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Marine Vehicle Ride Quality: A State-of-the-Art Assessment

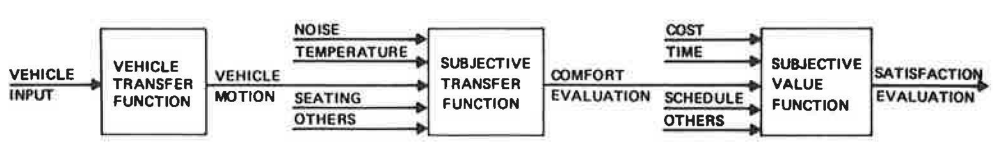
D.R. STARK

The riding characteristics of marine vehicles are affected mostly by low-frequency random accelerations. Specific examples of acceleration spectra illustrate the nature and variability of the motions encountered, and the human response data base is discussed in light of the riding characteristics shown. Ride quality evaluation methods are identified and correlated with subjective ride scales. Standard evaluation methodology that uses frequency weighting is recommended.

In a broad sense, vehicular ride quality technology encompasses human response/human tolerance to the vehicle environment, vehicle responses to the external environment as well as vehicle-induced environments, and the subjective value function that passengers and operators place on the overall ride. (See Figure 1.)

Marine vehicles, due to the nature of the sea environment in which they operate, have riding properties dominated by low-frequency random motions, with most of the energy occurring below 1 Hz. Although marine vehicles subject the passengers to the other physical environments, noise, temperature, seating, leg space, and such, these environmental factors are generally less important to the state of the art than the motion environment. Also, solutions for these environmental influences derive directly from a broader technology base. This assessment of the

Figure 1. Overall ride-quality system.



state of the art in marine vehicle ride quality focuses on the motion and acceleration aspects of ride quality. Included in the assessment are the human subjective response data base, the vehicular prediction and design data base, and some elements of the value function data base.

Motions and acceleration within the sea environment can only be characterized in statistical terms, and this characterization is further compounded by the long-term variability of the sea environment and by variations in ship operating conditions such as speed and heading. Thus, a comprehensive statistical approach to the evaluation of ship riding properties becomes an absolute necessity.

CHARACTERIZATION OF MARINE VEHICLE MOTIONS

The ride quality of advanced, high-speed marine vehicles is dominated for the most part by vertical and horizontal accelerations. Angular motions can also be appreciable, but their importance is frequently shown to be secondary to accelerations. In conventional ships, roll and pitch motion frequently takes on more importance, but the accelerations remain as dominant ride quality factors.

Vehicle accelerations and motions are random in nature because they are introduced by the random seaway in which the ship operates. The frequency range of the seaway-induced accelerations and motions is essentially bounded at the lower end by zero frequency and at the upper end by 5 Hz, with the major portion of the energy occurring below 1 Hz.

Figure 2. Typical vertical acceleration spectra for five headings relative to the sea.

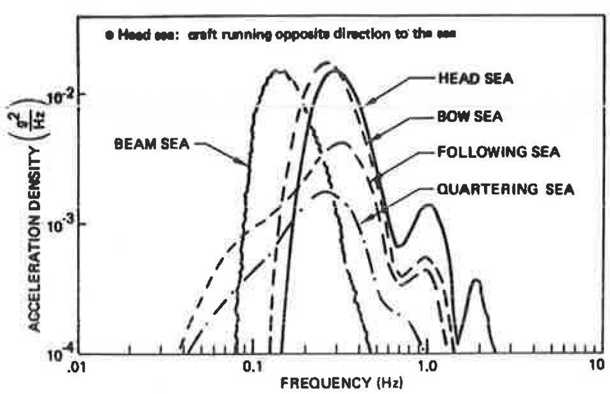


Figure 2 shows typical vertical acceleration spectra for a hydrofoil ship operating in a specific sea condition, Figure 3 shows a discrete spectrum for an air cushion vehicle (surface effects ship), and Figure 4 shows an acceleration spectrum for a conventional hull ship. Ship acceleration characteristics tend to vary with ship heading, as shown in Figure 2, and with speed (not shown).

The sea environment is also constantly changing in severity; hence, the acceleration responses differ from day to day and from locale to locale. Figure 5 shows typical long-term (annual) distributions of rms ship accelerations that will occur for operation in a given area; in this case, data are for a hydrofoil ship operating in the North Sea. Lateral accelerations, longitudinal accelerations, and angular motions, but with differing amplitudes and frequency content, are also related to the changing acceleration responses.

The vehicle riding properties can be seriously degraded by discrete, discontinuous, but repeated, acceleration and motion transients. In all ships, non-linear transients occur due to bow slamming. Hydrofoils and air cushion vehicles encounter similar discontinuous transients when the supporting element comes clear of the water surface and loses lift.

RIDE QUALITY EVALUATION

Human Response Data Base

Basic research must provide the fundamental response data on which ride quality criteria and evaluation standards can be based. It is well known that the human response data base for low-frequency random motions is lacking in many areas. Nevertheless, there are data that can guide the evaluation of ride quality. The more important elements of the data base are discussed in the following sections.

In regard to vertical acceleration frequency response data, the general format of the International Standards Organization ISO 2631 prevails in the frequency range above 1 Hz. (See Figure 6). The Shoenberger (1) and Miwa (2) equal-sensation studies provide data to extrapolate the human response below 1 Hz, whereas the O'Hanlon and McCauley (3) motion-sickness data provide a solid base for response in a narrow range of frequencies around 0.2 Hz.

Relative to horizontal acceleration frequency response data (see Figure 7), the ISO data above 1 Hz

Figure 3. Power spectral density of a surface effects ship.

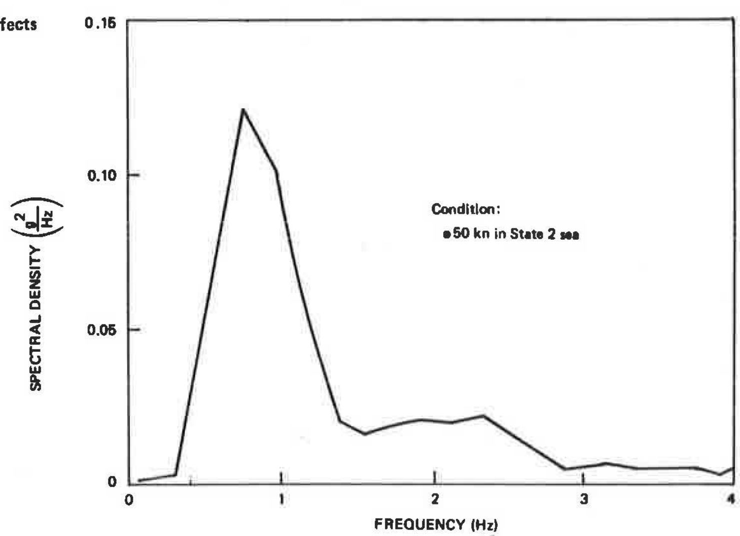


Figure 4. Typical vertical acceleration spectrum for a patrol boat.

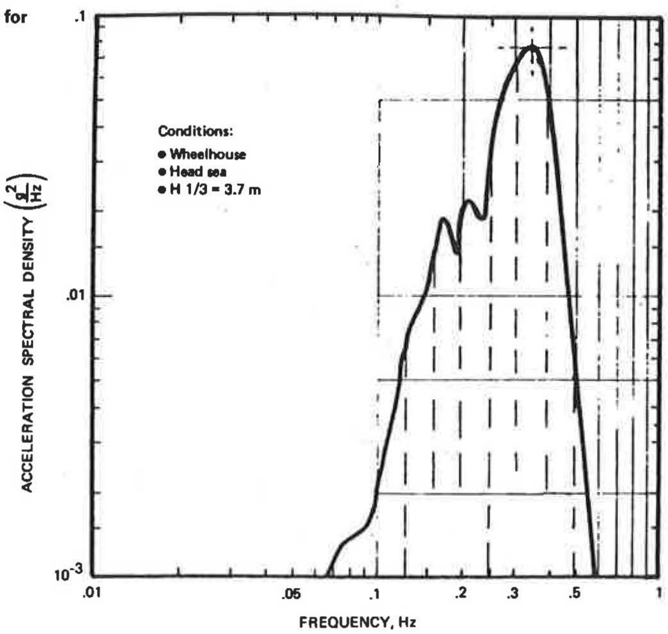


Figure 5. Predicted long-term acceleration distributions in the North Sea for a specific Hydrofoil ship.

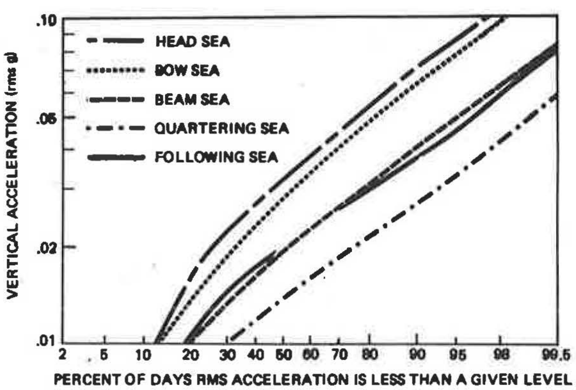


Figure 6. Data base for human response to vertical accelerations.

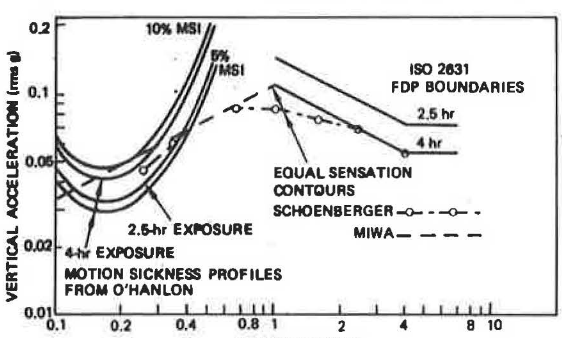
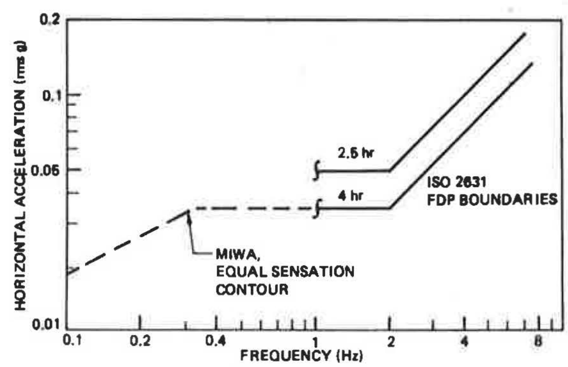


Figure 7. Data base for human response to horizontal accelerations.



and Miwa's equal-sensation data below 1 Hz provide the basic responses as a function of frequency. Shoenberger's equal-sensation studies did not include horizontal acceleration. Neither did the O'Hanlon and McCauley studies on motion sickness. Hence, for horizontal accelerations the extrapolation into frequencies below 1 Hz is primarily based on Miwa's limited work.

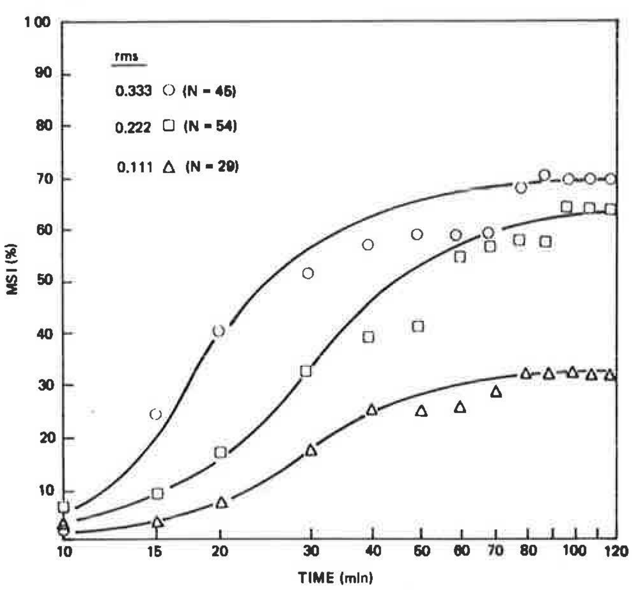
Angular motions--pitch, roll, and yaw--have an even shallower data base. Both Shoenberger (4) and Leatherwood and others (5) have explored responses to roll motions covering a range of frequencies from 1 to 10 Hz. Shoenberger's data would indicate that response is proportional to the first derivative of roll angular acceleration. Leatherwood's data show the response to be essentially proportional to roll acceleration over a range of frequencies tested (1-4 Hz), and Jacobson and Richards' (6) comfort formulas use roll rate as the measurement. Frequency response below 1 Hz has not been explored in any of these studies; however, McCauley and others (7) have explored roll and pitch motions in combination with vertical accelerations and found no significant additional contribution to motion sickness by the addition of angular motions. Thus the primary cause of motion sickness is vertical accelerations. In light of the above, it seems clear that with regard to roll motion the data base is full of conflicts and of little use to the marine community. Responses to pitch and yaw motions seem to be in similar disarray.

Much of the data base for low-frequency random motions is derived from sinusoidal motions. Unfortunately, in the real world one seldom or never encounters distinct sinusoidal motion. Hence, we must find a rational method of evaluating responses to random motions over a broad frequency band. For this we must rely on extrapolation of results in higher frequency studies. Dempsey and others (8) and Schoenberger (9) have conducted random acceleration and motion studies on their passenger ride simulators. From these studies, we can derive some relationships between random motions and sinusoidal motions in the frequency range above 1 Hz. Dempsey showed general agreement with the frequency trends of the ISO curves for sinusoidal vertical vibration. Furthermore, his studies with random vertical vibrations again showed general agreement with the frequency trends of the ISO standard over a range of center frequencies from 2 to 13 Hz.

Shoenberger, working with one-third octave random accelerations, also verified the general shape of the ISO curves above 1 Hz. A perhaps more important element is Dempsey's finding that for the same level of rms acceleration the test subjects found random vibration more uncomfortable than sinusoidal vibration. For lateral accelerations, Dempsey's studies show considerable variation from the ISO curves, both in frequency trends and magnitudes. This conflict leaves the evaluation of lateral accelerations somewhat up in the air. In truth, the data base for all motions other than vertical is somewhat questionable, even in the high-frequency regime. Therefore, considerably more research seems appropriate in these other areas of motion.

Until recently it was generally understood that discomfort or dissatisfaction with the ride increased with time, as indicated by ISO 2631. Clark (10), Richards (11), and Clevenson and others (12) independently have shown that discomfort or comfort ratings do not change appreciably with time. Clevenson's studies, in fact, showed a slight tendency toward less discomfort with time. Hence with regard to comfort and passenger acceptance, time (up to 6 h) is not considered a significant factor. With regard to motion sickness, however, a definite time dependency has been shown by the O'Hanlon and McCauley studies. (See Figure 8.)

Figure 8. Motion sickness incidence (cumulative percent emesis) as a function of time for three independent groups at one frequency (0.25 Hz) and three levels of acceleration.



Discrete Transients

A potentially important element of the ride quality characterization has been thus far neglected: the response and acceptance of the passenger and/or operator to discontinuous but repeated motion and acceleration transients. Most forms of transportation encounter such discontinuous motion transients, but to varying degrees. For example, the airline encounters a one-time-only shock transient on landing. In contrast, rail vehicles encounter frequent lateral and vertical transients due to rail discontinuities. Longitudinal and lateral transients are often introduced into surface transportation by operator inputs. Pavement or roadway discontinuities such as chuck holes and transition joints introduce similar non-continuous motion transients into the highway vehicle. All types of ships operating in a seaway experience severe discontinuous motion and acceleration transients associated with bow slamming. Some ships, such as hydrofoils and surface effect ships, encounter similar transients when the supporting element comes free of the water surface and loses lift. Automated transportation systems encounter similar motion transients due to guideway irregularities, breaking and speed control transients, and non-linear suspension systems. The response of a magnetically levitated vehicle when the suspension system bottoms out is an example of non-linear suspension transients.

Current measurements and evaluations of vehicle ride quality tend to be insensitive to discrete transients of the nature described. For example, a series of acceleration transients with peak amplitudes each in excess of 1 *g* might not raise the low level rms value, or the one-third octave rms value by as much as 1 percent. This is because the discrete transients, when averaged out over the total time period, do not represent a significant addition to the dynamic energy of the system. That is not, however, to say that they are insignificant to the ride quality assessment. Allen (13,14) and others have discussed this problem in some detail.

A comprehensive program is needed to develop and quantify human response and acceptance guidelines for discrete but repeated motion transients. The program should develop methodology and measurements for assessing the effects of transients on ride quality when taken by themselves and when taken in concert with other environmental factors such as vibration and noise. The program must develop measurement and evaluation methodology for transients that occur at random intervals, have randomly varying magnitudes, and random shapes and pulse widths.

EVALUATION METHODOLOGY

Background

When evaluating real-life vehicle motions or design alternatives, one seldom or never finds specific data in the literature directly applicable to the case at hand. Hence, it is necessary to develop standardized evaluation methods that cover as wide a range of the basic human response data base as possible.

The one-third octave evaluation format recommended by ISO 2631 has been applied extensively to marine vehicles with generally poor results. The frequency weighting method identified by ISO has also been used with generally good results by the hydrofoil community. Other methods have also been put to use, including the raw rms value, average frequency, and 1 octave evaluation.

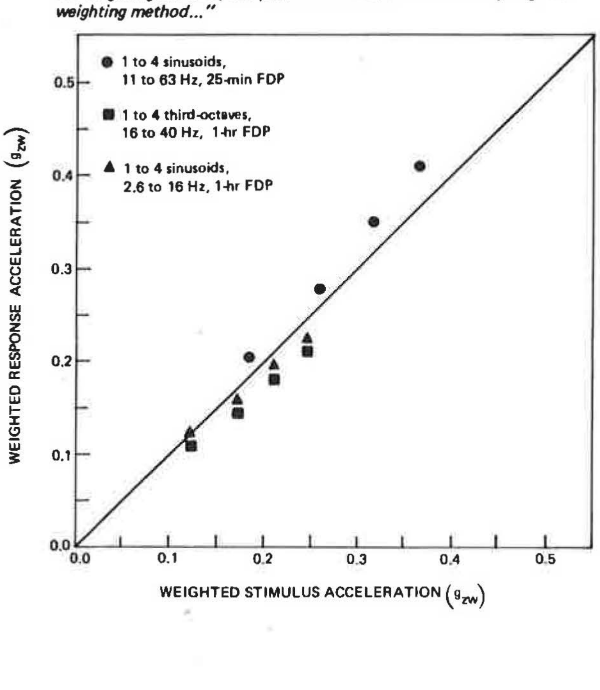
In performance of evaluations of vehicles of the same general type ISO now recommends a weight rms approach over the one-third octave band method. By now the separate one-third octave evaluation method recommended by ISO is currently in question where broad-band accelerations are concerned. Dempsey's data show no significant difference between 2 Hz and 5 Hz band-width data with the same overall rms acceleration levels, thereby supporting the "weighting method". Furthermore, Shoenberger's recent studies (9) clearly support the overall frequency weighting method (see Figure 9).

Fundamentally, the one-third octave format is unsuitable for evaluating marine vehicle ride quality in that its use can lead to erroneous conclusions about the acceptability of the vehicle riding qualities. Furthermore, when such erroneous conclusions are reached, they will always be more optimistic than the real-life situation. The basic problem is that the one-third octave approach divides the total vehicle accelerations into small frequency bands and the rms value in each band is evaluated separately against either a criterion or limit. There is no provision for consideration of the overall accelerations experienced by an occupant. Thus, where the acceleration spectrum is wide band relative to the one-third octave, the overall acceleration, and hence the overall ride quality, is ignored in the evaluation.

Figure 10 illustrates the differences between the one-third-octave approach and the overall frequency weighting methods. The one-third-octave accelerations are shown compared with the ISO curves and the O'Hanlon and McCauley motion sickness curves. Using the one-third octave method it appears that the ride satisfies the 8-h FDP (Fatigue Decreased Proficiency boundary) levels of ISO, and would cause less than 10 percent of the population to suffer motion sickness for 8 h of exposure. In contrast the frequency weighted rms acceleration level of 0.26 g rms (referenced to 1 Hz value) would satisfy only the 25-min FDP level of ISO. Also, when motion sickness is evaluated using the method recommended in Section 5.7, a motion sickness incidence of 10 percent would occur within 25 min as opposed to the apparent 8-h exposure.

Figure 9. Summary of Schoenberger's findings on frequency weighting method.

Shoenberger—1979 "The findings...indicate that the subjective intensity of nonsinusoidal vibration environments is more accurately reflected by the weighting method, and provide evidence in favor of adopting the weighting method..."



Another evaluation method used by Dempsey and others (8) and Leatherwood and others (5) in research and frequently by others for full-scale evaluations employs the rms acceleration and the center frequency. The rms level can be compared with curves such as the ISO boundaries at the measured center frequency. For low-frequency wide band acceleration, we have found this approach undesirable because small amounts of high-frequency energy can shift the center frequency upward with no appreciable change in the rms level. For marine vehicles, this frequency shift tends to move the center frequency closer to the 1-Hz peak in the human response curves, thus making ride quality seem improved by the added high-frequency energy. The idea that ride quality can be improved by adding accelerations at any frequency is categorically rejected.

Figure 11 illustrates this point with an actual ship acceleration characteristic using one-third octave values. The average frequency of the total is 0.9 Hz and the rms value is 0.108 g; the diamond in the middle of Figure 11 illustrates this data point. Interestingly, at 0.9 Hz, there is actually very little energy. Let us then consider the results if the higher frequency hump were removed (by redesign). The removal of the high frequency acceleration energy would only slightly reduce the rms acceleration (0.9 g) while the moving average frequency to 0.35 Hz (as shown by the square data point), thus making the ride appear much worse. Obviously, this evaluation method has a great potential for mischief and is definitely not recommended.

Recommended Method

Frequency weighting of the response combined with the evaluation of the total resultant frequency-weighted rms (FWRMS) value in each axis is offered as the most logical and comprehensive approach. This approach has been used extensively by hydrofoil and advanced marine vehicle work. The U.S. Navy hydrofoil design criteria (15) and the U.S. Air Force report, Military Specification--Flight Control Systems Piloted Aircraft (MIL-F-9490D), both require the use of FWRMS accelerations over the total frequency band of motions. ISO 2631 also identified frequency weighting as an alternative, and then recommended the one-third octave method. Draft Amendment I of ISO 2631 has since reversed that recommendation for comfort performance evaluations. Shoenberger's recent data also strongly support the frequency weighting method.

A separate evaluation of motion sickness is recommended that uses the same basic frequency weighting methods; however, the shape of the frequency weighting curve would be taken as the inverse of O'Hanlon's motion sickness curves.

Mechanics of Frequency Weighting

Frequency weighting is accomplished by multiplying the acceleration at each frequency by an amplitude multiplier with a frequency shape proportioned to the inverse of the human response characteristic. This method is discussed in detail in the ISO document 2631. (See Figure 12.)

Above 1 Hz, the ISO standard asymptotic curves are retained for both horizontal and vertical accelerations. Below 1 Hz, responses follow the general shape of the equal sensation curves developed by Miwa and Shoenberger. The shape of the curves below 0.2 Hz is a compromise. Miwa's equal-sensation data slope downward with decreasing frequency. If we extend the downward slope to zero frequency we arrive at a position where the allowable acceleration is zero. This is counter-intuitive! Motion sickness

Figure 10. Example of third octave analysis, compared with commonly accepted data base.

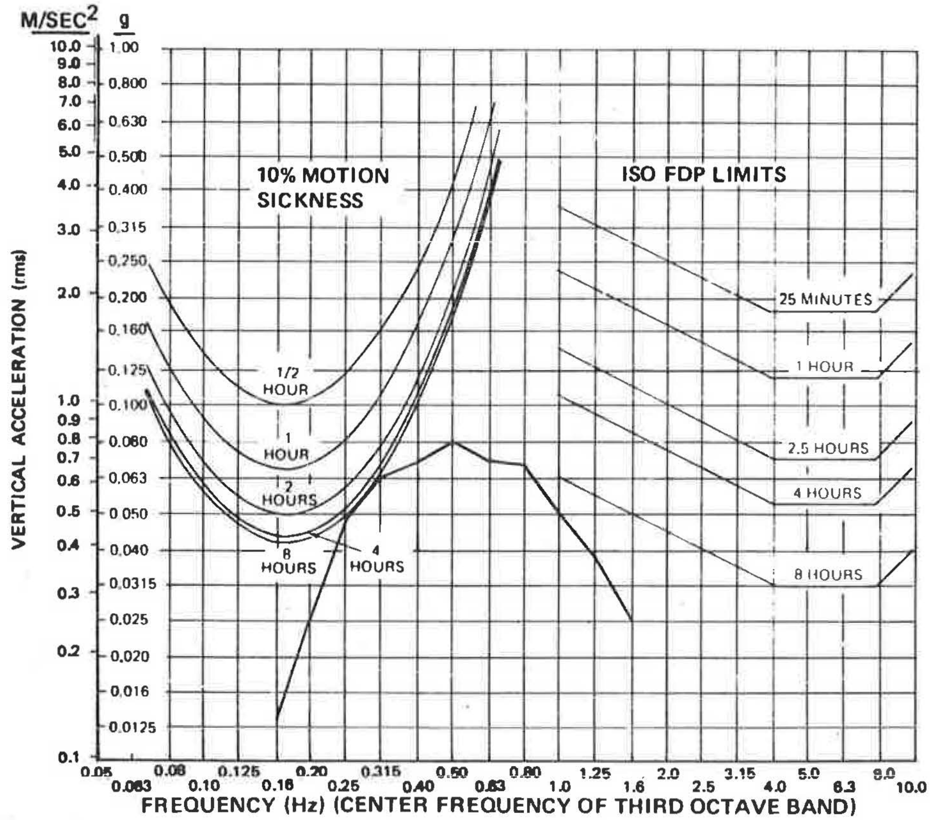


Figure 11. Example of third octave evaluation and evaluation using raw root mean square and average frequency.

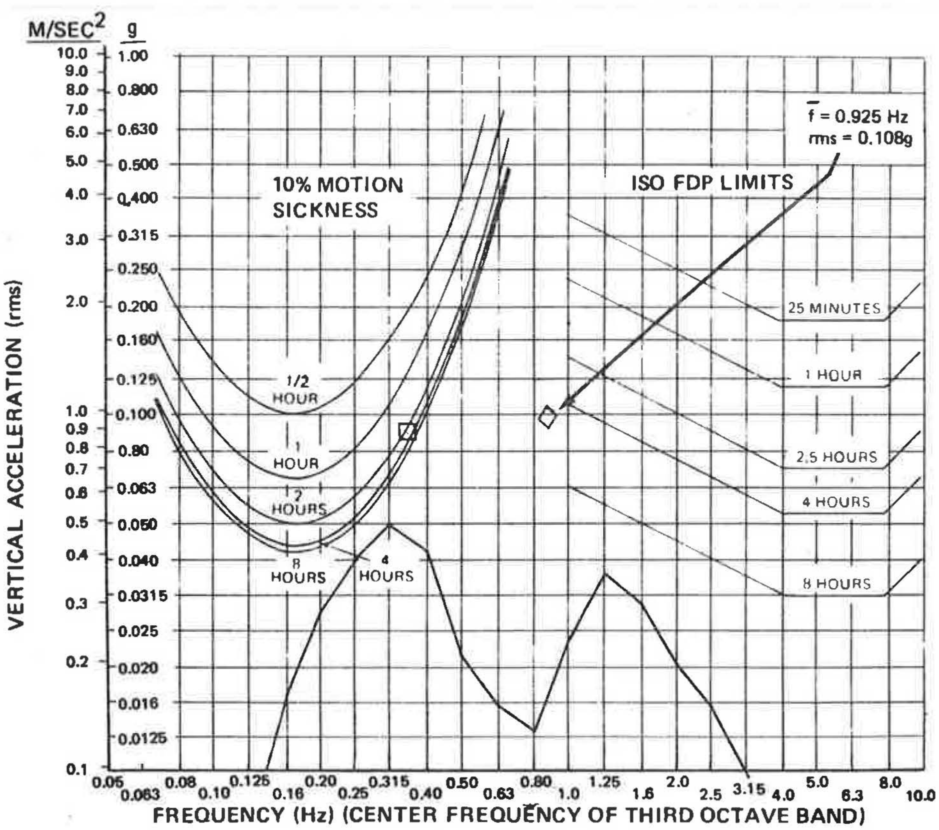


Figure 12. Composite estimates of human frequency response characteristic for a seated man.

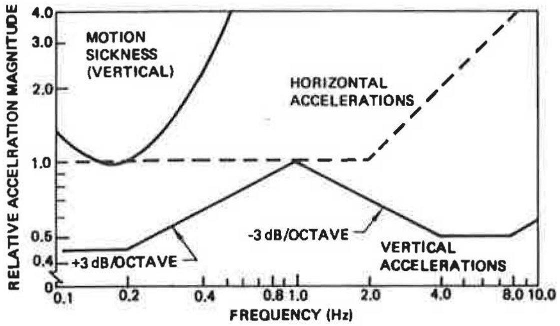


Table 1. Frequency weighting factors.

Frequency (Hz)	Acceleration Weighting Factors		
	Vertical	Horizontal	Motion Sickness
0.05	-	-	0.20
0.063	2.18	1.00	0.42
0.07	2.18	1.00	0.52
0.1	2.18	1.00	0.745
0.125	2.18	1.00	0.924
0.16	2.18	1.00	1.03
0.2	2.18	1.00	1.0
0.25	2.0	1.00	0.86
0.315	1.8	1.00	0.62
0.4	1.54	1.00	0.438
0.5	1.42	1.00	0.26
0.63	1.28	1.00	0.152
0.8	1.1	1.00	0.06
1.0	1.0	1.00	0.025
1.25	1.12	1.00	0.009
1.6	1.26	1.00	0.004
2.0	1.42	1.00	0
2.5	1.6	0.80	0
3.15	1.8	0.63	0
4.0	2.00	0.5	0
5.0	2.00	0.4	0
6.3	2.00	0.315	0
8.0	2.00	0.25	0

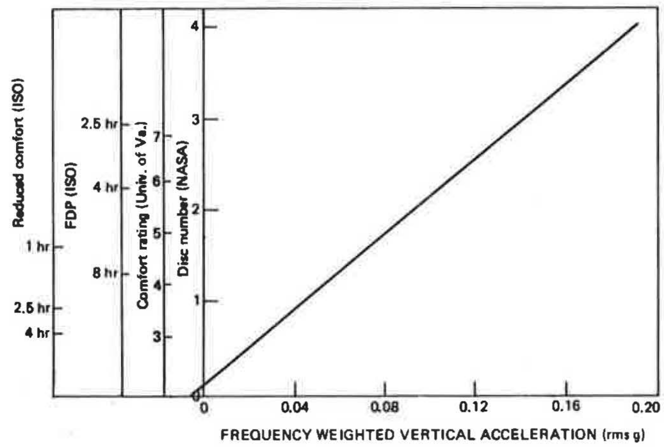
characterizations show that human response tolerance increases sharply below a frequency of 0.15 Hz. However, motion sickness phenomena involve an entirely different set of physiological factors than those associated with comfort or discomfort across the wider frequency spectrum. While motion sickness obviously results in discomfort and impeded performance, it is not necessarily true that the absence of motion sickness guarantees no discomfort. Therefore, as a reasonable compromise, both the vertical and horizontal response curves are made flat between 0.2 Hz and 0.0 Hz.

Discomfort and performance impediments associated with motion sickness are more of a good/bad nature than for other accelerations. Thus, it seems reasonable and prudent to evaluate motion sickness separately. This can be done by the same frequency weighting methods where the motion sickness weighting curve is taken as the inverse of the O'Hanlon and McCauley motion sickness incidence MSI characteristic, as refined by McCauley and others (7).

Table 1 lists the specific weighting factors in the format of ISO 2631 at each one-third octave center frequency. For vertical and horizontal accelerations, the frequency weighting factor (amplitude multiplier) is made equal to 1 at 1.0 Hz. The motion sickness weighting is unity at 0.2 Hz.

As suggested in ISO 2631, a filter with these amplitude response characteristics when inserted in

Figure 13. Vertical acceleration relationship to subjective ride-quality scales.



front of the rms measurement will provide the desired FWRMS values. Alternatively, frequency weighting can be applied after the fact to successive one-third octave measurements. In such a case, the weighting factor at each center frequency must be squared. The FWRMS value of an acceleration is then given by

$$\mathrm{FWRMS} = \left[\sum_{n=1}^{N} (\mathrm{RMS}_{1/3})^2 (\mathrm{FW}_n)^2 \right]^{1/2} \qquad (1)$$

Equation 1 is particularly important since it allows after-the-fact frequency weighted analyses of one-third octave data. It also points the way for the use of one-third octave data to better understand the frequencies involved, while the frequency weighted rms acceleration is used for evaluation of the overall ride quality.

SUBJECTIVE VALUE ASSESSMENT

For marine vehicles very little has been accomplished in the field of subjective value assessments. Hence the state of the art lies in relating quantitative measurements such as the frequency weighted acceleration measures to the qualitative value assessments developed for other vehicles, including aircraft and surface vehicles.

Stark (16) has tentatively anchored the frequency weighted rms measurements to several qualitative assessment scales in order to expand the understanding of the quantitative ride quality data. (See Figures 13 and 14.) FWRMS vertical and horizontal accelerations have been related to four qualitative ride scales: (a) the ISO reduced comfort boundaries, (b) the ISO fatigue decreased proficiency boundaries, (c) a University of Virginia seven-point comfort scale [Richards and Jacobson (17)], and (d) a NASA-developed discomfort scale [Dempsey and others (8)]. The NASA and University of Virginia scales are described in Table 2. These two scales were developed from random motions found in airplane studies, whereas the ISO scales derive principally from sinusoidal motion studies. Dempsey's data clearly show that the mean discomfort assessment for sinusoidal vibration differs from the mean discomfort assessment for random vibration of the same rms amplitude. Because of these differences, more confidence is placed in the relationship between the FWRMS measures and the NASA and University of Virginia scales.

In a similar manner, the motion sickness weighted

Figure 14. Lateral acceleration relationship to subjective ride scales.

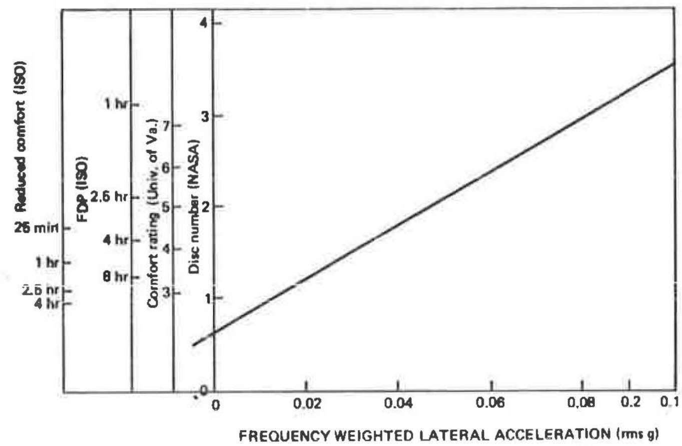


Figure 15. Effects of time and weighted acceleration on motion sickness.

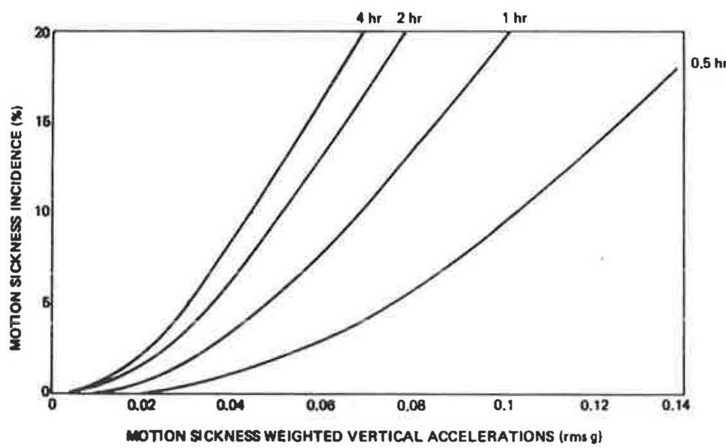


Table 2. Comfort and discomfort scales.

Comfort Scale (Univ. of Virginia)		Discomfort Scale (NASA)	
Comfort No.	Description	Disc No.	Description
1	Very comfortable	0	Zero discomfort
2	Comfortable	1	Threshold of discomfort
3	Somewhat comfortable	2	-
4	Neutral	3	-
5	Somewhat uncomfortable	4	-
		5	-
6	Uncomfortable	6	-
7	Very uncomfortable	7	-
		8	Maximum discomfort

acceleration values are related to time and the frequency of motion sickness in Figure 15. Motion sickness data were derived from the general formulas developed by O'Hanlon and McCauley.

Dempsey and Leatherwood (8) have also related roll and pitch accelerations to their discomfort scale, which can be used if roll and pitch are significant factors. However, recall that the shape of the response curve below 1 Hz is unknown.

VEHICLE RESPONSES AND DESIGN CAPABILITIES

Those aspects of ride quality associated with the development of the vehicle response characteristics are well known for most types of marine vehicles. The seaway characterization and historical data base are fairly well documented, and seem more than adequate to support design and analyses. With regard to the vehicle transfer function, both methodology and well-refined computer programs and model test facilities exist for most types of marine vehicles. The design community also is capable and knowledgeable of methods to design for ride quality. The evolution of the hydrofoil ship provides an example of how ride qualities have been progressively improved by specific design improvements. (See Figure 16.)

STANDARDS

Perhaps the greatest need in the ride quality area is the need for standardized measurements and assessments that adequately cover the low-frequency motions. It is recognized that knowledge of low-frequency ride characterization is still limited and that many serious questions remain unanswered. Nevertheless, standardization on evaluation methodology is considered long overdue and it remains for the larger community concerned with ride quality and ride properties to generate those standards. In the interim, the following recommendations on standardization are offered for consideration.

The FWRMS measurements of translational and rotational acceleration, with each axis taken separately, are recommended. Frequency weighting taken as the inverse of human response characteristics and normalized to values at 1 Hz is recommended for vertical and horizontal accelerations. Frequency weighting characteristics for angular accelerations need to be developed.

Figure 16. Hydrofoil ride-quality improvements through design and development.

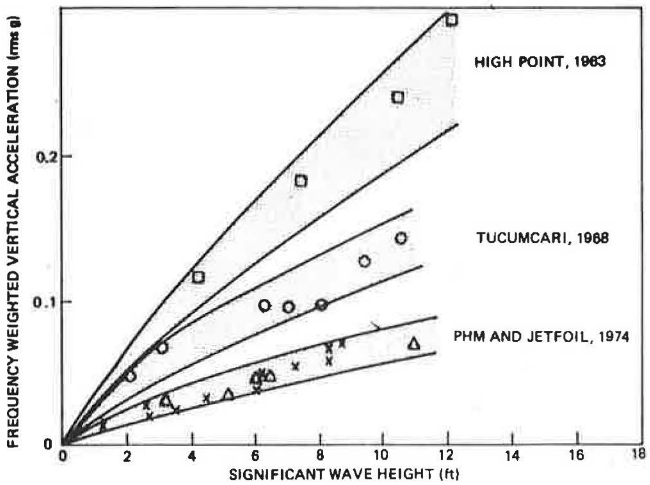
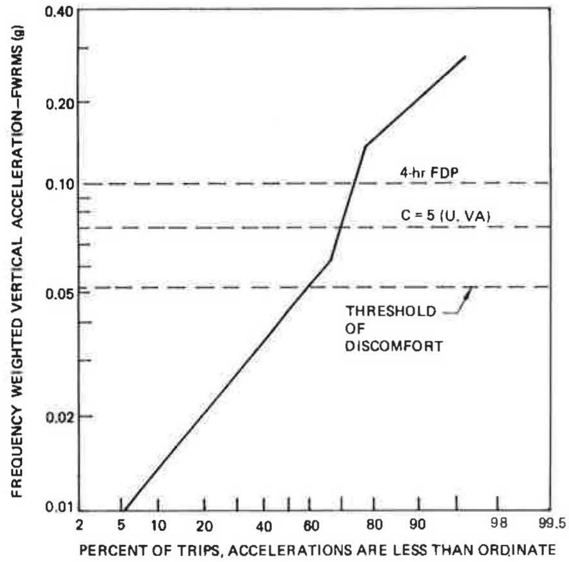


Figure 17. Long-term ride quality analyses.



Motion sickness evaluation should be accomplished by similar frequency weighting. The frequency weighting is taken as the inverse of the O'Hanlon and McCauley motion sickness response characteristic and normalized to 0.2 Hz.

In consideration of operators and crew, qualitative assessments such as the ISO Fatigue Decreased Proficiency Standard are of major importance to the marine industry. It is imperative that operator-related standards be expanded to cover the low-frequency regime and that those standards be related to the standardized measurements.

When considering passenger comfort and acceptance, specific standards seem less important. Instead, comfort/discomfort scales, such as those from NASA and the University of Virginia, are recommended; they delineate various degrees of discomfort. They also open the door for comparative evaluations across a broad spectrum of vehicle and passenger acceptance decisions. For instance, one can discern from Jacobson and others (17) that 70 percent of a set of airline passengers were satisfied with the ride for comfort ratings of 5 or less on the University of Virginia seven-point scale, and 50 percent were satisfied with comfort ratings of 6 or less.

Comparative evaluations are of major importance in decisions made by transportation system operators. The value and reliability of comparative evaluations would be greatly enhanced by comprehensive standards for ride quality evaluation. For most vehicles, little credence can be placed in any specific ride quality measurement because the ride properties can change in a few hours (or a few miles) as the environment changes. Properly, the total range and probability of riding properties needs consideration. Therefore, FWRMS acceleration distributions are recommended as a standard for comparison and evaluation (see Figure 17). Direct comparisons can be made in that same format for different vehicles operating in similar environments, and comparisons can be made with other operating systems by using the University of Virginia and NASA scales and literature. These quantitative methods permit basic comparative evaluations.

Finally, comprehensive methods for measuring and evaluating human response to (and tolerance of) repeated shock transients need to be developed and standardized. Those standards need to specifically consider the longer-period transients experienced by marine vehicles.

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Ride-Quality Evaluation and Technology in Intercity Rail Passenger Car Specifications

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Four factors—thermal comfort, noise, lighting, and vibration—that affect passenger comfort in rail passenger cars are reviewed. The effects of passenger seating and vehicle layout are also considered.

From the early 1950s until 1970 no new developments were made by American railroads or car builders in either state-of-the-art designs or passenger comfort for long-distance passenger cars. Therefore, ride-quality specification requirements for passenger cars must be addressed in light of the history of the U.S. long-distance passenger car industry.

Since 1970, Amtrak has operated almost all of the intercity passenger trains within the United States. When first formed, Amtrak owned neither rails nor rolling stock of any kind; it began to run passenger service in 1971 through contracts with private railroads using rolling stock acquired from those railroads. This rolling stock averaged between 20 and 30 years of age: all the cars with the exception of the Metroliners were built either before World War II or in the late 1940s and early 1950s. These cars were heated by steam and had electric power provided from axle-driven generators. Most of this old equipment was unreliable because of its age and because sufficient power could not be generated at the average slow passenger train speeds. Some of the most unreliable items were the air conditioning, lighting, and truck systems of the cars--three items that are of major importance to overall passenger comfort.

RIDE COMFORT SPECIFICATIONS

The process of buying rolling stock from the user's standpoint consists of writing the specification, selecting the car builder, and negotiating the final contract. The specification and contract cover the design parameters of the equipment as well as testing requirements. They also cover the program management terms and conditions. As far as passenger ride comfort is concerned, there are four major factors that must be addressed in all equipment specifications for passenger cars: (a) thermal comfort, (b) noise, (c) lighting, and (d) ride quality. In addition, there are several minor factors that are of secondary importance to passenger comfort: seating, space allocation, car interior color and material, toilets, food service, and the passenger's ability to see outside.

Of the four major factors mentioned above, the first three are relatively easy to specify, measure, enforce, test, and acquire reasonably accurate data based on the public's response. The fourth factor, ride quality, is the subject of much debate among scientists, car builders, and the public, as well as the operating railroads. The ride quality of long-distance rail equipment is difficult, if not impossible, to specify, test, measure, or analyze. It is also difficult to gather public response and reaction to the effects of ride quality.

The first Amtrak rolling stock specifications were prepared by consultants familiar with commuter equipment, federal government procurement practices, and test requirements as specified by the Urban Mass Transportation Administration. These specifications, although valuable, were hard to interpret and enforce.

Two types of specifications can be used to specify rolling stock equipment. The first and most lenient specification is called a hardware specification, wherein the buyer specifies to the builder only that he wishes a piece of equipment such as one that already exists (somewhat like purchasing a refrigerator or washer for the home). The second and most restrictive specification is called a requirements specification, wherein only the required parameters such as construction material dimensions, operating criteria, testing criteria, etc., are specified.

There are obvious problems with both types of these specifications. First, the builder can manufacture almost anything to satisfy the general hardware specification, whereas the buyer must have, or have access to, a large technical staff to ensure that the builder complies with the requirement spec-