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An Assessment of Current Shipboard Vibration Technology

E. F. Noonan, Member, NKF Engineering Associates, Inc., Silver Spring, Md.

ABSTRACT

The application of current vibration technology to the design of ships is still in the development stage and can be expected to continue to be for some years to come. Although much progress has been made in recent years shipboard vibration control must still be considered an art in which the designer freely applies his own approach and techniques to insure satisfactory performance. In this paper the author presents his approach to this complex problem as applied to current commercial and naval shipbuilding programs. Recent findings, solutions to problem areas and recommendations for future research efforts are also presented.

INTRODUCTION

Shipboard vibration problems have been with us for many years and continue to be a potential hazard, both to shipbuilders and operators. In recent years, the significant increase in power requirements has further magnified the problem and emphasized the need for effective design procedures. Unfortunately however, the problem is sufficiently complex so as to preclude the development of any simplified analysis procedure which could be effectively applied by inexperienced engineers or architects. It is the purpose of this paper therefore, to evaluate current technology in the field of shipboard vibration and to present the views of the author on his approach to a rational design procedure. Weaknesses in the available technology will be identified along with recommendations for future research efforts.

BACKGROUND

Shipboard vibration, for purposes of this presentation, will include both hull and machinery vibration, plus related dynamic considerations such as dynamic shaft stresses. Although the interest of the Ship Structures Symposium is primarily directed toward the ship structure, the interdependence of the hydrodynamic (hull form and propeller design), structural (configuration and structural details) and machinery (power plant and shafting design details) are so mutually interrelated as to necessitate consideration of the total shipboard vibration problem as a parallel effort.

As has occured in many other technical areas, the presence of serious vibration problems experienced aboard ship has been the stimulus behind the technological developments in shipboard vibration. Unfortunately however, most of the effort in this regard has been fragmentary, that is the studies have been initiated by a particular problem such as excessive vibration of the hull or deck house, a broken shaft, or unsatisfactory performance of shipboard equipment associated with vibration. It has also been true that because of the competitive nature of shipbuilding, and the fact that vibration aboard ship is normally considered to be a deficiency, the cooperation between designers and shipbuilders in this area leaves much to be desired. In this country, the principal research effort in this field has been sponsored by the Navy with technical support being provided by the Society of Naval Architects and Marine Engineers through their various research panels, such as HS-7 (Vibrations), M-20 (Machinery Vibration), and H-8 (Hydroelasticity).

Through the cooperative effort of the Vibration Panel (HS-7), the Hull Structure Committee and the Maritime Administration, initial shipboard studies were sponsored which resulted in the first "Code for Shipboard Hull Vibration Measurements," [1][1] in June 1964. Details of the "Code" and the Shipboard Vibration Research Program then underway, were presented at the 2nd International Ship Structures Congress, Delft, the Netherlands, in July 1964, [2]. This "Code" was revised in 1967 after procurement of the Maritime Administration instrumentation package. A third publication Code C-1, "Code for Shipboard Vibration Measurement", [3] originally scheduled for publication in 1970, was recently issued which includes in addition to hull vibration measurements, the measurement of longitudinal vibration of the main propulsion machinery. This latest publication represents a joint effort between S.N.A.M.E. Research Panels HS-7 and M-20. As stated in the Code, "The objective of this Code is to establish standard procedures for gathering, interpreting and presenting data on hull vibration and

[1] Numbers in brackets designate References at end of paper

longitudinal vibration of propulsion shaft systems and to provide a basis for design predictions, improvements, and comparison with vibration and reference levels or acceptance criteria". At this time the Code C-1 is only concerned with (1) the vibration of the ship girder excited by the propulsion system at shaft frequency, propeller blade frequency, harmonics of blade frequency and frequencies associated with the major components of machinery and (2) vibration caused by propeller excitation of propulsion shaft systems. An independent "Code for Shipboard Local Structures and Machinery Vibration Measurements" is in its final stages of review and should be ready for publication this year.

Through the cooperative effort of owners, designers and shipbuilders it is planned to publish ship vibration data through the S.N.A.M.E. for the technical use of the shipbuilding community and to assist in the development of vibration reference levels. The first set of sixteen shipsets of vibration data is expected to be issued shortly.

Parallel efforts are also underway on an international basis. At this time "A Proposed Code for the Measurement and Reporting of Shipboard Vibration Data" is expected to receive a final review at the September 1975 meeting of the International Organization for Standardization, Technical Committee 108, for Shock and Vibration, (ISO/TC108), in Amsterdam. This Code, patterned after C-1, has been developed by a Working Group on Ship Vibration which includes designers and members of the various ship classification societies. The standardization of techniques and procedures for the measurement and evaluation of shipboard vibration data will greatly enhance the amount and value of information obtained, and serve as a more reliable base for the development of the required prediction techniques.

Once having established a standard method of measurement and reporting of shipboard vibration data, criteria may be established and a basis formed against which ship vibration characteristics can be evaluated and design objectives established. The need for developing methods of improving design procedures which would permit the development of ships and machinery systems, free from excessive or damaging vibration was recognized over ten years ago. To do so however, it was considered necessary to close the large technical gap which existed, and to a degree still does exist, between the designer and research investigator. The first Conference on Ship Vibration, jointly sponsored by the Acoustics and Vibration Laboratory of the David Taylor Model Basin (NSRDC) and the Davidson Laboratory of the Stevens Institute of Technology was undertaken as a first step in bridging that gap. This conference was held at the Davidson Laboratory, Hoboken, New Jersey, in January 1965 and primarily served as a review of the "state-of-art" of Shipboard Vibration and included a program on "Vibratory Forces and Moments from Hydrodynamic Theory and Model Experiments" and a second program on "Vibratory Response Characteristics of Ships". It was the purpose of this conference, to exhibit, in understandable presentation, those findings of recent research and engineering studies in ship vibration which would be of immediate use to the practicing naval architect and marine engineer. The Proceedings of the conference on Ship Vibration were published by the David Taylor Model Basin [4]. Most recently, a review of Ship Vibration Prediction Methods and Evaluation of Influence of Hull Stiffness Variation on Vibratory Response [5], and a companion Bibliography [6] was published by the Ship Structure Committee.

Although many individual papers on various aspects of ship vibration have been published, both in the U.S. and abroad, since that first conference on ship vibration, there have been very few attempts to consolidate the research into a design procedure, which would be useful to the ship designer. One such attempt however, was presented by the author, in February 1970, at a meeting of the New York Metropolitan Section of the S.N.A.M.E. and was published in Marine Technology [7]. A more recent effort was presented by G. Volcy, [8] which discusses the approach made at Bureau Veritas. To be sure, many variations in the approach, extent of calculations and the degree of reliance on experience will vary widely between investigators. No attempt is made in this presentation to comment on the approach of others, but rather to present the approach to a rational design procedure, as employed in the development of the DD963 Destroyer Program for Litton Industries, and in the LNG Program for El Paso Natural Gas Company.

APPROACH TO A RATIONAL DESIGN PROCEDURE

A rational design procedure requires the following elements:

1. A set of design objectives or specifications
2. An analytical procedure which includes
    a. A suitable mathematical model of the mass-elastic system under consideration
    b. Input or forcing functions determined by theoretical analyses, model testing or a combination of both
    c. Appropriate damping coefficients
    d. Empirical factors to bridge missing functions, to efficiently simplify the analyses or to compensate for weaknesses or missing aspects of the theory.
3. Full scale test and evaluation program, to

a. Confirm the adequacy of the results, and

b. Obtain technical data to permit the continued development of improvement of empirical factors.

The assessment of current shipboard vibration technology employed in this presentation will include a review of the status of these basic elements, their availability and/or adequacy in the development of a rational design procedure and an identification of remaining problem areas in the application of these elements to modern shipbuilding. As a basis of judgment, this review will reflect the approach used in the development of two of the largest and most recent shipbuilding programs; the DD963 Destroyer Program, designed and built by the Ingalls Shipbuilding Division of Litton Industries for the Navy, and the 125,000 C^3 LNG Carriers under development for the El Paso Natural Gas Company. The LNG program currently includes a group of nine ships, three each, of three individual designs by Chantiers-France-Dunkerque, Avondale Shipbuilding of New Orleans, La., and Newport News Shipbuilding and Dry Dock Co. of Newport News, Virginia. These two design programs include widely varied characteristics, thus permitting an evaluation of the state-of-art from a high-speed, fine-lined destroyer to a large tanker. The principal characteristics of these ship designs are presented in Table 1.

It is unfortunate however, that the timing for this paper is a little premature in that, at the time of preparation, the final reports on the vibration tests conducted in February 1975 on the DD963 had not been issued, and that no time was available to obtain the necessary clearances for publication of the test results. In the case of the LNG Carriers, at the time of the writing, preparations were underway for an extensive test program on the first France-Dunkerque ship. It is expected however, that the tests will have been completed on the first LNG by the time this paper is presented and a qualitative evaluation of the vibration characteristics will be available.

Of particular note, in the development of these two radically different designs, is the operating characteristics of each. The destroyer must be capable of a wide range of operating speeds and of performing high speed maneuvers, while the LNG Carriers will normally operate continuously at a constant design speed. Also, the destroyers, equipped with twin CRP propellers and having a much lighter structure presented some unique problems while the LNG in turn, employed a power plant of some 25% greater than previously used on a single screw cargo ship. Some variations in the approach used in the vibration analyses of these two ships were therefore required.

Table 1 - Comparison of Characteristics of DD963 & France Dunkerque LNG Carrier

Ship Characteristics	DD963	LNG(F-D)
Length Overall, ft.	563.3	885.8
Length between perpendiculars, ft.	529.0	846.5
Breadth, ft.	55.0	136.5
Service draft, ft.	18.0	36.8
Depth, ft.		
to 01 level	42.0	—
moulded on trunk deck	—	90.0
Displacement, tons		
maximum load	7800	—
service	—	105,500
Machinery Characteristics		
Number of Shafts	2	1
Maximum horsepower per shaft	40,000	45,000
Maximum RPM	168	108
Number of struts per shaft	2	1
Type of propulsion	gas turbine	steam turbine
Propeller Characteristics		
Type	Controllable, reversible pitch	fixed pitch
Diameter, ft.	17.0	25.3
Pitch at 0.7 R, ft.	26.2	26.5
Pitch ratio	1.54	1.05
Developed area ratio	0.73	0.83
Number of blades	5	5
Total weight (in air), lb.	52,453	101,000

Design Objectives or Specifications

As in any other design study, it is necessary to have design objectives against which the analysis is judged. In the case of mechanical vibration, excited by a ship's propulsion system, the analysis of mechanical or structural components of the ship must satisfy fatigue, habitability or service requirements. In the application of these specifications it is also necessary to properly define the test and evaluation procedures. For this reason, the "Code for Shipboard Vibration Measurement" is most important and although it was published in 1975, it is very similar to the Navy Code [9] and the earlier S.N.A.M.E. Code for Shipboard Hull Vibration Measurements [1].

Early in 1971 a set of design objectives, in the form of Vibration Specifications was generated for the LNG design. These specifications were included in the requirements for both the Avondale and Newport News designs. Although the contract for the F-D design preceded these specifications, these requirements are being used for evaluation purposes and are included here for reference purposes.

A. Vibration Specifications for 125,000 M^3 LNG Carrier

1.0 General Requirements

The objective of this specification is to limit the vibration of the ship and within the ship, to those generally accepted levels which will not result in discomfort or annoyance to the crew, will not prove damaging to the main propulsion system, or precipitate damage or malfunction of other shipboard machinery and equipment. This specification establishes the criteria which will be used for purposes of evaluation as well as the procedures and methods of measurement to be employed in the evaluation. It shall be the responsibility of the builder to introduce corrective action where the established criteria is exceeded, or, if aspects of the design are not considered adequate to achieve the criteria herein established, recommend design changes, which, in their experience, are necessary to achieve the desired results. For convenience, the total ship is divided into the following five parts:

Part I Vibration of Hull Girder
II Vibration of Major Substructures
III Vibration of Local Structural Elements
IV Vibration of Shipboard Equipment
V Vibration of Main Propulsion System

The detailed requirements include the treatment of each of these parts.

2.0 Vibration of Hull Girder

The adequacy of the design with respect to the generation of the driving forces originating in the main propulsion system and the response of the hull girder is reflected in its vibration characteristics. These characteristics provide the base from which the response of the major substructures, local structures, and supporting systems for equipment may be judged.

2.1 Hull Girder Criteria

The design objective is to limit the vibration of the main hull girder to a velocity of $\pm.25$ in/sec vertically, and $\pm.15$ in/sec in the athwartship or longitudinal direction when tested in accordance with the "Code for Shipboard Hull Vibration Measurements", The Society of Naval Architects and Marine Engineers Bulletin No. 2-10. Amplitudes greater than 150% of these values ($\pm.375$ and $\pm.225$ in/sec) will be considered unacceptable. The selection of the propeller type, number of blades, skew and clearances should be compatible with the achievement of the desired vibration characteristics of the main hull girder and propulsion machinery. Structural design details, including but not limited to frame spacing, and dimensions, in the stern area of the ship, should be adequate to prevent warping or cracking due to propeller excited vibration. Foundations for the stanchions supporting the main deck house should be sufficiently rigid to prevent the amplification of the vertical motion of the hull in the deck house. Any failure of structural components, within the hull girder, which can be attributed to vibration, must be corrected by the builder, as required.

3.0 Vibration of Major Substructures

The response of major substructures reflects the dynamic behavior of those structural elements when subjected to the motions of the basic hull girder at the points of attachment. As a minimum, the vibration amplitudes and frequencies will correspond to those of the hull girder at the point of attachment. Some amplitude magnification generally may be expected as a result of flexibility and/or resonances present in these substructures. Examples of major substructures include deckhouses, uptakes, machinery platforms, decks, and bulkheads.

3.1 Criteria for Major Substructures

The criteria for the vibration of the major substructures occupied by the crew, is based on habitability requirements. As an objective, a maximum velocity of $\pm.30$ in/sec vertically and $\pm.20$ in/sec in the transverse (athwartship or longitudinal) directions is desired. Amplitudes greater than $\pm.45$ in/sec and $\pm.30$ in/sec in the vertical and transverse directions respectively, shall be considered unacceptable and must be corrected by the builder, as required. During ship trials, tests shall be conducted to demonstrate compliance with these requirements. Equipment and procedures called for in S.N.A.M.E. Bulletin 2-10 shall be used for evaluation purposes. To achieve these objectives, adequate supports to the main deck house and transverse (athwartship and longitudinal) bracing of the structure itself, will be required to prevent any significant amplification of the main-hull girder motion.

The criteria for the vibration of major substructures, not inhabited by the crew, is 0.1 g, provided this level of vibration is acceptable to equipment mounted thereon, including its supporting structure and mountings, if any. If the vibration of the equipment mounted on these substructures is considered excessive for the equipment, modifications of the substructure or the equipment supports, as necessary, will be the responsibility of the shipbuilder. In no case will structural damage attributable to this vibration be acceptable.

4.0 Vibration of Local Structural Elements

The vibration of panels, plates, or minor structural members are evaluated in terms of the vibration of the main structural members to which they are attached. The reference, therefore, could be the main hull girder at that point or a major substructure.

4.1 Criteria for Local Structural Elements

The criteria for local structural elements, if they are considered as a part of a habitable space in contact with the crew, such as a compartment floor, is based on habitability requirements. The same criteria apply, as in the case of major substructures, i.e., amplitudes greater than $\pm$.45 in/sec vertically, and $\pm$.30 in/sec in either transverse direction, shall be considered unacceptable and must be corrected by the builder.

The criteria for the vibration of structural elements, not in contact with the crew, and not supporting equipment, is $\pm$.25 g, provided no structural damage results or that noise generated by this vibration is not considered excessive (greater than 90 dBA). If damage to the structural element, or excessive noise in habitable compartments result, and can be attributed to the vibration observed, regardless of the level of vibration, correction will be required by the shipyard.

The criteria for the vibration of structural elements supporting vibration sensitive equipment must be limited to that considered acceptable to the equipment, as specified by the equipment manufacturer, or .25 g, whichever is the least. Structural damage or excessive noise generated in habitable compartments, must be corrected by the shipbuilder.

5.0 Vibration of Shipboard Equipment

This requirement applies to all auxiliary machinery and equipment installed aboard ship. It is applicable to both passive (not self-excited) and active (self-excited) equipment.

5.1 Criteria for Shipboard Equipment

Equipment selected should be designed to meet the environmental vibration requirements established for shipboard use. In this instance $\pm$.25 g should be used. Balancing and vibration tolerances for rotating machines should be representative of and must meet the accepted standards for good commercial practice. Installation details, including the choice of mountings, if used, should be checked to see that the equipment vibration, as installed, does not exceed that for which the equipment was designed.

In the case of self-excited equipment, such as engine generators, pumps, compressors, etc., the supporting structure and/or mountings if used, should be designed to prevent excessive vibration of the equipment or the generation of excessive vibration or noise in the compartment in which it is installed, or in adjacent habitable spaces. Excessive vibration is that above $\pm$.25 g or that level for which the equipment is certified by the manufacturer, whichever is the lesser. The vibration generated noise is excessive when it is over 90 dBA. Necessary corrections shall be the responsibility of the shipbuilder.

6.0 Vibration of Main Propulsion System

Main engines, shafts, couplings, reduction gears, propellers and related equipment are designed for structural adequacy under the conditions stipulated in the procurement specification. Vibration characteristics of the propulsion system must be controlled to avoid the presence of damaging vibration within the system and with the generation of severe hull vibration. Potential problems include balancing of components, lateral, torsional and longitudinal vibration of the propulsion system, and resonance of the hull structure when stimulated by propeller forces at propeller blade frequency or principal engine frequencies.

6.1 Balancing Requirements for Propulsion Machinery

All rotating propulsion machinery shall be balanced to minimize vibration, bearing wear, and noise. The types of correction, as shown in the table below, shall depend on the speed of rotation and relative dimensions of the rotor.

Type of Correction	Speed	Rotor Characteristics
Single-plane	0-1000	$L/D < 0.5$
	0-150	$L/D > 0.5$
Two-plane	> 1000	$L/D < 0.5$
	> 150	$L/D > 0.5$
Multi-plane		Flexible: Unable to correct by two-plane balancing

L = Length of rotor mass, exclusive of shaft.
D = Diameter of rotor mass, exclusive of shaft.

The residual unbalance in each plane of correction of any rotating part shall not exceed the value determined by:

$U = \frac{4W}{N}$ for speeds in excess of 1000 rpm

$U = \frac{4000W}{N^2}$ for speeds between 150 rpm and 1000 rpm

or $U = 0.177W$ for speeds below 150 rpm

where U = maximum residual unbalance in oz. - inches
W = weight of rotating part in lbs.
N = maximum operating rpm of unit

6.2 Torsional Vibration of Propulsion Machinery

The mass elastic system, consisting of turbines, couplings, reduction gears, shafting and propeller, shall have no excessive torsional vibratory stresses below the top operating speed of the unit nor excessive vibratory torque across gears within the operating speed of the unit. Excessive torsional vibratory stress is that stress in excess of

$$S_v = \frac{\text{Ultimate Tensile Strength}}{25}$$

Below the normal operating speed range, excessive torsional vibratory stress is that stress in excess of 1.75 times S_v.

Excessive vibratory torque, at any operating speed, is that vibratory torque greater than 75 percent of the driving torque at the same speed, or 10 percent of the full load torque, whichever is smaller.

A mathematical analysis of the system shall be prepared by the engine builder, design agent or shipbuilder to demonstrate probable compliance with these requirements. This analysis is to be forwarded to the El Paso N.G. Co. for review. During ship trials, measurements shall be performed, to demonstrate compliance with specified limits. These tests, conducted simultaneously with the hull vibration measurements called for in 3.1 are described in S.N.A.M.E. Code C-1, "Code for Shipboard Vibration Measurements". In this Code, longitudinal vibration measurements are called for at the following locations:

1. Thrust Bearing Housing
2. Forward End of Bull Gear Shaft. This position will require a probe and provision for access to the gear case.
3. Gear Case Foundation. On top of the gear case foundation under the shaft centerline.
4. Gear Case Top - Over shaft centerline.
5. High pressure Turbine. Attached to HP turbine casing at forward or after end.
6. Low Pressure Turbine. Attached to HP turbine casing at forward or after end.
7. Condenser - Mounted as low as practicable and as near the fore and aft centerline as possible.

6.4 Lateral Vibration of Propulsion Shafting

No critical frequency of lateral vibration of the propulsion shafting system shall exist below 115 percent of maximum rated speed. A mathematical analysis of the lateral vibration characteristics of the rotating propulsion shafting system shall be made to clearly demonstrate that the system is free from any lateral critical frequency below 115 percent of the maximum rated speed. This analysis shall be submitted to the El Paso N.G. Co. for review.

7.0 Design of Tailshaft

To avoid the possibility of a corrosion fatigue failure of the propeller shaft, in addition to meeting the ABS design requirements, the alternating bending stresses in the tail shaft shall be limited to ±6,000 psi when calculated by the following expression:

$$S = \frac{C(M_g + M_t)}{6,000} = \frac{I}{C}$$

where: S = Section Modulus = $\frac{I}{C}$

C = Service Factor = 1.75

M_g = Gravity moment due to overhanging propeller weight calculated from forward face of the propeller.

M_t = Calculated moment of eccentric thrust = .065 x Propeller Diameter x Rated Thrust

6000 = Maximum safe fatigue limit to be used for the assembly operating in the presence of a corrosive medium (psi).

B. Other Vibration Specifications and Criteria

While the preceeding specifications were developed for merchant LNG vessels they will not be found to differ substantially from those applied to the Navy DD963. It should also be noted that a wide circulation had been made of the LNG specifications, both here and abroad, with the response generally consisting of favorable comments. These requirements are presently under review and consideration as the basis for "Ship Vibration and Noise Guidelines", being prepared by the Vibration Panel (HS-7) under the direction of the Hull Structure Committee.

The scope of shipboard vibration in this paper concerns itself with hull and machinery vibration excited by the propulsion system. The normal criteria for the hull reflects habitability requirements while the components of the machinery system are generally controlled by fatigue characteristics. The habitability requirements of Major Substructures, paragraph 3.1 of the LNG specifications, and the Hull Criteria, given in paragraph 2.1 of the LNG specifications, prepared in February 1971, are shown in Figure 1. Superimposed on this figure is the Interim Guide-Lines for Habitability Criterion porposed by Working Group 2, "Ship Vibration" of ISO/TC108/SC2 in September 1974. The proposed ISO Criterion includes all ship types, both diesel and turbine drives. For turbine driven ships, as in the case for both the DD963 and the LNG, the constant velicity criteria used in this specification has subsequently been endorsed by Det Norske Veritas, with practically identical range of 4 mm/sec to 10 mm/sec for the shaded zone. For diesel driven ships, the constant acceleration criteria, in the low frequency range is considered appropriate. The levels used in the specifications were intended to relate to the "State of the art" of shipboard vibration as well as satisfying the requirements of human susceptability to whole body vibration. [10]

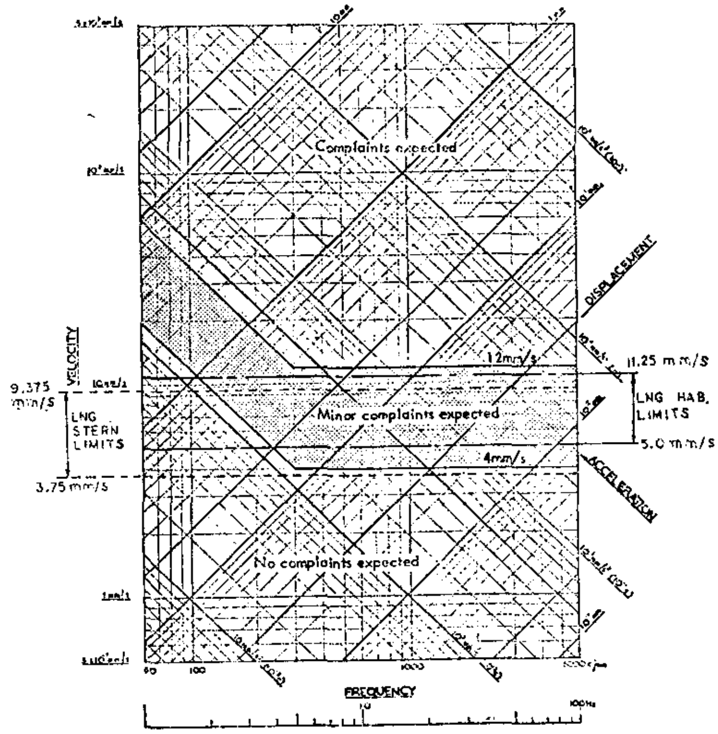


Figure 1 - Habitability Criteria

The requirements for vibration of *Main Propulsion Systems are consistent* with the technical standards developed by the Navy [11] and are based on potential damage or fatigue levels. The tailshaft design requirements are an outgrowth of studies conducted by the S.N.A.M.E. M-8 Panel on "Tailshaft Failures" and relate to designs employing shaft liners. Results of previous studies, on which this criteria, and the Navy shaft design procedure [12] are based, were discussed in the A.S.N.E. Transactions [13]. For oil-lubricated bearings, without shaft liners, this requirement has been reduced, but is presently under reconsideration as a result of the signficiantly large bending moment generated in some current designs.

The Analytical Procedure

The analytical procedure employed should include sufficient detail so as to enable the *designer to reasonably predict the response of* the system under study, for verification against the full scale measurements obtained during trials. Also, a number of signficiant points must be recognized, which have a bearing on the results:

1. The measurements relate to the maximum repetitive amplitude, necessitating an empirical factor, to account for the normal signal modulation obtained aboard ship. In the case of the Destroyer, the range of amplitude variation between maximum and minimum values, is about three to one, under the prescribed test conditions. For a large tanker of the size of the LNG, this factor would be in the order of two to one.

2. When high speed maneuvers are considered a significant operational requirement, as in the case of a destroyer, a further amplifying factor of approximately three and one-half has been obtained experimentally. This factor becomes significant when estimating the response of mechnical parts such as the dynamic load on thrust bearings or gear tooth stresses.

3. The calculated or measured forces and moments generated by the model propeller, operating in the measured wake, is assumed to represent an average or mean sinusoidal value.

4. Hull pressure forces are normally significant in the vertical direction and relatively light in the transverse direction.

5. The presence of propeller cavitation, in the range of 85% to 100% of full power may significantly increase the exciting forces, by factors up to ten to twenty to one, or higher [15]. Results of cavitation studies should be used to determine appropriate factors.

6. Viscous damping or Rayleigh type damping has been recommended for the prediction of hull response [16]. More recently, viscous damping factors which increase with frequency have been effectively used for naval ships [17]. A factor of $\frac{c}{\mu\omega}$ = .035, when used in the mid-frequency range of 5 to 7 Hz, when low modulation and little cavitation is present, has yielded good results for the prediction of hull girder response. For the higher frequencies, in the full-power range, both increased damping and propeller forces are required.

7. Superstructures would generally be expected to have principally hysteresis damping and proportionately higher response than the hull girder.

8. While it is true that the response of the total ship, and all its components, such as superstructure and propulsion shafting systems are related thru modal coupling, it is not necessarily true that the complete ship matrix is required to properly evaluate anticipated response. In most cases hull, main machinery and superstructures or equivalent substructures can be effectively studied, independently.

A. DD963 Program

Although the vibratory response of most main machinery items have been under specification control, in Naval application, through MTL-STD-167 [11], as a result of previous difficulties, the application of specific limitations to hull vibration, was an innovation in the DD963. Specific vibratory limits were placed on the hull, in the form of target and reject amplitudes. A detailed vibration program was developed [18], which included a "Preliminary Hull and Machinery Vibration Analysis" [19] which was primarily used to make early engineering decisions. During the detailed design development numerous supplemental analyses were performed culminating in the full scale vibration tests conducted in February 1975.

At this point, an insight is given on the effectiveness of the preliminary vibration analysis performed on the DD963 and the utility of the current state of the art in the prediction of hull and machinery vibration. Judgment on the effectiveness of the program, which leans heavily on the experience of the investigators, is best formed by an examination of the test results. However, because of time and classification restrictions the data presented is limited to the following points:

1. The full-power shaft RPM was chosen to fall between second vertical and athwartship hull resonances and below the fundamental torsional resonance of the hull girder, to avoid significant response of the hull girder when excited by dynamic or hydrodynamic forces at shaft frequency. Figure 2 shows the hull natural frequencies calculated during the preliminary design phase [19]. Table 2 shows a comparison of the calculated frequencies with those observed during the anchor drop tests.

Table 2 - DD963. Anchor Drop Tests Comparison of Observed & Calculated Hull Frequencies

Mode	Observed, Hz	Calculated, Hz
1st Vertical	1.2	1.2
2nd Vertical	2.4	2.5
3rd Athwart-ship	5.8	5.4

2. In this preliminary study the estimate of propeller forces and moments were obtained (1) by extrapolation from those calculated for similar ship types and (2) by calculation from an assumed wake and a standard propeller based on estimated propulsion characteristics. The thrust and torque fluctuations, thrust eccentricity and the horizontal and vertical bearing forces were calculated by a refined two-dimensional airfoil quasi-steady method originated by Burrill [20]. The hull pressure forces were assumed to be equal in magnitude to the vertical bearing forces and in phase with them. No additional allowances were made for cavitation. Propeller design analysis and cavitation studies were performed by Hydronautics, Inc.

Table 3 shows the estimated and calculated forces and moments in column 1 and 2 respectively. Column 3 shows the values used in the response calculations. Columns 4 and 5 show the corresponding forces and moments later calculated for the baseline propeller (20° skew) and the backup propeller (40° skew) operating in the DD963 wake [21]. For these calculations, the Breslin Program was used since the Burrill Program did not effectively treat propeller skew. The propeller data was developed by Hydronautics [22,23]. Although the calculation of propeller forces is a rather controversial subject, when referring to these forces on an absolute basis, the agreement between the values used in the analysis, column 3 and those calculated by the Breslin Program for the baseline propeller, column 4 was unusually good.

3. For the preliminary hull frequency analysis, the digital computer program developed at NSRDC [14,7] was used as the primary (conventional) procedure. For backup, and in support of the development of a simplified procedure to be applied in preliminary design, a simplified representation, as was used by Ali [24] in 1968 with fair results, was also used. The natural frequencies thus obtained are shown in Table 4.

The results of these frequency studies tend to bear out the potential gains to be made by developing the simplified approach for use in preliminary design.

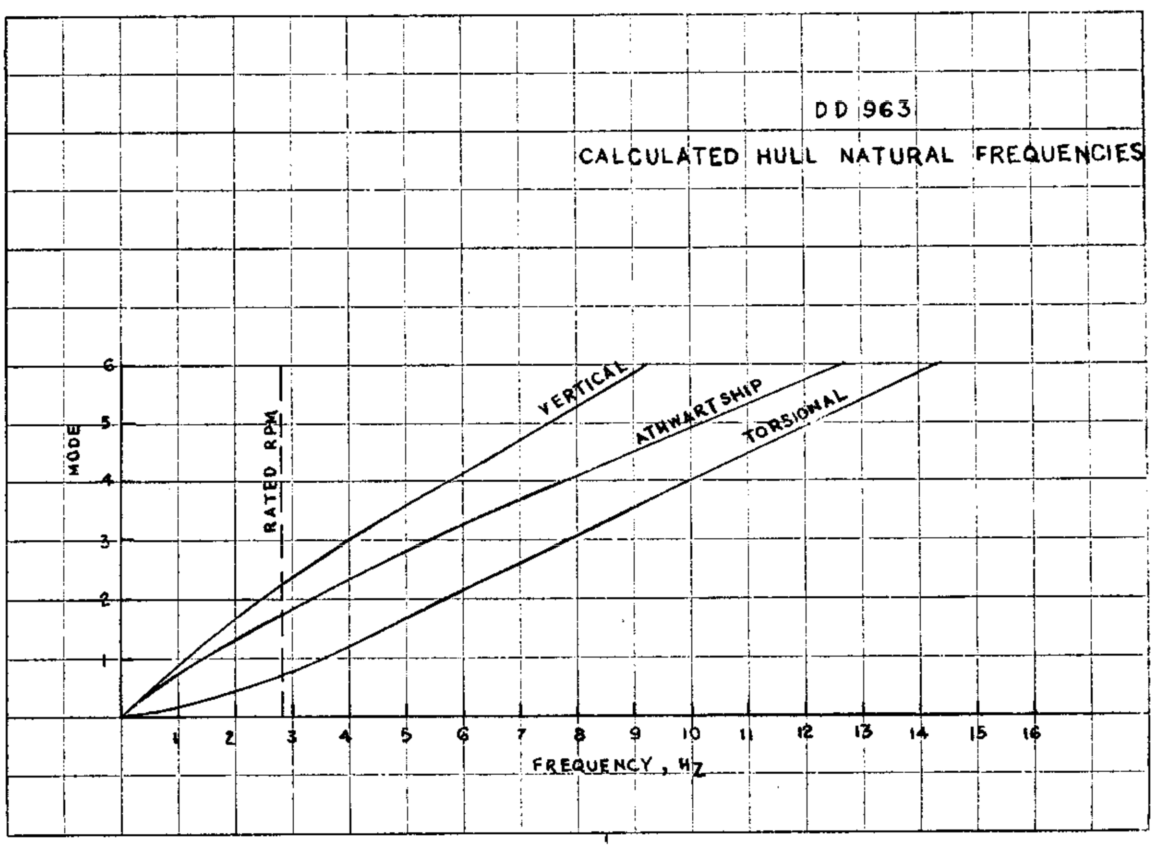


Figure 2

Table 3 - Summary of Propeller Force Calculations - DD963

	1	2	3	4	5
	Estimated from similar ships.	Calculated by Burrill method*	Values Used in response calculations	20° skew calculated by Breslin method**	40° calculated Breslin method**
Mean Thrust $\bar{T}$ lb.	284,000	284,860			
Alternating Thrust $\tilde{T}$ lb.	± 5,250	± 5,094	± 5,000	± 4,897	± 3,047
$\tilde{T}$ in % $\bar{T}$ at Blade Frequ.	± 1.85	± 1.79			
Mean Torque $\bar{Q}$ ft. lb.	1,236,400	1,237,100			
Alternating Torque $\tilde{Q}$ ft. lb.	±18,540	±15,566	±20,000	±19,524	±12,806
$\tilde{Q}$ in % $\bar{Q}$ at Blade Frequ.	± 1.50	± 1.26			
Mean Hor. Brg. Force $\bar{F}_h$ lb.	4,800	4,897			
$\bar{F}_h$ in % of $\bar{T}$	1.69	1.72			
Alternating Hor. Brg. Force $\tilde{F}_h$ lb.	± 4,970	± 4,077	± 3,800	± 4,002	± 3,115
$\tilde{F}_h$ in % of $\bar{T}$ at Blade Frequ.	± 1.75	± 1.43			
Mean Vert. Brg. Force $\bar{F}_v$ lb.	26,000	29,392			
$\bar{F}_v$ in % of $\bar{T}$	9.15	10.33			
Alternating Vert. Brg. Force $\tilde{F}_v$ lb.	± 3,000	± 2,854	± 2,800	± 2,931	± 1,538
$\tilde{F}_v$ in % of $\bar{T}$ at Blade Frequ.	± 1.10	± 1.00			

* Assumed Wake and Wageningen B-5 Series Propeller.
** DD963 Wake and DD963 Proposed Propeller Designs.

4. In calculating or predicting the hull response under actual operating conditions, the blade-frequency propeller forces are assumed to vary with the square of the shaft RPM and at full power are estimated at 2800 lbs in the vertical direction and 3800 lbs in athwartship direction for each propeller (Table 3). The vertical forces are multiplied by 4 to account for the two propellers and the pressure forces, which are assumed to be of equal magnitude. The horizontal forces are multiplied by 2 since there are no significant pressure forces in the horizontal direction. The digital program [14] was used to simulate the blade frequency flexural hull response at the stern, Station 19 1/2, to sinusoidal forces applied at the stern. A structural damping factor of $\frac{c}{\mu\omega} = 0.03$ was assumed at each mass. The resulting "theoretical response" may be compared to the expected response of the hull to a sinusoidal input of a vibration generator with forces equivalent to those predicted, but with the ship "dead-in-the-water", in a calm sea. The "predicted response" or actual response expected under trial

Table 4 - Summary of Natural Frequencies

Mode	Vertical Frequencies (Hz) Conventional	Vertical Simplified	Horizontal Conventional	Horizontal Simplified	Torsion Simplified
1	1.10	1.20	1.42	1.54	3.64
2	2.24	2.50	3.14	3.26	5.62
3	3.70	4.04	5.36	5.40	7.76
4	5.50	5.82	7.60	7.76	10.05
5	7.20	7.60	10.28	10.22	12.20
6	9.02	9.22	12.50	12.70	14.40
	Schlick	Burrill	Schlick	Burrill	Horn
1	1.2	1.3	1.8	1.9	3.8

conditions was developed from the "theoretical response" in the mid-frequency (5 to 7 Hz) range, multiplied by empirical service factors developed from previous destroyer designs. No allowance was made for cavitation. The maximum stern hull response obtained for blade-rate forces are shown in Table 5 and are compared to the target and calculated values.

5. Probably the most serious potential problem area, relative to the propulsion system was the longitudinal vibration of the propulsion system. Particular attention was paid to the starboard or short shaft since it would result in the higher frequency. Estimates, with a four-bladed propeller resulted in the fundamental critical occurring near power. For the initial study, the alternating thrust of ±5,000 lbs. at full power, shown in Table 3 was used. This value was assumed to vary as the square of the RPM and at the estimated critical of 130 RPM was ±3,000 lbs. A digital calculation of the system was performed, using a damping factor of 4,500 lb. sec./in for the CRP propeller. A service factor of 2.5 was used to obtain maximum peak values for steady running and a second factor of 3.5 was used to estimate the effect of hard maneuvers. A more detailed finite element analysis of the propulsion system was performed, after the final design was completed. A comparison between the results obtained in the preliminary and final analysis is shown in Table 6, together with the observed values obtained during the Builder's Trials.

   The close agreement obtained between the preliminary analysis, the more detailed finite-element analysis and the measurements obtained by the semi-conducter strain gage and telemetry system used during the trials was heavily dependent on empirical data obtained on similar ship types [27], and supports the requirement for additional empirical data on all ship types to improve the prediction of hull and machinery vibration.

6. Supplemental studies, including finite element analyses of major substructures, including gun and missile foundations were carried out to insure resonances at blade-rate frequencies were avoided.

Table 5 - Stern Hull Response, DD963
(Blade-Rate Maximum Velocity, inches/sec)

	Target	Calculated (19) Theoretical	Calculated (19) Predicted	Observed (25)
Vertical	.32	.15	.23 - .30	.17
Athwartship	.32	.15	.25 - .30	.21
Longitudinal	.32	-	-	.09

Table 6 - DD963, Starboard Shaft, Longitudinal Critical

	Calculated Preliminary [19]	Calculated Finite Element [26]	Observed [25]
Frequency, Hz	10.83	10.25	10.5
RPM Critical	130	123	126
Straight Course Alt. Thrust ±lbs.	40,000	43,000	45,500
Maneuvers, Alt. Thrust ±lbs.	134,000	150,000	132,000

Similarly, the support systems (foundations and mountings) were analysed for most equipment installation. As a direct result of the low levels of vibration present in the hull, and the absence of resonant magnification of this vibration in mounted equipment, the DD-963 was considered unusually free of troublesome vibration.

B. LNG Program

Unlike the case of the Destroyer design, little vibration experience was available to the designers and builders of the first 125,000 Cubic Meter LNG ship, having a single screw and 45,000 SHP. In 1970 performance guarantees could not be obtained above 36,000 SHP. Because of the potential impact of serious vibration problems on the program of the Owners, the El Paso Natural Gas Company, all reasonable effort to avoid such difficulties were required of the builders, Chantiers Atlantique, France-Dunkerque. The specifications referred to earlier, were invoked on subsequent contracts with Newport News Shipbuilding and Drydock Company and Avondale Shipyards, Inc.

At the time of this writing, preparations are underway for the vibration test program scheduled for the Builder's and Acceptance Trials, in July 1975. With the support of the Owners, it is expected that the results of these studies, and the correlation with design analyses, will be made available to the industry. This approach will contribute much useful data required to develop the empirical factors necessary for vibration prediction. The information presented here, will briefly review the more important steps taken during the design phase, to minimize the possibility of vibration difficulties.

1. The first step involved the selection of the stern configuration. For this purpose France-Dunkerque had three models tested at the Netherlands Ship Model Basin (NSMB):

Model 4141 - Modified Hogner - Figure 3
Model 4147 - Conventional - Figure 4
Model 4148 - Open Transom - Figure 5

Figures 6, 7 and 8 show the circumferential distributions of longitudinal velocity components obtained by NSMB for each model respectively. A preliminary analysis of the longitudinal vibration characteristics of the main machinery system indicated the maximum number of propeller blades required to insure the fundamental critical falling above the operating speed, would be five. Therefore, since an examination of the longitudinal velocity harmonics indicated a five-bladed propeller would be preferable to a four, the propeller parameters were developed in accordance with the Wageningen B-Series for five-bladed propellers. As in the case of the DD-963 the propeller forces and moments were developed for comparison purposes. Results taken from reference (28) are shown in Table 7.

Table 7 - 125,000 M^3 LNG Ships with 5-bladed propeller

Results of Calculations of Propeller Forces Based on NSMB Data

	Model 4141	Model 4147	Model 4148
V_s Knots	20.0	19.0	20.0
SHP_m	43,000	34,400	41,600
D ft.	26.64	25.0	24.5
T Thrust, lbs	635,800	472,900	451,600
$\tilde{T}$ ±lbs.	39,760	31,820	17,520
$\tilde{T}/T$ ±%	6.25	6.75	3.89
$\bar{Q}$ Torque, ft. lbs	2,370,000	1.754,000	2,053,000
$\tilde{Q}$ ±ft. lbs.	97,470	88,780	56,660
$\tilde{Q}/Q$ ±%	4.10	5.05	2.74
$\tilde{F}_h$ Brg. Force, ±lbs.	6,750	3,900	4,950
$\tilde{F}_h/T$ ±%	1.06	.82	1.11
$\tilde{F}_v$ Brg. Force, ±lbs.	3,190	1,660	2,134
$\tilde{F}_v/T$ ±%	.50	.35	.47

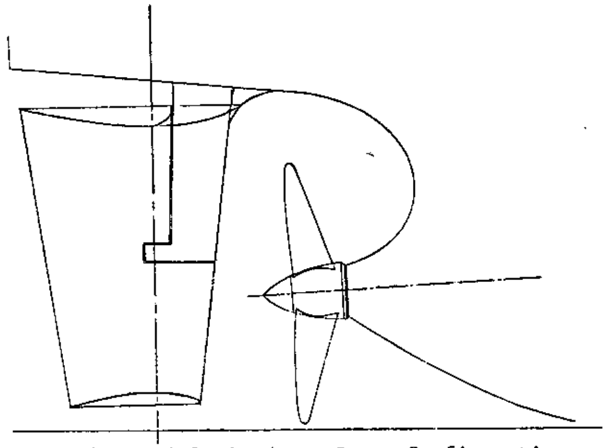


Figure 3 - Modified Hogner Stern Configuration

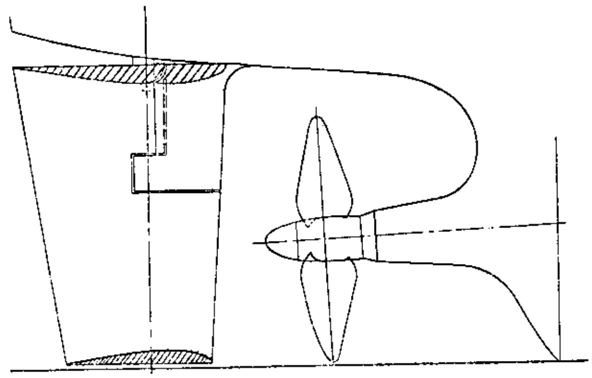


Figure 4 - Conventional Stern Configuration

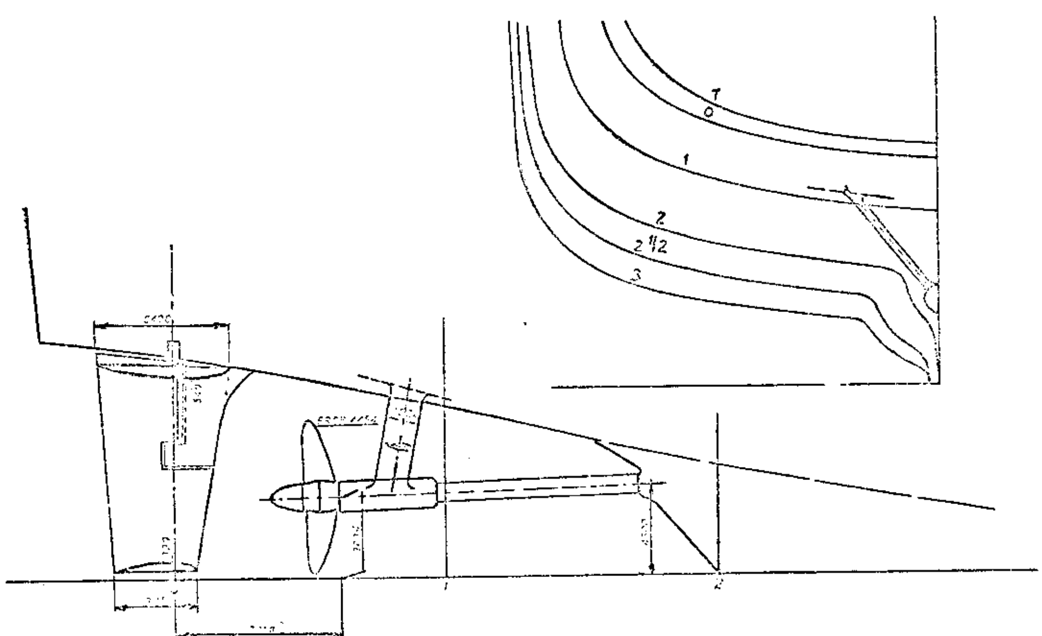
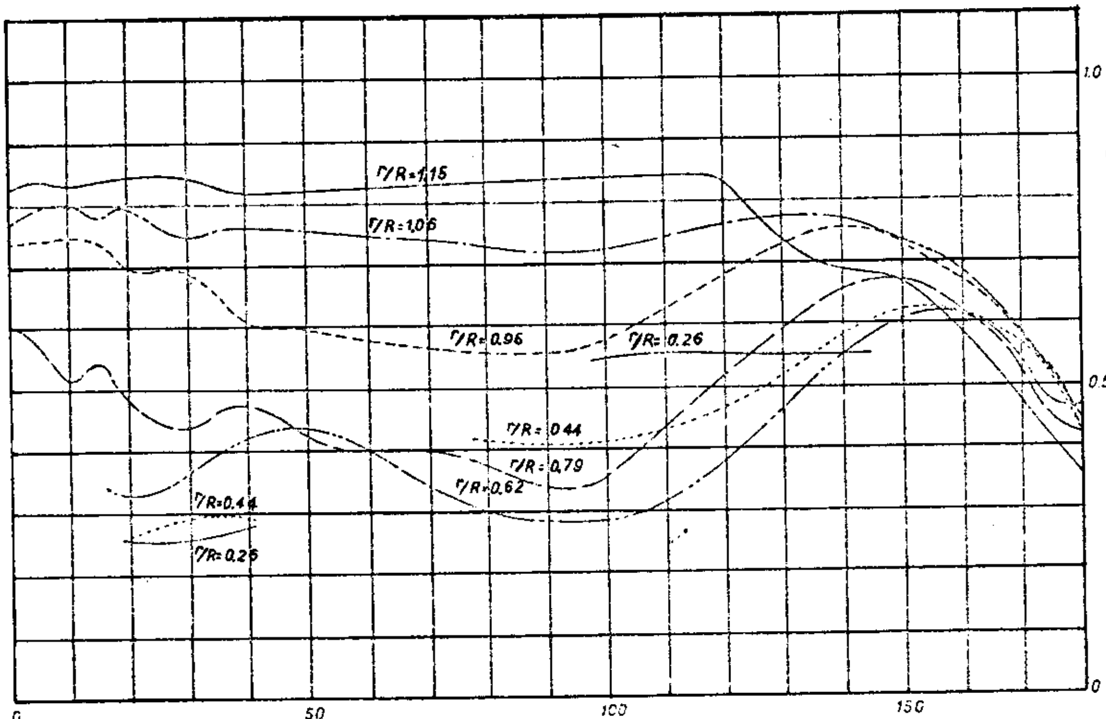
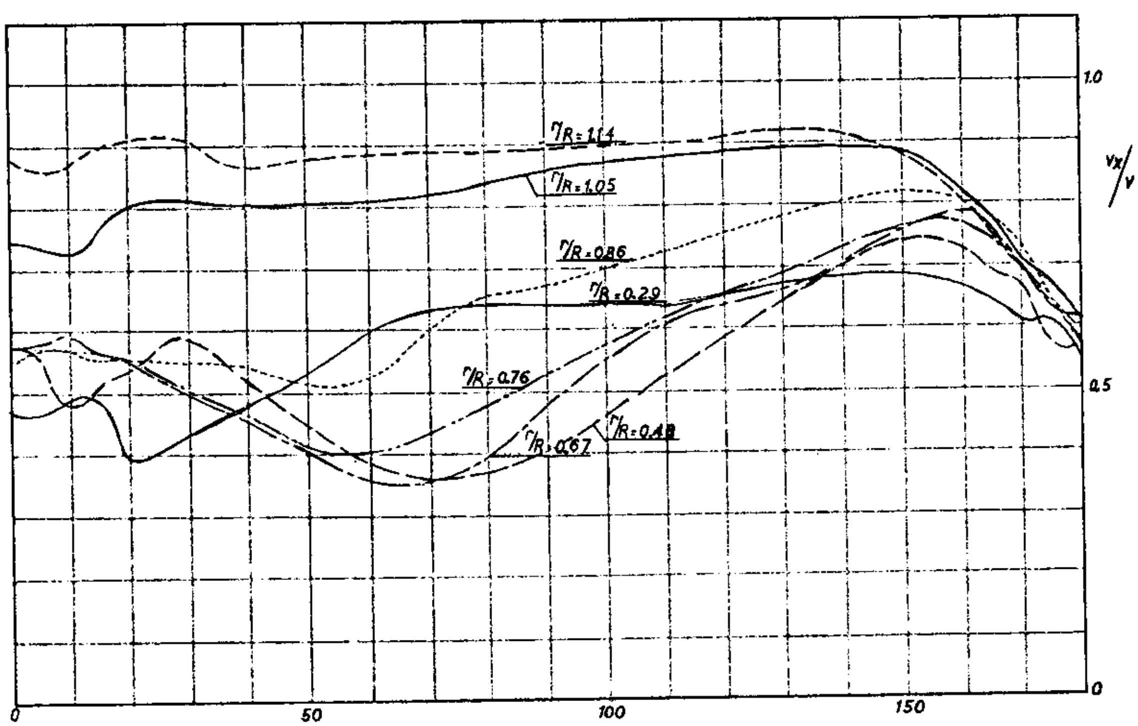


Figure 5 - Open Transom Stern Configuration



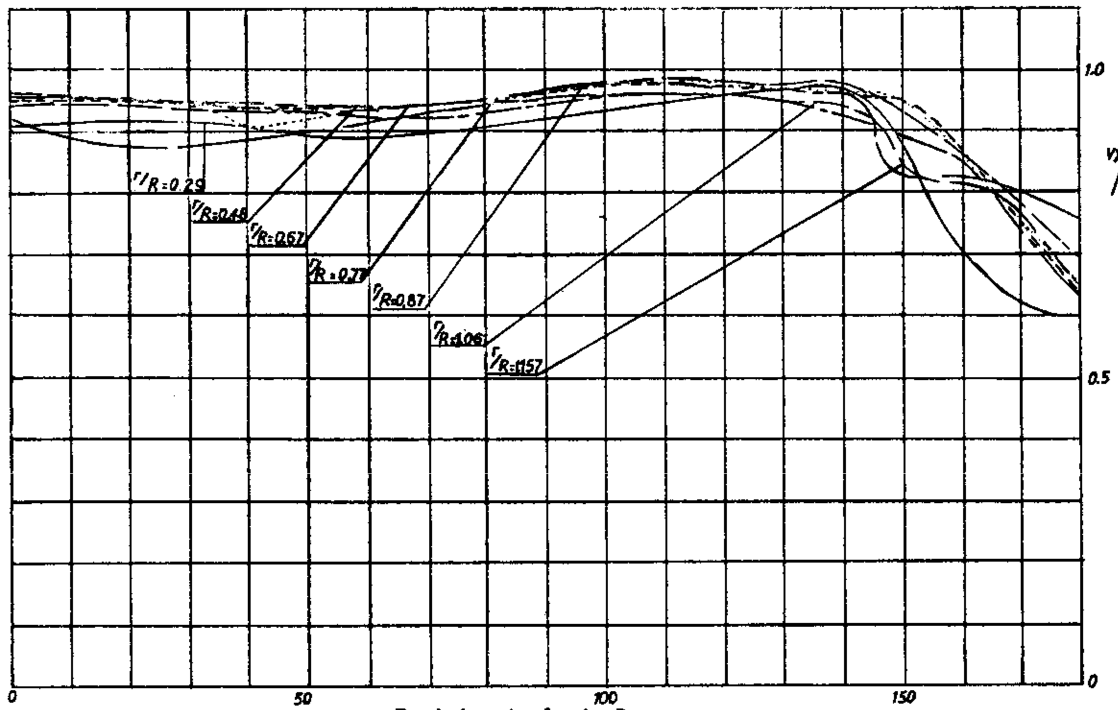
Position Angle in Degrees
Circumferential Distributions of Longitudinal Velocity Components

Figure 6 - Wake Components of Model 4141 - Modified Hogner Stern



Position Angle in Degrees
Circumferential Distributions of Longitudinal Velocity Components

Figure 7 - Wake Components of Model 4147 - Conventional Stern



Position Angle in Degrees
Circumferential Distributions of Longitudinal Velocity Components
Figure 8 - Wake Components of Model 4148 - Open Transom Stern

Based on the results of these studies, France-Dunkerque selected the open transom stern for their final configuration. This same configuration was also selected by Newport News Shipbuilding and Drydock Co. for the 120,000 M^3 LNG ships presently under construction for El Paso Gas Co.

2. The second important step was the prediction of the vibratory forces and moments on the final design, represented by Model 4221A and five-bladed Propeller Model 4522. These predictions were made at NSMB by direct measurement on a wooden model constructed for that purpose. The results of the measurements made by NSMB, taken from reference [29] are shown in Table 8, along with the calculations made by Det Norske Veritas (DNV) on Model 4171 (slightly longer than 4221A), taken from reference [30], and the original results given for the project hull, Model 4148, as previously shown in Table 7. The measured results are considered more reliable and are used for hull response calculations, when available.

Hull pressure forces and moments, with and without cavitation, were also provided by NSMB. They were based on model pressure measurements and were included in reference [29]. The horizontal and vertical hull forces are normally the most significant in regard to hull vibration. In this instance, on the open transom stern without cavitation, the horizontal force was negligible, while the vertical hull pressure force at blade-frequency was ±2,600 lbs, approximately equal to the bearing force ±3,310 lbs shown in Table 8. Without cavitation, only the first harmonic was important and when combined vectorially with the bearing force, the resultant vertical force was ±2,260 lbs just a little smaller than the bearing force alone. Referring back to the DD963, one will recall, we assumed these two forces equal, but in the interest of conservatism, assumed they were in phase.

Of particular interest was the hull pressure forces with cavitation. The horizontal forces remain negligible, but the vertical hull pressure forces increase substantially for the first three harmonics, as follows: F_{v1} from ±2,660 lbs to ±19,200 lbs., F_{v2} from ±180 lbs, to ±13,900 lbs, and F_{v3} from ± 130 lbs to ±1,540 lbs. When vectorially combined with the vertical bearing force, the first three harmonics are equivalent to ±16,980 lbs, ±13,700 lbs and ±1,540 lbs. These values indicate the strong influence of cavitation on hull vibration.

3. To minimize the effect of cavitation on the hull, supplemental studies were conducted by F-D on the final propeller design, at the vacuum water channel, at Gotenburg, Sweden. Details of the propeller design and testing program were presented by Latron, in reference [31]. Correlation between theoretical force prediction, measured forces, and actual forces which may be deduced from full scale studies should contribute much to the evaluation of cavitation forces in the design stage.

Table 8 - F-D 125,000 M^3 LNG Ship with 5-Bladed Propeller

Comparison of Measured and Calculated Propeller Forces and Moments

	Model 4221A Measured NSMB (29)	Model 4171 Calculated DNV (30)	Model 4148 Calculated NKF (28)
V_s Knots	20	20	20
SHP_m	40,500	45,000	41,600
D Ft.	25	25	24.5
$\bar{T}$ Thrust, lbs.	460,760	520,290	451,600
$\tilde{T}_1$ ± lbs.	7,050	9,040	17,520
$\tilde{T}_2$ ± lbs.	880	7,500	
$\bar{Q}$ Torque, ft. lbs.	1,938,440	2,292,860	2,053,000
$\tilde{Q}_1$ ± ft. lbs.	23,150	33,270	56,660
$\tilde{Q}_2$ ± ft. lbs.	1,450	26,760	
$\bar{F}_v$ Vert.Brg. Force, lbs.	37,480	970	16,500
$\tilde{F}_{v1}$ ± lbs.	3,310	1,240	2,134
$\tilde{F}_{v2}$ ± lbs.	220	970	
$\bar{F}_h$ Hor.Brg. Force,lbs.	16,090	15,450	3,700
$\tilde{F}_{h1}$ ± lbs.	3,530	460	4,950
$\tilde{F}_{h2}$ ± lbs.	440	290	
$\bar{M}_{tv}$ Vert.Moment,ft. lbs.	475,930	318,980	
$\tilde{M}_{tv1}$ ± ft. lbs.	104,160	97,650	
$\tilde{M}_{tv2}$ ± ft. lbs.	9,400	60,760	
$\bar{M}_{th}$ Hor.Moment,ft. lbs.	528,010	73,600	
$\tilde{M}_{th1}$ ± ft. lbs.	26,760	31,100	
$\tilde{M}_{th2}$ ± ft. lbs.	1,450	23,150	

4. Finite element analysis of the hull for structural response was performed by Bureau Veritas. Although estimates of vibratory amplitudes were made, they were based on conservative damping coefficients and "maximum expected response" was determined, rather than predicted amplitudes. The major value of these calculations were to identify possible structural resonances, which might prove objectionable. One such potential problem area which was identified and corrected was the fore-and-aft response of the strut support for the propeller shaft bearing. Model characteristics of the deck house also provided the basis for stiffening, if required.

5. A vibration generator, which produced 13,200 pounds force at 9 Hz, was installed on the aft deck of the "Paul Kayser", the F-D LNG to physically determine the presence of structural resonances in the deck house and the aft portion of the hull. This work was done dockside in the shipyard.

   No structural deficiencies were determined by this process.

6. The vibratory characteristics of the main propulsion machinery were determined by both finite-element analysis and by conventional design procedures. Good agreement was observed between the investigators, for torsional, longitudinal and lateral shaft vibration. As is generally the case, the torsional critical was determined low in the shaft speed (42 RPM) and the longitudinal critical was determined to be above the operating speed, at approximately 145 RPM. The lateral shaft resonances were determined to fall in the range of 83 to 98 RPM, per reference [32].

   The subject of lateral shaft vibration requires special attention at this time. The presence of shaft whirl or lateral vibration of the shaft excited by propeller-blade frequency, has been calculated to fall in the upper speed range of a number of ships, and has generally been considered acceptable. Recent experiences gained on other large ships employing oil lubricated bearings and propulsion systems similar to that employed on the LNG Carrier has prompted an in-depth study of the misalignment and lateral shaft vibration characteristics of such designs. These studies have indicated that in some cases, the angular misalignment between the axis of the shaft and bearing, can exceed the tolerances of a long, fixed bearing, and the vibratory response of the shaft within the bearing can exceed the clearances of the bearing, in the vicinity of the lateral resonances. Further investigations are underway on this problem. In the meantime, however, recommendations have been made to avoid lateral criticals within ±15% of normal operating speed.

7. Full scale trials, schedules for July 1975, during the Builder's and Acceptance Trials of the "Paul Kayser" will include:

   Hull and Machinery Vibration by NKF
   Hull Pressure Forces by F-D
   Propeller Stress Measurements by IRCN
   Underwater TV of Cavitation by DNV
   Vibration and Noise Habitability by F-D & NKF

   It is expected that these studies will contribute significantly to correlation between theoretical predictions and actual underway vibratory characteristics of the LNG ships.

C. Avondale and Newport News LNG Designs

The following LNG ships include the Avondale design which is a conventional hull, approximating Model 4147 and the Newport News design which is also open transom stern, similar to Model 4148 and the F-D design. Both of these designs were studied in the new Vacuum Tank at NSMB.

Three cavitation tests were conducted on the Avondale Model. The first, with propeller model 4756 produced a vertical hull pressure force, of 40,250 lbs. The second, with an improved propeller (model 4833A), produced a force of 30,120 lbs. The third test included the improved propeller and the addition of a tunnel to improve the flow into the propeller. This resulted in a force of 7,700 lbs [33]. These modifications provided reductions of 25% and 80% respectively, from the original hull pressure force of 40,250 lbs.

The Newport News model, although having an open transom stern similar to that of the F-D, produced generally lower forces and moments than the F-D model, as well as a lower vertical hull pressure force [34]. A portion of this difference may be attributed to the difference in test conditions. The F-D model was tested in the open basin, while the N.N. model was tested in the new Vacuum Tank, both at NSMB.

The total test program, planned for all three designs, together with the extensive analyses conducted, should materially contributed to the understanding of the problems associated with the measurement and prediction of hull vibration on ships of this type. Of course, programs of this type, which ultimately rely heavily on empirical factors, require many more ship studies. It is on such data that the test program and publication of ship vibration data, recommended by Panel HS-7 and supported by the Hull Structure Committee of the S.N.A.M.E depends.

GENERAL OBSERVATIONS

An assessment of current shipboard vibration technology, with particular reference to the work carried out on the DD-963 and LNG programs, leads to some general observations, the more important of which are:

1. The primary effort to control shipboard vibration (hull and machinery) should be directed at the exciting forces, the major forces generally being related to those at propeller-blade frequency or harmonics of propeller blade frequency.

2. Having limited the exciting forces to acceptable levels, structural and/or mechanical resonances should be avoided in the important operating speed range.

3. Since many other design factors contribute to the final configuration of hull or machinery, technical impacts between hull and machinery characteristics must be considered, such as hull criteria and shaft RPM or the number of propeller blades and propulsion system resonances.

4. For a given ship design, one stern configuration could prove superior to another, as noted in the earlier LNG studies.

5. Design details of a given stern configuration can significantly influence the forces generated.

6. The presence of significant cavitation can magnify the hull pressure forces by factors greater than ten to one or increase forces response greater than at resonance.

7. Theoretically determined propeller and hull forces and moments may be used effectively in preliminary design.

8. The propeller forces and moments, obtained by measurement on the ship model, are considered more reliable than theoretically derived values.

9. Hull pressure forces and moments to assess cavitation effects can best be obtained in a vacuum tank.

10. The response of the hull girder and main machinery system can be estimated by the application of the propeller forces and moments applied to a suitable model by the inclusion of damping estimates and/or the application of service factors.

11. Considerable full scale testing, correlated against design predictions, are required to develop more reliable damping and/or service factors.

12. Finite element analyses are considered most useful for the design evaluation of major substructures and propulsion systems.

In a more general context it may be noted that in many cases in the past, the presence (or absence) of serious vibration aboard ship has been a matter of chance and serious vibration if present only corrected by major surgery, if at all. Although presently we are still a long way from the ultimate objective, there are many examples whereby problem areas have been eliminated or reduced to acceptable levels by improved design approaches. Some of the more common of such problems include torsional and longitudinal vibration of propulsion systems, dynamic balancing, shaft bending stresses and hull vibration caused by dynamic or hyrodynamic unbalance and cavitation. At this time, it seems safe to say, that our present technical knowledge has not been fully integrated into a design procedure. Too much is frequently left to chance retained in company files, or never fully evaluated for the purpose of improving our approach. Most of us could cite many examples of such design or management deficiencies which actually inhibit the development of improved techniques.

The initial steps for improvement are now underway. Detailed performance requirements have been specified in a number of cases, such as cited for those ships referred to in this paper. Such requirements not only identify vibration or stress levels which would normally be objectionable from habitability or stress point of view, but also provide a basis by which design approaches may reasonably be included in the cost of the ship. While vibration studies are not always specifically defined, we can already recognize the progress toward a more standardized approach.

FUTURE RESEARCH

The current test plan scheduled for the "Paul Kayser" of the LNG program, as identified earlier, includes:

(a) Hull and Machinery Vibration

In addition to the conventional hull and machinery vibration measurements prescribed by Code C-1, the "Code for Shipboard Vibration Measurement" [3], which will be used to correlate actual ship and machinery response against predictions, the following supplemental measurements will be made.

1. Alternating thrust in the propeller shaft.

2. Fore and aft vibration of the strut

3. Shaft motion within the strut bearing (both ends)

4. Oil pressure to the bearing and sea water pressure in the oil seals to the strut bearing.

(b) Hull Pressure Forces for correlation with predicted forces obtained by calculation and model testing.

(c) Propeller Stress Measurements plus alternating torque and thrust to correlate actual propeller forces against calculated and measured values.

(d) Cavitation studies by underwater TV, for correlation with laboratory model studies.

(e) Vibration and Noise Habitability measurements for comparision with existing or proposed standards.

This program, which is largely supported by the El Paso Gas Company, will contribute much to an understanding and evaluation of current design procedures. However, an in-depth study of a single hull is inadequate and the extension of the test program to the follow-on designs is needed to develop reliable design data applicable to the LNG Carriers. Similar programs of study are considered necessary on other basic designs to gain sufficient empirical data required to obtain the ultimate design procedures required. In this regard industry-wide support of the HS-7 Vibration Panel's program for obtaining and publishing, in standard format, the vibration characteristics of all new ships, is strongly recommended.

In the hydrodynamic area, it is considered necessary to obtain the preferred configuration for a given ship class, to optimize the design details, to minimize the adverse effects of cavitation, and to obtain reliable input forces and moments to be used for dynamic analysis. While it may be said that the means for carrying out these studies are available in one form or another, the application of this information, by the average designer appears to be somewhere between an art and a research program. The development of a standard or recommended procedure which will provide the desired results at minimum cost, is also strongly recommended.

Another significant contribution to ship vibration research was the "Highly-Skewed Propeller Research Program" recently carried out on the San Clemente Class Ore Bulk Oil (OBO Carriers) [34]. This program primarily sponsored by the Maritime Administration, explored the use propeller skew as a means of reducing hull and machinery vibration. As was concluded, "Skewed Propellers are useful tools for reducing vibration problems but they are not a panacea that can be used blindly". Further study is recommended on this subject to determine when and how to apply the skewed propeller to advantage. It is suggested however that highly skewed propellers might appropriately be limited to those applications in which conventional design techniques will not achieve the desired results or to those in which minimum vibration and noise are a requirement.

The HS-7 Panel compiled a list of seven individual recommended research projects, which were subsequently endorsed by the Hull Structure Committee in 1972. These projects, which would also include the efforts of the Hydrodynamics and Machinery Committees are identified under the following titles:

- HS-7-1 Vibration Specifications
- HS-7-2 Vibratory Propeller Forces
- HS-7-3 Hull Frequency Determinations
- HS-7-4 Dynamic Response of Ship Hulls
- HS-7-5 Dynamic Response of Main Machinery Systems
- HS-7-6 Vibration Measurement and Analysis Procedures
- HS-7-7 Design Guide for Shipboard Vibration Control (Interim)

The objective, plan of action and end product has been defined in each case. At this time the HS-7 Panel is engaged on the first project "Vibration Specifications". A similar effort is underway by the M-20 Panel (Machinery Vibrations). While the research panels of the S.N.A.M.E. have accomplished much in the past conducting research by part time contributions of panel members is painfully slow. It is recommended, that more aggressive action be taken by the industry as a whole, in support of these projects.

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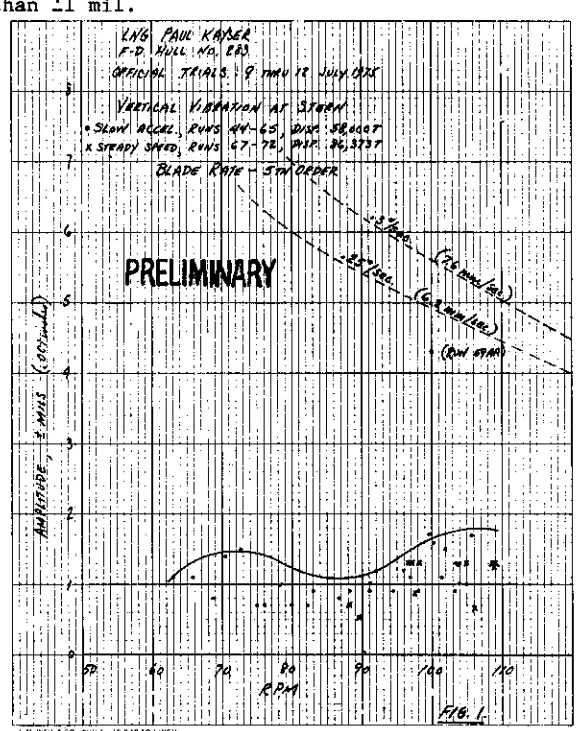
SUPPLEMENT

Preliminary Report on Vibration Survey El Paso (Paul Kayser)

At the time this paper was prepared, the LNG Carrier, El Paso PAUL KAYSER, had not yet put to sea. The vibration trials were conducted during the Builder's and Acceptance Trials, in July 1975. Results of that trial, based on the Preliminary Report of August 12, 1975 are briefly reported here, for information purposes.

Evaluation of Hull and Superstructure Vibration

The vibration of the hull did not exceed 50% of the NKF recommended criteria, when delivering 45,000 SHP. This information was based on the performance, at 86,000 tons displacement, with clean bottom, obtained during the Official Trials, at a speed of approximately 21.2 knots. This data was used and based on the "Steady-speed" runs, conducted during the Official Trials, as noted by "X" on Figure 1, which shows the Vertical Vibration at the Stern. This location, close to the aft-perpendicular, is normally considered as the basis of judgement for evaluation of hull vibration. Both the Athwartship and Fore and Aft (Longitudinal) vibration measured at this location, were less than ±1 mil.



Supplemental data were also taken during the slow acceleration runs, from 63 to 106 RPM, to provide "fill-in" data. The complete analysis to all vertical vibration data obtained at the stern, for both the steady-speed runs, and the slow-acceleration runs, shows a single point (Run 59AA) in which the maximum amplitude exceeded all other points by a factor of better than two. This point is shown on Figure 1, together with all the other data, and the NKF recommended criteria for the hull (.25"/sec) and major substructures, such as the deckhouse (.30"/sec). For evaluation purposes, the forced vibration of the hull was assessed as being less than 50% of the suggested criteria.

The only other significant hull-superstructure vibration noted, was associated with the fore-and-aft (longitudinal) motion of the deckhouse, at the bridge level. These data are shown on Figure 3 and indicate a very sharp resonance of the deck-house, based on the response of the wheelhouse and starboard bridge wing fore-and-aft pickups. Superimposed on this Figure, are a few points observed on the base of the Radar Antennae Platform, after the significant vibration of this location was noted, in the vicinity of 100 RPM.

Figure 5 shows the fore-and-aft rocking motion of the deck house, coupled with the vertical flexure of the hull, as deduced from the test and Hull Vibration Analysis using the 20 Station Beam Model. The phase relationship between the vertical vibration of the hull forward of the deck-house and that measured at the stern corresponds with the calculation. A deduced vertical amplitude of ±.7 mils, forward of the deck-house would result in a vertical amplitude at the aft end of the deck-house of ±4.2 mils and a fore-and-aft rocking amplitude of approx. ±6 mils. Since the maximum fore-and-aft motion exceeds this value, some wracking of the deck-house is implied.

Figure 6, taken from Reference (b) shows a forced response for the vertical motion of the stern (read the maximum values) obtained by the beam analysis. Also shown is the measured values obtained during the Official Trials and the corresponding full-power amplitude at the same location, from Bu. Veritas calculations. For comparison purposes, the test data gives <u>maximum</u> values while the calculations give an <u>average</u> value, which would be about two-thirds as large.

The tentative conclusions reached on the hull and superstructure vibration, were as follows:

1. The general level of forced hull vibra-

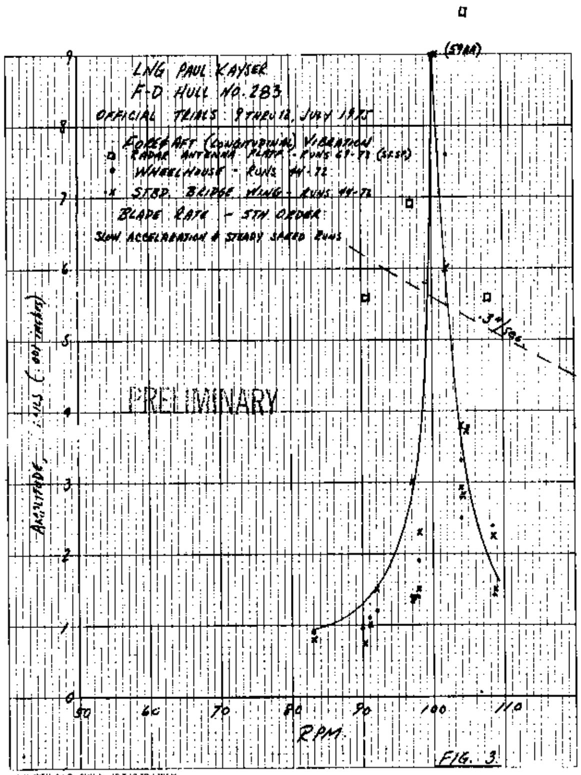
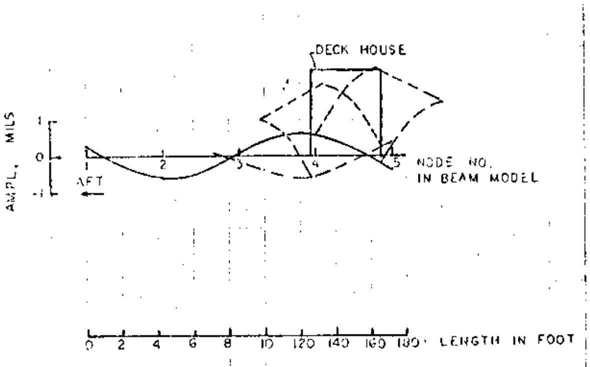


FIGURE 5

DERIVED COUPLED VERTICAL-FLEXURAL MOTION OF DECK HOUSE

REF: OFFIAL TRIAL DATA & BEAM ANALYSIS, REF.(b)



tion, as measured at the aft perpendicular, is well within the previously recommended hull criteria.

2. A sharp resonance of the deck-house, in the fore-and-aft direction, occurs at 100 RPM and is coupled with the vertical response of the hull.

3. The torsional motion of the radar platform reflects a sympathetic resonance to the sharp fore-and-aft resonance of the deck-house.

4. The sharp resonance associated with the coupled fore-and-aft motion of the deck-house and the vertical vibration of the hull can be readily avoided and would not appear to pose a problem in normal operation.

5. To more effectively evaluate this sharp resonance, and avoid it, if possible, in future designs, a more detailed measurement on the second ship may be required.

Longitudinal Vibration of Main Machinery

The full power alternating thrust, maximum values, are shown on Figure 7. These values represent peak values and indicate the 5th order resonance to be above operating speed, as predicted. As a preliminary estimate, allowing for signal modulation and resonant magnification of two to three, on the forward end of the resonance slope, a more detailed analysis would confirm the predicted alternating thrust of approximately $\pm 7{,}000$ pounds.

FIGURE 6

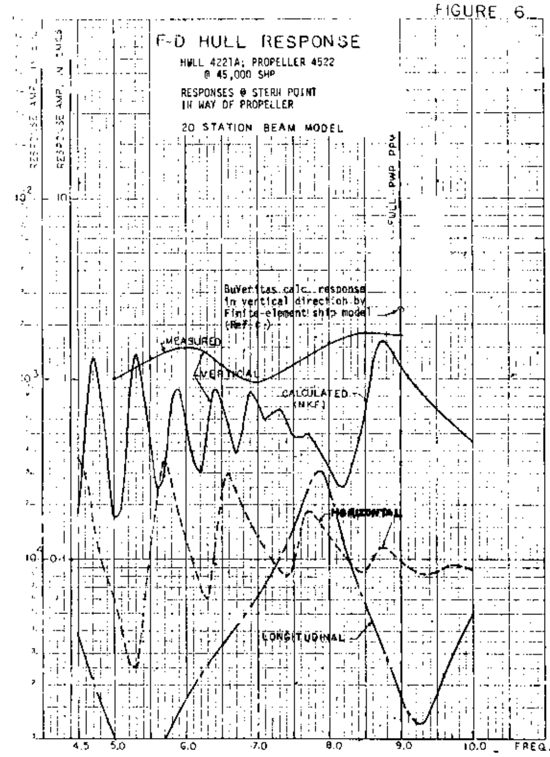
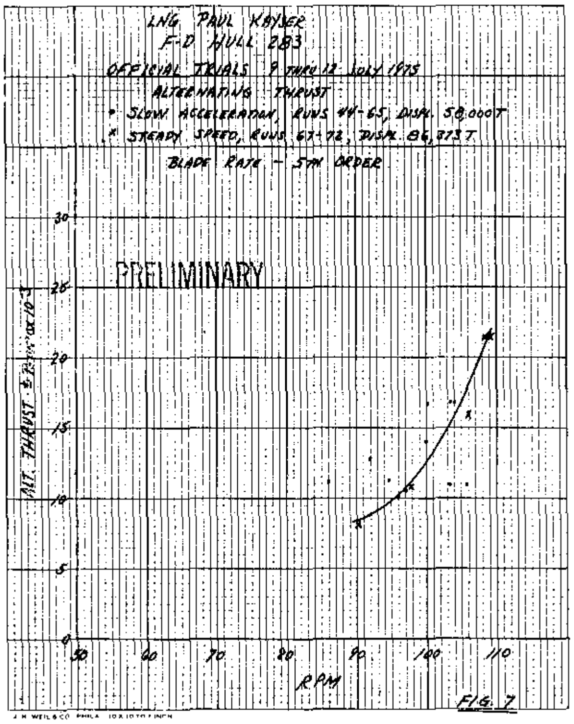
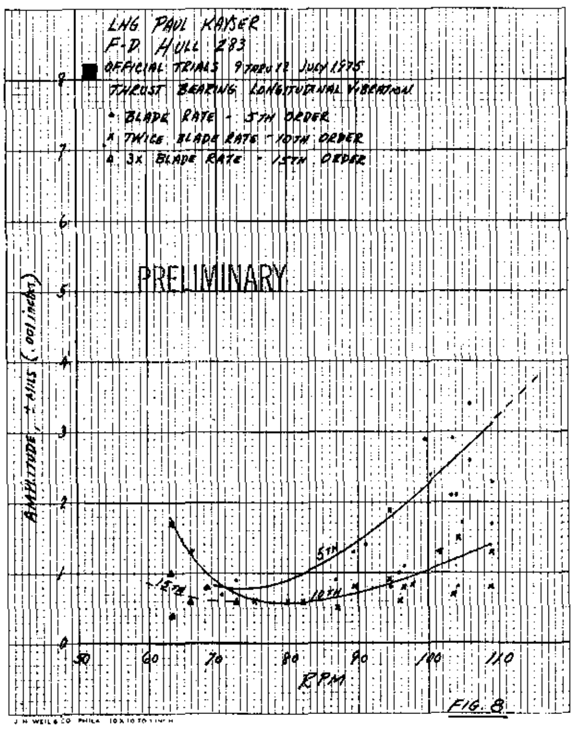


Figure 8 shows the longitudinal vibration of the thrust bearing and also indicates the 5th order longitudinal resonance of the propulsion system is above the operating speed, as predicted. The amplitudes are considered acceptable. However, further analysis is required to correlate predictions with response.

Proximity Shaft and Oil Pressure Measurements

As previously noted, the special studies involving the shaft motions within the strut bearing and the oil and water pressures in the strut bearing were lost. It was noted however, before the cable carried away, that blade-rate was present in the strut bearing proximity gages, while only shaft frequency was noted at the forward end of the stern tube bearing. Further information, if available, will be included in the final report on this study.





Supplemental Studies

Preliminary results of the Hull Pressure Forces obtained by IRCN & F-D, the Underwater Photography obtained by DNV, and the Vibration Measurements obtained by NKF were reviewed at a meeting held in Amsterdam in September. This meeting was also attended by Bu. Veritas and NSMB representatives and plans were generated for supplemental data requirements necessary for correlation studies between analyses and test results. Plans were also initiated for further work considered necessary to eliminate the superstructure resonance in this ship, if possible, and in the follow-on hulls. The propeller stress measurements were not accomplished at this time but are planned for the future.

Discussion

G. C. Volcy, Member

Having the pleasure to know Ed Noonan personally and professionally for several years I must congratulate him heartly for this paper and thank him for the invitation to contribute.

His paper is very timely indicating the present flaws in the state of the art in ship vibrations as well as the way to overcoming them, backed by concrete examples.

In fact it is a gold mine of technical information coming from his long experience in the concerned field and I seize this opportunity to pay homage to him for all his strenuous efforts in the struggle against ship vibrations, which are the "cancer of mechanics" in different specialized panels in which he is for a long time active. One of them is ISO Ship Vibration Working Group, where we are working together and where I could observe and appreciate his activity which has contributed in a large extent to accelerate the completion of several documents such as a Code for Measurements and Reporting Shipboard and local vibrations, Interim guide lines for acceptable vibrations limits,...

In this paper, as in the daily behaviour, Ed Noonan is using frank and plain language, having the courage to clearly explain his point of view even if this does not please to all present persons. I agree fully with him when he says that in spite of the tremendous progress in the study of ship vibrations it is still an art. I think also that being now liberated from burdensome calculations, now accomplished by speedy computers, the most important thing is the good understanding of the physical side of encountered phenomena and preparation of correct input data. If not and if "garbage" is put into computer "garbage" must come out... and deceive the involved people. It is my personal experience which has led me to introduce the notion of forced vibration resonators, the corresponding philosophy being exposed in a publication, see Reference (1).

In the described studies Ed Noonan has been lucky since for his destroyer theoretical studies he was in possession of correlation data used for evaluating input data for his calculations. In fact it is of utmost importance to have the results of previous calculations of input data related to stiffnesses, equivalent masses, empirical correction factors concerning excitation forces and results of consecutive measurements executed on similar or not very different ships. In my Special Study and Research Section of BUREAU VERITAS we have executed hundreds of simplified calculations, which results have been rather satisfactory and this because we could provide my Calculation Team with reliable input data deduced from previous measurements executed by my Measurement Team. But for cases which differ greatly from conventional ones, I did arrive to the conclusion that it is better not to over rely on extrapolation. This also, because for such cases, that I have had and continue to have the chance to meet, the clients come with controversable results executed on basis of simplified and even over-simplified calculations, what is throwing them into confusion and compelling them to take a decision which is full of potential, technical and financial consequences.

If you are requested to give a binding advice and execute calculation for ships never built before such as SL 7 container ships, 350 K dwt or 500 K dwt and more tankers, or three engine driven ro-ro ships equipped with biggest in the world (42 500 HP) cp propeller or even unconventional LNG tankers to be equipped with overlapping propellers such as the one mentioned by the author in Reference (8), honestly speaking you will hesitate to make hasty considerations.

Then the answer must be the most reliable possible hence the necessity of sophisticated and not oversimplified finite element calculations in order not to deceive the clients and friends.

This was also the case of FRANCE DUNKERQUE EL PASO ships the structure of which has no similitude with previously known to us ships and which steel-work especially in way of thrust bearing foundations, put at first on simple bottom, have been radically corrected by the Shipyard, and with endorsement of EL PASO technical staff, according to my recommendation well before starting the finite element calculations.

These calculations, to which the Author refers in item 4 of page 39, and mentioned also by Mr. Latron, see (31) of paper References, concerned the detailed modelisation of aft part, engine room and superstructures of concerned LNG tankers, the forward cryogenic part being included also into free and forced vibration calculations as equivalent elements, have been executed following our philosophy exposed in (1) and according to calculation program related to the integral treatment of static and vibratory phenomena of engine room and propulsive plants presented at ATMA in 1974, see Reference (2). The results, including also the calculation of free and forced vibrations of lateral and precession (whirling) and longitudinal vibrations of the propulsive plant have been presented in five distinct reports. I am also glad to learn that the calculations of free shafting vibrations executed by NKF, using our thrust and shafting bearing stiffnesses have been in general agreement with those obtained by us. Moreover, the results of sea trials of EL PASO PAUL KAYSER which look to give satisfaction to Shipowner and Shipyard have confirmed the efforts, time and money involved into these studies, executed by calling for our philosophy, have not been lost.

I agree with the Author that such complete studies cannot be executed at the early stage of the project when lot of crucial decisions must be taken. But, having at first executed a considerable number of simplified vibration calculations for conventional ships, and afterwards enough numerous complete calculations of aft part and engine room of big VLCC and ULCC, LNG and Ro-Ro container ships, we could obtain a better insight on the mutual interaction of different hull and machinery sub-assemblies, backed in particular by the results of shipboard measurements.

Now, on the basis of previous experience, we are preparing the so-called "compact model" for vibration calculations which should maintain a sufficient degree of credibility and provide their quicker and cheaper execution. But besides such model, we feel, like the Author, the urgent need of more reliable data related to different types of damping coefficients. Such studies are also under way in my Special study and Research Team.

Before ending I would like also to add my agreement on the Author's opinion in respect to reliability of hydrodynamic excitation values, a subject on which unfortunately it seems to exist some doubts and even confusions.

One does concern the influence on hull surface efforts of cavitation phenomenon. It is impossible to work when the different hydrodynamicists are showing you that this type of effects may vary from two to fifteen times! I have had the occasion to assist at such discussions and frankly speaking it looked to me that often they are not understanding themselves, not speaking of the same thing.

As to results to all our complete calculations of forced vibrations, where the hydrodynamic excitations have been determined by NSMB, even with cavitation, the participation of hull surface efforts to the vibratory level of superstructures has never exceeded 30%, the rest being due to the six components of propeller efforts. Regarding the last one there is also an other confusion! Due to the action of the propeller in the ship's wake it occurs, besides hull surface efforts, the six components of propeller efforts (forces and moments) and not the bearing forces as it is often written. In fact, the bearing forces are function not only of propeller efforts but also of dynamics of line shafting and characteristics of bearings and associated hull steel-work. These bearing forces are, of course, different from the propeller forces, evaluated by hydrohynamicists, and I would rather object to the oversimplification of hydrodynamic excitation phenomenon, as mentioned in the paper, by substracting them (even by taking into consideration of their respective phases). For example of calculation of bearing forces being the function of propeller efforts please see Reference (8) of the paper.

Once more my best compliments for this very valuable and timely paper.

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F. Everet Reed, Member

Mr. Noonan's paper is excellent and raises a lot of points that must be faced relative to ship vibration. It often seems that the progress in predicting and avoiding ship vibration is far behind the available technology. However, it must be recognized that the vibration level on the ships that are now being built indicates that either we are applying more knowledge than we realize or else many of our past concerns about vibration were unwarranted. However, there are some ships in service that vibrate excessively, and these are a constant concern and expense to their operators.

The author lists in good order the procedures to be followed in a vibration analysis. Hopefully as these procedures are applied, the indefinite empirical factors (2.d) will become continually less important.

I am pleased to see that the author has proposed that vibration acceptance levels be expressed in terms of vibratory velocity. Not only are the limits of human response most simply expressed in terms of the harmonic velocity of vibration, but also the vibratory stress as I will endeavor to show by a simple model.

Consider an idealized uniform Euler beam simply supported at the ends. The equation of the deflection of this beam when subjected to harmonic excitation is:

$$\gamma = A \sin \frac{\eta x}{\ell}$$

Where

$$\eta = \sqrt[4]{\frac{m\,\omega^2}{E\,I}}$$

m = mass per unit length

ω = circular frequency of vibration

E = modulus of Elasticity

I = Moment of Inertia of the cross section

The bending moment in the beam becomes

$$m = EI \frac{d^2\gamma}{dx^2} = -AEI\eta^2 \sin \frac{\eta x}{\ell} \cos \omega T$$

and the stress

$$\sigma = \frac{MC}{I} = -AEC\,\eta^2 \sin\frac{\eta x}{\ell} \cos\omega T$$

$$= =AEC\omega\sqrt{\frac{m}{EI}} \sin\frac{\eta x}{\ell} \cos\omega T$$

C is the distance of the remote fiber from the neutral axis.

Now - $A\omega\cos\omega T$ is the vibratory velocity, υ, of the bean and so the maximum vibratory stress where $\sin \frac{\eta x}{\ell} = 1$ is

$$\sigma_{max} = \upsilon_{max}\, EC\sqrt{\frac{m}{EI}}$$

This applies for all natural modes.

To find out how large $EC\sqrt{\frac{m}{EI}}$ is, I have run through a rough calculation for a 720 ft. container ship, assuming midship properties to suit ABS rules and find that σ is approximately equal to 700υ. Similar checks for other structures will show that for most vibrations the vibratory velocity is a good measure of stress and that the stresses are generally low. Thus, the acceptable levels of vibration that are proposed are those imposed by the tolerance levels at which people can work effectively.

In the stated design objectives and acceptance levels it is difficult to understand why higher vibration levels can be accepted in living quarters which are in major substructures than in the steering gear flat which is a part of the hull girder. In general, although the hull girder may be reasonably well defined in a multicompartmental combatant ship, on a commercial ship it is much more difficult to know what should be called the vibration of the hull girder. There can be a wide variation in vibration amplitude across the breadth of the ship even over a bulkhead. It would appear reasonable to forget all distinctions as to structural locations and then specify the acceptance vibration level in terms of the tasks and equipment located in a specific location. It would appear reasonable to reflect the length of time that personnel are working in a particular location.

It would seem desirable to give the shipbuilder as much freedom in meeting operational requirements as are acceptable to passenger, crew, equipment and structure. Although the procedures for estimating the vibration in well-compartmented combatant ships are fairly well developed, the same cannot be said for commercial ships and a shipbuilder faces problems in bidding on tight specifications for vibration in high-powered ships.

The N-20, Machinery Vibration, Panel is developing a code for acceptable vibrations in machinery. This is somewhat more detailed than the specification suggested by the author and is intended to serve as a guide to machine manufacturers in the design of machinery for ships.

E. Mogil, Visitor

The author has well stated the status of current technology for design, analysis, and measurement of shipboard vibration. He is to be commended on an excellent presentation.

The success of the DD963 in achieving a low level vibration environment was largely due in part to the support provided by NKF Engineering and to a positive vibration design approach. As outlined by the author the DD963 program included specific limits to satisfy design objectives. Consistent with this, preliminary hull and machinery vibration analysis during the system design phase minimized changes during detailed design and resulted in final test results with little or no corrective action required.

One of the more significant recommendations cited in the paper is the need for positive design criteria and limits. Criteria such as "Excessive Vibration" rely too much on subjective interpretation and,without a quantitative value for guidance, creates too broad range of acceptable limits for design and compliance demonstration. The limit of ± 0.25 g as being the lower bound for "excessive vibration" as defined for the LNG carrier appears to be a resonable value for machinery self-excited vibration.

Continued effort in standardizing the design criteria, analysis techniques, and measurement methods (compliance demonstration) for controlling shipboard vibration is highly desirable and should be pursued aggressively.

. The collection and dissemination of related studies on shipboard vibration from a central source (or data bank) would also benefit all users and interested organizations. As indicated in the paper, there are numerous codes, panels, and international groups pursuing similar and closely related vibration studies. The seven research projects indicated in the paper are highly endorsed by this reviewer. Design guidance manuals can be a strong cost-effective tool for those ship designers who must meet the more stringent vibration requirements of todays military and commercial ships.

Paris Genalis, Member

Ship designers and analysts owe the author thanks for his paper documenting the application of vibration technology to ship design and illustrating it with remarks about the DD-963 and the 125000 CM LNG carrier design.

This discusser would only like to contribute a remark of caution regarding the authors comments on the use of finite element analyses during the design cycles to predict the vibratory response of ship substructures.

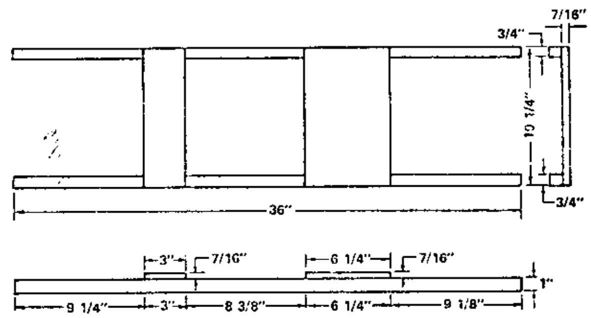
There is no question in this discusser's mind that the finite element technique is a powerful tool capable of handling extremely complicated problems which would have been impossible to approach without the availability of this method.

The finite element method though, requires as an input the geometry of the structure to be analyzed. In early design stages such detail as required for a proper vibratory response analysis is not available. As a result one is forced to make assumptions about the structural details of geometry and boundary conditions. Based on those, the finite element method predicts the response. The results are therefore, by necessity, only as good as the input assumptions.

If on the other hand a detailed structural geometry exists, indicating that the design has progressed to quite an advanced stage, the analysis can be quite accurate, but quite expensive. As an example let me site the analysis of the ASR-21, performed for static loads. Figure 1 (from Ref. 1) shows the finite element approximation, which, complicated though it appears, is really quite crude compared to the real structure. Approximately six man-months were required to produce numerical data for this idealization. More modern techniques could no doubt cut some time off this effort, but still, the data preparation time is extensive, even if one does not consider the man-hour effort of data preparation.

As an alternative, it has been suggested that simpler equivalent structures be analyzed, again by finite elements. Figure 2 (from Ref. 2) shows the idealization of the same ship in a much cruder way, and Table 1 (Ref. 2) shows the comparison of measured and computed frequencies.

Figure 1



SPLIT CROSS-STRUCTURE CATAMARAN MODEL

Fig. 2

The measured frequencies were for a physical model, of a geometry identical to that of the mathetical model.

TABLE 1

Frequency Number	Experimental Hz	Computed Hz
1	108	106
2	169	169
3	225	227
4	381	398
5	477	481

The table clearly shows that the finite element method can predict frequencies very well. Space does not permit me to show the agreement of the modes, but that also was excellent.

Note however that the prediction was for the model, not the full-scale ship. The question still remains: how do you simplify the real ship to something as simple as shown in Figure 2?

This discusser believes that a designer would rather deal with a simple structure (as in Figure 2) during the early design cycles if he knew how to simplify the real structure. Such analysis is much cheaper and fits the early design cycle much better, since no structural arrangement is available yet.

With this in mind, it is recommended that some effort be expanded in understanding the processes of simple mathematical model formulation in the early stages of the design. Finite element analysis of such simple models can guide the designer through the preliminary design stages. When structural detail becomes available, a final detailed check of the vibratory response can be carried out, also by the finite element.

References

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NSRDC report 3780

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Some Examples of Vibratory Response of Long Ships and Special Craft
Paper presented at the International Symposium on the Dynamics of Marine Vehicles and Structures in Waves
1-5 April 1974, University College London

A. Zaloumis, Member

As expected, Mr. Noonan has presented a logical, straightforward approach to the problem of considering the vibration aspects of a ship while in the design stage. His ability to do so is, or course, enhanced by his many years of experience in this field and his knack of being able to convey his thoughts in uncomplicated language. It should be noted that at the David W. Taylor Naval Ship Research and Development Center (DTNSRDC) a very similar design analysis procedure is followed for all ship designs.

The Navy's criteria for allowable vibration is contained in MIL-STD-1472A (May 1970) and accounts for the effects of vibration in terms of comfort, proficiency and safety. In attempting to relate the MIL-STD criteria to the proposed criteria by Mr. Noonan, the MIL-STD Comfort levels were chosen. (The allowable levels for Proficiency considerations are 3.15 times those for Comfort; the allowable levels from a Safety standpoint are 6.3 times those for Comfort.) Figure 1 shows a comparison between the Navy's criteria and the author's proposed criteria for the LNG and DD 963. It is

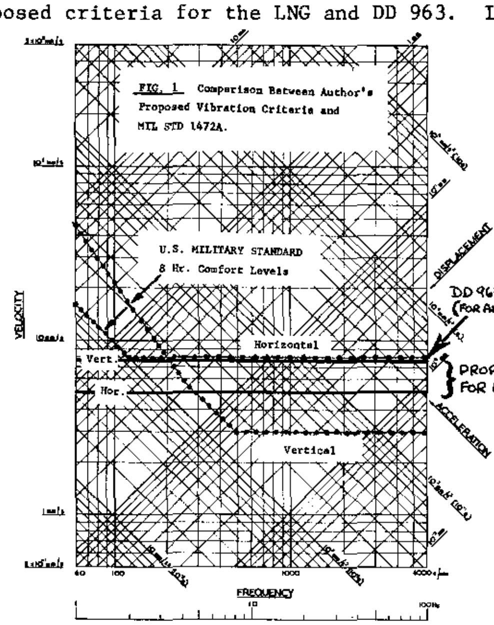


FIG. 1 Comparison Between Author's Proposed Vibration Criteria and MIL STD 1472A.

interesting to note that the Navy criteria allows more vibration for the horizontal direction than for the vertical direction for frequencies in excess of 3 Hz, whereas the author's proposed criteria for the LNG is just the reverse. For the DD 963, the author's target levels are the same for any direction of motion. It would be appreciated if the author would elaborate on this point.

I would now like to comment on Mr. Noonan's remarks concerning the degree to which the ship's hull should be modeled for a vibration study.

In conducting a vibration analysis of a hull at DTNSRDC, the mathematical model used is the conventional 20 station lumped-mass type. This method is considered quite adequate for the lower modes, say up to the fourth or fifth mode. Anything beyond that is quite academic since the hull no longer responds as a free-free beam but instead goes thru a series of local vibrations, which theoretically, are infinite in number. Any attempt to model that kind of dynamic behavior would be futile and quite unnecessary, even if possible, not to mention the extensive time and cost. Of course, major sub-systems such as the propulsion system or deck-houses should also be analysed to ascertain their vibration characteristics but these can be done on an independent basis.

With the advent of large-capacity, high-speed computers, there appears to be a trend towards 'overkilling' the mathematical model of a ships hull such as shown in Figure 2. There are several reasons why I disagree with such an approach.

Figure 2

1. First of all, for a vibration study to have any utility, it should be performed in the early design states when changes can be made without significant impact on other design considerations. As such, a ship's hull would not have the detailed structural definition in the early stages that would be needed for a finite-element representation. In the later stages when the hull is more structurally defined, it is usually too late for such an analysis.

2. Propeller forces,which act on the propeller bearing and on the stern through pressure pulsations, are at best extremely difficult to predict with a reasonable degree of accuracy. It would be foolish to develop a very elaborate finite-element model and subject it to a single-point force excitation - which in itself is quite often an educated guess. Furthermore, very little is known about the damping of such finite elements especially at the interfaces of the various subsystems with the hull, not to mention damping due to entrained water.

3. In most cases, sub-systems such as propulsion systems, struts, decks, etc., have impedances that are sufficiently different from the hull to permit treating them as independent systems. Furthermore, this enables the designer to analyse a given sub-system when enough information is available on it rather than waiting for all systems to be sufficiently detailed for a complete finite-element study.

4. The state-of-the-art on hydro-elasticity precludes the consideration of the effective mass of entrained water in a manner that would be commensurate with the degree of the structural finite-element model itself. Additional problems with mass loads arise when attempting to account for the myriad of machinery, equipment cargo, etc. that is dispersed throughout a ships hull.

Perhaps in years to come, finite-element analyses of a complete hull may be a commonplace tool for vibration studies of ships in the design stage, but as of now they should be given very limited application.

Norman O. Hammer, Member

Every once in awhile, the wine industry has what is known as a "vintage year". That is, a year in which wine is produced of outstanding quality relative to that produced in other years. In August 1975 at the STAR Symposium the paper entitled "Highly Skewed Propeller for San Clemente Class Ore/Bulk/Oil Carrier Design Considerations, Model and Full-Scale Evaluation" was presented. And today, the paper entitled "An Assessment of Current Shipboard Vibration Technology" has been presented. I think those of us involved in ship vibration technology can say that 1975 will be known as a "vintage year".

While each paper reports on work that has been underway for several years and outlines two independent efforts to reduce unwanted ship vibrations, I believe each paper will have a profound affect on future vibration technology. Also, I believe that each of these two outstanding projects had a number of common elements contributing immensely toward their respective successful outcomes that needs to be highlighted here today.

First each effort had the full support of all participants in the work from the start, most importantly the shipyard and the shipowner. Second each effort was comprehensive in nature including: (1) model experimental work, (2) analytic work and (3) extensive full-scale verification work. In fact the full-scale verification work outlined by Mr. Noonan on the El Paso "Paul Kayser" is one of the most outstanding efforts of underway ship testing that has ever been accomplished on any merchant ship. About the only thing missing from this comprehensive full-scale test program was the measurement of long-term propeller blade erosion due to cavitation, which is a research program in its own right.

I think based upon examination of the paper presented today a number of questions can be posed, as follows:

(1) Are those of us involved in the design/construction process doing enough to increase the chances of successful performance of new ships with regard to vibration?

(2) Are the efforts to reduce vibration comprehensive in nature (e.g. covering model work, analytic work, and full-scale work)?

(3) Are the U. S. model test facilities and response times adequate to insure that needed model experimental work can be accomplished domestically?

(4) Are individual shipowners, and others in the maritime industry willing to support the work needed to achieve relatively vibration-free ships? This means support in terms of advice, cost sharing and availability of ships.

We have only to look to the recent past to find many examples of failure. It is hoped that based upon the work reported today in the "vintage year" of 1975 we will recognize that it is possible to increase the number of successes if the effort is made at the start of precontract negotiations and at time contract signing for new vessels.

Author's Closure

I would like to thank Mr. Volcy for his discussion which, I feel, adds to the value of the total material presented here today. In general, I interpret Mr. Volcy's remarks as being in general agreement with the material presented, supplemented by his own experiences. From the remarks presented, I have identified two points on which I might offer some additional comment.

The first concerns simplified or over simplified calculations and extrapolations. To be sure, if all the necessary inputs are available, a reliable program exists, the design can be considered conventional, and confirming full-scale tests on similar designs are available, there should be little difficulty in performing the analysis and reliably predicting the results. Unfortunately, however, this is seldom the case in the preliminary design phase. In new, or unusual designs, in which much of the required input data is either missing or subject to change it is necessary to rely on simplified approaches to make early decisions. The value of the results obtained by such simplified analyses is, however, largely dependent on the experience of the investigator. The application of this experience to new or unusual designs or problem areas is, therefore, referred to as an art. To employ extensive analyses when inadequate data is available, will produce, as Mr. Volcy indicates, only garbage.

More significantly, in the case of the LNG, in which Bureau Veritas performed very valuable finite analyses of the ship's afterbody, these analyses were performed after the hull form was selected and the structural details were completed. I believe the important point to make here, is that the vibration characteristics of a design are dependent on many factors and that, at this time, it is not sufficient to satisfy the problem needs for a shipbuilder or designer to obtain a wake survey, establish propeller characteristics, select a structural arrangement and pour it into a computer. The cookbook approach is not yet available, nor will it be until he can calculate, test and consistently confirm the calculations.

The second comment concerns the exciting forces. To avoid any misunderstanding, we refer to "Bearing Forces" and "Hull Pressure Forces". Bearing forces, are, as Mr. Volcy indicates, the six components of propeller effort (three forces and three mements) which enter the ship mechanically thru the shaft and bearings. Depending on the model and the analysis used, these forces and moments may be directly applied, or corrected by the dynamic magnifier generated by the intervening structure. The hull surface forces are another matter. Approximations can be made, based on experience, or approximations can be estimated by computer programs. In either case, however, the true forces are largely altered by the effects of cavitation and the order of magnitude can be so significant (10 to 20 times the estimated or calculated values are not unusual) as to render the total input forces questionable in many cases. Considerably more effort is considered necessary in predicting the input forces with the degree of cavitation expected. When is the data useful and when is it garbage? This is probably the most significant problem present in ship vibration at this time.

Mr. Reed's discussion is a welcomed contribution to the subject of Shipboard Vibration Technology which we are concerned with at this time. His contribution supports the use of vibration levels defined in terms of vibration velocity.

Mr. Reed suggests it would appear reasonable to forget all distinctions as to structural locations and then specify the acceptance vibration level in terms of the tasks and equipment located in a specific location. This I can concur in, as long as we are dealing solely with capability of men and machines of effectively dealing with the vibratory environment. There is, however, another consideration, and that is, as stated in the Code for Shipboard Vibration Measurement, "to provide a basis for design prediction, improvements, and comparison with vibration reference levels or acceptance criteria". In this respect, the vibratory response of the hull girder, as measured at the aft-perpendicular, is still considered as the most effective basis for such evaluation. The forced response at the stern is still, in the opinion of the author, as the best measure of the "efficiency" of the design, in regard to the hull form, propeller characteristics and resulting girder response. Any other location is significantly effected by local structural design details.

Discussions in this area can generally emphasize the "art" in shipboard vibration engineering. The stern measurements are considered basic to any investigation of excessive or troublesome structural response. It is the opinion of this investigator, that the girder motion, when judged against a set of guide-lines, and the local response of substructures or structural elements, with reference to this motion, are basic requirements to the problem identification and to the determination of the effectiveness of alternate solutions.

In dealing with the code of acceptable vibrations in machinery, caution is recommended. At this time considerable work is underway in the Shock and Vibration Committee of the International Organization for Standardization. In most cases, vibration limits are based on internally excited forces and the effect of these forces on the life of machine components, particularly bearings. The effect of a vibratory environment, superimposed on these forces, is not too well known at this time. It is recommended that close collaboration be maintained with the ISO program in this area.

I would like to thank Mr. Mogil for his comments and strong support for the content and recommendations offered in the paper. Of particular interest is Mr. Mogil's support of vibration criteria or specifications relative to the control of vibration. Although most builder's are inclined to object to the restrictions imposed by such specifications this is the only effective method of obtaining the improvements desired. It is important, however, that the specifications be realistic and practical to achieve. Since a responsible shipbuilder will normally plan to conduct some reasonable vibration control program, such specifications tend to define the scope of the effort, enables them to place reasonable cost estimates on the work and avoids the handicap that may result when a competitor omits the item and "takes his chances". When this happens, the problem is frequently resolved by lawyers, rather than solved by engineers.

The discussion presented by Dr. Genalis is extremely welcomed. It does two things. First, it demonstrates complete agreement with the observations made, and the examples given in reference 7 of the paper, and in the DD-963 program, that finite-element analysis is not warranted during the preliminary design cycles. Secondly, it points up a possible problem of interpretation. Perhaps this can be clarified by enlarging on item 12 of my general observations relative to the use of finite element analyses of major substructures and propulsion systems.

It was the intent of this point to convey the understanding, that when the substructure or propulsion system design has been firmed up, that a finite element analysis of the limited system, as opposed to the complete hull, substructure and propulsion system, can provide more useful information than the analysis of the complete package, which some designers prefer. It has been our experience that the total package is too large, too expensive to run, and the results less accurate than those obtained by analyzing the individual substructures in which greater detail is possible. Examples of this were the support structures for the gun turrets on the DD-963 and the shafting systems of both the DD-963 and the LNG, as calculated by NKF.

In this regard, Dr. Genalis also agrees with Mr. Zaloumis and Mr. Volcy, that the finite element analysis is not the answer to a Maiden's prayer, and should be used with caution. In this I would whole-heartedly agree.

Dr. Genalis has suggested that in the preliminary design stage a simplified model is required. This is concurred in as suggested in Reference 7, and as used on the DD-963 and on the LNG, the 20-station beam model produced good results. This is not to imply that this method cannot be improved upon, rather, that it has been an effective tool. Further development in this area would be most welcome, and I would propose that the Ship Structure Committee give this suggestion serious thought as a possible research project.

Mr. Zaloumis' comments are most welcomed. They permit me to clarify a few misunderstandings which are frequently encountered in respect to a shipboard vibration habitability criteria. First let me point out, that the MIL-STD-1472A (May 1970) apparently is based on the International Standard, ISO/DIS 2631, "Guide for the Evaluation of Human Exposure to Whole Body Vibration", and, in this case the Navy Criteria allows more vibration in the horizontal direction, than in the vertical direction above 3 Hz, as noted by Mr. Zaloumis. It should also be noted that the 1968 draft of this ISO standard showed the comparative allowable horizontal level was less than the vertical level. At that time the allowable horizontal level was given as 60% of the vertical level and is referred to in reference 7. This 180° turn was made at the ISO meeting of 1969 in Dusseldorf, Germany. This was discussed in dpeth in a proprietary report to NSRDC entitled "A Proposed Criteria for Hull Vibration Based on Habitability Requirements", dated 31 August 1971. The following paragraph is taken from that report.

> "The most recent ISO document (1970) introduced a drastic change in the limits for horizontal vibration. In the current version, the criteria for horizontal vibration is a constant velocity from 2 Hz to 80 Hz and a constant acceleration in the 1 Hz to 2 Hz range. In the 8 Hz to 80 Hz range, in which the ISO proposes a constant velocity criteria, the most recent ISO horizontal criteria is now about 8.5 dB higher than the vertical criteria, where previously it was about 3 dB lower. This represents a net change of about 12 dB, from an allowable velocity of 6 mm/sec to more than 24 mm/sec. The earlier ISO horizontal criteria (1968) appears to more accurately reflect shipboard experience and was used as a principal basis for establishing the recommended limits."

To support this viewpoint I am appending a series of curves entitled "Ship Vibration - Interim Guide-Lines for Habitability Criterion (September 1974), Comparison with Various Criteria (Peak Values)". This compilation was made by Mr. Viner of Lloyd's Register and one of the British members of Working Group 2, of ISO/TC108/SC2 "Vibration of Ships". In all cases, except the German VDI plot which does not differentiate between Horizontal and Vertical, the Horizontal Criterion is lower than the Vertical Criterion. These include the B.S.R.A., Bureau Veritas, IRCN, Japanese (1970) Proposal, and Lloyd's Register.

With respect to the DD-963 Target (for all directions) and the proposed LNG Specification, the following table will show their comparison with those I recommended to the Navy in the previously cited report of 31 August 1971.

You will note these values apply to the hull girder, and, in addition to meeting minimum requirements for habitability, are intended to reflect the state-of-the-art. The recommended values represent a general set of values. The LNG specification values represent the author's assessment of a reasonable set of values for that particular ship, which were handily achieved. The DD963 values were set by the Navy. The midship values compare well with the LNG stern values, while the stern values for the destroyer employs the recommended vertical limits for both vertical and horizontal directions.

As this point it is appropriate to bring to your attention the most recent version of the ISO "Interim Guide-Lines for Hull Vibration Criterion", which was approved by the Ship Vibration Working Group, at their September 1975 meeting in Amsterdam. You will note that it is similar to the 1974 version, which is included in the report, except that the upper constant velocity line has been reduced to 9 mm/sec, approximately the value proposed for the LNG. This interim guide-line does not differentiate between vertical and horizontal vibration, because the ISO prefers to develop their guide lines by the use of reliable data, based on their proposed code, which was also approved by the working group in September, 1975. However, preliminary data shown by the VID, suggests we will have a lower criterion for horizontal vibration.

Vibration Velocities, mm/sec

	Recommended*	
	Vert.	Hor.
Objective	7.5	5.0
Limit	11.25	7.5

*All ships of the Navy

	LNG	
	Vert.	Hor.
Objective	6.25	3.75
Limit	9.375	5.625

	DD963(Midship)	
	Vert.	Hor.
Objective	6.0	3.6
Limit	8.0	4.8

	DD963(Stern)	
	Vert.	Hor.
Objective	8.0	8.0
Limit	11.25	11.25

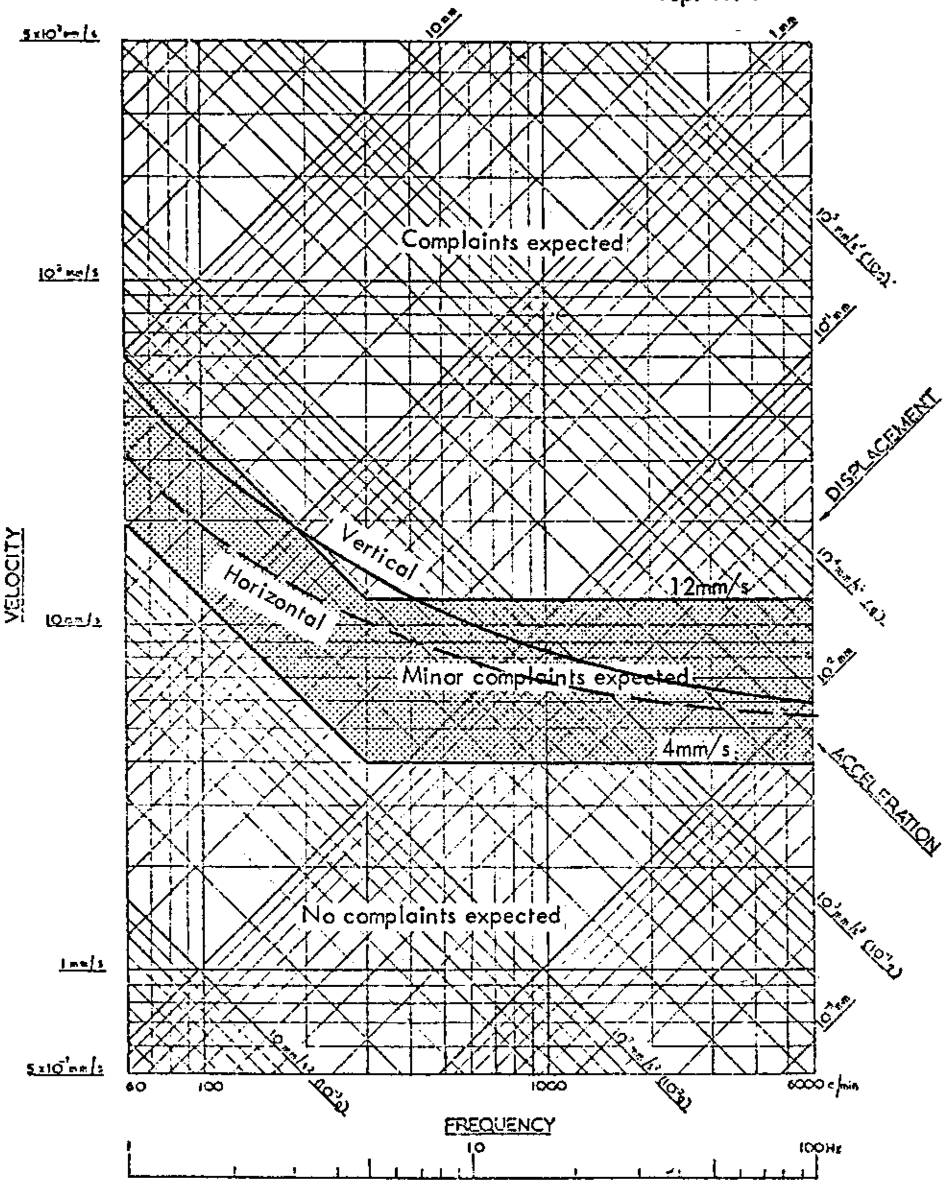
As a final point on the low-frequency values, a constant velocity criterion is considered appropriate for turbine driven ships. The constant acceleration curve, in the low-frequency range, is considered appropriate for diesel driven ships, because of their large unbalanced forces and moments. This is reflected in the Norske Veritas proposal and should be borne out when sufficient data is collected.

In his remarks on structural modeling, Mr. Zaloumis points out that NSRDC uses a similar analysis procedure with the conventional 20 station lumped-mass type but doubts the credibility of the response above the 4th or 5th mode. Since he also questions the credibility of the finite-element technique, I am not sure what NSRDC does at this point. I am confident, however, that an improved approach is possible, for use in preliminary design. That is another paper, however.

To clear the air, I would refer Mr. Zaloumis to my response to Dr. Genalis, who raised somewhat similar remarks concerning the application of the finite-element technique. If you refer to my remarks closely, you will recognize that we all three hold similar positions on this point. In my paper, however, I have avoided making specific recommendations, because, as in the case of the hull vibration criterion, too many factors are involved and the application of criterion, specifications, computer techniques, etc. are all tools. The application of these tools in the design process is still an art. That is the theme of the paper.

ISO / TC108 / SC2 / WG2

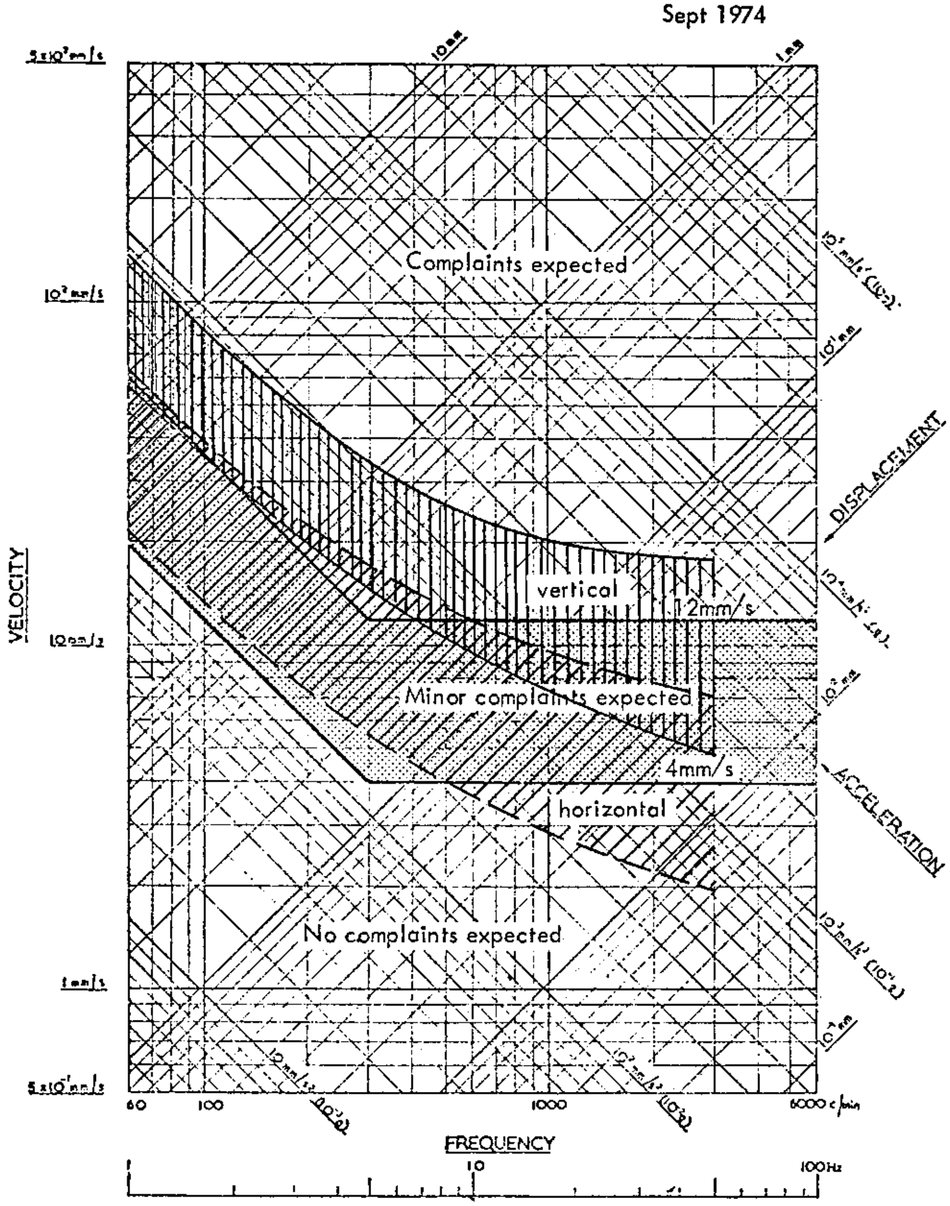
Sept 1974



SHIP VIBRATION

INTERIM GUIDE-LINES FOR HABITABILITY CRITERION

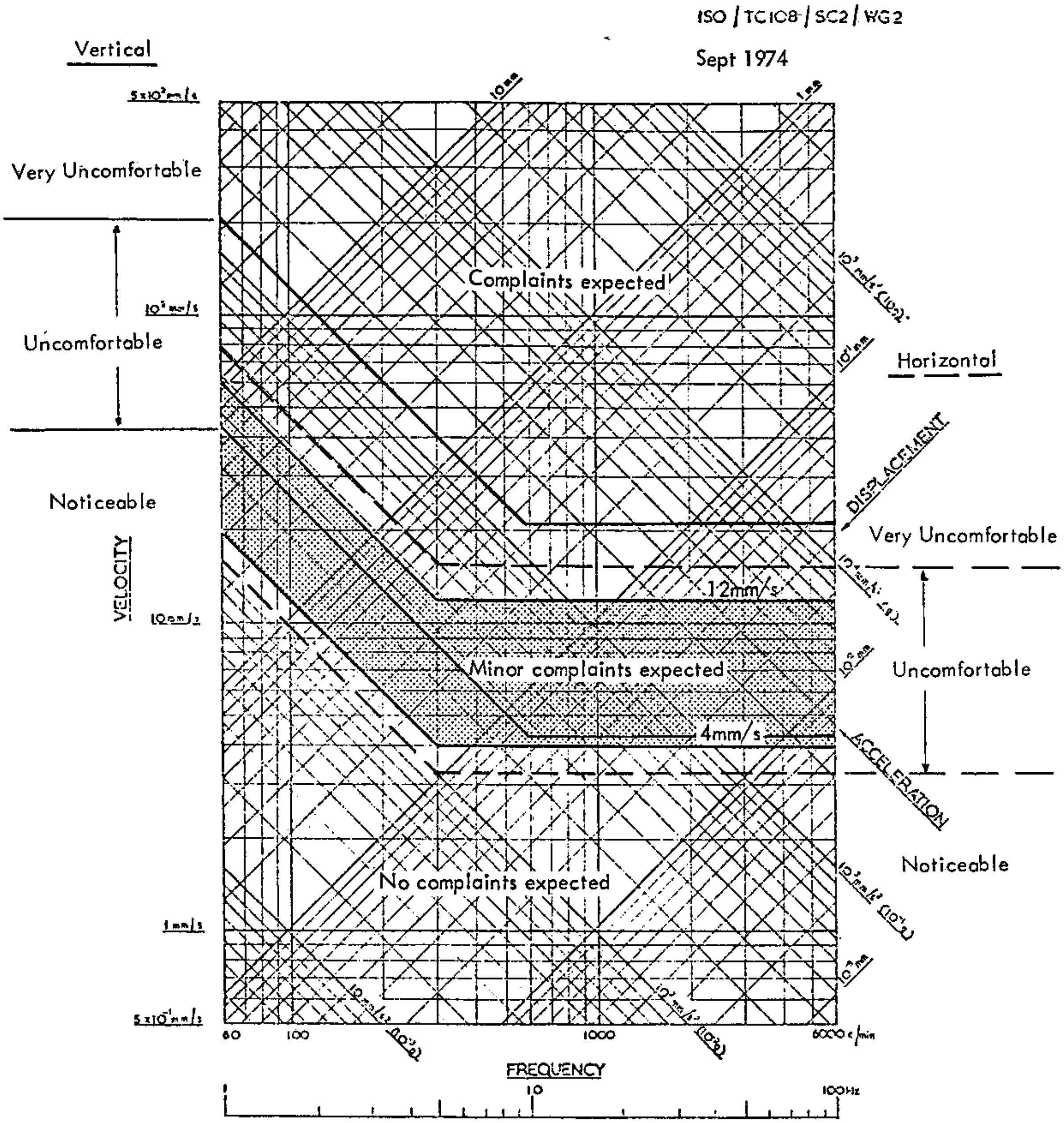
Fig.1 COMPARISON WITH B.S.R.A. LIMITS



SHIP VIBRATION

INTERIM GUIDE-LINES FOR HABITABILITY CRITERION

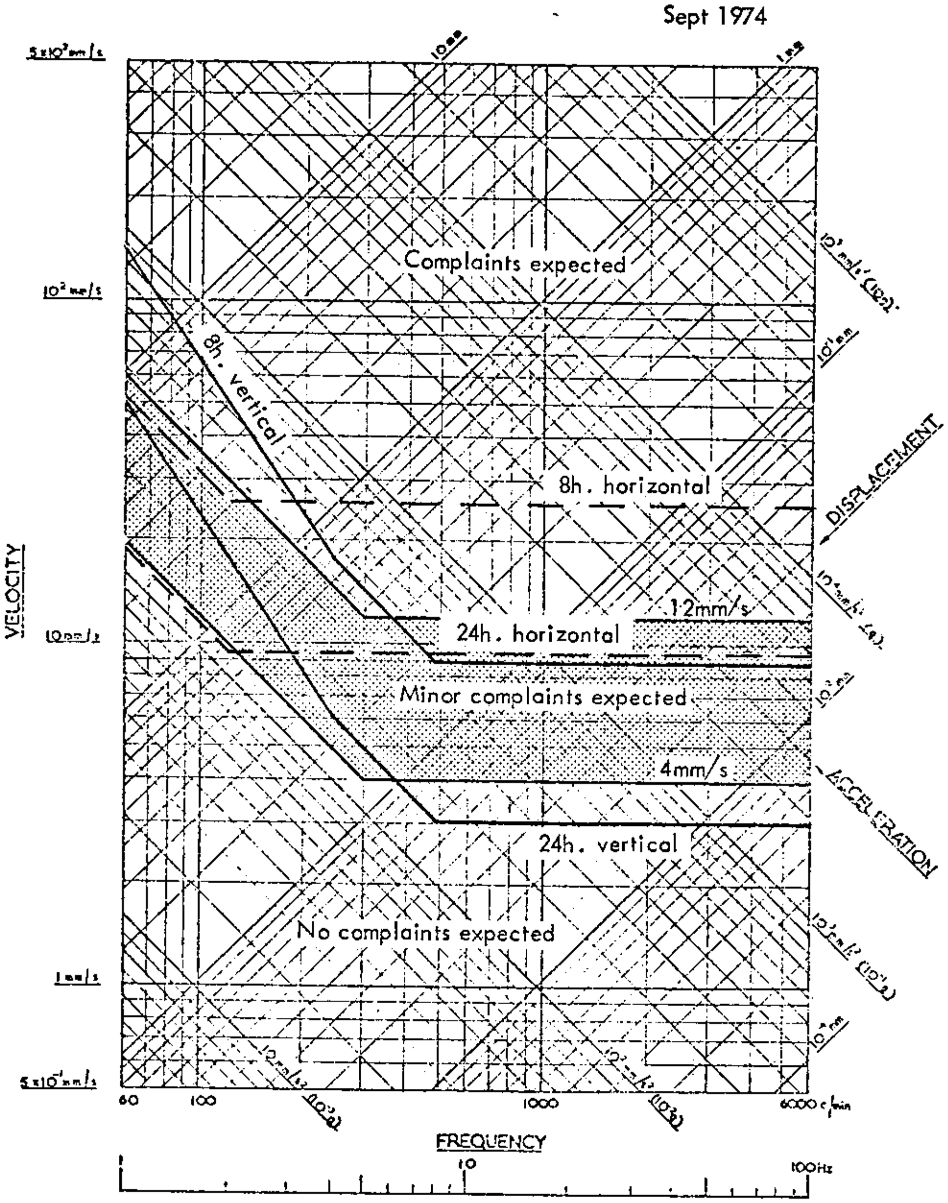
Fig. 2 COMPARISON WITH BUREAU VERITAS VIBRATION LIMITS FOR CREW



SHIP VIBRATION

INTERIM GUIDE-LINES FOR HABITABILITY CRITERION

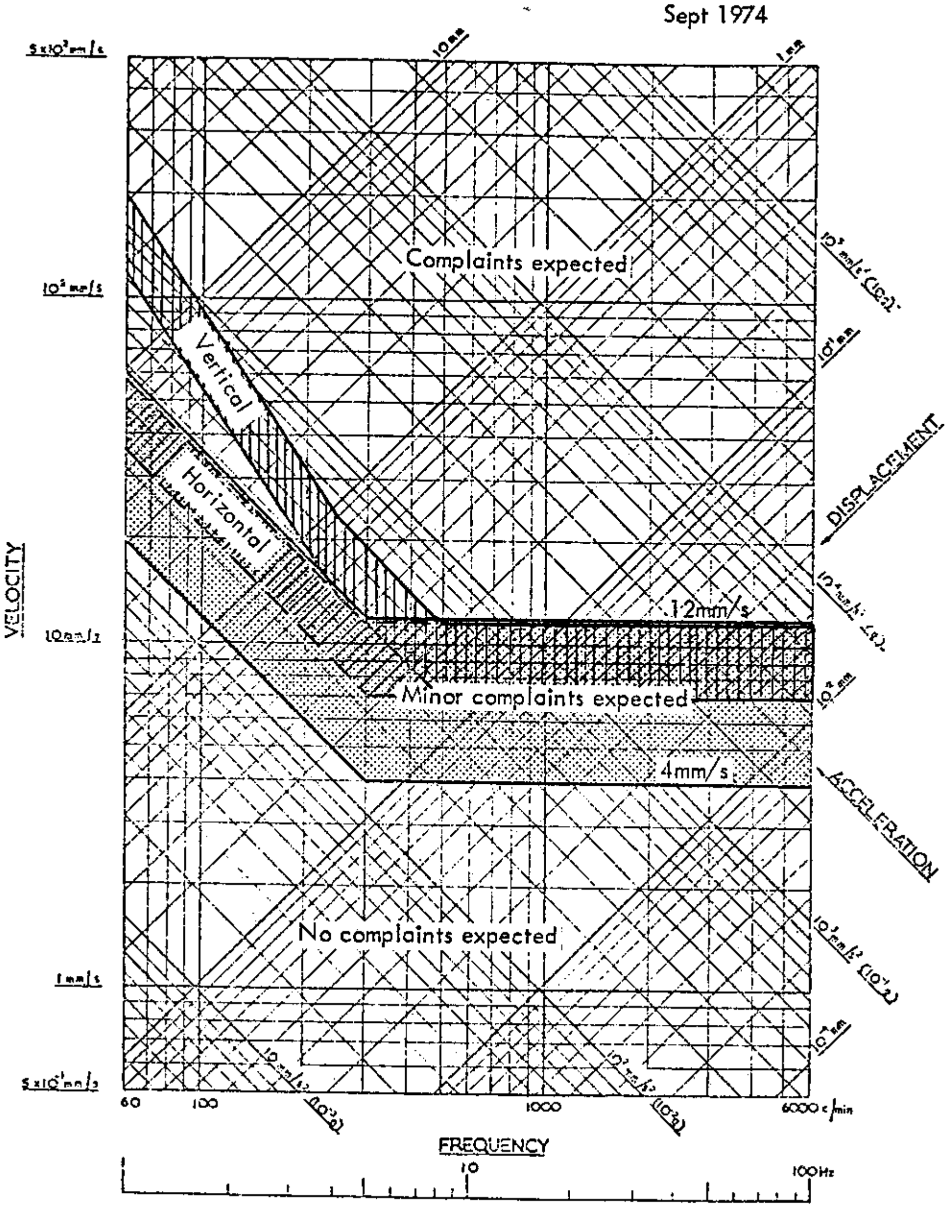
Fig. 3 COMPARISON WITH IRCN CURVES



SHIP VIBRATION

INTERIM GUIDE-LINES FOR HABITABILITY CRITERION

Fig.4 COMPARISON WITH ISO/DIS 2631 FATIGUE-DECREASED PROFICIENCY BOUNDARY



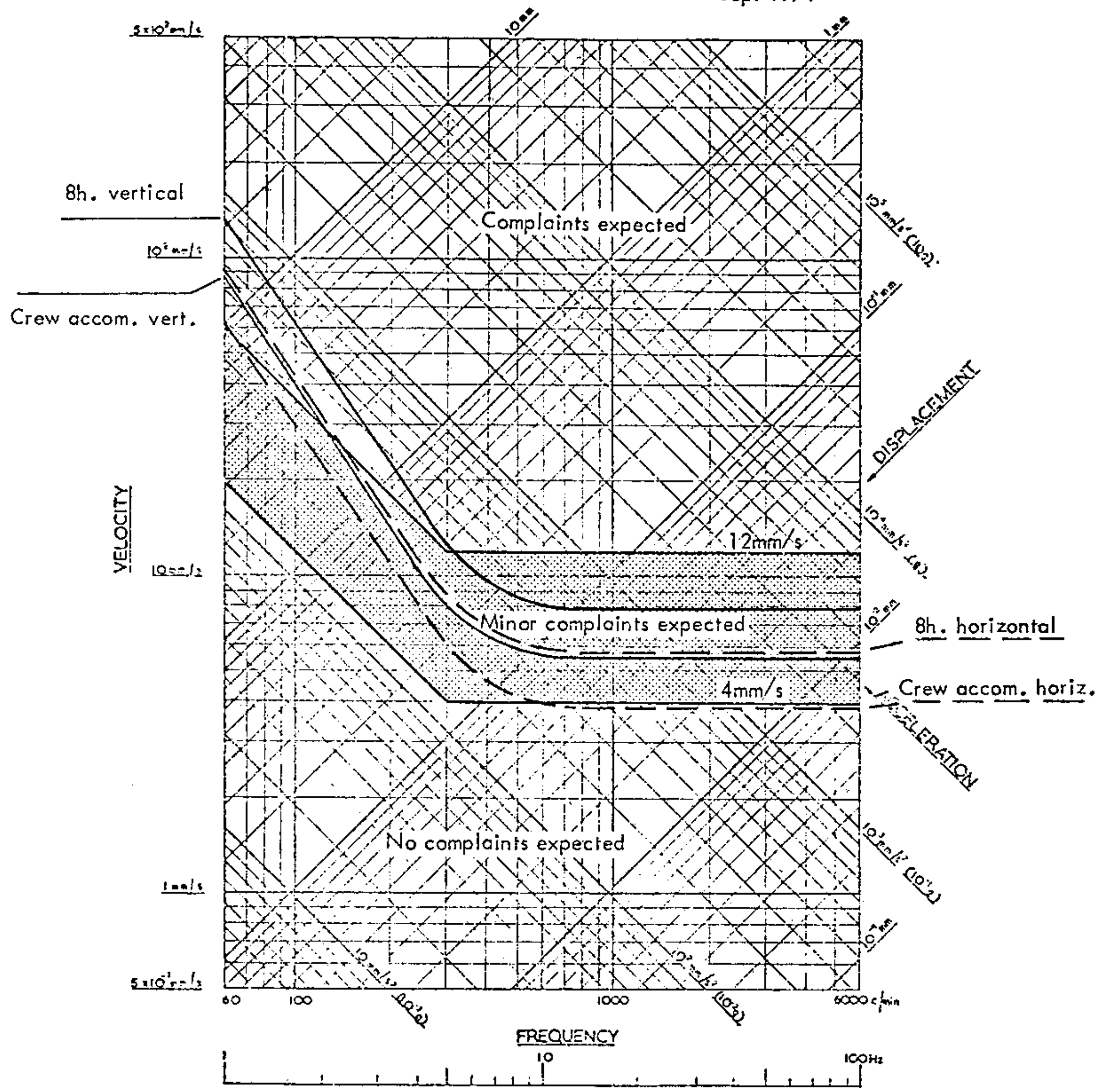
SHIP VIBRATION

INTERIM GUIDE-LINES FOR HABITABILITY CRITERION

Fig.5 COMPARISON WITH JAPANESE 1970 PROPOSAL

ISO / TC108 / SC2 / WG2

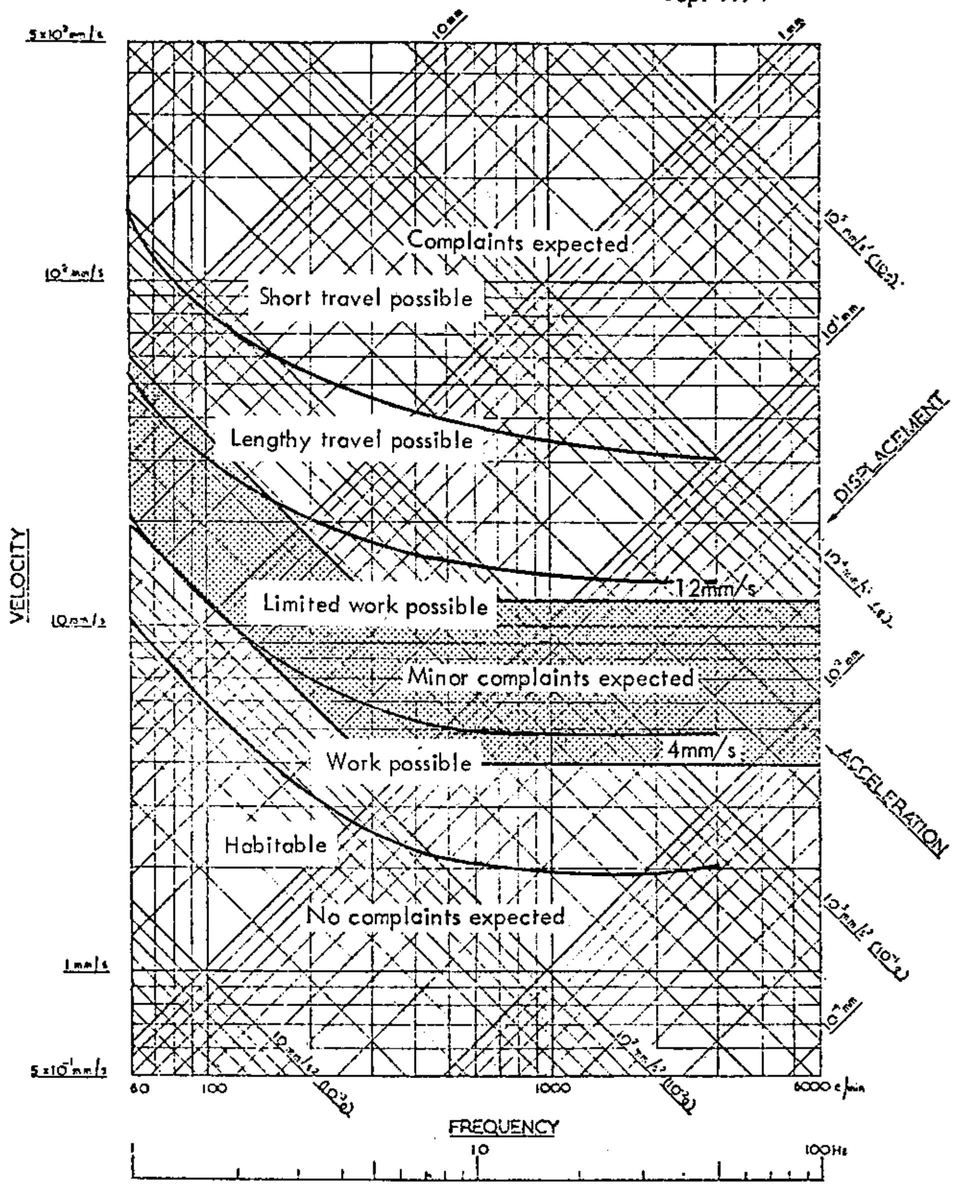
Sept 1974



SHIP VIBRATION

INTERIM GUIDE-LINES FOR HABITABILITY CRITERION

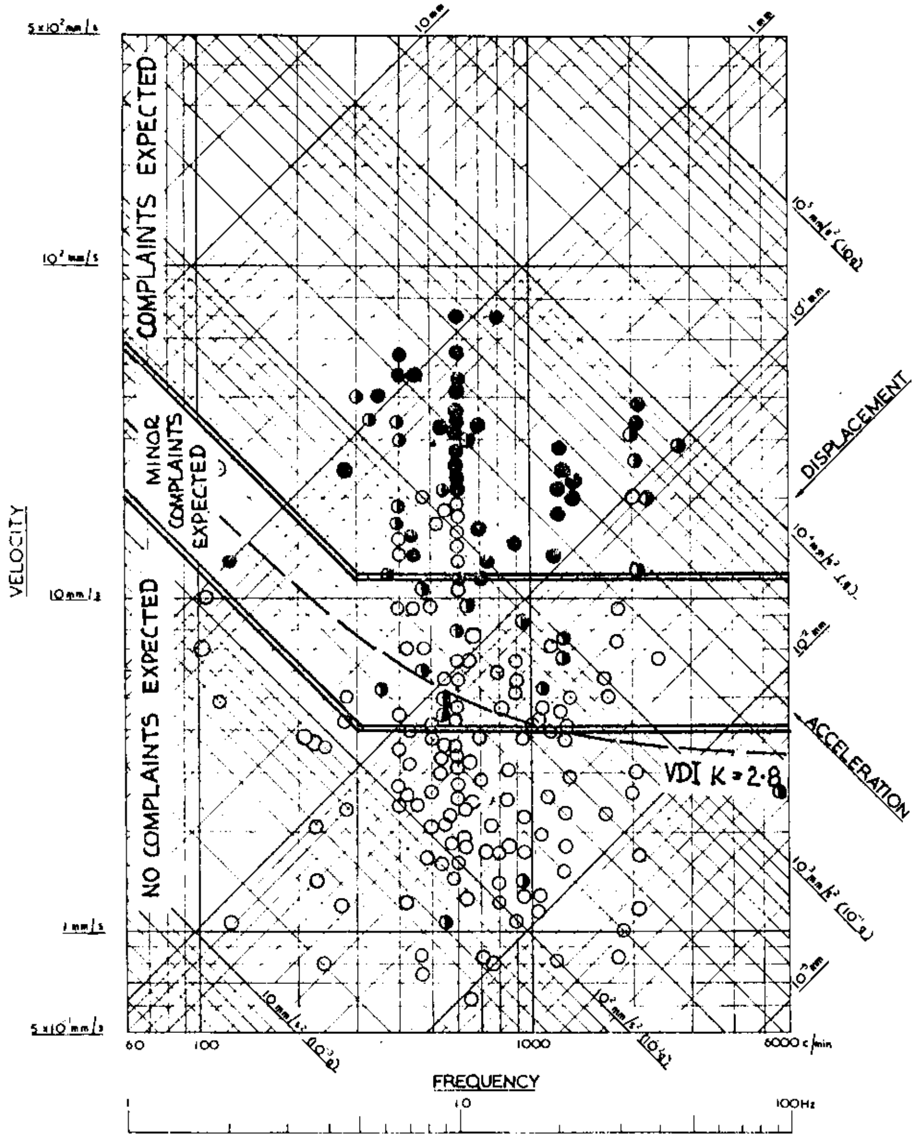
Fig. 6 COMPARISON WITH LLOYD'S REGISTER VIBRATION LIMITS FOR CREW COMFORT



SHIP VIBRATION

INTERIM GUIDE-LINES FOR HABITABILITY CRITERION

Fig.7 COMPARISON WITH VEREIN DEUTSCHER INGENIEURE CURVES



DETAILS OF DATA PRESENTED

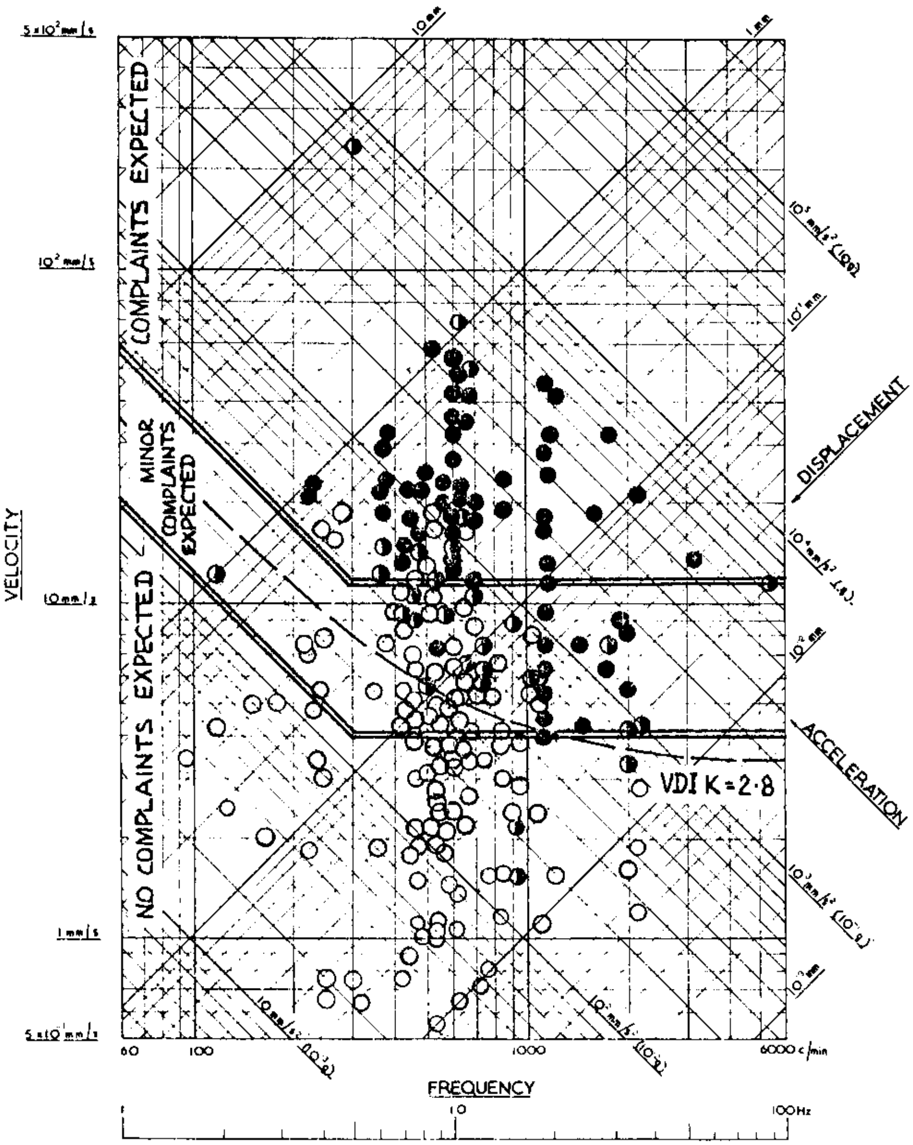
TO SEP 1975

VERTICAL VIBRATION : PROPOSED ACCEPTABLE LEVELS AND MEASURED DATA.

ISO SHIP GROUP —— VDI K = 2·8

O NO COMPLAINTS ◑ SLIGHT COMPLAINTS ● STRONG COMPLAINTS

SHIP VIBRATION DATA



TO SEP 1975

DETAILS OF DATA PRESENTED

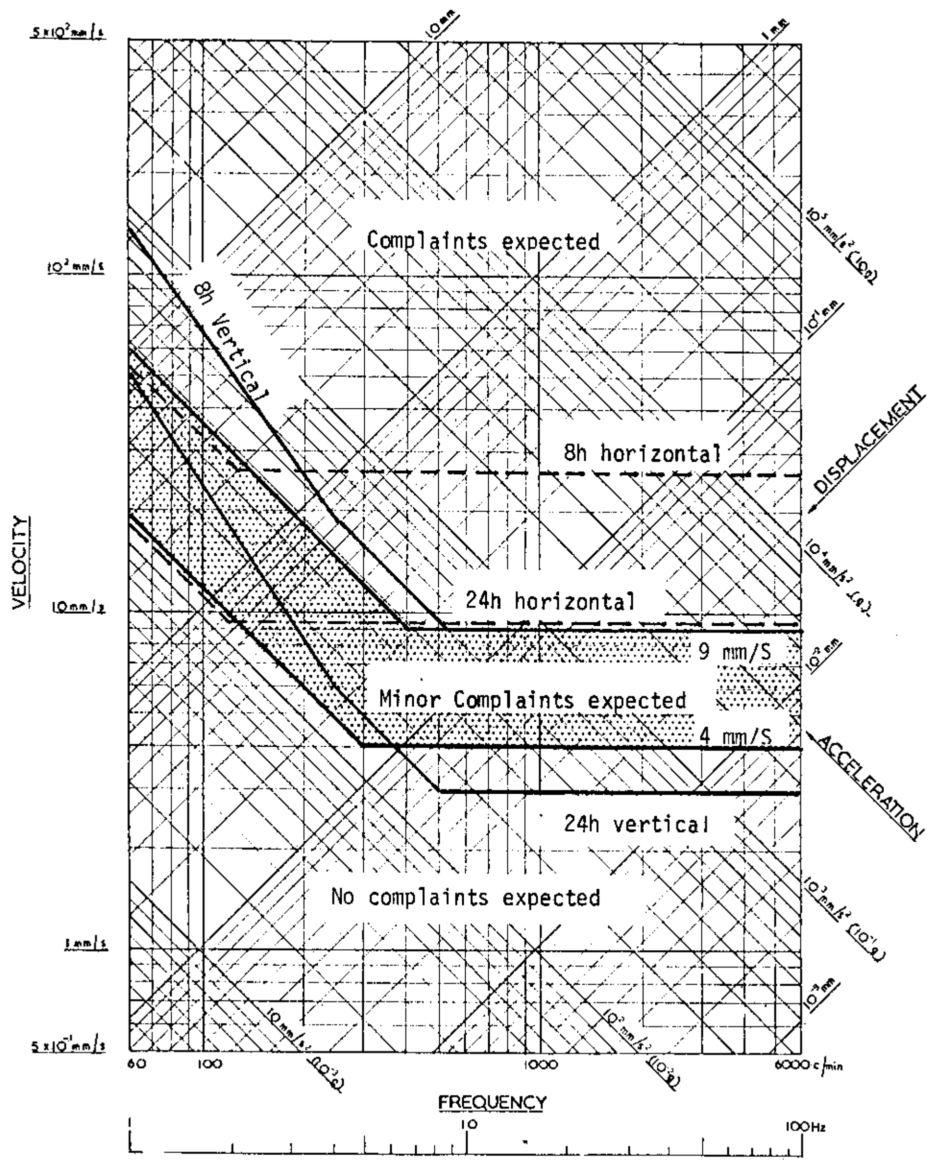
HORIZONTAL VIBRATION : PROPOSED ACCEPTABLE LEVELS AND MEASURED DATA.

═══ ISO SHIP GROUP — — — VDI K = 2·8

○ NO COMPLAINTS ◑ SLIGHT COMPLAINTS ● STRONG COMPLAINTS

SHIP VIBRATION DATA

ISO / TC108 / SC2 / WG2
September 1975



SHIP VIBRATION

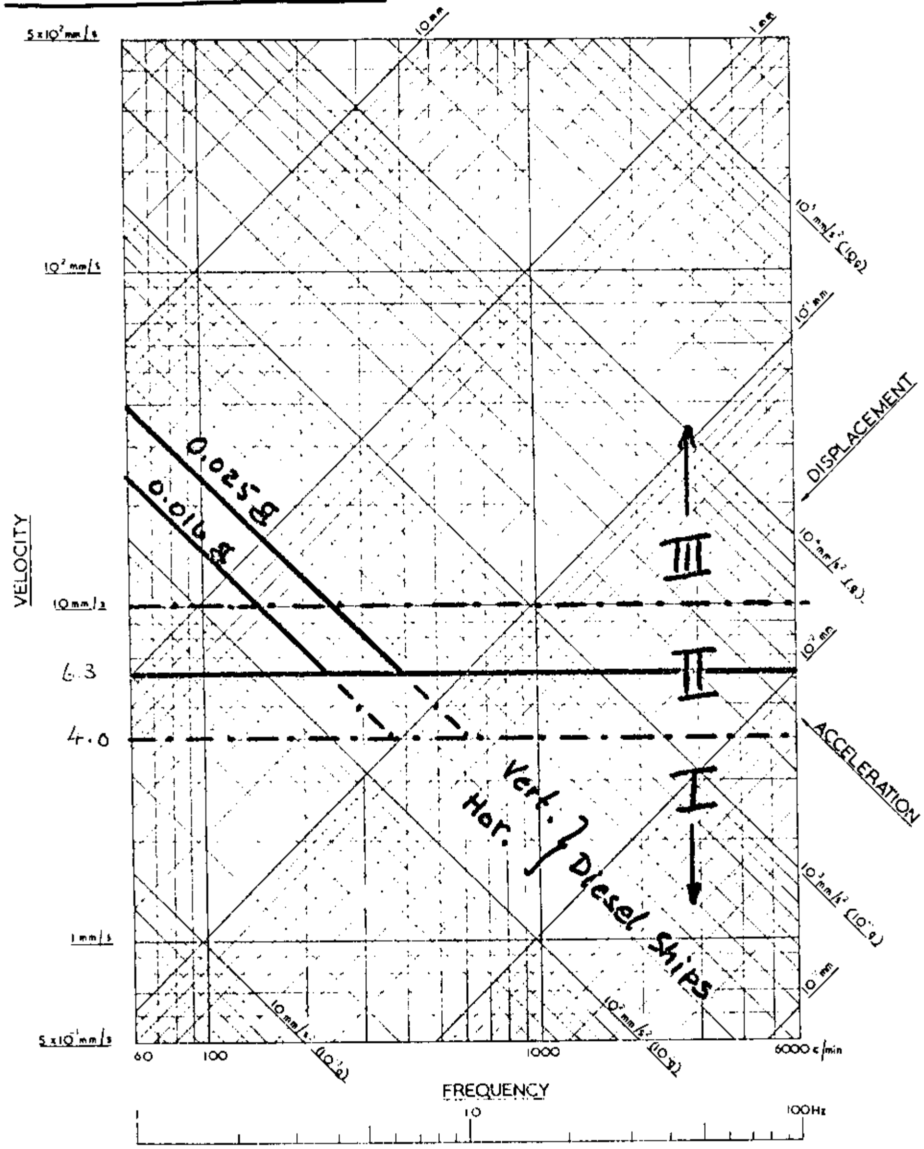
INTERIM GUIDE-LINES FOR HULL VIBRATION CRITERION

Fig. 1 - COMPARISON WITH ISO/DIS 2631 FATIGUE-DECREASED PROFICIENCY BOUNDARY

Appraisal of Ship Vibration

Norske Veritas Proposal for Interim Guide-Lines

ISO / TC108 / SC2 / WG2



DETAILS OF DATA PRESENTED

SHIP VIBRATION DATA

Mr. Hammer's remarks are a welcomed supplement to this paper. He has raised several significant questions on the adequacy of the approaches used, in many cases, in our efforts to limit and control shipboard vibraton. Mr. Hammer has also answered some of these questions, which are primarily directed to the designers, builders and owners. For the sake of the record, however, I would offer a few general remarks.

First, I believe, that in most cases, an adequate program to minimize vibration is neither contemplated nor funded. It is also evident, in the two programs cited, that good vibration characteristics can be achieved, if the effort is made.

As a second point, a greater willingness on the part of industry, to exchange technical information on given designs is necessary if we are to improve our position in this area. At the present time we have made progress, thru the Research Panels of the S.N.A.M.E. What we need now is more direct cooperation by the owners and builders, in conducting full-scale studies on new designs, as outlined in the recently published "Code for Shipboard Vibration Measurements", and the inclusion of the test results in the S.N.A.M.E. data bank.

Finally, I consider the fact that most designs are tested in Europe to represent an admission of the fact that the U.S. model test facilities, response times and costs to be less atrractive than those of our European colleagues. I would raise a personal question to Mr. Hammer, as to what steps are being taken by the Maritime Administration to narrow this gap?