THE SOCIETY OF NAVAL ARCHITECTS AND MARINE ENGINEERS
One World Trade Center, Suite 1369, New York, N.Y. 10048
Paper to be presented at the Ship Vibration Symposium
Arlington, Va., October 16–17, 1978

State of the Art for Shipboard Vibration and Noise Control

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ABSTRACT

The continued demand for higher powers, the use of gas turbine propulsion systems and the invocation of more stringent vibration and airborne noise criteria have placed increased emphasis on the control of shipboard vibration and airborne noise on both Commercial and Military ships. The authors present a brief background review of the specifications invoked in recent shipbuilding programs and the approach employed in the development of the DD963 and LNG designs. The paper is presented in two parts, the first dealing with vibration control and the second with noise control.

Part I includes the Approach to Vibration Control, Vibration Specifications, The Measurement and Evaluation of Shipboard Vibration, Propeller Excited Forces, The Prediction of Hull and Machinery Vibration, Recent Findings and Recommended Research Efforts.

Part II deals with Current Commercial and Military Airborne Noise Specifications, The Prediction of Compartment Noise, and Techniques Employed in the Control of Compartment Noise.

INTRODUCTION

Vibration and Noise problems have plagued the shipbuilding industry for many years. The types of problems are varied and include those associated with both hull and machinery vibration and the interaction between the two. Unfortunately, when problems arise during new construction programs, the lack of adequate standards, design procedures and flexibility for corrective action frequently results in lengthy litigation between the shipbuilder and his client, a compromised settlement and then on to the next job. Obviously, the lessons learned by experience contribute heavily to our understanding of the problems encountered and form the basis for our future design approach. However, problem solving, after the fact, is extremely expensive, and at best usually represents a compromise solution. Quite naturally therefore, all parties to this process are interested in improving our capability for avoiding such difficulties before they are built into the ship.

As if these factors were not enough, the rapid increase in powers, with their unknown impact on the vibration and noise problem, and the recent introduction of vibration and noise specifications, have placed a greater burden on designers and shipbuilders. It becomes mandatory at times for the industry as a whole, to take a close look at the problem and determine their capability of meeting current requirements with presently available technology. This is one of those times and this paper is intended to provide a brief overview of the state-of-the-art, in the development of a rational design procedure aimed at the limitation and control of shipboard vibration within generally accepted criteria. As was the case of the First Conference on Ship Vibration [1][2], which was held at Stevens Institute of Technology in 1965, it is intended that this Symposium bring together representatives of the Maritime Industry in a free exchange of information on all aspects of ship vibration, noise, and hull/machinery incompatability.

STATE OF THE ART FOR SHIPBOARD VIBRATION - PART I

BACKGROUND

This conference, which is jointly sponsored by the interagency Ship Structure Committee and the Society of Naval Architects and Marine Engineers is referred to as the second in the scheduled series of Symposia, jointly sponsored by these two organizations. The first was the Ship Structures Symposium [2], held in October 1975. However, this conference may also be referred to as the Second Conference on Ship Vibration, the First Conference as previously noted, having been held in 1965.

The first conference, unlike this international symposium, was attended almost exclusively by U.S.A. representatives. However, like this symposium, it had a definite objective which was to bridge the large technical gap, which existed at the time, between the research investigator and the designer-shipbuilder, for the purpose of making each more responsive to the needs of the other. At that time, two of the more important facilities in this country, which were involved in Ship Vibration Research were the David Taylor Model Basin (D.T.N.S.R.D.C.) and the Davidson Laboratory of the Stevens Institute of Technology. The program, jointly sponsored by these two

[1] NKF Engineering Associates, Inc., Silver Spring, Maryland.

[2] Numbers in brackets designate References at end of paper.

facilities was conveniently divided into two sessions, the first dealt with Exciting Forces and the second with Vibratory Response. At the completion of each session a summary paper was presented on the Application of Presented Results to Ship Design. While that conference obviously did not provide a resolution of all our problems, just as this symposium will not, it did provide an increased awareness of the basic problems associated with the design and development of vibration-free ships, and thru the efforts of some of the attendees at that conference, significant impacts were registered on the programs of the Research Panels of the S.N.A.M.E., on the Ship Structures Committee, on the International Ship Structures Committee, on the International Standards Organization and stimulated the sponsorship of numerous research projects. In addition, all the Classification Societies, Towing Tanks, Designers and Shipbuilders have, in the interim become much more sophisticated in their design process. In due respect to the total shipbuilding community, it must also be acknowledged that the current trend toward the protection of the environment (habitability) of the crew, the development of vibration and noise criteria by regulatory bodies such as OSHA has focused the attention of the owners to the problems of vibration and noise aboard ship and has led to the incorporation of specific requirements in ship specifications.

Since this paper is intended to provide an overview of the State of the Art for Shipboard Vibration and Noise Control, it is appropriate that we review some of the significant accomplishments that have been achieved since the time of the earlier Shipboard Vibration Conference, primarily to acquaint the reader with the programs of the sponsors and the availability of the pertinent publications. Emphasis is of course, placed on the work carried out in this country, particularly thru the efforts of the S.N.A.M.E. research panels HS-7 (Vibration), 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, shipboard vibration studies were sponsored which resulted in the first "Code for Shipboard Hull Vibration Measurements" [3] 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 [4]. The primary purpose of this latter step, was to stimulate the development of an international effort in the exchange of information and the possible development of an accepted method of evaluating shipboard vibration.

In 1965 the Maritime Administration procured a shipboard vibration measurement instrumentation package to permit implementation of the Code, and made it available to the industry on a loan basis. The Code was revised in 1967 to include an expanded section on instrumentation. This revised "Code for Shipboard Hull Vibration Measurements" was replaced in 1975 by the current S.N.A.M.E. Code C-1, "Code for Shipboard Vibration Measurements" [5]. This effort was the product of a joint venture by HS-7 (Vibration) and M-20 (Machinery Vibration Panels) and included measurement procedures on main propulsion machinery but was primarily directed at the longitudinal vibration problem related to geared-turbine propulsion drives. Although published in 1975, this Code was originally submitted for review in 1970.

In 1970, at the Geneva meeting of the Organization for International Standardization, the Shock and Vibration Committee recognized the need for an international standard on Shipboard Vibration Measurements. To implement this decision, a Ship Vibration Working Group was established under Subcommittee 2, which deals with Vibration in Machines, Vehicles and Structures. This working group, ISO/TC108/SC2/WG2, included members of all classification societies and has produced a Draft Proposal 4867, "Code for the Measurement and Reporting of Shipboard Vibration Data" [6], which at this writing has been approved by the Shock and Vibration Committee (TC108) and is currently under review by the member nations of the ISO for approval as an ISO document. This document was based, in part, on the S.N.A.M.E. Code C-1. It was expanded however, to include consideration of some of the machinery vibration problems associated with diesel drive systems.

In December 1976, S.N.A.M.E. issued the T&R Code C-4, "Local Shipboard Structures and Machinery Vibration Measurements" [7], which was followed in 1977 by the ISO Draft Proposal, "Code for the Measurement and Reporting of Shipboard Local Vibration Data" [8]. S.N.A.M.E. also published in 1976, T&R Code C-5, "Acceptable Vibration of Marine Steam and Heavy Duty Gas Turbine Main and Auxiliary Machinery Plants" [9].

In 1977 the ISO also circulated a Draft Technical Report, "Interim Guidelines for the Evaluation of Vibration in Merchant Ships," [10] while the S.N.A.M.E. HS-7 Panel had prepared "Ship Vibration and Noise Guidelines" [11]. It is also important to note that Subcommittee 4 of the ISO/TC108, which concerns itself with "Human Exposure to Mechanical Vibration and Shock," has developed an International Standard "Guide for the Evaluation of Human Exposure to Whole Body Vibration" [12], and has recently (1975) prepared a Draft Proposal, "Vibration Acceptable to Crew Aboard Ships [13] as an addendum to the standard [12].

At this point, it becomes increasingly obvious that the development of standards, specifications and "guidelines" for the measurement and evaluation of shipboard vibration and noise has brought us forward at a rapid pace in the last few years. You might also say; "Now that we know what we want, and know how to measure what we get, do we have the necessary means of achieving that objective?" The answer to that basic question is what this paper will attempt to do.

APPROACH TO A RATIONAL DESIGN PROCEDURE

Designers and Shipbuilders are particularly interested in the early development of a design procedure by which they may proceed with the orderly design and construction of a vessel with assurance that the operation of the ship does not produce damaging or annoying vibration problems which result in expensive modifications and/or lengthy and expensive litigation. The state-of-the-art for the design of a ship which is free from troublesome vibration problems does not lend itself to a cookbook procedure at this time, nor is it anticipated in the near future. However, it is the opinion of the authors that most of the ingredients are available, and if properly employed, can lead to successful designs, provided of course, that the necessary studies and tests are planned for and a suitable budget is included in the program.

For convenience, we will briefly review the approach used in the development of the largest and most recent shipbuilding programs with which we have been associated; the DD963 Destroyer Program, designed and built by the Ingalls Shipbuilding Division of Litton Industries for the U.S. Navy, and the 125,000 CM 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 de France Dunkerque, Avondale Shipbuilding of New Orleans, Louisiana, 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-the-art from a high-speed, fine-lined destroyer to a large tanker. Some details of these two programs were included in the Ship Structures Symposium '75 Paper on "An Assessment of Current Shipboard Vibration Technology" [14], and will not be repeated here, except as necessary to demonstrate the approach used or in support of a technical viewpoint.

The two programs differed in one basic point, the hull form of the Destroyer was selected, based on model testing performed at the David Taylor Research and Development Center and reflected the total experience of the Model Basin in selecting the optimum hull form for the twin-screw ship. In the case of the LNG Carrier, a single screw vessel of 45,000 SHP, the optimum hull form was a major unknown, and the selection of the configuration of the stern was the first order of business. This power represented a 25% extension of the state-of-the-art from 36,000 SHP to 45,000 SHP. In all other respects, a similar design procedure was employed which included the following basic elements:

1. A set of design objectives or specifications.
2. An analytical procedure which includes:
   a. A suitable math 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.
   b. Obtain technical data to permit the continued development or improvement of empirical factors.

The judgment of the adequacy of the design is based on the evaluation of the vibratory characteristics of the ship against the specifications or criteria established. The adequacy of the analytical procedure is based on the ability to reliably predict the vibratory characteristics observed.

In the application of this approach, the total ship system may be conveniently divided into the following parts:

Part I - Vibration of Hull Girder - The most fundamental requirement pertains to the response of the hull girder. The adequacy of the design, principally the stern configuration and the propeller design, which control the forces generated, and the response of the hull girder to these forces are reflected in the vibration characteristics of the hull. These characteristics provide the base from which the response of major substructures, local structures and supporting systems for equipment may be judged.

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

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

Part IV - Vibration of Shipboard Equipment - Equipment should be designed to meet the environmental requirements established for shipboard use. Balancing and vibration tolerances for rotating machines should be specified. 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 is designed, and in the case of self-excited equipment, the supporting structure should be such as to prevent the generation of excessive vibration or noise from a habitability point of view.

Part V - Vibration of Main Propulsion Systems - Main engines, shafts and propellers are designed for structural adequacy under the conditions stipulated in the procurement specifications. 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 vibration of the hull. Potential problems include dynamic unbalance of components, lateral, torsional and longitudinal vibration of the propulsion system, and the generation of hull structural resonances when stimulated by propeller forces or shaft and engine frequencies.

During the preliminary design phase, the Vibration of the Hull Girder, Part I, and the Vibration of the Main Propulsion System, Part V, directly effect each other. Therefore, the principal purpose of the study is to determine the anticipated vibratory characteristics of the hull girder and the main propulsion machinery system of the proposed design and to provide a detailed evaluation of the influence of the various parameters which affect these characteristics. The scope of the study should include an estimate of propeller exciting forces, an estimate of the principal hull criticals of vertical, athwartship and torsional modes of vibration and a prediction of the response and the importance of the various modes of vibration, relative to the acceptance criteria or design specifications. A detailed evaluation of the lateral, torsional and longitudinal vibration characteristics of the propulsion system should also be provided, together with suitable recommendations for the optimization of the hull and machinery system parameters, to minimize the estimated vibratory response.

Since many of the calculations performed in the preliminary design phase may be based on assumptions and estimates, detailed design studies will be required in the detailed design phase, to confirm the earlier predictions, to provide a basis for the test and evaluation studies and to permit continued improvement of design procedures. Also during the detailed design phase, when the necessary information is available, the Vibration of Major Substructure, Part II, and the Vibration of Local Structural Elements, Part III, can be more effectively evaluated.

DESIGN OBJECTIVES OR SPECIFICATIONS

To insure ships are built free from excessive or damaging vibration, it is necessary to invoke technical requirements in the form of design objectives or specifications. These requirements or criteria represent the basis against which the adequacy of the design are evaluated. Design studies should be carried out in the preliminary design phase to verify the likelihood that the proposed design will meet the requirements, to permit reasonable modifications to the controlling parameters, when required, and to provide a suitable basis for the improvement of the prediction technique employed. Full-scale studies should be carried out to confirm the design analyses, and when feasible, supplemental tests should be conducted to obtain technical data on which improvements to the state-of-the-art may be based. Early in 1971 a set of design objectives, in the form of Vibration Specifications, were generated for the LNG design. These specifications reflect the current state-of-the-art, were invoked for the more recent Avondale and Newport News LNG Carrier designs, were originally published in the paper, "An Assessment of Current Shipboard Vibration Technology" [14] and are repeated here for ready reference purposes.

Vibration Specifications for 125,000 CM 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 established 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 ±.25 in/sec vertically, and ±.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 (±.375 and ±.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 ±.30 in/sec vertically and ±.20 in/sec in the transverse (athwartship or longitudinal) directions is desired. Amplitudes greater than ±.45 in/sec and ±.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 is 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 ±.45 in/sec vertically, and ±.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 ±.25g, 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 results, 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 ±.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 ±.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 (RPM)	Rotor Characteristics
Single-plane	0-1000	$L/D \leq 0.5$
	0-150	$L/D > 0.5$
Two-plane	>1000	$L/D \leq 0.5$
	>150	$L/D > 0.5$
Multi-plane		Flexible: Unable to correct by two-plane balancing.

L = Length of rotor mass, exclusive of shaft
D = Diam. 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, oz/in.

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 3/4 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. In the event the analysis does not indicate probable compliance, a torsiograph test of the system will be required prior to acceptance.

6.3 Longitudinal Vibration of Propulsion Machinery

The dynamic response of the propulsion system shall have no excessive alternating thrust within the operating speed range. In no case however, shall the displacement amplitude of longitudinal vibration of the propulsion machinery, including the main condenser and associated piping, be sufficient to adversely affect the operation of the propulsion unit or precipitate fatigue failure.

Excessive alternating thrust is defined as:

a. Main and Turbine Thrust Bearings

Excessive alternating thrust occurs when the single amplitude alternating thrust, measured at the main and turbine thrust bearings, exceeds the mean thrust at that speed or exceeds 50 percent of the full power thrust, whichever is smaller.

b. Propulsion Reduction Gear

Excessive alternating thrust in the reduction gear occurs when the vibratory stress in the gear teeth exceed the allowable limits established by the gear manufacturer.

c. Excessive Longitudinal Vibration

Excessive longitudinal vibration of the main propulsion system units (including condenser, piping, etc.) occurs when the vibration exceeds ±.25 g, or that level certified as satisfactory by the equipment manufacturer, whichever is the least.

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 2.1 are described in S.N.A.M.E. Code C-1, "Code for Shipboard Vibration Measurements" [5]. In this Code, longitudinal vibration measurements are called for at the following locations:

a. Thrust Bearing Housing.
b. Forward End of Bull Gear Shaft. This position will require a probe and provision for access to the gear case.
c. Gear Case Foundation. On top of the gear case foundation under the shaft centerline.
d. Gear Case Top. Over shaft centerline.
e. High Pressure Turbine. Attached to HP turbine casing at forward or after end.
f. Low Pressure Turbine. Attached to LP turbine casing at forward or after end.
g. 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 Natural Gas Co. for review.

HABITABILITY CONSIDERATIONS

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, but is used to define acceptable vibration levels for ship structures rather than to define the vibration levels acceptable to man, 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 the Interim Guide-Lines for Habitability Criterion proposed 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 velocity 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 as shown in Figure 2. 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 [12]. More recently, WG2 of ISO-TC108/SC2 adjusted their "Interim Guide-Lines for the Evaluation of Vibration in Merchant Ships" [10] to permit better agreement with the ISO Standard 2631 as shown on Figure 3. The upper limit shown is also consistent with Curve 2 on Figure 4, taken from the proposed S.N.A.M.E. "Ship Vibration and Noise Guide-Lines" [11]. Only minor adjustments will be required in future LNG specifications.

For comparison purposes a series of curves entitled "Ship Vibration Interim Guide-Lines for Habitability Criterion (September 1974), Comparison with Various Criteria (Peak Valued)," was compiled by Lloyd's Register. These curves, Figures 5 thru 11, show the alternate criterion used by B.S.R.A., Bureau Veritas, IRCN, etc., plotted against the 1974 ISO Proposed Criterion as shown in Figure 1 with the LNG limits. It should be noted that the proposed "Interim Guide-Lines" [10] and [11] do not differentiate between vertical and horizontal vibration, because the ISO WG members prefer to develop their guidelines by the use of reliable data obtained from shipbuilders and operators. However, preliminary data shown by the VDI, suggests we will have a lower criterion for horizontal vibration, as shown in Figure 12 and 13. This would appear to be more consistent with the previously used criterion, Figures 5 thru 11, than with the ISO Standard 2631 [12].

Reference to S.N.A.M.E. Bulletin 2-10 [3] should be replaced by the revised "Code for Shipboard Vibration Measurement" [5] or, in the near future, by the ISO Standard, "Code for the Measurement and Reporting of Shipboard Vibration Data" [6]. The requirements for vibration of Main Propulsion Systems are consistent with the technical standards developed by the Navy [15] and are based on potential damage or fatigue levels. Previous reference to tailshaft design requirements [14], was an outgrowth of studies conducted by the S.N.A.M.E. M-8 Panel on "Tailshaft Failures" and relates to designs employing shaft liners. Results of previous studies, on which this criteria, and the Navy shaft design procedure [16] are based, were discussed in the A.S.N.E. Transactions [17].

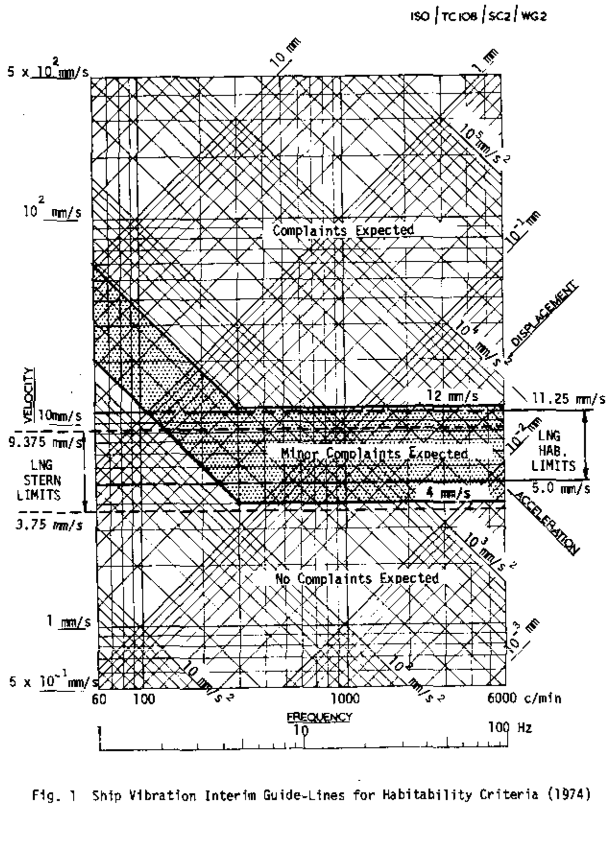


Fig. 1 Ship Vibration Interim Guide-Lines for Habitability Criteria (1974)

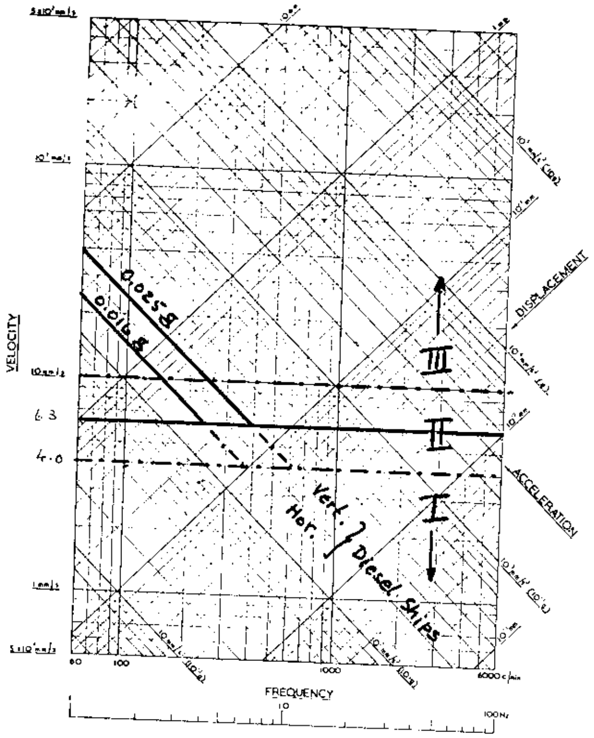
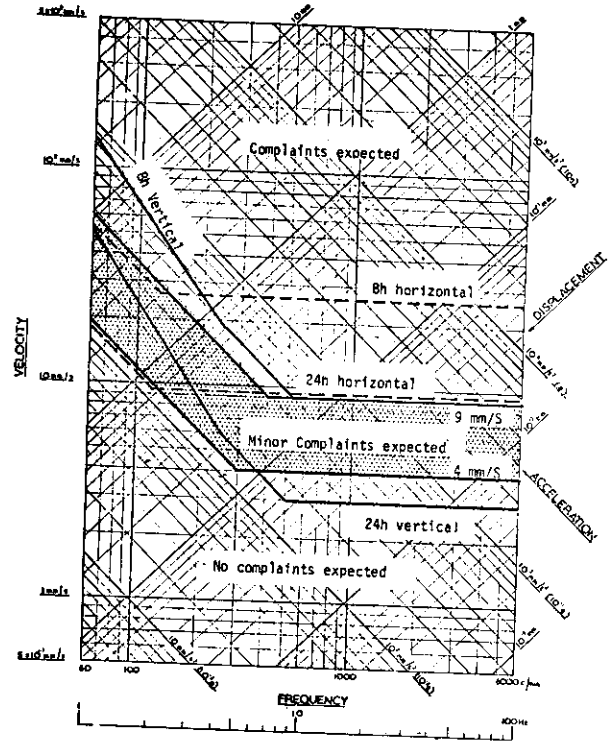


Fig. 2 Appraisal of Ship Vibration
Norske Veritas Proposal for Interim Guidelines



SHIP VIBRATION INTERIM GUIDE-LINES FOR HULL VIBRATION CRITERION (1974)

Fig. 3 Comparison with ISO/DIS 2631 Fatigue-Decreased Proficiency Boundary

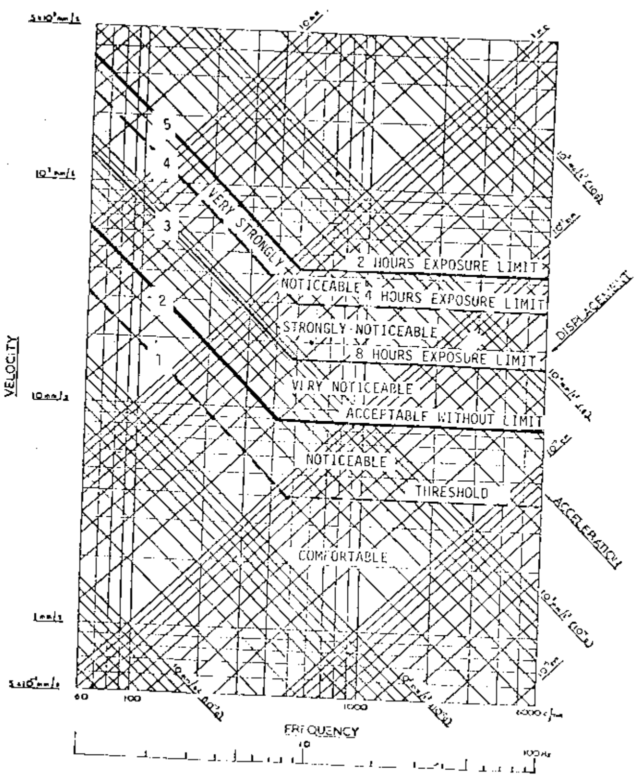
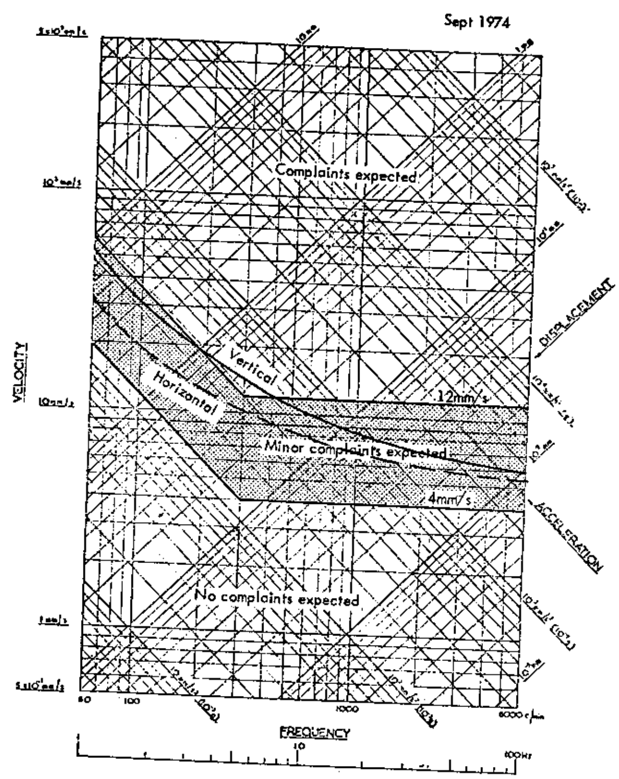
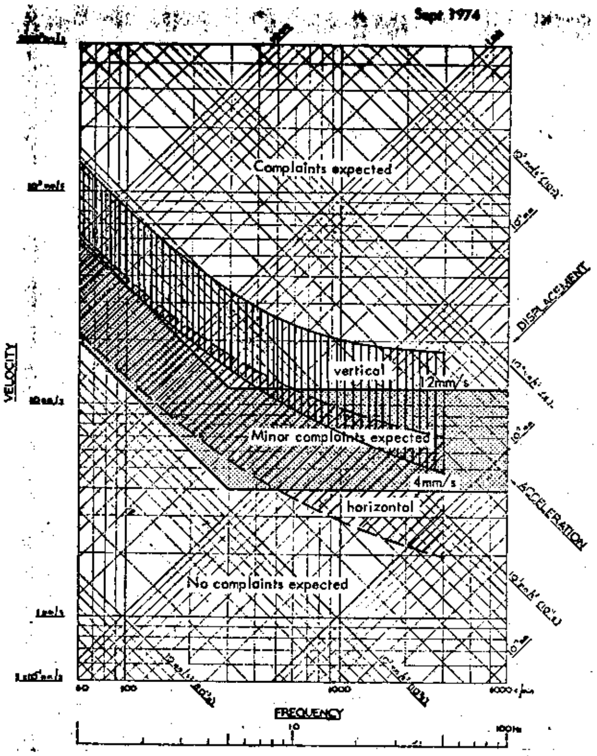


Fig. 4 Human Exposure Guidelines for Vibrations on Ships (Vertical and Horizontal)



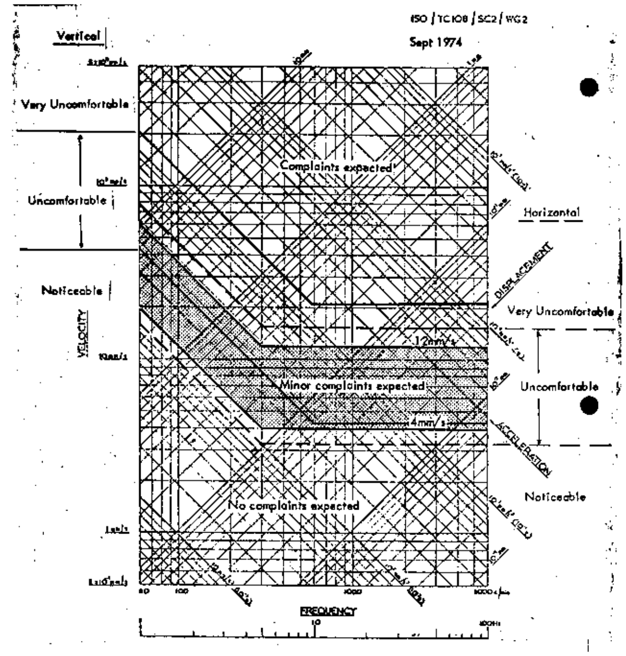
SHIP VIBRATION INTERIM GUIDE-LINES FOR HABITABILITY CRITERION (1974)

Fig. 5 Comparison with B.S.R.A. Limits



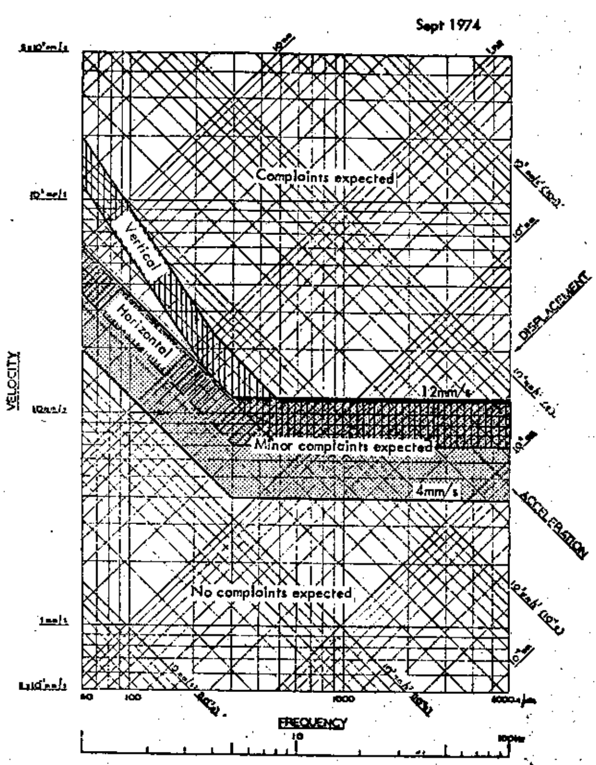
SHIP VIBRATION INTERIM GUIDE-LINES FOR HABITABILITY CRITERION (1974)

Fig. 6 Comparison with Bureau Veritas Vibration Limits for Crew



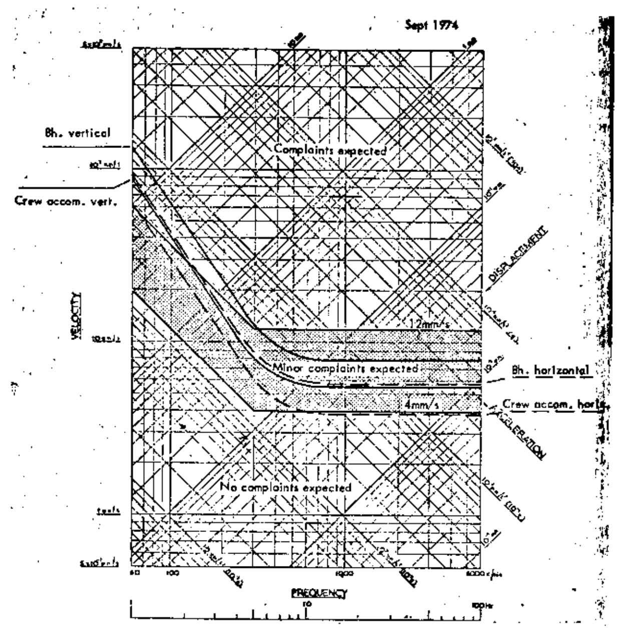
SHIP VIBRATION INTERIM GUIDE-LINES FOR HABITABILITY CRITERION (1974)

Fig. 7 Comparison with IRCN Curves



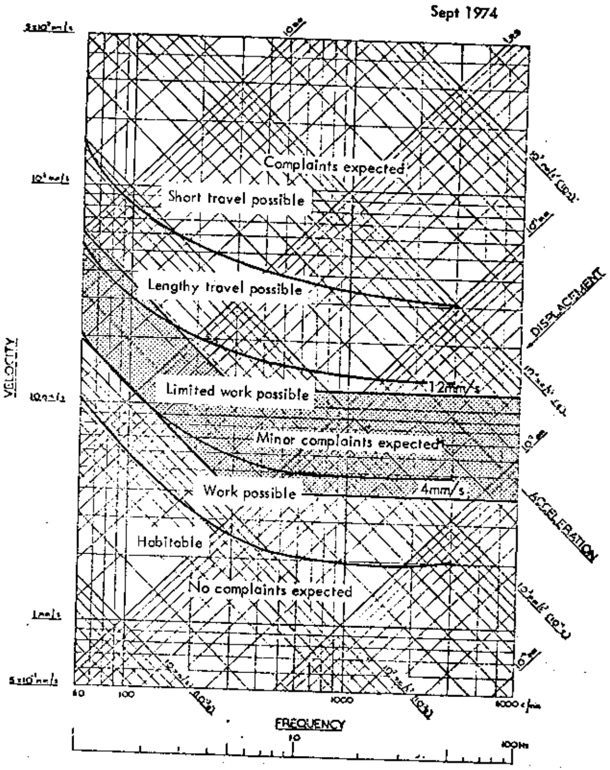
SHIP VIBRATION INTERIM GUIDE-LINES FOR HABITABILITY CRITERION (1974)

Fig. 8 Comparison with Japanese 1970 Proposal



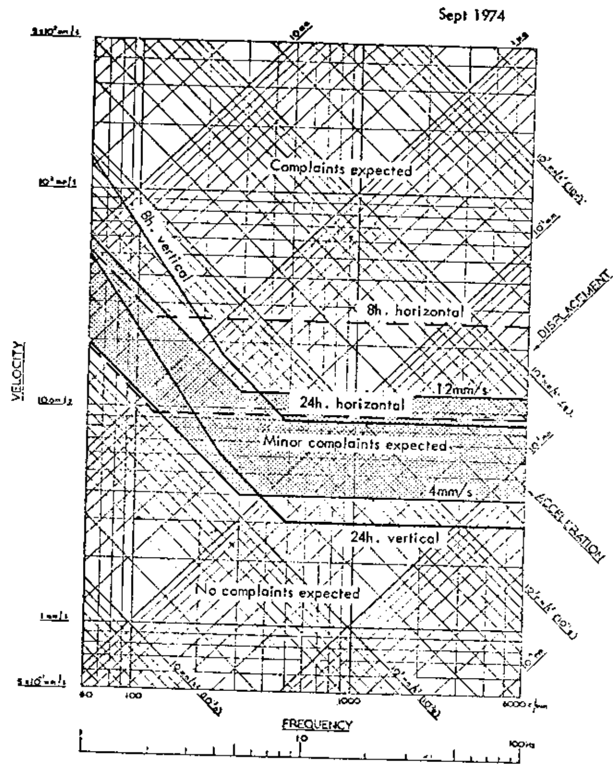
SHIP VIBRATION INTERIM GUIDE-LINES FOR HABITABILITY CRITERION (1974)

Fig. 9 Comparison with Lloyd's Register Vibration Limits for Crew Comfort



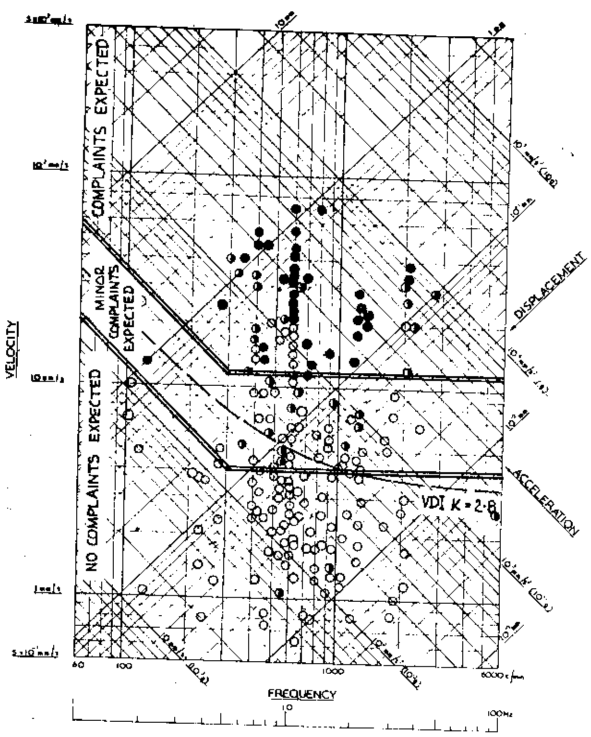
SHIP VIBRATION INTERIM GUIDE-LINES FOR HABITABILITY CRITERION (1974)

Fig. 10 Comparison with Verein Deutscher Ingenieure Curves



SHIP VIBRATION INTERIM GUIDE-LINES FOR HABITABILITY CRITERION (1974)

Fig. 11 Comparison with ISO/DIS 2631 Fatigue-Decreased Proficiency Boundary



DETAILS OF DATA PRESENTED

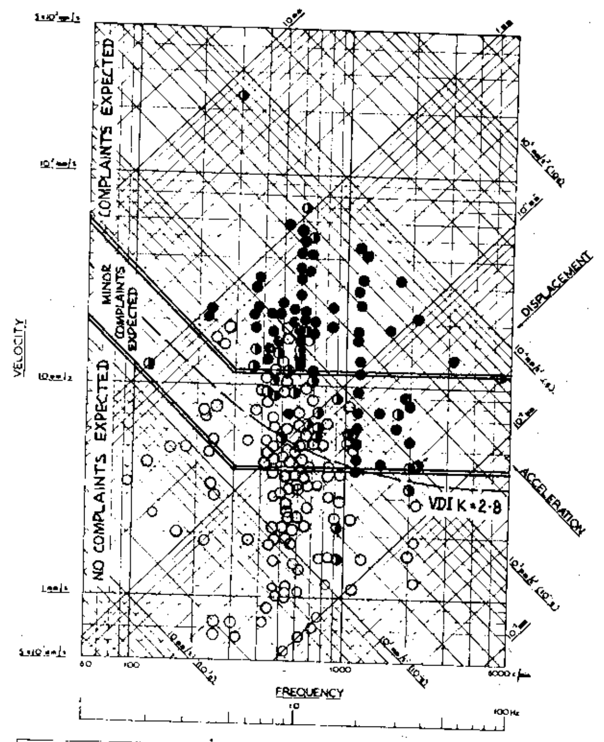
TO SEP 1975

VERTICAL VIBRATION : PROPOSED ACCEPTABLE LEVELS AND MEASURED DATA.

ISO SHIP GROUP — VDI K = 2·8

O NO COMPLAINTS ◑ SLIGHT COMPLAINTS ● STRONG COMPLAINTS

Fig. 12 Ship Vibration Data



DETAILS OF DATA PRESENTED

TO SEP 1975

HORIZONTAL VIBRATION : PROPOSED ACCEPTABLE LEVELS AND MEASURED DATA.

ISO SHIP GROUP — VDI K = 2·8

O NO COMPLAINTS ◑ SLIGHT COMPLAINTS ● STRONG COMPLAINTS

Fig. 13 Ship Vibration Data

THE MATHEMATICAL MODEL

The mathematical model used in the prediction of hull response is generally based on the availability of the technical data required, the status of the design, cost and time available. Recent advances in computer technology in handling large and complex structures have resulted in wide usage of the finite-element method for the prediction of ship hull response. This method however, requires structural details which are not normally available in the preliminary design phase. Also, the time required to perform such an analysis, the high cost for modeling and computation, plus the limitations of the input forcing functions, damping characteristics and empirical factors necessary to estimate actual ship vibration, would not appear to justify the use of the finite-element method, in the preliminary design phase.

An alternate method of predicting the response of the hull girder has been successfully used by NSRDC, for preliminary design studies. This method is referred to as the 20-Station free-free Beam Method, is described in NSRDC Report 1317 [18] and is based on Timoshenko's differential equation for the free lateral vibration of prismatic bars, and the differential equation for torsion [19]. In this analysis, the ship is represented by a non-uniform, continuous, 20-station free-free beam having the same mechanical and elastic properties as the physical ship. The ship's structural weight, machinery, cargo and added masses (entrained water) are lumped at the half-stations. These masses are connected by beam segments which possess the same elastic properties as the corresponding ship sections. A detailed description of the development of the parameters for this program is presented for a Coast Guard Icebreaker in Marine Technology [20].

Natural frequency and hull response calculations can be carried out on existing programs such as NSRDC General Bending Response Code 1 [21], NASTRAN, STARDYNE or by use of the Electrical Analog [22]. Natural frequency calculations are useful in selecting optimum shaft RPM to avoid vibration of the lower hull modes at important operating speeds, when excited by unbalanced forces at shaft frequency. The natural frequencies are obtained by the use of a unit exciting force to an undamped system. Hull response calculations may be predicted by driving the mechanical system by the predicted propeller forces and system damping, discussed later. Specific examples in which the 20-Station Free-Free Beam Method have been successfully used, include the DD963 Class Destroyer [23] and the 125,000 CM LNG Carrier [24]. Results of full scale tests and comparison with calculations were briefly reported in Ship Structures Symposium '75 by Noonan [14].

In the finite-element method, the aft part of the ship structure, including the propulsion system and superstructure, is modeled in utmost detail, using numerous plate and beam finite-elements and lumped weights. The forebody is modeled by a continuous beam having the same elastic properties as the corresponding ship structure, in a manner similar to that used for the free-free beam method. Thus hull or deck platings are represented by the plate finite-elements; deck plating stiffeners, hull plating stringers, and supporting stanchions, etc., are represented by the beam finite-elements; the propulsion system is modeled by beam elements simulating shafting and bearing supports, and by lumped weights simulating propeller, turbine and gearings, etc. as described by Pauling [25].

The hull response of the ship can be computed by exciting the finite-element model using the blade frequency bearing forces input at the propeller and the blade frequency pressure forces input at hull surface area in way of the propeller.

The finite-element method for ship structure modeling represents a promising potential of advancement in ship vibration prediction in that it not only is capable of predicting modes of vibration of the main structure superstructure and the propulsion system, it also is capable of predicting the local vibration of the major bulkheads, deck plating or the support structure for the major machinery, provided a good representation or modeling is affected. However, its requirement for detailed structural design, for a proper modeling of the real ship, limits its application in the preliminary design stage. The finite-element method of analysis is however, considered particularly well-suited for final design analysis in which structural details are established and at which time the vibratory characteristics of major substructures and local structural elements are evaluated.

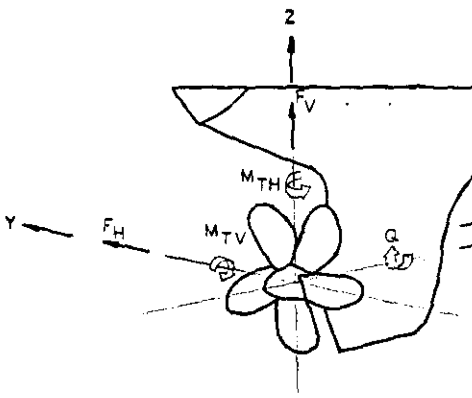
INPUT OR FORCING FUNCTIONS

In addition to the basic design purpose of generating steady thrust for the ship's propulsion, the marine propeller also generates fluctuating hydrodynamic forces and moments due to its operation in a non-uniform wake and due to the passage of the blades close to the hull and appendages. These fluctuating forces and moments are usually referred to as propeller forces and are of blade frequency and its higher harmonics, although the higher harmonics are normally of secondary importance. These propeller forces are in turn categorized into two groups, the bearing forces and the hull pressure forces.

1. Bearing Forces

The bearing forces originate from the non-uniformity of the wake in the plane of the propeller disc. The strength of the various harmonics of the wake offsets the magnitude of the bearing forces and influences the choice of the number of propeller blades. The relative strength of the various orders of wake harmonics indicates the relative strength of the blade frequency forces. The wake in turn is influenced by the design of the hull form. An optimum design of hull form would reduce the non-uniformity of the wake, thereby reducing the magnitude of the bearing forces. The bearing forces excite the ship through the propulsion shafting/bearing system, and are fully described by six components as illustrated in Figure 14. As shown in Figure 14, with the origin of axes at the center of the propeller, these components are the thrust and torque in and about the longitudinal or fore-aft axis; the horizontal bearing force and the vertical bending moment in and about the horizontal or athwartship axis; the vertical bearing force and horizontal bending moment in and about the vertical axis.

The vertical and horizontal bearing forces result from the propeller torque, while the vertical and horizontal bending moments are due to the propeller thrust eccentricity from the center of the propeller.



X, Y, Z axes = Fore-Aft, Athwartship and Vertical axes, respectively

T = Thrust

F_H = Horizontal Bearing Force

F_V = Vertical Bearing Force

Q = Torque

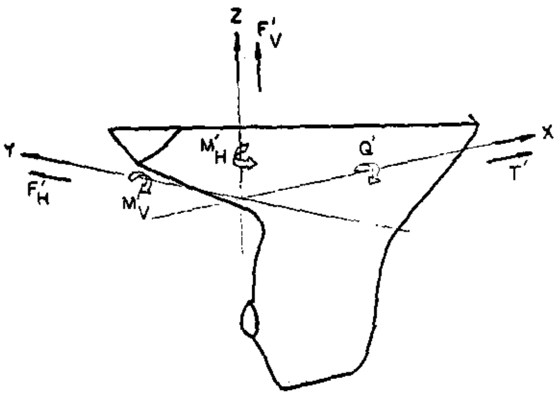
M_{TV} = Vertical Bending Moment

M_{TH} = Horizontal Bending Moment

Fig. 14 Description of Bearing Forces & Moments

2. Hull Pressure Forces

The hull pressure forces originate from the pressure variation caused by the passage of propeller blade tips close to the hull and appendages. The hull pressure forces are expected to be affected by propeller-hull clearance by blade loading and by changes in the local pressure field around the blade. Thus the occurrence of blade cavitation will drastically increase the pressure forces. In some cases, a 20 to 40 times increase of hull pressure forces due to cavitation has been observed in experimental measurements, as compared to non-cavitating condition [26]. The pressure forces excite the ship thru the hull bottom surface in way of and adjacent to the propeller. The pressure forces are fully described by six components, namely, the longitudinal force and moment in and about the fore-aft axis; the horizontal force and vertical moment in and about the athwartship axis; and the vertical force and horizontal moment in and about the vertical axis, as illustrated in Figure 15.



X, Y, Z axes = Fore-Aft, Athwartship and Vertical axes, respectively

T' = Longitudinal Force

F'_H = Horizontal Force

F'_V = Vertical Force

Q' = Longitudinal Moment

M'_V = Vertical Moment

M'_H = Horizontal Moment

Fig. 15 Description of Hull Pressure Forces & Moments

3. Shaft Frequency Forces

In addition to the aforementioned blade frequency hydrodynamic forces, the propulsion system also generates some low-frequency mechanical forces which are associated with shaft rotational speed. These shaft frequency forces may result from one or more of the following causes:

a. Shaft unbalance;

b. Propeller unbalance;

c. Propeller pitch error;

d. Engine unbalance (for Diesel driven ships);

e. Misaligned shafting;

f. Bent shafting;

g. Journal eccentricity.

The most likely causes of shaft frequency forces are those due to a, b, c, and d above. The other possible causes are not as likely to occur if reasonable specifications, workmanship and inspection procedures are exercised.

Shaft frequency forces occur within a low-frequency range. They are, however, of considerable concern since they may be of large magnitude and may excite one of the lower hull modes at or near full power.

4. Effect of the Propeller Forces

The alternating blade frequency thrust of the bearing forces provides the principal excitation to the propulsion system in the longitudinal mode, while the blade frequency torque constitutes the principal excitation to the propulsion system in the torsional mode. The blade frequency vertical bearing force, when vectorially combined with the blade frequency vertical pressure force, provides the total vertical force which excites the hull in the vertical direction. Similarly the horizontal bearing forces, when combined with the blade frequency horizontal pressure forces, provide the major contribution for exciting the hull in the horizontal direction. The vertical and horizontal forces and the distance from the neutral axis of the hull combine to excite the hull torsionally. Longitudinal hull pressure forces and alternating thrust entering the hull thru the thrust bearing will combine to excite the hull in the longitudinal direction. Shaft frequency forces, generally assumed to equal the allowable unbalance tolerances, will excite those lower hull modes which occur within the operating speed.

FACTORS AFFECTING THE PROPELLER FORCES AND THEIR CONTROL IN THE PRELIMINARY DESIGN STAGE

In most designs, the principal vibratory forces will be at blade-frequency and harmonics of blade-frequency. These frequencies will generally result in forced, rather than resonant vibration of the hull. It is important, therefore, to minimize the input forces generated by the propeller and, at the same time, avoid objectionable resonant responses in the propulsion systems and in the major substructures or structural components. To optimize the design, by minimizing the input forces to the hull, the following factors should be closely examined, preferably by model studies:

1. Stern Configuration

The stern configuration significantly influences the wake field which in turn effects the propeller forces generated. Unless previous studies clearly indicate the inherent advantages of a given stern configuration for minimizing the propeller forces generated, wake studies should be conducted on competitive models. Typical examples of such studies are shown by Hadler [27] and Noonan [14]. The total forces generated necessarily also include the hull pressure forces. Therefore, estimates of hull pressure forces and the effects of cavitation on these pressure forces must also be included in the decision making process.

2. Hull Form

The details of the hull form selected can also significantly effect the propeller forces generated. The longitudinal velocity component of the wake generally follows the buttock line and represents the main contribution to the wake harmonics which in turn influence the magnitude of the bearing forces. Therefore, to optimize the hull form, care should be taken to insure the buttock line does not produce any blocking effect to the water flow. The more uniform wake will result in a reduction of both the bearing forces and pressure forces. The effect of improved flow was dramatically shown on a single screw LNG carrier, in which a fin was added to the model, as a substitute for improved hull form which was already committed [14]. In this instance, a reduction in hull pressure force of 80% was obtained.

3. Propeller Clearances

Insufficient propeller-hull clearance will induce excessive hull pressure forces. The longitduinal clearance, the distance between the trailing edge of a strut or skeg and the leading edge of the propeller blade, is more important than the hull-tip clearance. A minimum tip clearance of .2D and a clearance of .5D between the strut and propeller-blade should be used as design objectives.

4. Cavitation

Propeller cavitation inception will increase the pressure force tremendously. It is known from numerous model tests, and confirmed by available full-scale trial data, that a cavitating propeller will produce vertical blade-rate pressure forces ten times as large as the corresponding vertical bearing force. Comparatively, the blade frequency pressure force at non-cavitating condition may be of the same order of magnitude as the vertical bearing force. As a larger excitation force implies a higher level of hull vibration, it is therefore mandatory that due attention be placed on the prevention or suppression of propeller-blade cavitation. This would include effective design of the buttock line to insure a more uniform wake and the prevention of excessive blade thrust loading.

5. Number of Propeller Blades

An harmonic analysis of the wake should be performed as a guide to the selection of the number of propeller blades which would minimize the strength of the blade frequency bearing forces. The number of propeller blades selected should not coincide with strong components of wake. Before selection however, care should be taken to insure that the number of blades best suited to minimum bearing forces and corresponding minimum hull vibration, are acceptable to the vibratory characteristics of the propulsion system, particularly in longitudinal vibration. In open designs, such as an open-transom design, the strength of the wake harmonics will generally reduce with an increase in the number of propeller blades, and the optimum choice will be dictated by the predicted response of the propulsion system. An odd-number of blades will tend to minimize the reinforcement of two vertical blades operating in a non-uniform wake and result in lower blade-frequency thrust and torque. For unskewed propellers, an even number of blades will generally produce lower vertical and horizontal bearing forces [28].

6. Propeller Skew

Among the more recent developments in propeller design is the application of significant amounts of skew. Although still considered to be in the development stage, highly skewed propellers, which have good cavitation and vibration characteristics, have been successfully used to ameliorate the occurrence of serious vibration and cavitation problems, [28] and [29]. In new designs, however, the considerations previously discussed should be employed as the primary approach to the limitation or propeller generated vibratory forces. The application of high skew should be considered for further improvement of a good design and not as a substitute for good design procedures.

PREDICTION OF PROPELLER FORCES

1. Blade Frequency Forces - Calculation Methods

In the prediction of the blade frequency bearing and pressure forces, various theoretical calculation methods as well as experimental methods are available. For the theoretical calculation of bearing forces, typical approaches include a two-dimensional quasi steady-state method and a method based on the three-dimensional unsteady lifting surface theory. The former method involves the assumption that the frequency of the oscillations of the inflow velocity is sufficiently small to allow a quasi steady-state analysis, [30] and [31]. The latter method attempts to account for the unsteady effects attending the three-dimensional flow generated by the blades as they move thru the spatially non-uniform wake, including the effects of interferences between the blades [32]. These calculation methods utilize the wake and the propeller characteristics as input data and give as output the mean and alternating bearing forces, including the blade frequency and the higher harmonic forces. Boswell gives a good presentation of the various calculation methods and their application to design [33]. Research work on predicting the blade-frequency hull pressure forces is not as advanced as that for estimating bearing forces. References [34] and [35] present some attempts in this aspect. However, the effect of propeller cavitation, which has only recently been recognized as capable of tremendously magnifying the pressure forces, is not included in these investigations. Since propeller blade cavitation is generally an invariant occurrence for modern high powered ships, the ignorance of the cavitation effect on the pressure forces puts doubts on the accuracy of these prediction methods. Noordzij investigated the pressure field induced by a cavitating propeller [36]. His work, when combined with that by Breslin and Eng, [34], or by Breslin and Tsakonas, [37], is capable of predicting hull pressure on the aft body hull bottom area near the propeller. Based on a single ship data, theoretical prediction of hull pressure using Noordzij's method does not show satisfactory comparison with experimental data obtained by towing tank measurement. It's comparison with full-scale trial data is unavailable due to lack of data.

2. Blade Frequency Forces - Experimental Method

Experimental methods for predicting blade frequency bearing and pressure forces utilizes towing tank facilities and scaled models of ship and propeller. Until the recent entry of the large NSMB depressurized towing tank in the early 70's [38], open water towing tank and depressurized cavitation tunnels were invariably utilized for the experimental measurement of bearing and pressure forces, respectively. The depressurized towing tank can be used to measure both the pressure and the bearing forces. In this tank the pressure can be lowered to truly simulate the actual cavitating condition for the ship/propeller model. In addition, it represents an important improvement over the conventional cavitation tunnel in that the dimensions of the tank are large enough to avoid unpredictable scale effects on the ship's hull with respect to blade cavitation and flow separation phenomena.

For the experimental measurement of bearing forces the ship propeller model is towed in the open or depressurized towing tank at the simulated operating condition. The bearing forces and moments are measured by means of six-component balance installed on the propeller shaft. Data reduction then yields the mean and unsteady bearing forces and moments, at blade frequency and higher harmonics.

For the experimental measurement of pressure forces, either the ship/propeller model is towed in the depressurized towing tank or the scaled model of the ship's aft body, together with the propeller is built into the cavitation tunnel. In both cases, the cavitating condition as expected for the actual operating condition is simulated by reducing the air pressure in the tunnel or the towing tank. Hull pressure on the stern area or the aft body is recorded by placing pressure transducers at various strategic locations. These data are further analyzed to give blade frequency pressure and its higher harmonics. Integration of these pressure data over the appropriate hull surface area will give rise to blade frequency pressure forces and moments, and their higher harmonics.

Recent correlation studies on propeller forces for the LNG Carriers have indicated reliable accuracy of the experimental methods used to measure the blade frequency bearing and pressure forces. Considering all factors, the model studies of the bearing and hull pressure forces, including the effects of cavitation are considered more reliable than theoretical calculations.

HULL DAMPING COEFFICIENTS

Damping plays a very important role in the study of ship vibration since the resonant amplitudes of a hull depend not only on the magnitude and location of the exciting forces, but also on the magnitude of the damping present in the ship and surrounding water. Comprehensive understanding and reasonable representation of damping will minimize the discrepancies between computed and measured hull responses.

Ship damping may originate from various sources which include structural or hysteretic damping associated with internal friction in the hull structure material; friction due to relative slipping and sliding between dry surfaces;and viscous damping due to interaction between the hull and the surrounding water. Due to the complex nature of the hull damping, empirical treatment of the subject is utilized by various investigators, [39] and [40]. McGoldrick [39] suggested that for flexural vibration the damping coefficient C is increasing with frequency, where C represents a damping force per unit velocity, per unit length. Kumai reported damping factors for various ships in vertical vibration and suggested the frequency dependency of the logarithmic decrement for vertical vibration [40].

Within this context of empiricism, Foster and Alma conducted anchor drop tests to excite transient vibrations of the ship's hull at low frequencies [40]. Their experimental observations indicated that damping factors varied with frequency and had proposed an empirical formula for damping factor as $c/\mu\omega = 8.5 \times 10^{-4}\omega$. This finding may be considered as representative of the state-of-the-art as far as knowledge of damping is concerned. The damping factor is the reciprocal of the resonant magnification factor Q, that is

$$Q = \frac{1}{c/\mu\omega}$$

This factor, shown on Figure 16, has been successfully used in the prediction of hull vibration in both the free-free beam model and in the finite-element calculations. When predicting the response of major substructures or local elements, a hysteretical damping factor, equivalent to five percent critical is recommended.

EMPIRICAL FACTORS

The measurements and evaluation of shipboard vibration concerns itself with maximum repetitive amplitudes since it is this value which is pertinent, whether we are concerned with the fatigue of metal, malfunction of equipment or physiological response to environmental vibration.

The purpose of design calculations is to afford the design engineer the opportunity to evaluate the anticipated vibratory characteristics against the measured characteristics. To effectively do this, a number of empirical factors will be required since design calculations presume a sinusoidal input, steady flow conditions, and do not properly account for the impact of cavitation, ship maneuvers,

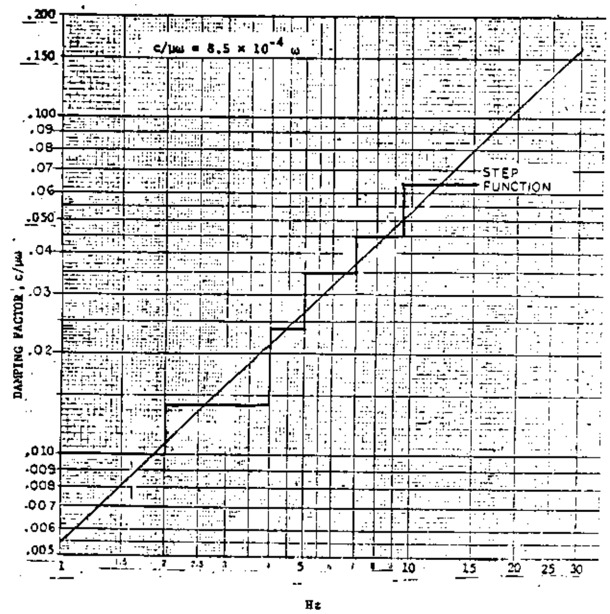


Fig. 16 Hull Damping Versus Frequency

or the normal modulation of the shipboard vibration records.

Although existing data is extremely limited a few of the more important factors which have been satisfactorily used in the preliminary design phase are given here for information purposes:

1. Signal Modulation

Shipboard vibration measurements portray a significant signal modulation, even under the most ideal sea conditions. Under sea conditions specified for trials in the vibration test codes [5] and [6], a factor of two to three has been found to exist between an "average" or estimated "sinusoidal" signal and that observed as the "maximum repetitive" value. These values pertain to straight ahead, steady-speed operation, with minimum rudder and a "Sea State 3 or less." The factor of two is based on data observed on large tankers and the factor of three has been found to be representative of Destroyers.

2. Ship Maneuvers

Unlike commercial ships, Navy Combatant types, particularly Destroyers, frequently are required to conduct sharp, high-speed maneuvers. Under such conditions the ship is likely to require all vital equipment to perform effectively. Under high speed hard-turn maneuvers, the magnification factor of the steady-speed runs can be expected to reach a factor of three or better over the steady-speed conditions noted above, in the case of Destroyers. For larger combatant types, this factor may be reduced to a factor of two for large ships of dimensions of a carrier. A factor of two may also be expected in large single-screw commercial vessels.

3. Cavitation

As has been discussed earlier, the presence of cavitation, although difficult to predict, can result in a significant rise to the vibration levels noted aboard ship in the range of 85 to 100 percent of full-power rating. In the case of Destroyer types, on which most data is available, a magnification factor of three has frequently been observed at full-power, over that response obtained by assuming the vibratory forces to follow an RPM^2 function. In particular cases such as in the case of the U.S.S. SPRUANCE, DD963, in which particular emphasis was placed on the avoidance of cavitation [42], a lower factor would be in order. In that case, in the preliminary design study, the hull pressure forces were assumed to equal in magnitude to the vertical bearing forces, and in phase with them and no additional factor was applied.

4. Longitudinal Vibration of Machinery Systems

To effectively relate theoretical predictions of the vibratory characteristics of shipboard systems, to the actual underway performance characteristics, the correlation of calculations with full-scale shipboard studies is required, on a statistical basis. This approach offers significant opportunities to simplify the preliminary design procedure. As an example, it has been reported by Zaloumis and Antonides [43], that on a representative number of Navy Surface Ships, that the resonant magnification factor for longitudinal vibratory response of the main propulsion system, varied between nine and twelve. The value of data of this type is readily apparent and points the way toward a more efficient preliminary design process.

FULL-SCALE TEST AND EVALUATION

The method of measuring and evaluating shipboard vibration has evolved over many years and is reflected in the recent S.N.A.M.E. "Code for Shipboard Vibration Measurement" [5]. The procedures and methods of measurement and evaluation presented in this document have been accepted universally and have been used as the basis for the proposed ISO Standard "Code for the Measurement and Reporting of Shipboard Vibration Data," [6], which is expected to appear shortly as an International Standard. The latter document includes additional measurements, particularly related to hull pressure measurements, lateral vibration of shafting systems and vibration measurements specifically related to diesel engine drive systems.

Complete full-scale studies are required for the evaluation of "First of Class" designs. These studies should be conducted in accordance with the prescribed codes. Limited studies are required on follow-on ships. The purpose of these studies is intended to:

1. Confirm the adequacy of the design, relative to the design requirements or specifications;
2. Determine corrective action, where required;
3. Obtain technical data to permit the continued development of the design procedure and the improvement of empirical factors.

A review of the factors given, demonstrates the complexity of the problem of predicting response characteristics of shipboard vibration. As a direct result, it becomes increasingly apparent that the total problem is not amenable to a "cookbook" design procedure, but rather is dependent on a collection of research data and a judicious application of that data by experienced vibration engineers. Full-scale testing is therefore a major factor in developing and improving the design procedure. Many supplemental studies conducted in connection with routine full-scale studies on new designs, some of which have been outlined in [14] and [44], are considered important in the ultimate objective of developing and improving a simplified preliminary design procedure.

THE DD963 AND THE EL PASO LNG CARRIER PROGRAMS

The DD963 Program

The application of specific limitations to hull vibration was an innovation in the development of the DD963. Specific limits were placed on hull vibration in the form of target and reject amplitudes. A detailed vibration program was developed [45], which included a "Preliminary Hull and Machinery Vibration Analysis" [46] 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 test program conducted in February 1975.

In the earlier paper, "An Assessment of Current Shipboard Vibration Technology" [14], limited data was presented because of classification restrictions. However, the information presented did provide an insight 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. Judgement on the effectiveness of the program, which leaned heavily on the experience of the investigators, is best formed by an examination of the data previously presented and the test results. Because of current security restrictions, no additional data is presented in this paper. However, the following observations may be readily made from the data previously presented:

1. Predicted hull frequencies were closely confirmed by test.
2. Simplified method of predicting hull frequencies [47] agreed well with the 20 station "beam model" [20].
3. Good agreement was achieved between estimated and calculated propeller forces.

4. Target levels for hull vibration were readily met.

5. Observed levels for hull vibration fell between theoretical values (based on sinusoidal response) and predicted amplitudes (which included empirical adjustments required to conform to test requirements).

6. Very good agreement in the longitudinal vibration characteristics of the propulsion system was achieved between the preliminary design analysis [46], the finite-element analysis [48], and that observed by strain gage measurements observed during trials [49].

In general, the design procedure employed to predict the vibration characteristics of the hull and propulsion machinery systems of the DD963 was similar to that described in this paper. However, based on accumulated data available at the David Taylor Ship Research and Development Center, the hull form was established and the optimization of the details of struts, rudders, etc. were carried out at the Navy Facility. For preliminary estimates of propeller forces, calculations were based on an assumed wake taken from a similar hull form and a standard series propeller. Final calculations were based on actual wake survey and propeller details. In this instance and in the absence of an adequate prediction procedure, 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 since particular efforts were made to avoid cavitation on this ship.

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. Similarly, the support systems (foundations and mountings) were analyzed for most equipment installations. 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 DD963 was considered unusually free of troublesome vibration.

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 was 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.

The design approach described earlier was found to be quite effective in the development of the El Paso LNG Program, which at that time, was an advancement of the state-of-the-art. The earlier report [14] described in some detail the development of the initial design. Some of the more important steps taken in that design are briefly discussed here for purposes of evaluating the approach used.

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 (NSMD):

   Model 4141 - Modified Hogner - Figure 17
   Model 4147 - Conventional - Figure 18
   Model 4148 - Open Transom - Figure 19

   The circumferential distributions of Longitudinal Velocity Components obtained by NSMB for each model were shown in the earlier report [14]. 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 Wagenigen B-Series for 5-bladed propellers. As in the case of the DD963 the propeller forces and moments were developed for comparison purposes. Results taken from reference [50] are shown in Table I.
   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 Dry Dock Co. for the 125,000 CM 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 5-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 [51] are shown in Table II, along with the calculations made by Det Norske Veritas (DNV) on Model 4171 (slightly longer than 4221A), taken from reference [52], and the original results given for the project hull, Model 4148, as previously shown in Table I. 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 [51]. The horizontal and vertical hull forces are normally the most significant in regard to hull vibration. In this instance, on the open

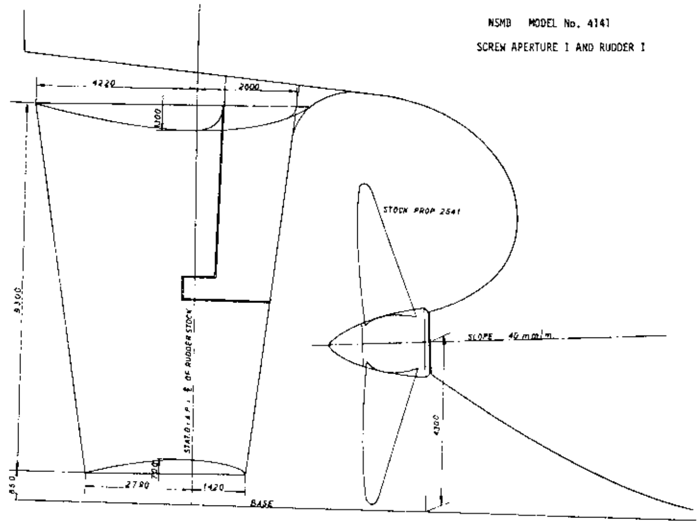


Fig. 17 Stern Configuration of Model 4141 DRAWING N° M4141-6

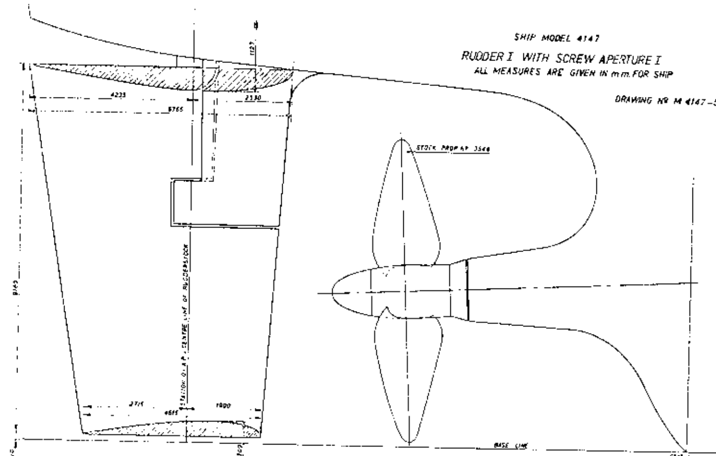


Fig. 18 Stern Configuration of Model 4147

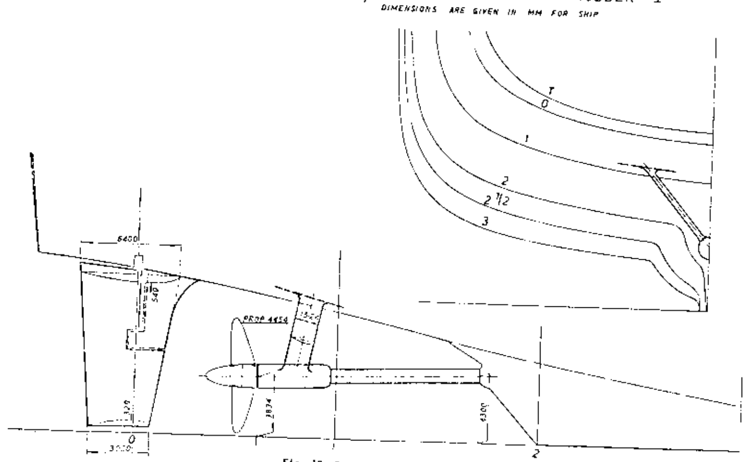


Fig. 19 Stern Configuration of Model 4148 DRAWING NO M4148.8

TABLE I

125,000 CM 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
$\bar{T}$ Thrust, lbs.	635,800	472,900	451,600
$\tilde{T}$ ±lbs.	39,760	31,820	17,520
$\tilde{T}/\bar{T}$ ±%	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
$\tilde{Q}/\bar{Q}$ ±%	4.10	5.05	2.74
$\tilde{F}_h$ Brg. Force, ±lbs.	6,750	3,900	4,950
$\tilde{F}_h/\bar{T}$ ±%	1.06	.82	1.11
$\tilde{F}_v$ Brg. Force, ±lbs.	3,190	1,660	2,134
$\tilde{F}_v/\bar{T}$ ±%	.50	.35	.47

TABLE II

F-D 125,000 CM LNG Ship with 5-bladed Propeller
Comparison of Measured and Calculated Propeller Forces and Moments

	MODEL 4221A MEASURED NSMB [51]	MODEL 4171 CALCULATED DNV [52]	MODEL 4148 CALCULATED NKF [50]
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	450	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	

transom stern without cavitation, the horizontal force was negligible while the vertical hull pressure force at blade-frequency was ±2,660 lbs, approximately equal to the bearing force ±3,310 lbs shown in Table II. Without cavitation, only the first harmonic was important and when combined vectorially with the bearing force, the resultant vertical force was ±2,620 lbs just a little smaller than the bearing force alone. Referring back to the DD963, you 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:

$\tilde{F}_{v1}$ from ±2,660 lbs to ±19,200 lbs,

$\tilde{F}_{v2}$ from ±180 lbs to ±13,900 lbs and

$\tilde{F}_{v3}$ 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 [53]. 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.

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 was 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. Modal 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 [54]. 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 were conducted in July 1975, during Builder's and Acceptance Trials of the "PAUL KAYSER" and included:

   Hull and Machinery Vibration Measurements by NKF.
   Propeller Shaft Strain Gage Measurements by NKF.
   Hull Pressure Forces by F-D.
   Underwater TV of Cavitation by DNV.
   Vibration and Noise Habitability by F-D and NKF.
   Proximity Shaft and Oil Pressure Measurements by NKF.
   Vibration Generator Studies by F-D and NKF.

   Results of the trials, based on the Preliminary Report of August 12, 1975, were included as a Supplement to the earlier paper on "An Assessment of Current Shipboard Vibration Technology" [14]. The final report for the PAUL KAYSER [55] was published in November 1975.

The principal results indicated:

a. The level of forced hull vibration, as measured at the aft perpendicular, was well within the recommended design objectives. The vibration of the hull did not exceed 50% of the recommended criteria, when delivering 45,000 SHP.

b. The calculated amplitudes of hull vibration, using the 20 Station Beam model [56] showed good comparison with test results and that predicted by Bureau Veritas by finite element analysis.

c. A sharp resonance of the deck house, in the fore-and-aft direction, coupled with the vertical response of the hull, was determined to be related to resonances of the radar antenna and mast. Subsequent shaker studies conducted on the El Paso SONATRACH led to identification of the problem and corrective action and was reported in the SONATRACH vibration report [57].

d. Significant increases in vertical hull vibration during Builder's Trials (300%) were noted and related to increased cavitation thru the underwater TV studies conducted by DNV [58]. The large increase was originally assumed to be related to a section of the launching cradle, which was still attached to the bow during the Builder's Trials. Later studies on the SONATRACH indicated the difference in amplitude between Builder's Trials and Official Trials was reduced to a factor of 2:1 vs 3:1 observed on the KAYSER.

e. Hull pressure measurements reported by IRCN [59] and vibration generator studies conducted during the trials, provided correlation between hull pressure forces, cavitation and hull vibration [60].

f. The maximum alternating thrust, measured by strain gage was observed to be ±21,500 lbs. This represented a peak value, which allowing for the estimated amplification present, would confirm the estimated sinusoidal input force of approximately ±7,000 lbs.

g. The fundamental longitudinal frequency of the main propulsion system was predicted to be well above the maximum operating speed of 108 RPM. The test results confirmed that this was the case but the actual resonant frequency was of course, not determined by test.

h. Special studies were conducted on both the PAUL KAYSER and SONATRACH to gain insight into potential problem areas reflected in recent failures experienced on oil-lubricated strut bearings and seals on high-horsepowered ships. Special measurement transducers included:

   Velocity gages on the strut bearing to measure vibratory displacement.

   Non-contact proximitors to measure relative motions between shaft and bearing.

   Pressure gages to measure sea-water and oil-pressures on both sides of bearing seals.
   These measurements were met with limited success, but were not particularly pertinent to the evaluation of the design procedure under discussion.

THE AVONDALE AND NEWPORT NEWS LNG DESIGNS

The following El Paso 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 [61]. 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 [62]. A portion of this difference may be attributed to the difference in test conditions. The F-D model was tested in open basin, while the N.N. model was tested in the new Vacuum Tank, both at NSMB. Recent trials conducted on the El Paso SOUTHERN, the first of the Newport News hulls, in March of 1978, did not appear to support this difference. Preliminary results indicated hull vibration to be similar to that observed on the France-Dunkerque ships.

The total test program planned for all three designs, together with the extensive analyses conducted, should materially contribute 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 the HS-7 Panel and supported by the Hull Structures Committee of the S.N.A.M.E. depends.

GENERAL OBSERVATIONS (Repeated from "An Assessment of Current Shipboard Vibration Technology" [14])

An assessment of current shipboard vibration technology, with particular reference to the work carried out on the DD963 and LNG programs, leads us to some general observations, the more important of which are:

1. The control of shipboard vibration (hull and machinery) indicates the primary effort should be directed at the control of 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 criticals 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 forced response greater than 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, is 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.

13. More simplified analyses, such as the 20-Station beam model, have been found useful in preliminary design studies.

In a more general context we may note that in many cases in the past the presence (or absence) of serious vibration aboard ship has been a matter of chance and was only corrected by major surgery, if at all. Although we are still a long way from our ultimate objective, we have 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 hydrodynamic unbalance and cavitation. At this time, it seems safe to say that we have not fully integrated our present technical knowledge into our 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 are now underway. Specifications or requirements have been laid on in a number of cases, such as 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. True, the vibration studies are not always specifically defined. However, we can already recognize the progress toward a more standardized approach.

FUTURE RESEARCH

The current test plan scheduled for the LNG program, as identified earlier, includes:

1. Hull and Machinery Vibration.
   In addition to the conventional hull and machinery vibration measurements prescribed by the "Code for Shipboard Vibration Measurements" [5], and which will be used to correlate actual ship and machinery response against predictions, the following supplemental measurements should be made:

   a. Alternating thrust in the propeller shaft.

   b. Fore and aft vibration of the shaft strut.

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

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

2. Hull Pressure Forces for correlation with predicted forces obtained by calculation and model testing.

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

4. Cavitation studies by underwater TV for correlation with laboratory model studies.

5. Vibration and Noise Habitability measurements for comparison 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 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 [63]. This program, primarily sponsored by the Maritime Administration, has explored the use of 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 <u>not</u> 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 it might appropriately be limited to those applications in which conventional design techniques will not achieve the desired results or to those in which <u>minimum</u> 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 have been defined in each case. At this time the HS-7 Panel has drafted "Ship Vibration and Noise Guidelines"[11] and the M-20 Panel (Machinery Vibrations) has produced Code C-5, "Acceptable Vibration of Marine Steam and Heavy Duty Gas Turbine Main and Auxiliary Machinery Plants"[9]. 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.

STATE-OF-THE-ART, 1978

Referring to the Design Approach discussed in this paper, and as applied in the development of the DD963 and El Paso LNG Designs, we may make the following observations relative to the State-of-the-Art for the Design Prediction of Hull Vibration, as it exists in 1978:

1. We have reasonably well established suitable hull vibration criteria which reflects the state of art of the shipbuilding industry and the physiological requirements of the crew.

2. These criteria, when applied to the vibratory characteristics of the hull girder, can be readily used as design objectives or basis of judgement of the hydrodynamic adequacy of the hull and propeller design configuration.

3. By considering the hull-girder criteria as a reflection of the characteristics of the hull-propeller design configuration, suitable criteria may be developed for other major substructures or local structural elements as have been included in this presentation.

4. Criteria which directly impacts on the structural adequacy of propulsion machinery components, or on the reliability of shipboard equipment has, of necessity, been previously established as the problems have been encountered and resolved, such as torsional or longitudinal shafting problems.

5. We have in 1978 reasonably well defined the methods of testing and evaluating the end product against the design criteria.

6. We have reasonably well identified the exciting forces generated by the propulsion system, particularly the propeller, and in the last ten years, learned much in the control of these forces.

7. We have at our disposal a number of programs and model measurement techniques by which we may estimate these forces.

8. We have not yet developed adequate techniques for convenient full-scale evaluation of the true forces present, which is required for confirmation and/or improvement of estimation procedures.

9. We have mathematical models useful in response predictions but lack enough collective experience at this point in the use of these models to reduce the

problem to its simplest form.

10. Damping coefficients still represent a challenge to the total design process but again, can only be improved by a firm understanding of the input forces and the correlation of these forces with full-scale studies and design analyses.

In summary we may conclude that we have a good handle on the beginning and the end of a rational design procedure, we know much about the factors inbetween, but need considerably more full-scale data and correlation studies on such programs as the DD963 and the El Paso LNG Carriers. In attempting to fill in the middle however, we cannot expect the owners or designers to underwrite the required R&D effort on an individual basis. This represents a group problem and, if we are to improve our prediction procedures, a group effort will be required.

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BACKGROUND

Prior to the 1960's and 70's little or no attention was given to the subject of noise in the design and construction of ships, both military and commercial. It was merely taken for granted that the working environment in powered ships was noisy. Just as the coal miners were expected to have "black lung," the ship engine crew were expected to be a "little deaf."

In the last decade or so, we have seen a revolution in the awareness of the total environment around us. Not only has man become concerned over the pollution of our air and water, but he has become increasingly vocal about the noise around him. The term "noise pollution" was coined only in recent years.

We have all witnessed the public outcries against the evils of the noise created by jet aircraft. Anyone living in the flight pattern of a busy airport suddenly realized that noise was causing him headaches, fatigue and a whole host of ills including deficiencies to the yet unborn.

In this type of environment it was inevitable that the awareness of the effects of noise should spread to the industrial work community. The steelworker, millworker, and forge operators awoke to the fact that noise was not their unique inalienable right. They became aware of a cause-effect relationship between daily exposure to high noise levels and partial deafness or extreme fatigue. The result has been compensation claims for work-related deafness damage running into hundreds of millions of dollars.

It was not until 1969, less than nine years ago, that the Office of Safety and Health Administration promulgated the now well known OSHA Noise Limits; and it is only in the last ten years that the concern over noise in ships has spread to the maritime industry.

What does all this mean to the ship designers and ship builders?

I shall attempt, in this Paper, to identify Noise Level Criteria applicable to the shipboard environment, and current methods available to the designer and builder for meeting these criteria. I shall also present a limited comparison between predicted and measured noise levels in ships' spaces. And finally, I shall identify areas where additional research and development are required to improve the current state-of-art for noise prediction.

NOISE LEVEL CRITERIA

Prior to the 1950's there were no known quantitative noise and vibration limits included in Shipbuilding Specifications. Such limits were sometimes included in innocuous qualitative requirements, such as: "The ship shall be free of any undesirable noise and vibration," or some similar statement.

Such qualitative specifications were virtually unenforceable from a contractual or legal standpoint.

In the mid 1950's the U.S. Navy initiated airborne noise limits in the General Specifications for Ships of the U.S. Navy [1]. Noise Limits were specified in each of five different space categories, depending upon the functions to be performed in these spaces. These categories ranged from Category A, where intelligible speech communications were necessary; to Category D, where high noise levels were expected and personnel deafness avoidance was of prime consideration.

Since that time a number of changes to the Navy noise limits have been promulgated. The current Noise Criteria, as specified in Section 073 of the January 1974 issue of the General Specifications are reproduced in Table I.

In addition to the noise limits specified in Table I, in areas where high intensity noise levels are expected, the Navy also has invoked a deafness avoidance criteria specified by BUMED INSTRUCTION 6260.6B [2]. For steady-state high intensity noise this requirement is identical with the current OSHA limits.

In general, with judicious care taken in the design phase, there is little difficulty in meeting acceptable noise level requirements in operational control and living spaces. This is often not the case in machinery spaces.

With the increasing power concentrated within machinery spaces, it is becoming increasingly difficult to meet an 85 to 90 dbA noise limit unless extreme measures of noise reduction are included in the design. It is in regard to these high noise level areas that I shall discuss the applicability of the OSHA noise limits.

The OSHA noise limits [3] are summarized in Figure 1. These limits are defined in terms of noise level in equivalent dbA and allowable exposure times; ranging from 8 hours at 90 dbA to 15 minutes at 115 dbA.

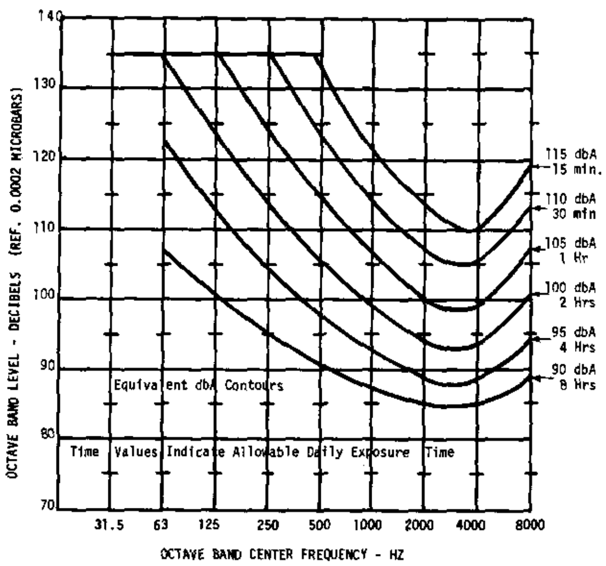

Fig. 1 OSHA Deafness Risk Criteria

TABLE I

AIRBORNE NOISE LEVELS (IN DECIBELS)

Airborne Noise Category	Center Frequencies of Standard Octave Bands (c.p.s.)									SIL Value
	32	63	125	250	500	1000	2000	4000	8000	
A	115	110	105	100	SIL value requirement			85	85	64
B	90	84	79	76	73	71	70	69	68	--
C	85	78	72	68	65	62	60	58	57	--
D	115	110	105	100	90	85	85	85	85	--
E	115	110	105	100	SIL value requirement			85	85	72
F	115	110	105	100	SIL value requirement			85	85	65

Airborne Noise Categories

Category A: Spaces, other than category E spaces, where intelligible speech communication is necessary.

Category B: Spaces where comfort of personnel in their quarters is normally considered to be an important factor.

Category C: Spaces where it is essential to maintain especially quiet conditions.

Category D: Spaces or areas where a higher noise level is expected and where deafness avoidance is a greater consideration than intelligible speech communication.

Category E: High noise level areas where intelligible speech communication is necessary.

Category F: Topside operating stations on weather decks where intelligible speech communication is necessary.

Speech Interference Level (SIL)

Measure of the effect of airborne background noise on intelligible speech communication. Numerically, it is the arithmetical average of the sound pressure level, in decibels, in the octave bands with center frequencies of 500, 1000, and 2000 c.p.s.

Note: Table I extracted from General Specifications for Ships of the United States Navy, January 1974.

In applying such criteria to the shipboard environment, it is important to understand the significance of the OSHA criteria. The principal objective of the OSHA limits is to provide a measure of protection to the industrial worker who may be subjected to high noise levels on a daily repetitive basis. The noise limits were established on the basis of lengthy physiological studies of human exposure to noise. Statistical studies indicated that when a group of workers were exposed to a 90 dbA noise field for 8 hours daily over their working lives of about 20 years, approximately 25 percent of that group would experience an occupational related hearing loss of about 25 db in the 500 to 2000 Hz range. Obviously, at higher noise level exposures the hearing loss would become more severe. Therefore, the limits prescribe lower exposure times for higher noise levels.

It is also important to recognize that the OSHA limits assume that the exposed worker also has daily relief periods in relatively quiet environments of 60 to 70 dbA.

How do the OSHA limits apply to shipboard exposure? For the engine crew stationed directly in the engine room a 90 dbA environment would be indicated for an 8 hour watch. With a 4 hour on and 8 hour off cycle the OSHA limit would permit a level of 95 dbA. These levels assume that during off-duty periods the recreational and living quarters provide a quiet environment of about 60 to 70 dbA.

In the case of machinery areas where personnel are protected by means of Enclosed Operating Stations (EOS), the situation is quite different. Within a suitably designed EOS the noise levels should be well below deafness risk levels. Therefore, the machinery space itself will be manned only for observation and maintenance purposes. Based on the OSHA limits, occasional exposures (less than 15 minutes) of as much as 115 dbA would be permissible. However, if engine spaces were permitted to be this noisy, it may become impracticable to achieve a quiet environment in adjacent EOS areas.

In order to develop a recommended set of noise criteria for all manned spaces aboard ships, an examination has been made of noise criteria and specifications established by other National and Foreign regulatory bodies. A study was also made of existing noise data in shipboard spaces. Because of obvious practical considerations, the recommended criteria will be based on a trade-off between desirable noise levels and achievable levels within the current state-of-art.

Figures 2, 3, and 4 present a comparison of representative noise specifications for Bridge Control Spaces, Living Spaces, and Machinery Spaces respectively. Table II provides a summary, compiled by the SNAME HS-7 Panel [4], of noise limits in dbA established, or being considered, by various Government Regulatory Agencies. Also included are recommended limits proposed by NKF ENGINEERING ASSOCIATES, INC. and the HS-7 Panel.

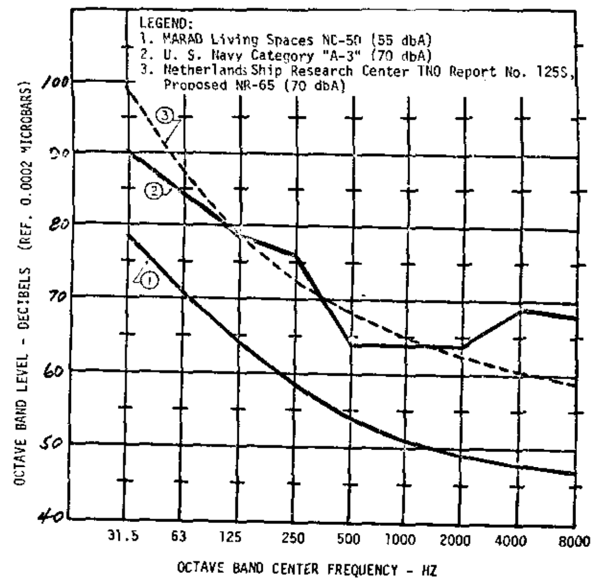

Fig. 2 Comparison of Noise Specifications for Shipboard Bridge Control Spaces

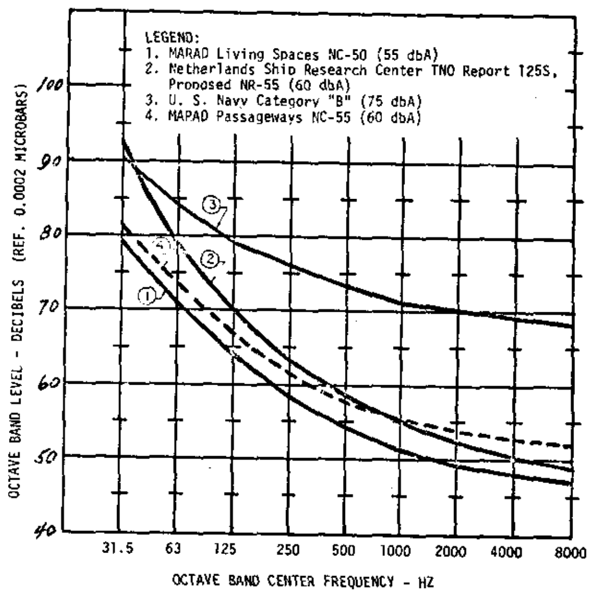

Fig. 3 Comparison of Noise Specifications for Shipboard Living Spaces

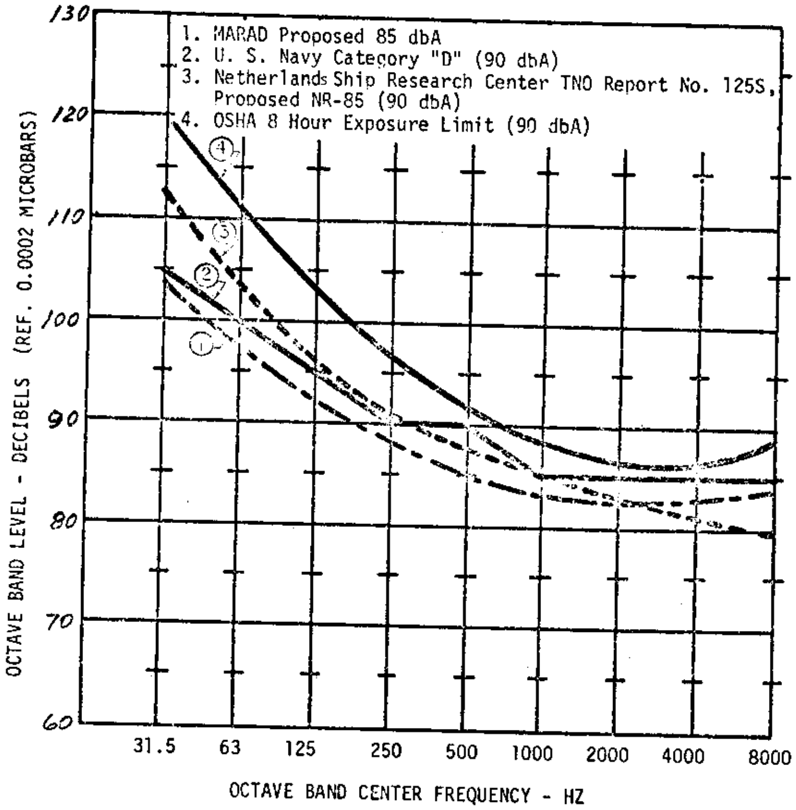

Fig. 4 Comparison of Noise Specifications for Shipboard Machinery Spaces

TABLE II

COMPILATION OF INTERNATIONAL SHIP NOISE CRITERIA

SNAME HS-7 PANEL

	SWEDISH REG.	DANISH	WEST GERMANY	ISRAEL	USSR	DSRK EAST GERMANY	PRS POLISH REG. OF SHPG.	NORSK MARITIME DIRECTORATE	MARAD	USN	NKF RECS.	HS-7 RECS.
ACCOMMODATIONS												
CABINS, OFFICES	55	60	60	55	50	60	60	60	56	60	60	60
GALLEY, PANTRY, HOBBY ROOM	65	70						70	–			65
MESSES, DAYROOM	65	65	65	65	55 (DR)	60 (DR)	60 (DR)	70	56 (DR)			65
PASSAGEWAYS									62	75	80	NO REC.
BRIDGE												
BRIDGE WINGS	70	70	65					70	–			NO REC.
WHEEL HOUSE	65	65	60	60	55	60	65	65	–	55	60	60–65*
RADIO ROOM	65	65	60	60	50	60	60	65	–	60		60*
MACHINERY SPACES												
MANNED W/O CONTROL RM.	85	90	90	85	80	–	90	90	85	85	90	85
CONTROL ROOM	70	75		70	65	80	75	75	70–75		75	75
WORKSHOP & STORES	75	85		75	65	90	90	85	–			85

NOTE: All Noise Levels shown in dbA.

*65 w/Doors Open
60 w/Doors Closed

Figures 5 through 8 present a summary of data reported of noise measurements made on many different ships. The data presented are typical, or median levels of a great many measurements. The actual data show a spread of about ±10 db around the median curves shown.

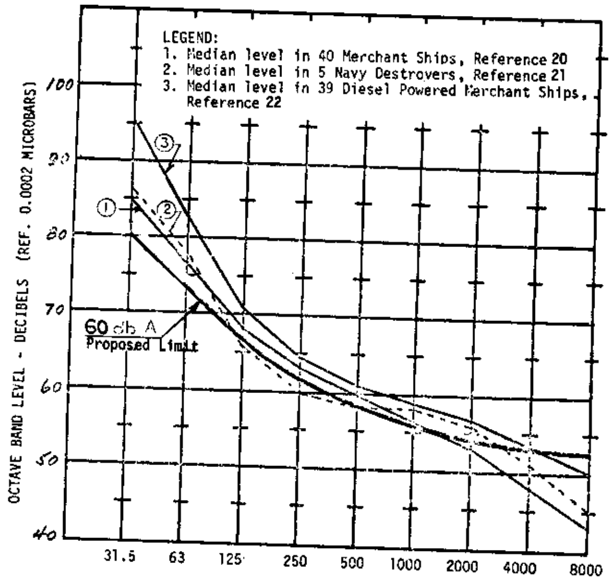

Fig. 5 Measured Noise Levels in Bridge Control Spaces

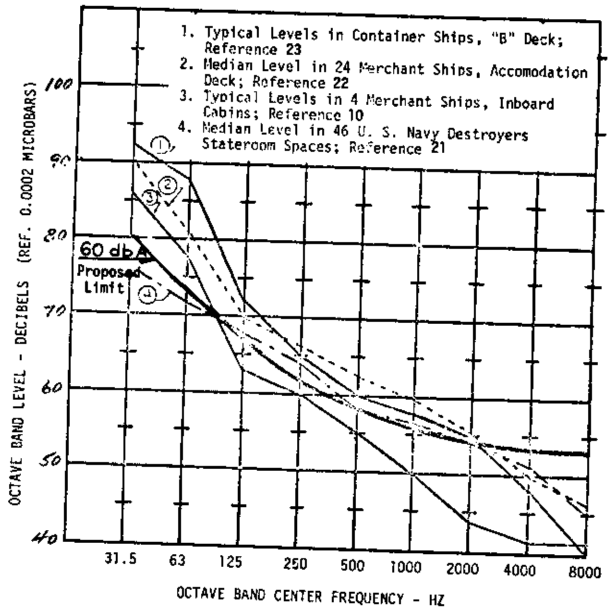

Fig. 6 Measured Noise Levels in Shipboard Living Spaces

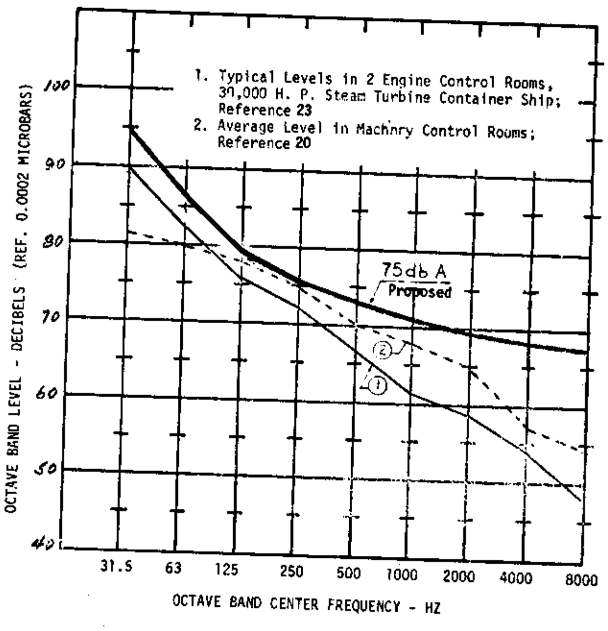

Fig. 7 Measured Noise Levels in Engine Room Enclosed Operating Stations

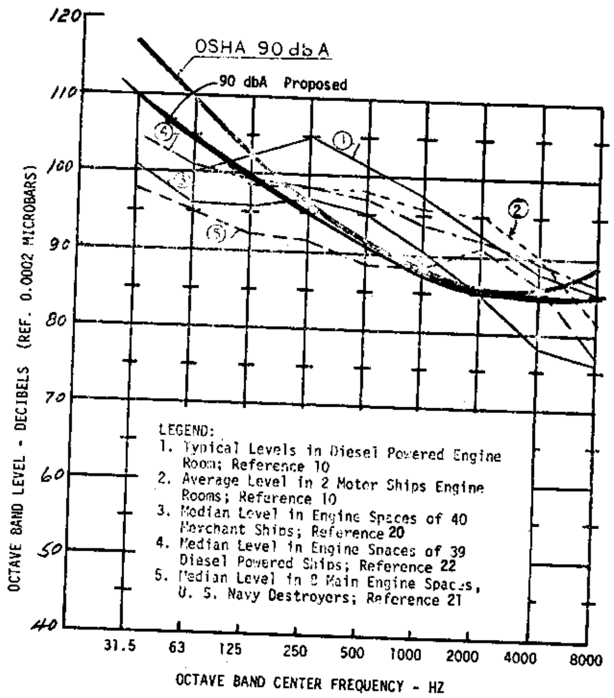

Fig. 8 Measured Noise Levels in Shipboard Main Engine Spaces

Considering the varied sources of data presented in Figures 5 through 8, a fairly narrow spread of median values was observed for similar types of spaces. Since the data is representative of noise levels in ships where no special noise control measures were incorporated, it may be assumed that with additional attention to noise reduction, the noise environment in new ships can be significantly reduced.

Based on a trade-off between desirably low noise levels and current practicable state-of-art, a set of noise level limits for shipboard spaces has been developed herein. These proposed limits have been superimposed on Figures 5 through 8. The following recommended noise levels are proposed for each of five space categories:

Category A - Operational Control and Living Spaces

It is desirable that a relatively comfortable environment in which good speech communication is possible be established in Control Spaces, Office Spaces and Living Quarters. Examination of Figures 5 and 6 indicates that the range of median levels for Living Spaces is similar to the levels observed in Control Spaces, falling between NR-50 and NR-60 curves. The NR values refer to ISO Rating Curves shown in Figure 9 [5].

It is recommended that a maximum noise level of 60 dbA, which is equivalent to NR-55, be established for Operational Control and Living Spaces.

With reasonable care in space arrangements and acoustics design, a level of 60 dbA should be readily achievable.

Category B - Enclosed Operating Stations

The principal purpose of an Enclosed Operating Station is to provide a more habitable environment for engine crew than the Engine Room. It also provides a space where speech communications are possible.

It is recommended that a maximum noise level of 75 dbA, equivalent to NR-70, be established for EOS spaces.

If the noise levels in adjacent machinery spaces are kept within the limits of Category D, there should be little difficulty in achieving a level of 75 dbA in the EOS.

Category C - Passageways

Passageways are occupied only for intermittent relatively short periods of time. Therefore, personnel can tolerate a considerably higher noise level than for the living space environment.

However, since passageways can be used effectively as a buffer zone to attenuate noise from a high noise level area to nearby "quiet" spaces, a limit is advisable. The noise levels in passageways may fall somewhere between the Category A and Category D levels.

It is recommended that the noise level in Passageways be limited to 80 dbA, equivalent to NR-75.

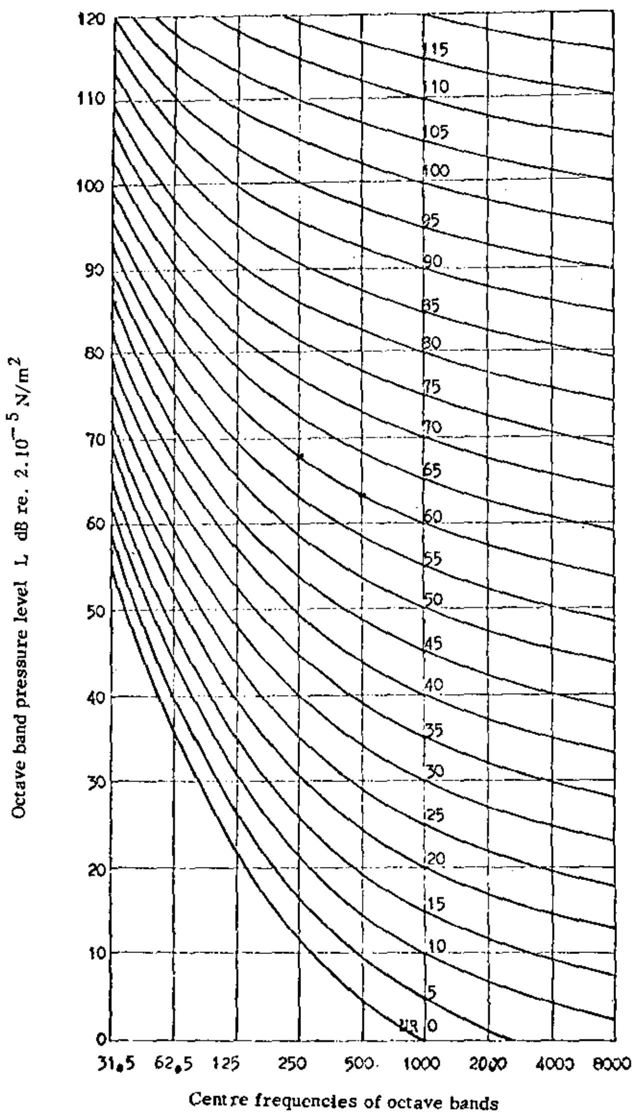

Fig. 9 Noise Rating Curves
(From ISO Standard 1996)

Category D - Machinery Spaces

Examination of Figure 8 indicates that, under normal current shipbuilding practices, the noise levels in main engine spaces often do not meet the 90 dbA OSHA limits. This is particularly so in diesel-powered ships.

It is noted from Table II that the various countries have recommended limits of from 80 to 90 dbA for engine spaces. In order to meet these levels, extreme noise reduction measures may be required, such as engine and gear enclosures and vibration isolation. Therefore, a somewhat different approach is proposed.

For those engine spaces where personnel are stationed within the space, it is mandatory that the noise levels be kept within the 90 dbA limit.

For those spaces where Enclosed Operating Stations are provided, the noise limits in the engine rooms may be relaxed to some degree. From Figure 1 it may be seen that for short period exposures of less than 15 minutes the OSHA limits would permit levels as high as 115 dbA. However, if engine spaces were permitted to reach such levels, it would be extremely difficult to provide adjacent EOS levels of 75 dbA.

Therefore, it is suggested that the Category D noise levels for machinery spaces be separated into two sub-groups:

1. Engine Spaces without EOS - Noise Levels should be limited to 90 dbA.
2. Engine Spaces with EOS - Noise Levels should be limited to 100 dbA.

Category E - Unmanned Machinery Spaces

For normally unmanned spaces such as pump rooms, forced-draft blower spaces, etc., which are only entered occasionally for maintenance and inspection, it is recommended that a maximum short-term exposure limit of 115 dbA be established.

All areas which exceed 90 dbA should be designated as CAUTION - HEARING DAMAGE AREAS. Ear protectors must be provided to all personnel entering such spaces.

Figure 10 summarizes the Airborne Noise Limits proposed herein. These limits are shown both in terms of octave-band limits and allowable overall dbA levels.

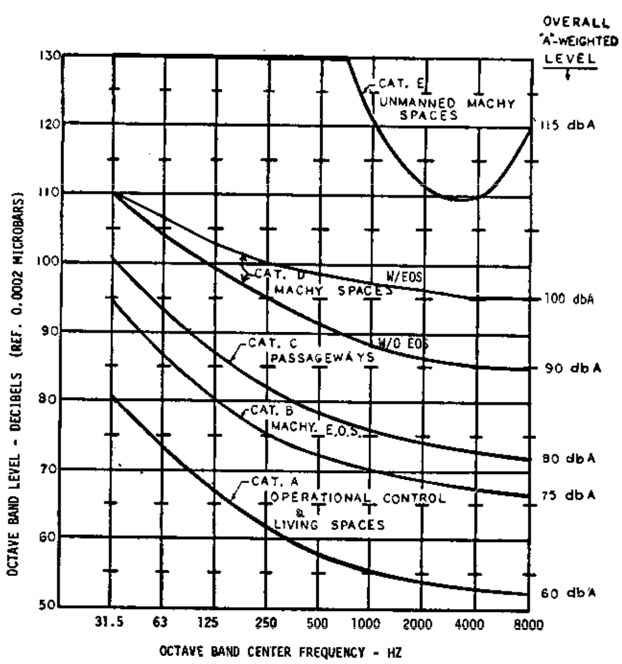

Fig. 10 Proposed Airborne Noise Limits

NOISE LEVEL PREDICTION

The purpose of this Section of the Paper is to examine the state-of-art of shipboard noise prediction and to discuss the degree of accuracy to be expected in such predictions.

Many references can be found in current literature which provide detailed methods for calculating airborne noise levels within a room. However, practically all these references apply to the prediction of noise levels in architectural spaces where the principal sources of noise are generally associated with interior ventilation systems, exterior environmental traffic noises, industrial noise exposure, etc.

Shipboard acoustics differ from architectural acoustics principally in the amount of metal structure used. Light-weight uninsulated metal bulkheads are more efficient acoustic radiators than plaster walls. Steel structures are more efficient propagators of vibration than wood or concrete.

For shipboard spaces, the total Sound Pressure Level (SPL) will depend on the airborne Power Level (PWL) of sources within and around the space, and on the Vibratory Power external to the space. Therefore, in predicting the noise level within a shipboard space, consideration must be given to not only the airborne noise sources close to the space, but to structureborne sources which may be relatively remote from the space.

Only an overview of prediction techniques is presented herein. Details of analyses will be discussed in other Papers of this Symposium, and can be found in referenced literature, such as: References 6 through 14. A comprehensive detailed procedure has been developed under the joint sponsorship of the U.S. Navy and U.S. Coast Guard, entitled the "Handbook for Shipboard Airborne Noise Control" [15]. This document, which was presented and discussed at the INTER-NOISE '74 Conference and at the Acoustical Society of America meeting of November 1976 [16, 17 and 18], contains the methodology used herein by the author.

The conventional methods for predicting noise in a space utilize the well-known Source-Path-Receiver approach. A flow-chart of this Procedure is shown in Figure 11. The principal difference between this procedure and those generally used in architectural acoustics is the introduction of the contribution from structureborne sources.

In the Source-Path-Receiver approach the first step in the prediction process is to determine the source Sound Power Level (PWL). In the absence of measured data, a number of empirically developed formulae for estimating the PWL of machinery can be found in the referenced literature. As shown in Figure 12, a baseline PWL_B and L_A (Acceleration Level) is determined by empirical formulae, and adjustments are made to determine the octave-band spectral distribution.

Table III presents a number of PWL formulae for typical shipboard machinery, that have been taken from Reference 15. These formulae have been empirically adjusted based on a limited number of test measurements. The estimated range of accuracy varies from about ±3 db to ±10 db, depending upon the type of machine considered.

It is important to note that very little has been done to date towards verifying the accuracy of various empirical formulae used for determining PWL values. Therefore whenever available, measured Sound Power Level values should be used.

The second step in the prediction process is to estimate the amount of attenuation in the path (or paths) between the source and the receiver. Figures 13 and 14 identify the principal attenuation factors to be considered in the Airborne and Structureborne Paths respectively.

The final or third step in the prediction process is to estimate the Sound Pressure Level (SPL) in the Receiver Space. The overall PWL in the Receiver Space is determined by summing

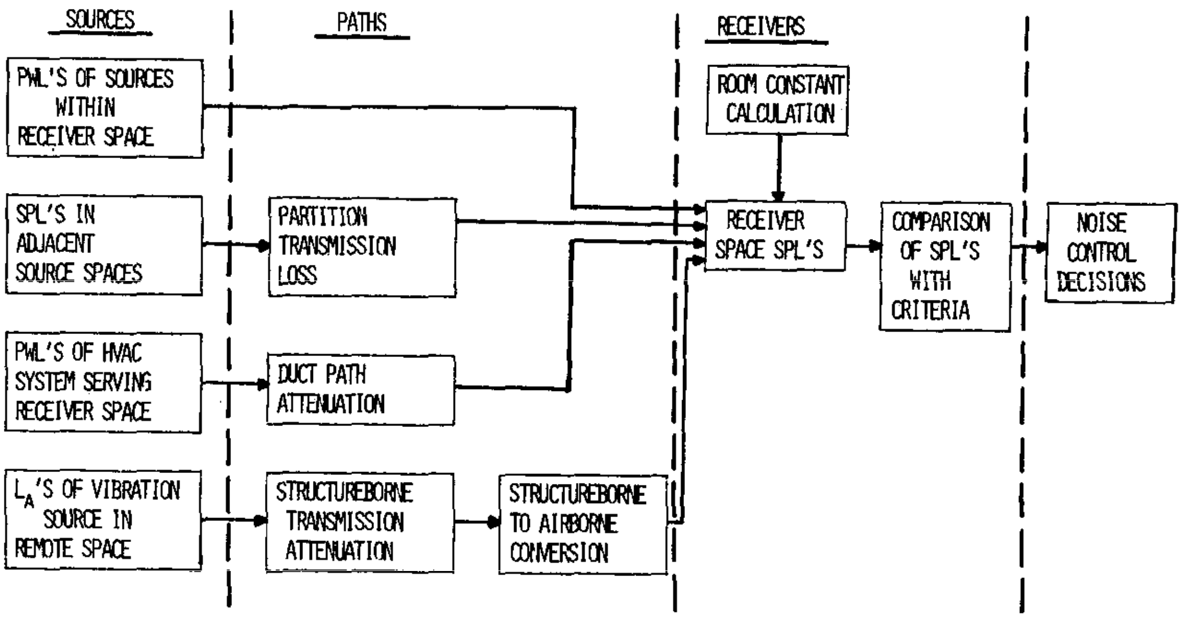

Fig. 11 Noise Prediction Procedure

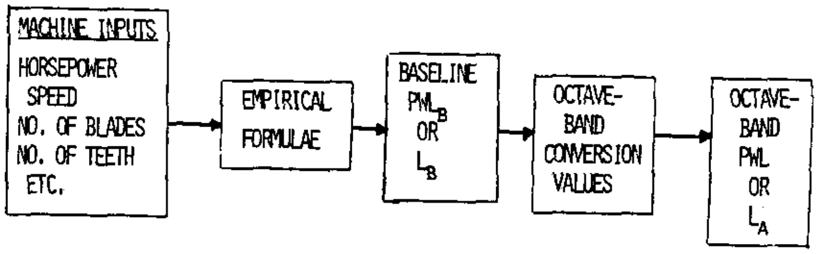

Fig. 12 Procedure for Estimating Source Levels

TABLE III

SOURCE POWER LEVEL ESTIMATION FORMULAE*

		Octave	31.5	63	125	250	500	1K	2K	4K	8K
CENTRIFUGAL PUMPS											
AIRBORNE**	PWL_B = 0.7 log (horsepower) + 62 + CV	CV	-5	+1	-2	+8	+10	+11	+6	0	-3
STRUCTUREBORNE	L_B = 10 log (horsepower)+ 50 + CV		10	15	21	27	33	39	46	39	33
REDUCTION GEAR											
AIRBORNE**	PWL_B = 3.4 log (horsepower + 3.4 log (low speed shaft RPM) + 77 + CV. Also add 10 db to the appropriate octave for the high speed shaft frequency = RPM/60		-3	0	0	8	+11	+13	+11	0	-2
STRUCTUREBORNE	L_B = 10 log (horsepower) + 60 + CV		3	9	15	15	15	15	15	15	15
FANS											
AIRBORNE**	PWL_B = 5.7 log (cu.ft/min) + 11.4 log (static pressure rise) + 60 + CV. Add BPC to appropriate octave for blade passage rate = no blades x RPM/60	TYPE A (Sizes 1/4 - 2) BPC = 7	-	-	-12	-6	+4	+5	+5	0	-5
		TYPE C (Sizes 1/2 - 5) BPC = 3	-	-4	+4	-2	+1	-2	+1	-5	-14
DIESEL ENGINES											
AIRBORNE											
Inlet & casing	PWL_B = 10 log (horsepower) + 57 + CV		19	24	26	24	26	26	24	20	14
Exhaust	PWL_B = 10 log (horsepower) + 71 + CV		44	40	46	42	34	30	24	14	6
STRUCTUREBORNE	L_B =		94	101	108	113	118	123	124	123	115
	Also add 8 db to the appropriate octaves for each of the following frequencies RR = RPM/60, 2 x RR, 3 x RR, FR = RR x No Cylinders x 2 - No STROKES, 2 x FR and 3 x FR										

NOTES: * This is a sample of typical estimation formulae; for a more comprehensive compilation of equations and octave conversion values, see Reference 15.

** Indicates formula and/or CVs adjusted by NKF.

All PWL re 10^{-12} watts; All L_B re 10^{-3} cm/sec^2

the combined acoustic powers from all sources, both airborne and structureborne. Since the noise level within a space is affected by the reverberant characteristics of the space, an adjustment must be made for room absorption. The SPL is then determined by applying the Room Constant (R) correction in the well known equation for Sound Pressure Level:

$$SPL = PWL - 10 \log R + 16$$

In addition to the more commonly used Source-Path-Receiver approach for prediction, other approaches should be mentioned. Attempts have been made to predict the noise in a space by experimentally determining a Transfer-Function between the noise source and the receiver space. This may be done by measuring the noise in a space due to a selected group of machines operating. With the machinery secured, simulated airborne or structureborne noise is introduced in the machinery space, and the airborne noise again measured in the receiver space. A separation can then be made between the airborne and structureborne noise components, and Transfer-Functions determined between the Source and Receiver. Relationships determined in this manner are valid when applied to similarly designed ships. Valuable attenuation information may also be available from such experimental testing. It is noted that this approach is quite costly because of the experimental testing required.

Another approach in estimating noise levels in ship spaces utilizes a similitude analysis. In this approach it is assumed that the machinery characteristics and the receiver space noise levels are known by measurement. Changes in machinery and changes in structure are then evaluated in terms of their effective changes in noise level. This approach is only valid when the design of ships is similar, and when the relative contributions from airborne and structureborne sources are known.

PREDICTION VERIFICATION

Because of the increased use of quantitative noise limits in shipbuilding specifications, it is expected that there will be an increased effort in analytical prediction of noise during the design process. It is, therefore, important that we understand the accuracies and limitations of the prediction procedure.

An examination of the literature indicates numerous examples of analytical modeling, scattered measurement data, and methods for

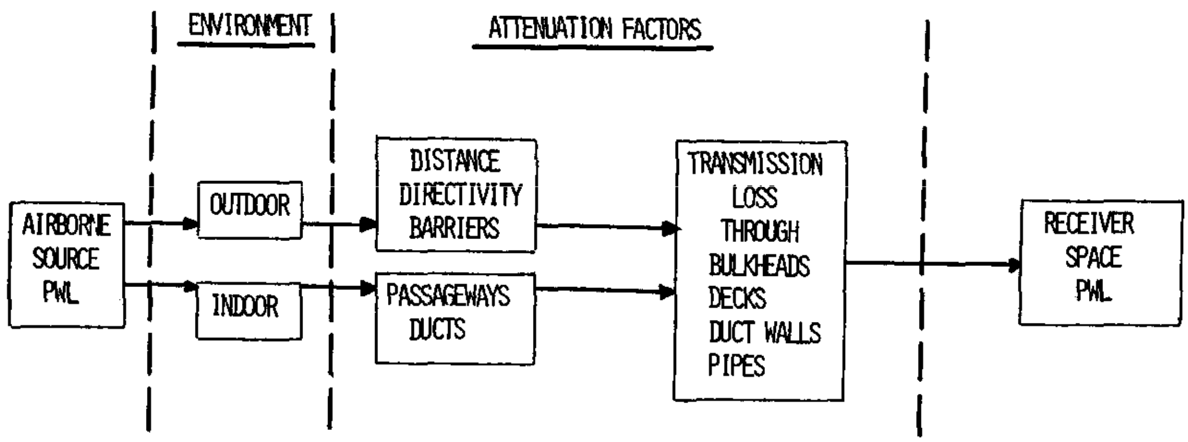

Fig. 13 Attenuation in Airborne to Airborne Paths

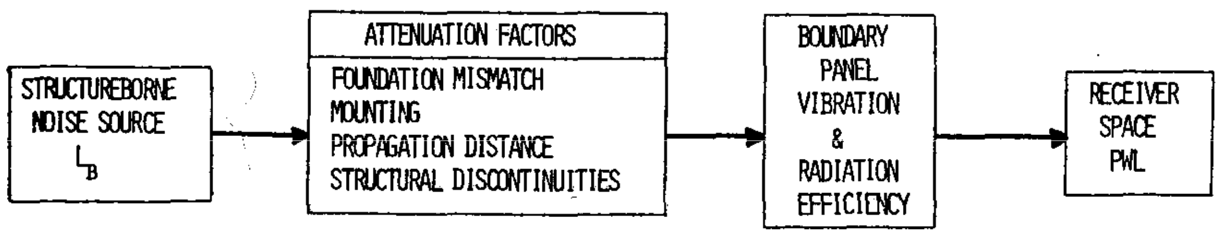

Fig. 14 Attenuation in Structureborne to Airborne Paths

noise reduction. However, very little has been found in which sufficient prediction versus measured noise data could be compared systematically.

One detailed study was conducted on a U.S. Navy Destroyer, in which analytical predictions were made of the noise levels in manned spaces; and then followed up by measurements during sea trials. This study afforded an opportunity to compare measured SPL against design predictions in a reasonable number of samples.

Based on this Destroyer study, direct comparisons were made in 9 spaces in which octave-band analyses were available, and in about 40 spaces in which dbA levels were available [18].

Figure 15 summarizes the SPL deviation observed between predicted and measured octave-band levels for a sample of 9 spaces. At the lower and upper ends of the spectrum, the predictions were within ±10 db of measured. At the mid-frequencies the predictions were about ±5 db with a -5 db bias.

Figures 16 and 17 show the distribution between predicted and measured levels, for a sample of 43 spaces in which dbA levels were available. On the average, approximately 40% of the estimated levels fell within 3 dbA and 75% fell within 6 dbA. For this sampling, the estimated levels were almost equally divided between low estimates and high estimates.

With respect to the data just presented it is important to note that:

1. Much of the predictions were based on measured source Sound Power Levels, rather than estimated by empirical formulae; and,
2. On this ship all major machinery was resiliently mounted on soft mountings.

Since source PWL's were measured, the accuracies of prediction shown are probably better than would be expected if source PWL's were calculated. Also, because of the soft mounted machinery, the major contribution of noise in the spaces was airborne related, with little contribution from structureborne sources. Therefore, the predictions would again be more accurate than if structureborne noise was a major factor.

Thus, if the prediction process was based on analytical models only (with no measured data), the prediction accuracy would probably be greater than ±10 db.

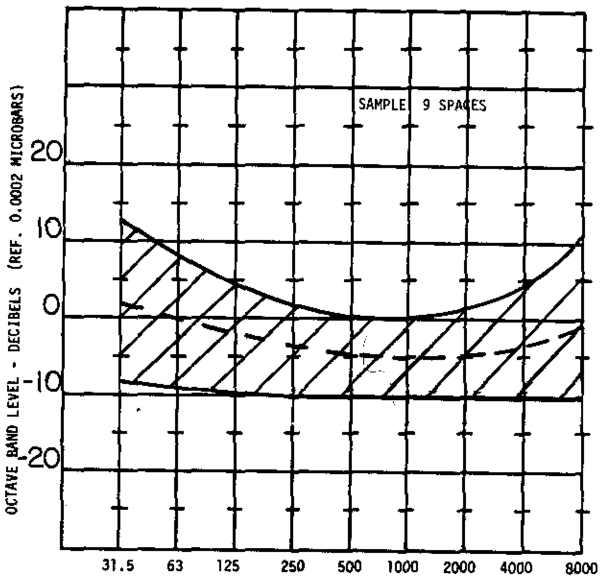

Fig. 15 Range Between Predicted and Measured SPL in 9 Destroyer Spaces

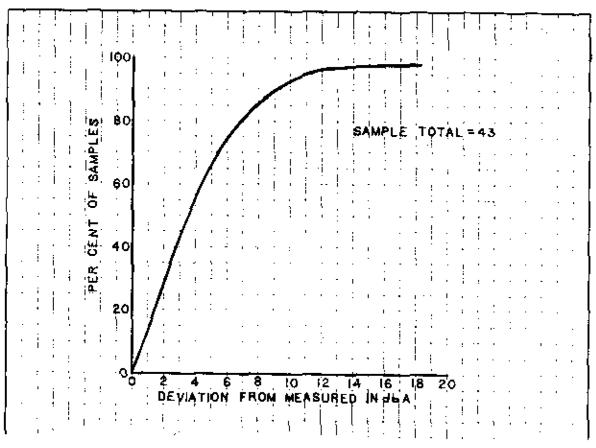

Fig. 16 Distribution of Estimated vs Measured SPLs in 43 Destroyer Spaces

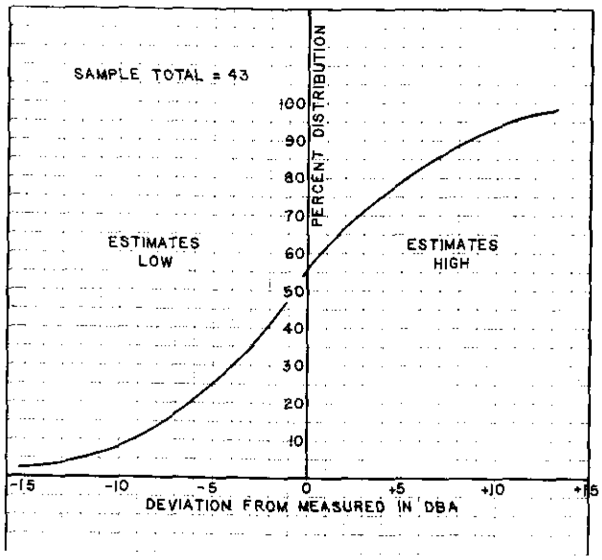

Fig. 17 Distribution of Estimated vs Measured SPLs in 43 Destroyer Spaces

SHIPBOARD NOISE REDUCTION

Considerable information on noise reduction techniques can be found in the open literature. Shipboard noise reduction will also be discussed in greater detail by follow-on Papers. Therefore, only a brief overview of shipboard noise reduction treatments will be discussed herein.

In order to effectively control the noise environment aboard ship, it is mandatory that noise control considerations be included in the ship design process. Experience has shown that the past policies of correcting a noise problem only after the ship has been completed can be a very costly procedure.

The first, and often most effective, noise reduction measure can be implemented through judicious ship space arrangements. Wherever possible spaces such as passageways, store rooms, and infrequently manned workshops should be located between noisy machinery areas and operational or living spaces. Such arrangements provide convenient buffer zones with considerable acoustic attenuation.

At the present time the U.S. Navy imposes specific noise limits on equipment and machinery to be installed aboard ship [19]. This would be a good practice to follow in order to keep the source levels down to manageable limits. Therefore, it is recommended that noise limits be imposed on vendor-supplied machinery.

Where previous experience, or prediction analyses indicates the need for noise reduction, specific treatments such as the following should be considered:

1. Ventilation System Silencers or Duct Treatment
2. Bulkhead and Decking Acoustic Treatment
3. Machinery Vibration Isolation
4. Floating Deck Structures.

The subject of floating accommodation spaces and resiliently-mounted superstructures will be discussed in a follow-on Paper.

In deciding which of the noise reduction treatments to apply, it is mandatory that the relative noise levels of the contributory sources be known. It is also necessary to establish whether the principal noise sources are airborne or structureborne related. The highest noise level sources and the principal paths of transmission must be given primary consideration.

When the principal noise is airborne related additional acoustic attenuation in the bulkheads and/or decks are indicated. In the case of predominant structureborne noise, the treatment would call for machinery isolation or structurally resilient arrangements.

Although good noise control may be initiated in the design phase, the effectivity of such control can be totally nullified by poor quality control in the construction phase. Airborne noise leakage paths and structural "shorting" of vibration isolators are common fabrication faults.

The effect of a high acoustic transmission loss bulkhead or deck can be lost by the introduction of small openings or the inclusion of an overhead sheet metal ventilation duct. The effect of a costly machinery isolation system can be voided by the installation of a rigid pipe or electrical conduit.

Thus, for effective noise control it is important not only to incorporate acoustic treatments in the design, but to also follow through the fabrication stage with effective quality control inspection.

SUMMARY

In summary, I have herein discussed Noise Level Criteria, Noise Level Predictions, and in a cursory overview, Noise Reduction Treatments.

For commercial ships, it is recommended that consideration be given to including the Noise Level Limits shown in Figure 10 in the Shipbuilding Specifications. All efforts should be made to meet these limits through a suitable acoustic design. It is believed that the current state-of-art permits the achievement of these noise limits.

With regard to the current state-of-art on Noise Level Prediction, it appears that much still remains to be done towards verifying the limits of accuracy achievable.

I have demonstrated that where the sources of noise are predominantly airborne, and when the source Sound Power Levels are fairly well known, the noise in a shipboard space may be predicted within about 5 to 10 dbA. However, when the Sound Power Levels must be empirically estimated, and/or where the predominant noise is structureborne, the degree of prediction accuracy is still somewhat questionable, and probably greater than 10 dbA.

It is therefore recommended that additional study be devoted to the following areas:

1. Improved source Sound Power Level prediction capability. In this area of development, it would be extremely helpful if machinery vendors were to measure airborne Sound Power Levels and structureborne Acceleration Levels. These parameters could be used to generate more accurate empirical source level relationships for different types of machinery.
2. Improved Noise Prediction Technology, particularly with regard to Structureborne sources.
3. Additional full-scale verification studies aimed at improving the confidence level in the prediction process.

The means for improving the prediction accuracy, and the tools for implementing improved noise control are available. However, noise control must become part of the ship design process; and additional studies are required towards improving and verifying the prediction methods.

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