechanical shock mitigation. The curve on the left is the unfiltered deck and seat cushion data for one wave impact recorded in a high-speed craft operating in rough seas. The black curve is the unfiltered vertical deck acceleration recorded at the base of the shock isolated seat. Its content includes both rigid body heave acceleration and local deck vibrations (hence the fuzzy appearance of the record). The red curve on the left is the unfiltered vertical acceleration recorded on the seat cushion (i.e., below the seat payload weight). The smooth appearance of the red curve on the left demonstrates how the seat spring-damper-cushion assembly removes most of the vibration content observed in the deck acceleration.

The unfiltered data shown on the left side of Figure E-5 was then subjected to a 20 Hz low-pass filter to estimate the rigid body content in the two signals. The low-pass filtered signals are shown on the right side of the figure. The blue curve is the estimated rigid body heave acceleration at the deck location and the red curve is the estimated heave acceleration on the seat cushion. The red filtered curve on the right is almost identical to the red unfiltered curve on the left, illustrating that the red unfiltered seat cushion data on the left has little or no vibration content. The red curve on the left is therefore primarily attributed to rigid body heave motion. In the left plot the large reduction in the high frequency deck peak acceleration (black curve, about 7 g) compared to the cushion peak (red curve about 3.6 g), and the removal of the “fuzzy” character of the black deck data (compared to the smooth red cushion data) is an example of vibration mitigation (e.g., 3.6g / 7 g is a 48% reduction due to vibration attenuation).

The rigid body acceleration curves on the right in Figure E-5 are used for evaluating shock mitigation. Effective shock mitigation occurs when the rigid-body peak acceleration (i.e., the blue curve on the right in Figure E-5) is reduced in amplitude, the rate of acceleration application (i.e., jerk) is reduced, and the duration of the shock pulse is increased.

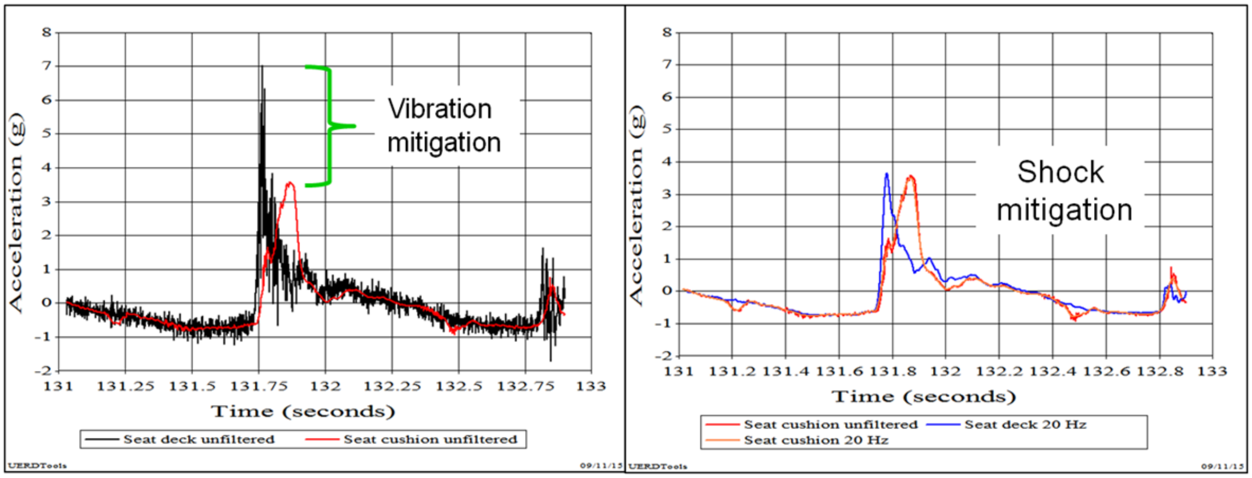

Figure. E-5. Examples of Vibration Mitigation and Shock Mitigation

SRS Mitigation Ratios

The existence of the International Standards Organization (ISO) requirements [E2 – E4] for the evaluation of human exposure to whole body vibrations introduces the question whether or not an 80 Hz low-pass filter could be used when computing SRS mitigation ratios when shock isolation seats are subjected to single impacts (either an individual wave impact in a craft or a laboratory shock test). To address this question acceleration data recorded on a shock isolation seat during high-speed trials was analyzed. The results are tabulated in Table E-1. It lists computed MR_{SRS} values for twenty-four individual wave impacts using acceleration data recorded on the deck, the seat pan, and the seat cushion (i.e., pad). The first column tabulates the wave slam number which is the approximate time in seconds that the wave impact occurred in the record. Eight impacts with 20 Hz MR_{SRS} values were selected in each of the following categories: MR_{SRS} less than 1.0, MR_{SRS} = 1.0, and MR_{SRS} >1.0. The values in red correspond to the impacts when seat bottom impact occurred. The deck, pan, and pad accelerations were also low-pass filtered using an 80 Hz cutoff frequency and mitigation ratios were computed. They are listed in Table E-1 along with the MR_{SRS} values obtained using the 20 Hz low-pass filter.

The values listed in the table are compared in Figure E-6. The plot shows that the MR_{SRS} values for the 80 Hz data are within -5% to +3% of the 20 Hz values.

Figure E-6 shows that using the 80 Hz low-pass filtered data would be satisfactory for evaluating the mitigation performance of shock isolation seats. For MR < 1 the 80 Hz values are roughly 5% lower (more mitigation) than 20 Hz values, and for MR > 1 the 80 Hz values are up to 3% higher (more amplification) than 20 Hz values. The reason the 20 Hz and 80 Hz MR values are similar is because the MR_{SRS} was computed using an 8 Hz lumped mass model with 22% damping[14]. In the calculation the presence of the roughly 25 Hz to 80 Hz vibration content in the deck input acceleration is isolated by the 8 Hz mathematical model, so the relative displacement response across the 8 Hz spring is roughly the same (within -5% to +3%) as the response to the 20 Hz input acceleration.

[14] This is the lumped mass model used in the Dynamic Response Index (DRI) calculation.

Table E1. 80 Hz and 20 Hz SRS Mitigation Ratios

Slam Number	Fixed Seat SRS Mitigation Ratio				Isolated Seat SRS Mitigation Ratio			
	Pan 80 Hz	Pan 20 Hz	Pad 80 Hz	Pad 20 Hz	Pan 80 Hz	Pan 20 Hz	Pad 80 Hz	Pad 20 Hz
2	1.01	1.01	1.62	1.59	0.97	1.00	0.83	0.85
17	1.17	1.17	1.35	1.37	1.51	1.51	1.51	1.50
26B	1.29	1.28	1.73	1.68	1.03	1.04	1.35	1.32
27	1.18	1.18	1.51	1.48	0.66	0.69	0.64	0.67
31	1.17	1.16	1.62	1.59	0.54	0.56	0.65	0.67
32	1.09	1.08	1.55	1.51	0.81	0.83	0.82	0.85
39B	1.20	1.18	1.67	1.62	1.00	1.03	1.17	1.17
55	1.11	1.12	1.49	1.47	0.92	0.93	1.02	1.03
81B	1.07	1.08	1.29	1.30	1.39	1.38	2.04	1.98
85	1.11	1.11	1.59	1.55	0.73	0.76	0.62	0.65
112	1.19	1.18	1.59	1.55	0.85	0.87	0.81	0.83
124	1.22	1.22	1.72	1.67	0.87	0.88	0.86	0.88
145	1.07	1.07	1.54	1.50	1.31	1.31	1.49	1.49
151	1.19	1.18	1.72	1.67	0.90	0.92	0.86	0.88
159	1.18	1.18	1.68	1.64	0.87	0.89	0.85	0.87
194B	1.22	1.21	1.65	1.60	1.02	1.03	1.05	1.06
237	1.06	1.08	1.69	1.66	1.00	1.03	0.93	0.96
238	1.27	1.25	1.69	1.64	1.33	1.35	1.30	1.32
246	1.15	1.16	1.58	1.55	0.88	0.89	0.93	0.94
335	1.13	1.11	1.33	1.32	0.68	0.69	0.58	0.62
403	1.06	1.06	1.77	1.73	0.75	0.78	0.63	0.66
434	1.11	1.10	1.61	1.58	0.70	0.72	0.65	0.67
482	1.22	1.22	1.51	1.51	1.32	1.31	1.44	1.44
528	1.10	1.11	1.65	1.62	1.07	1.08	1.09	1.10

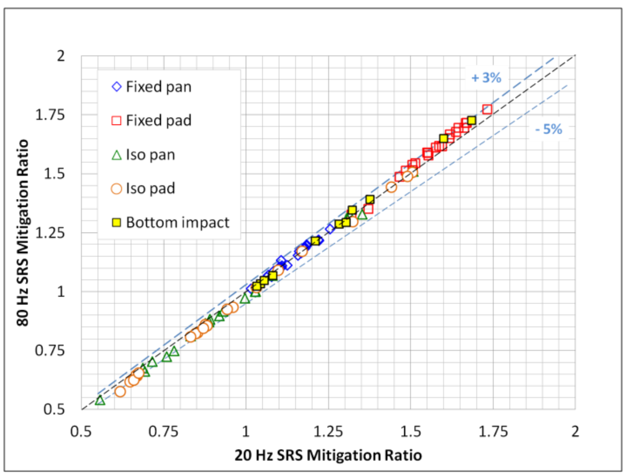

Figure E-6. 80 Hz vs. 20 Hz SRS Mitigation Ratios

This is illustrated further in Figure E-7. The same color scheme is used for the input and predicted response curves. The lower plot shows the unfiltered deck, pan, and pad accelerations that were used as inputs into the 8 Hz lumped-mass model. The curves in the upper plot are the predicted acceleration responses of the mass in the 8-Hz 22% damped model for each of the unfiltered inputs.

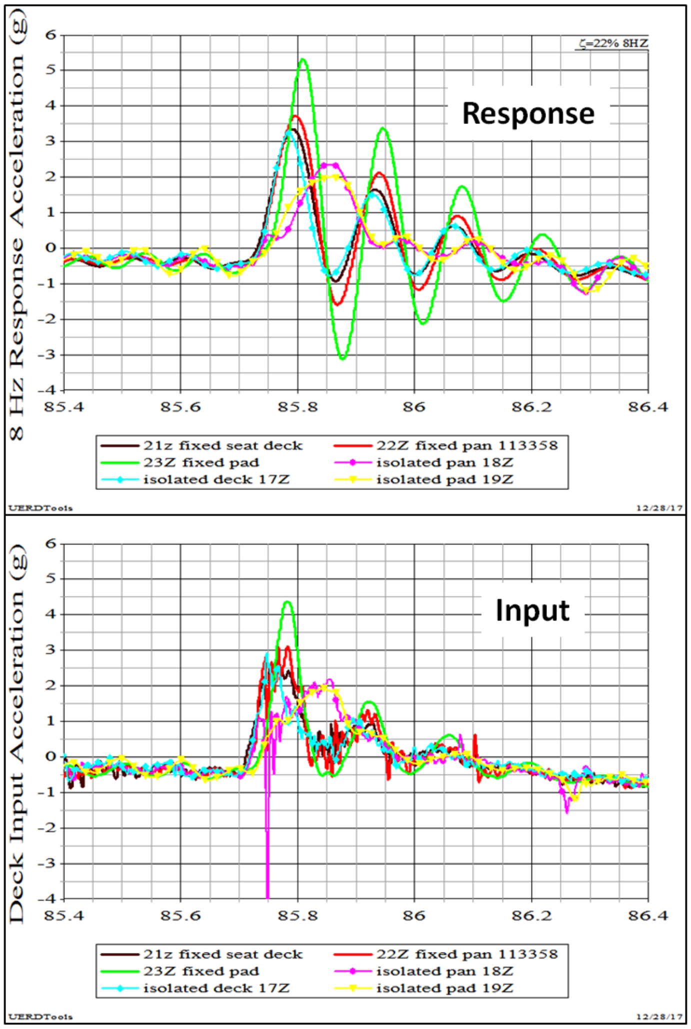

Figure E-7. 80 Hz Filtered Deck Inputs and 8 Hz Spring, 22% Damped Responses

Visual inspection of the predicted response curves in the upper plot in Figure E-7 shows that the vibration content in the input deck and in the pan signals has been removed by the high compliance (i.e., low stiffness) of the 8-Hz lumped-mass model. This is confirmed in Figure E-8 by the Fast Fourier Transform (FFT) plots of each time history plot in Figure E-7. The high frequency content up to roughly 110 Hz, observed in the lower plot, is not present in the upper plot. High compliance systems (i.e., low frequency) do not transmit high-frequency low-displacement acceleration signals (i.e., vibrations), but they do transmit low-frequency large-displacement acceleration signals (e.g., shock pulses).

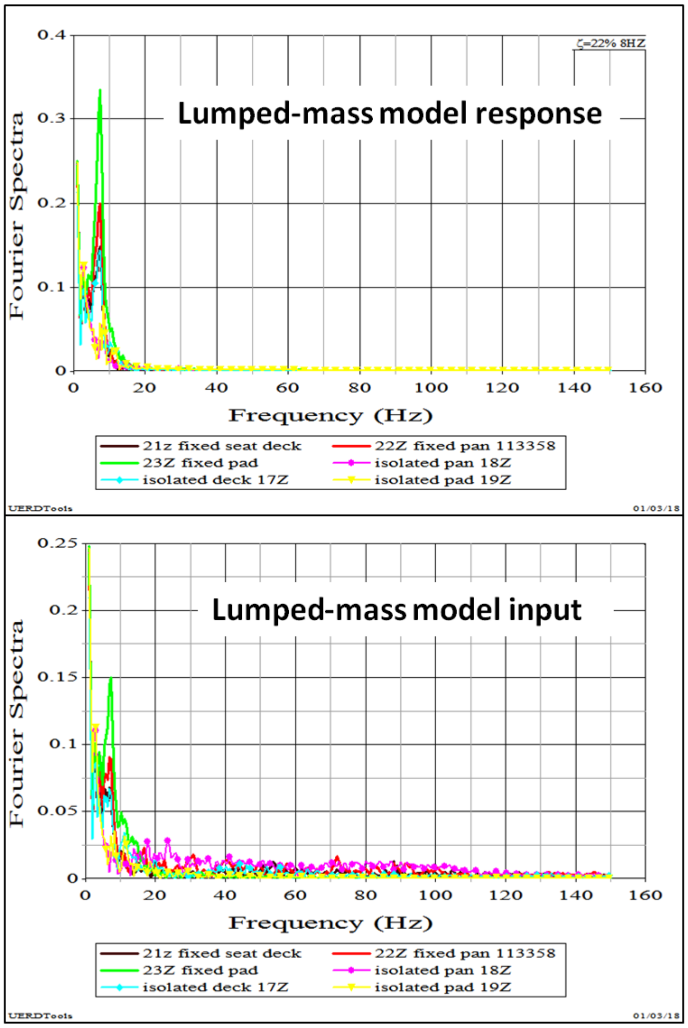

Figure E-8. FFT of 8 Hz Model Input and Response

The implication of the results shown in Figures E-6, E-7, and E-8 is that the roughly 20 Hz to 80 Hz vibration signals in the deck and pan input accelerations are not relevant for evaluating the response of high compliance systems (i.e., low stiffness less than or equal to 8 Hz). The 20 Hz to 80 Hz vibration signals in the deck and pan input accelerations are therefore not relevant for evaluating the mitigation performance of marine shock isolation seats to single impacts. The 8-Hz lumped-mass model isolates the vibration content observed in the deck and pan accelerations.

In the real world Figure E-9 shows that the seat cushion isolates the vibration content in the deck and the pan acceleration records. The figure shows FFT plots of three unfiltered 10-minute long acceleration records recorded on the deck at the base of a shock isolation seat, on the isolated seat pan, and on the isolated seat cushion. The curves show how the seat cushion begins mitigating the deck and pan vibration signals at about 12 Hz, and at 40 Hz and higher most of the vibration content is significantly attenuated.

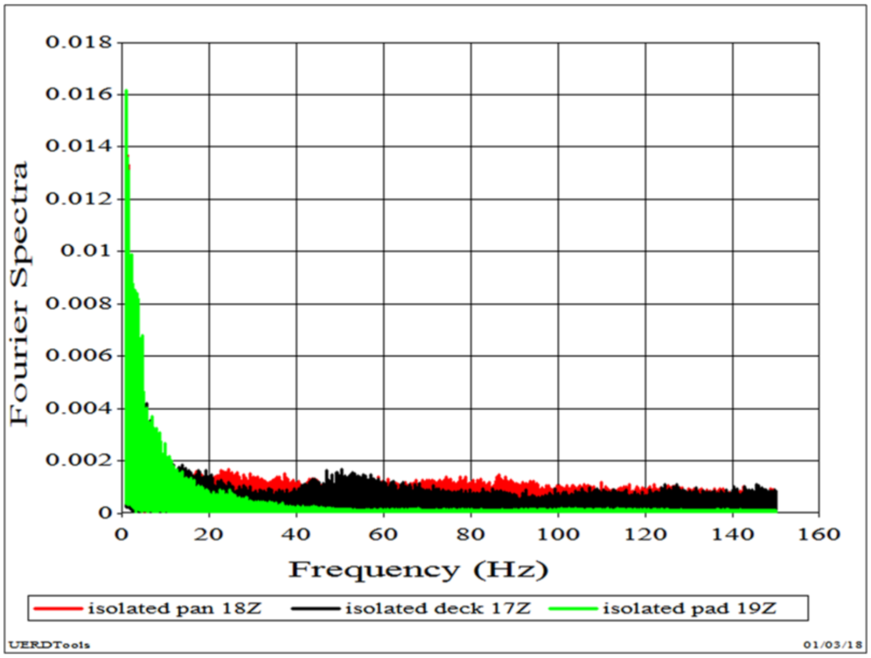

Figure E-9. FFT Plots of Deck, Pan, and Pad Acceleration Data

Even though the 20 Hz to 80 Hz vibration content is not relevant for an 8 Hz 22% damped lumped-mass comparison model, the computational results show that it would be acceptable to specify the use of 80 Hz bandwidth data for MR_{SRS} calculations because the results are within roughly 5 percent of values using 20 Hz low-pass filtered data. But unfiltered data or data filtered with an 80 Hz low-pass filter should not be used when evaluating shock pulse peak-G amplitudes or for comparing recorded shock pulses with allowable envelopes. This includes evaluating whether or not an acceleration shock pulse fits within the allowable acceleration envelopes for either laboratory test data or craft data.

Mass Effect

Understanding how relative motions between large and small masses affect each other is another helpful way to understand the effect of deck vibrations. This involves the topic of mass participation in dynamic environments. The participating mass associated with a structural motion determines the transfer of shock force during motions caused by impact. This is explained further in the next paragraph by considering the mass of a vibrating deck in a craft, a 200-lb mass positioned on the deck, and the mass of the entire craft at a specific cross-section.

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 was then installed next to the accelerometer location the deck vibrations would not drive the mass to vibrate at the same frequency or amplitude as the previous scenario. This is because the 200-lb mass is large relative to the mass of the contiguous deck plating that is vibrating. The presence of the 200-lb mass would change the deck vibration pattern, amplitude, and frequency content, but it would not change the wave impact shock amplitude or pulse duration of the rigid body heave acceleration on the deck next to the mass. This is because the 200-pound mass and the mass of the contiguous deck plate is relatively small compared to the mass of the craft at the cross-section where the accelerometer is positioned. The 200-lb mass is not large enough to change the heave motion of the craft. Vibrations in craft do not transmit wave impact shock force to equipment or people. Rigid body heave motions (e.g., the rapid change in heave acceleration) transmit shock load.

Appendix E Reference

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