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Computer Techniques for Use in Ship Hull Vibration Analysis and Design

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ABSTRACT

The prediction of propeller excited ship hull vibration and the development of designs to minimize propeller excited hull vibration are complex problems that at the present time are far from fully developed. All analytical procedures require (1) the prediction of the excitation at the propeller and on the hull as a consequence of the pressure field surrounding the propeller, and (2) the prediction of the response of the ship to these excitations. In the past several years, many computer programs have been written to predict the excitation and response of a ship to propeller-induced vibrations. This usually involves applying the propeller forces and pressures computed in a separate hydrodynamic program to a beam or finite element model of the ship's structure. In general the model, if reliable, is quite large and so the program should be capable of handling a large system. This paper presents information on the computer programs commonly used in ship vibration analyses and observations based upon surveys and experience.

INTRODUCTION

The level of propeller-excited ship hull vibration has defied analytical prediction until recently. Methods for estimating the natural frequencies of the fundamental vertical bending frequency with reasonable accuracy based on experience with previous similar ships of different dimensions have been available since the Schlick formula was first presented in 1894 (1). The prediction of the bending frequency using available analytical methods applied to a nonuniform beam was unreliable until the effects of water inertia were explained by Frank M. Lewis (to whose memory this Symposium is dedicated) in 1929 (2) and J. Lockwood Taylor in 1930 (3). The methods for determining the natural frequencies were difficult, requiring iterative processes and clearing higher modes from the components of lower modes using the Stodola and Rayleigh-Ritz techniques.

The first analyses were based upon a beam modeled by point masses and bending flexibility only. Timoshenko (4) developed the differential equation for the beam that included bending and shear deflections and point and rotary inertias. In 1945 Prohl (5) proposed a procedure similar to the Holzer Method for predicting bending natural frequencies, and Myklestad (6) applied the process to coupled bending and torsion.

During the period from 1945 to 1960 the Prohl-Myklestad methods were applied to the bending of a number of ships, including shear deflection, by McGoldrick and the DTMB staff, and the predicted results were compared with experiments, using shakers and in normal operation. It was found that whereas the fundamental bending mode could be predicted with reasonable accuracy if shear effects were neglected, the frequencies of higher modes became increasingly influenced by the shear deflections. It was also found that although some ships such as lake ore carriers showed a uniform sequence of frequencies to as high as the ninth mode, most ships showed an erratic sequence of frequencies and corresponding irregular mode shapes above the third or fourth mode. These deviations from uniformity were found to be due to local and cross-sectional resonances. Studies of ship bending vibration (7) showed that the dynamic characteristics of the propeller-shaft system strongly influenced the response of the ship to propeller excitation. At about 1960, procedures were being developed for predicting, by calculation from wake data and the geometry of propellers and hull, the propeller generated excitation at the shafting and on the hull. All of these influences are reflected in the calculation procedures discussed in the remainder of this paper.

CALCULATION PROCESSES

For perspective in discussing the calculation procedures and methods used in hull vibration prediction, reference is made to Figure 1, which is the flow chart for propeller excited hull vibration related to analytical prediction procedures proposed in a current study for the Ship Structure Committee (8).

Attention in this presentation will be concentrated on the process covered by blocks 4, 6, 7, 8, 9, 10, 11, 14, 15, 16, 17, 18, 19, 20, and 21. It will be noted that the procedure shows both analytical and experimental techniques for determining the excitation. In general the calculation procedures are less expensive and less time consuming than the experimental model test procedures and give a broader picture of the excitation. Therefore, it is probably desirable to carry them through even if detailed model tests are run. The processes will be discussed, the available methods for covering the material will be presented, and a brief discussion will be given of the strong points and weaknesses of each.

For convenience of the reader, specific information about each program is presented in the appendices, and the appendix number corresponds to the appropriate box in Figure 1. Since all boxes or procedures are not discussed in this paper, the appendices are not necessarily numbered consecutively.

EXCITATION CALCULATIONS

Estimate Propulsion System Longitudinal Vibration Frequencies - [Box 4]

In general, to keep propeller and hull excitation forces low, it is desirable to use many blades on the propeller. The number of blades chosen is set primarily by the natural frequency of the shafting and propeller in longitudinal vibration. To ascertain the probable frequency that will be found after the design of the propulsion system and its supports are developed, it is useful to have a plot of natural frequency versus foundation stiffness. Using values of the probable range of foundation and thrust bearing stiffness, the probable range of shaft longitudinal frequency is determined. The number of blades in the propeller is chosen so that, preferably, the excitation frequency is less than 80 percent of a possible propulsion natural frequency. A less desirable, but sometimes necessary, solution is to make the excitation frequency about 30 percent above the longitudinal natural frequency.

For making these predictions, the power and RPM of the plant must be defined. From this the approximate propeller diameter and propeller weight and the water inertia associated with longitudinal vibration can be established. Also the approximate diameter of the tailshaft and lineshaft can be established. The simplest procedure is to predict on the basis of a one-degree-of-freedom system consisting of the propeller and water inertia plus a portion of the shaft weight vibrating against the stiffness of the thrust bearing and its foundation. Since the shafting weighs considerably more than the propeller and adds flexibility, this procedure is not very good.

An improved procedure is to model the propeller and shaft as a series of concentrated masses and elastic elements and use a Holzer process for frequency prediction. With this degree of complication it becomes desirable to use a computer.

If a computer is used, many of the complications of defining the mass-elastic system can be avoided by using a finite element program such as ANSYS, STARDYNE, etc. However, a more accurate result can be obtained with less input (and, therefore, less chance for error) if the shaft is represented by a continuous mass and elasticity distribution. A computer program for predicting the natural frequencies of propeller and shafting systems defined by concentrated and distributed masses and stiffnesses is presented and discussed in Appendix 4-1.

Compute Propeller Forces - [Box 6]

Prior to about 1960, the only determination of propeller forces was by measurements on models, primarily by Frank M. Lewis (9,10). In the late 1950's estimates began to be made on a quasi-steady-state basis using the procedures developed by Burrill (11) for evaluating the loading and efficiency of propellers whose circumferentially averaged wake varied along the propeller radii. A computer program for calculating the harmonic forces and moments generated by the propeller working in varying wakes based upon this quasi-steady-state procedure was applied by Hinterthon (12). A similar computer program, also based on Burrill's procedure but including as well the Theodorsen effects (i.e., the inertia of the fluid in responding to circulation changes resulting from changes in angle of attack), was developed by CONESCO (13). The first tends to give high values of harmonic forces and moments and errors in their phase because the inertia effects are neglected. The latter program tends to give low values of harmonic forces and moments because the steady-state solution allows flow over the tip, between blade sections, and interaction between blades that are not developed in the unsteady flow. Both of these programs have been superseded by improved analyses of the problem.

In 1958 Ritzer and Breslin developed a theory for the unsteady thrust and torque of a propeller in a ship wake based upon unsteady airfoil theory. This work has been continued by Tsakonas

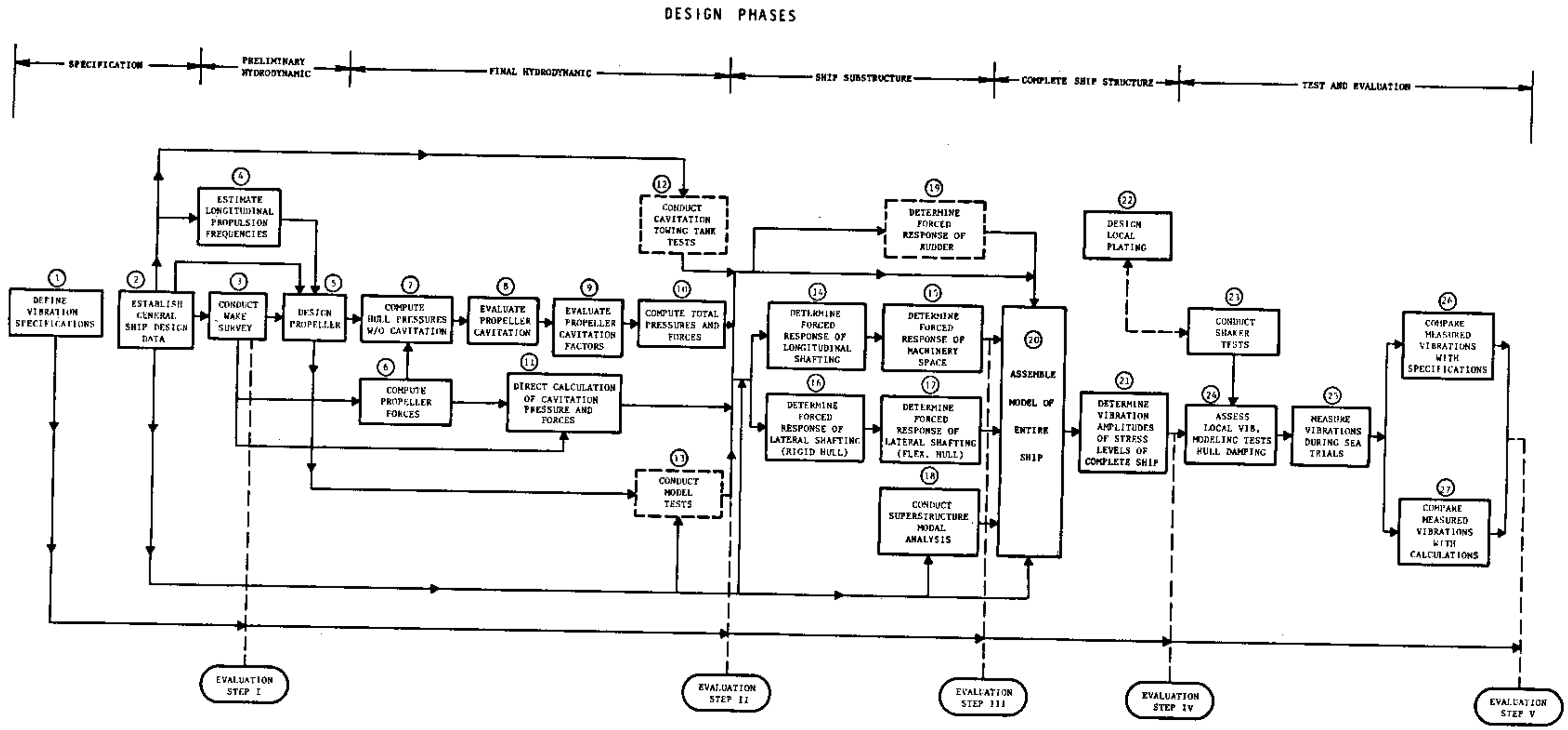

FIGURE 1. FLOW DIAGRAM OF SHIP DESIGN PROCEDURES TO MINIMIZE PROPELLER-INDUCED VIBRATIONS

and Jacobs (14) and is now a fully developed program for predicting the harmonic forces and moments exerted by a propeller on its supporting shaft, when working in the wake behind a ship. This work is based upon lifting surface theory. Although the computations are long, they are handled easily by computer. A description of this program is given in Appendix 6-1. This program is widely used both in the United States and abroad.

The Department of Naval Architecture and Marine Engineering at MIT has also been active in the prediction of the harmonic forces and moments transmitted by a marine propeller to its supporting shaft. Using unsteady flow theory with the propeller blade represented as a lifting line, Neal A. Brown developed relations for determining the periodic propeller forces (15). Several computer programs based on this theory have been developed. They are presented in Appendices 6-2 and 6-3.

More recently, Professor Justin E. Kerwin is approaching the unsteady force problem with another procedure (16). A discussion of the computer program using his approach is presented in Appendix 6-4.

Similar procedures to those developed in the United States have been developed in Europe. M. T. Murray and J. E. Tubby at the Admiralty Research Laboratory developed a computer program for determining the unsteady shaft forces from propellers (17). Information on this is presented in Appendix 6-5.

Compute Hull Pressures and Excitations - [Boxes 7, 8, 9, and 10]

The procedures for determining the excitation on the hull are varied. It has only recently been definitively shown by experiments (18) that variable cavitation on a propeller influences the hull pressures and the hull forces very strongly. The ideal way to determine the propeller generated hull forces would be by integrating the pressures generated by the cavitating propeller, if this were not too difficult, because a knowledge of the pressure distribution is useful for the ship design. At the present time this is not possible. It requires both a prediction of the cavitation growth and decay on the hull pressures. It appears that the intensity of the pressure is related to the second time derivative of the cavitation bubble volume. The problem is being studied, and some of the papers at this symposium indicate the state of the progress in this field.

At the present time the procedure is round about and consists of predicting the hull forces generated by the propeller in the absence of cavitation and modifying the results by an empirical factor to account for cavitation effects. The factor is determined by (1) experience with full-scale measurements, (2) model tests in a cavitation towing tank, and (3) model tests in a cavitation tunnel of sufficient size to include modeling of a portion of the ship.

In a sense, the use of a factor applied to the pressures determined in the noncavitating condition is theoretically unsound because the pressures are generated by another mechanism other than that responsible for the pressures generated in the noncavitating case. The hull pressures are generated as the sum of three different processes. The first is the pressure due to propeller loads, i.e., the difference in pressure on the face and back of the blades. The second source of pressure generation is the passage of the propeller blade bulk through the water. Generally the pressures from these two sources are approximately equal in amplitude, but can be quite different in phase. The third source, cavitation, is the growth and decay of cavitation bubbles as the blade moves into high wake regions. Since the growth and decay of a volume radiate pressure much more effectively than moving a volume from one place to another or introducing a flow from a source to a sink, the pressures from small cavitation volume changes can be large.

The determination of propeller generated hull forces can be made by two processes: (1) estimation of the hull pressure and (2) an integration process involving Green's function which yields the total excitation force. Either process involves many engineering approximations for a reasonable solution. Generally the hull pressure process involves determining the pressure that would be generated by the loading and thickness of the propeller in a free field and multiplying this pressure by a factor, usually 2, to represent the pressure of the hull. This process is entirely inadequate for estimating differential pressures across narrow surfaces such as skegs or rudders. The Green's function process requires an estimate of the added mass of the hull surface for motions corresponding to each of the components of force and moment that are required (19). Theoretical processes for predicting pressure differences across wedge and cone shaped surfaces (20,21) are available, but not yet programmed. A theoretical approach, the Smith-Hess procedure, for predicting the hull pressure is available, but the calculation is so long that it has not at the present time been programmed.

Procedures for predicting hull pressures and forces are presented in Appendices 7-1 and 7-2. The procedure for predicting the hull force is presented in Appendix 7-3.

The evaluation of propeller cavitation [Box 11] is discussed in Reference (22). It does not appear from this paper that a general computer program for the calculations is available.

It will be apparent that the prediction of the excitation from a propeller is far from an exact process at the present time (23). However, by a combination of calculations and experience, amplitude and phase of the propeller harmonic longitudinal forces and harmonic torque about the rotational axis can probably be estimated to a 90 percent probability that the predicted value will lie between 85 percent and 120 percent of the correct value. The accuracy for forces and moments referred to axes normal to the rotational axes is not as good--maybe a 90 percent probability that the predicted value will lie between 75 percent and 140 percent of the correct value. The main source for error lies in the wake values used as input to the computer programs.

The values of propeller excited hull pressures and propeller excited hull forces and moments cannot be estimated as well. In the absence of cavitation, the pressures on the hull surfaces adjacent to the propeller tip can be estimated with, say, 90 percent probability that the predicted value will lie between 65 percent and 150 percent of the correct value. Horizontal hull forces and moments about a vertical axis in the noncavitating condition are generally not predicted at present. If the prediction were of sufficient importance to justify development of computer programs, the accuracy would probably be about the same as the prediction for vertical forces and moments about the horizontal axis.

When cavitation is present, assuming that the amount of cavitation is not excessive from the viewpoint of propeller durability and efficiency, the accuracy of hull force predictions would probably be such that 80 percent would lie between 50 percent and 200 percent of the correct value.

RESPONSE CALCULATIONS

The prediction of excitation, it will be noted, is unique to ship vibration problems. For this reason the procedures for predicting excitation are not fully developed, but the procedures that are developed are generally used.

In the prediction of response, the procedures that can be used have wide application for structures other than ships and have generally been developed for such structures as aircraft and buildings. As a consequence, there are many computer programs that are suitable for predicting ship response, but they are not widely used. There have been some programs developed for particular aspects of ship vibration problems.

The procedure recommended in Reference (8) is to assure that if the component substructures of the ship have suitable response characteristics, then the ship composed of these substructures will have suitable characteristics. Consider now the computer programs available for analyzing the structures discussed in the several boxes.

Determine the Forced Response of the Shafting by Longitudinal Excitation - [Box 14]

By previous calculations the longitudinal exciting force at the propeller will have been determined. The purpose of this study is to find whether the vibration level generated by this excitation will be acceptable. The propeller and the length and diameter of the shafting will be known, and the thrust bearing will probably have been selected. The unknown quantity will be the stiffness of the thrust bearing foundation. The amplitude of motion at the thrust bearing as a function of frequency for different values of foundation stiffness is required.

The stiffness of the thrust bearing foundation must be determined to assure that a shaft longitudinal vibration resonance does not fall in the operating range. For preliminary analyses, the foundation static stiffness can be determined by the use of several methods. The simplest process is to represent the foundation and bottom as a combination of frustums of wedges and beams. This procedure is described in Reference (24). A process requiring less engineering judgement is to use finite element methods, assuming that the machinery double bottom is supported at its edges. It is also possible to represent the machinery space double bottom as an anisotropic plate.

Generally, it will be found that the natural frequency of the bottom structure will not be far removed from the propeller blade frequency. If this happens, the propeller through longitudinal vibration of the shafting will excite engine room vibration even though the natural frequency of the shaft in longitudinal vibration determined from statical stiffness considerations appears to be suitable. This aspect is considered in the following section.

Several types of computer programs are suitable for this analysis. The system can be broken down to a sequence of masses connected by springs. This can be analyzed by a Holzer Table program, the kind developed for torsional vibration, or by a standard finite element program such as ANSYS, MARC, STARDYNE, NASTRAN, SESAM, etc. However, the shafting, whose distributed weight is several times that of the propeller with its associated water inertia, consists of long lengths of constant diameter. This characteristic is encouraging to a program that represents the shaft as distributed mass and elasticity, and a few computer programs have been developed which utilize this property. In such a case the system can be defined with a minimum of input variables, thus saving time and improving accuracy and reducing the probability of erroneous inputs.

The following appendices taken from Reference (25) indicate that the Maritime Administration has a program (Appendix 14-1) based upon the Holzer Method for determining longitudinal vibrations; that J. J. McMullen has a program (Appendix 14-2) for determining longitudinal vibration where the shaft is modeled as lumped masses; and that Newport News has a program (Appendix 14-3) that can represent the shaft as a distributed mass system. A Littleton Research program utilizing lumped and distributed masses and elasticities is described in Appendix 14-4. Reference (26) contains results of a survey for ship structure computer programs made in 1974.

Determine Forced Response of Machinery Space - [Box 15]

The shafting system is connected in longitudinal vibration to the machinery space double bottom through the thrust bearing. Thus, vibrations of the shaft will be coupled with those in the machinery space, and vibrations in the machinery space bottom structure can be strongly coupled with longitudinal vibration of the shafting. Although it might be desirable to model the double bottom as an anisotropic plate with variable inertias for the same reasons that the distributed mass-elasticity procedure is used for the shafting, this type of model has not been developed, and it is necessary to use finite element modeling. It is desirable that the computer system that is used be compatible with that used for the complete ship. If the final ship is to be modeled by finite element procedures, the same system should be used for the machinery space, which can then be incorporated in the full model as a substructure. If the complete ship is to be modeled as a Timoshenko beam with sprung masses, any convenient finite element model can be used for the machinery space.

Determine Forced Response of the Shafting in the Lateral Direction Assuming a Rigid Hull; i.e., Rigid Pin Support at Bearings - [Box 16]

The shaft responds laterally to the harmonic force and moment excitations about axes normal to the rotational axis. If the lateral natural frequencies of the propeller and shaft system coincide with the blade frequency excitation, the input to the hull through the bearings can be strongly amplified. Calculations of ship response generally show peaks associated with lateral frequencies of the shafting. It is, therefore, desirable to design the shafting system so that these resonances will not occur at the normal operating speeds. As with the longitudinal vibrations, these studies are successively made on models of increasing complexity. The first studies are applied to the shaft simply supported at the bearings (either at the forward and after edges or one-third of the distance from the rear of the stern bearing). Since it is known that the bearings are relatively flexible, this model will generally give a frequency that is high so that if the lowest lateral frequency is less than, say, 30 percent above the full power blade frequency, it will probably be wise to consider relocating the bearings or modifying the shafting to raise the frequency. A finite element computer program is suitable for this analysis. It is also possible to use beam programs which include the effects of hull flexibility. The supporting structures can be modeled by making the supports very stiff or rigid. These programs are discussed in the next section.

Determine the Lateral Responses of the Shafting Including the Effects of Hull Flexibility - [Box 17]

If gyroscopic effects are neglected (they are important for whirling, but relatively unimportant at blade frequencies) and the supports are of equal stiffness in all directions, the natural frequency and response of the shaft will be the same in all directions. If the structure is symmetrical about a vertical axis and gyroscopic effects are neglected, the shaft will have two natural frequencies and corresponding mode shapes, one vertical and one horizontal. If the structure is not symmetrical, the fundamental normal modes will be skewed to the vertical, but at right angles to each other. The moment restraint at the bearings can have a significant influence on the shaft frequency.

The amount of structure to include in the stiffness calculation is a matter for the analyst's judgement. The object is to evaluate the stiffness to a region of a large hull mass. For a single screw ship, this may involve the structure from the after peak bulkhead and up to the steering gear flat. For shafts supported by struts, it will include the struts and their backup structure. If the complete hull is analyzed using a finite element analysis [Boxes 20 and 21], the validity of the modeling can be tested.

Since the structure supporting the shaft bearings is complicated, the use of finite element methods is the most feasible way of determining the support stiffness. The stiffness between the shaft and bearing of a stave bearing can be quite low if the staves are rubber. The stiffness of an oil film bearing is such that a bearing force introduces a motion having a component perpendicular to the load.

If,as a result of the calculations of shaft response, it is found that there are no shaft resonances near the operating speeds of the ship, the shafting can be considered satisfactory for this level of refinement. Later analyses of the whole ship will confirm its suitability. If, on the other hand, lateral resonances appear close to the operating speed, then by changing one or more of the following, a new propulsion system can be developed which has resonances properly located:

1. The overhang of the propeller beyond the stern bearing.
2. The span between the last two bearings supporting the propeller shaft.
3. The diameter of the propeller shaft.
4. The support of the propeller shaft bearing:
    (a) The skeg and stern tube structure for a single screw ship having a skeg supported bearing.
    (b) The angles, size, attachment to the bearing barrel of the arms carrying a strut bearing for open screw ships.
    (c) The structure supporting strut arms.
    (d) Other changes as indicated by the calculations.

Such changes are frequently required, and good judgement, often using analyses of simple models, is required to discover the optimum solution rapidly and inexpensively.

Appendices 17-1 and 17-2 present information on two computer programs used for the analysis of transverse propeller shafting vibration.

<u>Conduct Superstructure Modal Analysis - [Box 18]</u>

In addition to substructures of the shafting and machinery spaces, it is desirable to make a study of the superstructure as a subsystem since resonances in this region are a frequent cause of vibration troubles. Finite element methods are generally most suitable for modeling this structure by considering it to be attached to a rigid strength deck. With the high superstructures common on container ships and very long ships, some superstructures vibrate fore and aft as a cantilever beam. On others the decks vibrate symmetrically within the sides, while on still others the decks vibrate anti-symmetrically (port up, starboard down) so that the finite element model should not be too coarse to suitably represent the complexity of possible modes. Clearly the most suitable programs for these analyses are finite element programs.

It may be desirable to make substructural studies of other portions of the ship such as the rudder horn subsystem or systems involving stern crane or elevator carriers. It might also be desirable to combine two or more smaller subsystems. For example, the lateral shaft and substructure subsystem can be connected to the longitudinal shaft and machinery space subsystem to form a larger combined subsystem.

When the subsystems have been designed so that it is expected that they will be free of vibration resonances, it is time to make a vibration analysis of the complete ship. This analysis of the full ship fulfills two important functions:

1. It checks and confirms the validity of the boundaries assumed for the substructures.
2. By modeling the ship as a whole, it is possible, with the proper damping, to predict the vibration levels in all parts of the ship as a function of frequency. Comparing these predictions with established acceptable levels allows an assessment of acceptability of the ship at a point in construction where corrections and changes to overcome serious difficulties can be determined and incorporated in the design.

<u>Assemble Model of Entire Ship and Determine Vibration Amplitudes and Stress Levels of the Complete Ship - [Boxes 20 and 21]</u>

Reference (8) indicates that under some conditions the complete ship may be satisfactorily modeled as a beam structure using a Timoshenko beam as a base.

For vertical vibration the ship may be modeled as parallel beams elastically connected and carrying sprung weights. For lateral vibrations the bending is strongly coupled with torsion, and so a coupled model is required. Strictly speaking, the vertical bending should be coupled with axial vibration of the ship, and this is probably advisable if the blade frequency approaches an estimated longitudinal frequency of the ship.

Where these conditions cannot be met, it is necessary to model the ship by finite element methods in order to obtain reliable estimates of response.

Although heavy vibrations of twice and three times blade frequency can be measured on ships, the methods of analysis that are presented in this paper are feasible only for the range of the fundamental blade frequency, except for unusual cases.

As an example of a ship modeled in terms of beams, consider Figure 2, an uncoupled vertical vibration model, and Figure 3, a coupled lateral-torsional vibration model. A finite element model for half a ship (because of symmetry, only half need be represented) is shown in Figures 4 and 5.

For the analysis of structures as complex as shown in Figures 2 and 3, the computer program GBRP (General Bending Response Program) developed at the Naval Ship Research and Development Center (27) should be used. In the analysis of the ship hull, it can treat vertical as well as coupled lateral-torsional vibration. These capabilities and other features of the program are described in Appendix 20-1.

To define the elastic properties of the structure, it is necessary to define the cross-sectional elastic properties. A program and procedure for computing

I_y	(Moment of inertia about transverse axis)
I_z	(Moment of inertia about vertical axis)
I_{yz}	(Product of inertia relative to horizontal and vertical axes)
A	(Cross-sectional area)
$K_{xz}A$	(Shear area constant vertical plane)
$K_{xy}A$	(Shear area constant transverse plane)
J_x	(Torsional area constant about longitudinal axis)
$\bar{z}$	(Vertical coordinate of the neutral axis)
$\bar{y}$	(Transverse coordinate of neutral axis)
y', z'	(Coordinates of the shear center of the section)

is given in Reference (28). This program calculates the equivalent beam parameters for the ship section properties using data tabulations obtained from hull plans by a preestablished orderly procedure.

U.S. Steel Engineers and Consultants, Inc., developed a program that represents a ship as a beam on an elastic foundation. Information on this program is given in Appendix 20-2. Other vibration programs based upon modeling the hull as beams have been developed by Lloyd's Registry of Shipping and by Dr. Ing. E. Metzmeier of the Institut für Schiffstechnik in Berlin, Federal Republic of Germany. These programs are briefly described in Appendix 20-3.

The factors that enter into the choice of the number and location of the subdivisions of the hull structure are considered in Reference (8). The advantage of representing a ship by a beam model is that the computer analysis is more direct and more easily interpreted and is considerably less expensive than that with a finite element analysis for a structure that is as well defined. The disadvantages are that for many ship vibration problems, particularly where the decks are open so that the vibration across the width of the ship is important, the beam representation of the ship is inadequate and a finite element process is required for satisfactory modeling.

The use of finite element methods for predicting ship vibrations is becoming widespread. Some organizations have developed finite element programs specifically for ship applications. Among these should be mentioned SESAM-69 (Appendix 20-3) developed by Det norske Veritas, the Norwegian classification society (29), and DASH (Appendix 20-3) developed by the Netherlands Ship Research Center. Bureau Veritas, the French classification society, has made many finite element analyses of ship structures. They do not supply information on the characteristics of their program.

The Electric Boat Division of General Dynamics Corporation developed and maintains a finite element computer program GENSAM for use on submarine vibration problems. Information on this is presented in Appendix 20-3.

Since the cost of developing, maintaining, and updating a large finite element computer program is high, it is common to apply general purpose computer

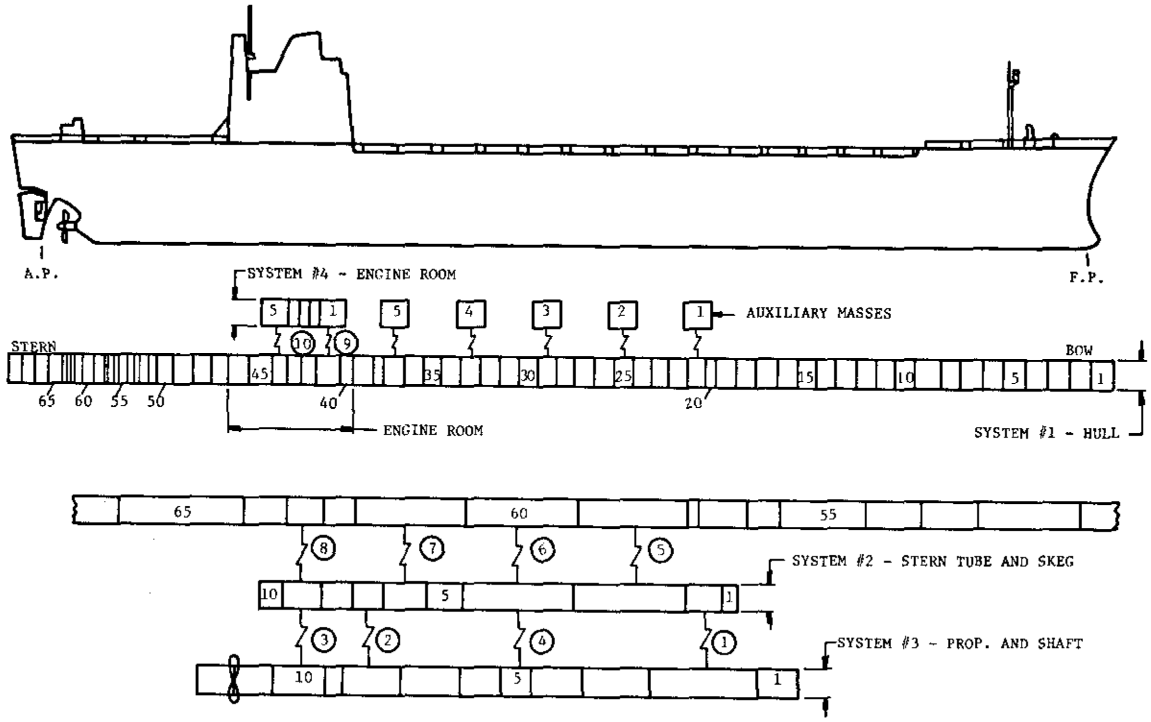

FIGURE 2. UNITIZED AND CONTAINERIZED SHIP, VERTICAL VIBRATION MODEL

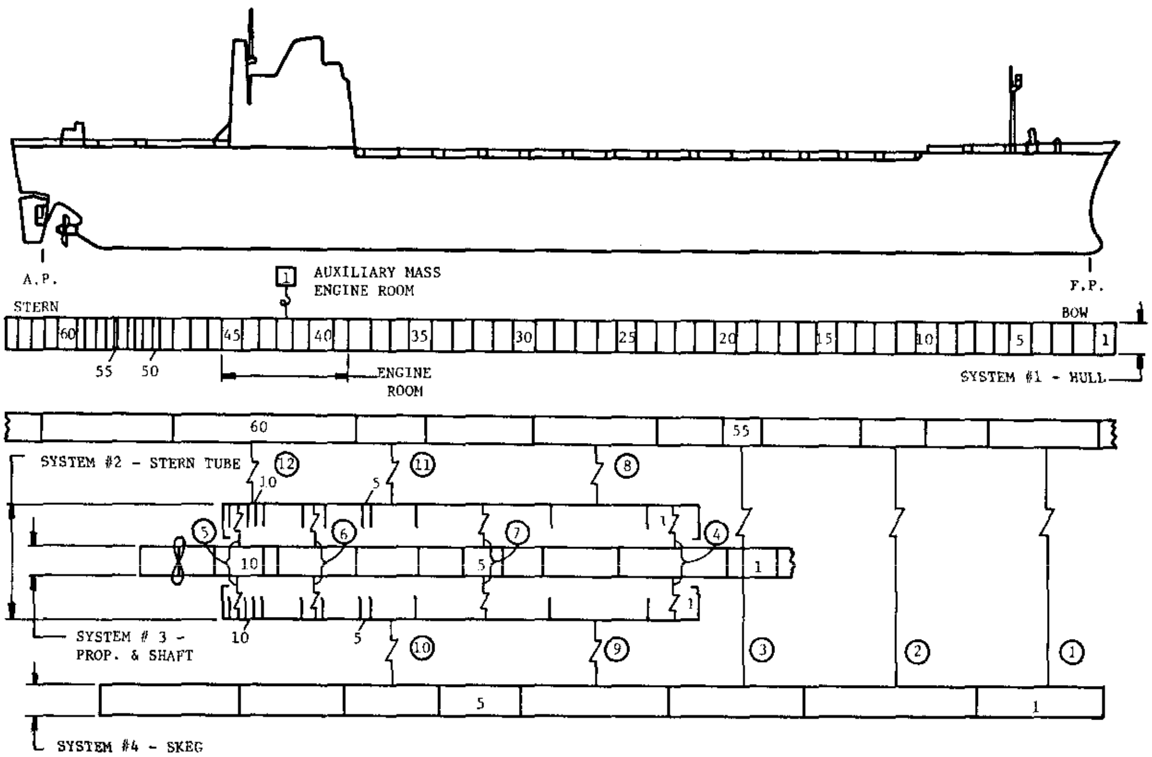

FIGURE 3. UNITIZED AND CONTAINERIZED SHIP, TRANSVERSE VIBRATION MODEL

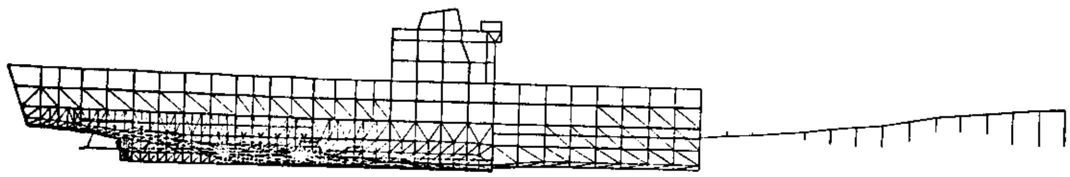

FIGURE 4. ELEVATION VIEW OF FINITE ELEMENT MODEL

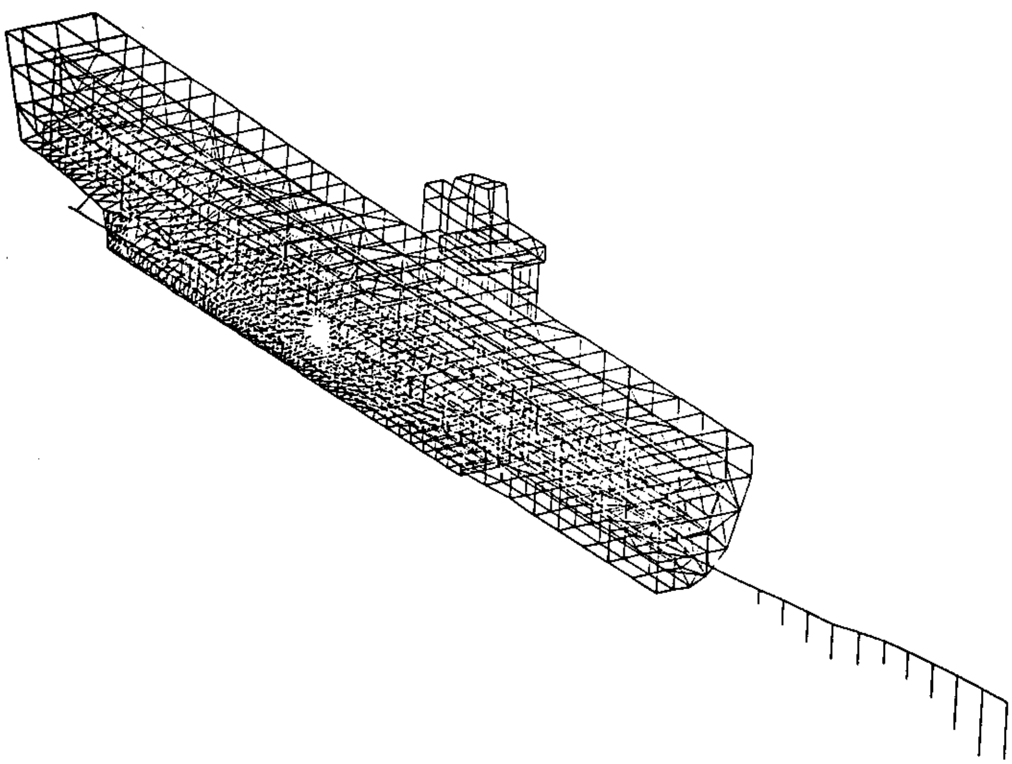

FIGURE 5. ROTATED VIEW OF FINITE ELEMENT MODEL

programs to ship vibration problems. In particular, NASTRAN is used by Lloyd's, the American Bureau of Shipping, Littleton Research and Engineering Corporation, and probably by other organizations. This program was developed for the analysis of large structures and readily applies to ships. It is being continually updated to improve the representation of structural elements and the processing efficiency. Since ship applications constitute only a small part of NASTRAN usage, the program does not contain a subroutine for determining an added mass matrix associated with the entrained water. Because it was developed for large projects, NASTRAN carries high overhead structure. Most ship finite element studies involve models that are large enough to benefit from the generality provided by the overhead, but for many small studies that will not later be incorporated in the large model, it may be desirable to use other finite element programs such as STARDYNE, ANSYS, MARC, STRUDL, or SAP.

SUMMARY

This paper has shown that there are many computer programs available for predicting the loading and response of a ship due to propeller-induced excitations. The choice of a particular program is largely a matter of availability, user experience, and degree of sophistication desired in the analysis.

For the most part, the present programs can adequately predict the excitation and response, with the exception being in the analytical prediction of cavitation pressures on the hull surface. Unfortunately, these cavitation effects can be significant, and continued research in this area is required.

In the area of overall design philosophy, there seems to be some reluctance on the part of shipbuilders to fully employ the analytical techniques available or on the part of shipowners to insist that the techniques be used to minimize vibrations in the design stage. These analytical procedures must be utilized on a routine basis with a feedback mechanism to compare measured and predicted calculations and response so their accuracy can be evaluated and, if necessary, improved. Only through a logical design process can ships be consistently built which possess acceptable vibration characteristics.

ACKNOWLEDGEMENTS

The authors wish to thank all individuals and companies which furnished information about their computer programs. Because this particular field is the subject of many active research programs throughout the world, it is not possible to recognize each on an individual basis. A special appreciation is extended to the Ship Research Committee which has offered many valuable comments and suggestions throughout the conduct of the project associated with this paper.

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APPENDIX 4-1. LONGITUDINAL VIBRATION OF SHAFTING, I

Program Developed and Used by Littleton Research and Engineering Corporation

Output Information

A plot of natural frequency of the shafting system as a function of the stiffness of the thrust bearing and its foundation. Alternatively, if the thrust bearing stiffness is known, the variable may be the foundation stiffness.

Input Information

1. Shafting arrangement, diameters, and lengths.
2. Propeller mass, diameter, number of blades, pitch, and developed area ratio (or mean width ratio).

Basis for Calculation

The propeller is represented by its weight and the weight of entrained water as given by Lewis and Auslaender. The shaft is represented by a distributed mass and elasticity. The gearing and turbines forward of the thrust bearing are not included.

This is a proprietary program not developed for general distribution.

APPENDIX 6-1. PROPELLER MEAN AND VIBRATORY FORCES PROGRAM

Program Developed by Davidson Laboratory, Stevens Institute of Technology, under U.S. Navy contracts; widely used.

Output Information

1. Steady and time-dependent blade loading distribution at multiples of shaft frequency.
2. Mean and blade-frequency force and moment components in coefficient form for:
   (a) Thrust/$\rho n^2 d^4$
   (b) Torque/$\rho n^2 d^5$
   (c) Transverse force/$\rho n^2 d^4$
   (d) Vertical force/$\rho n^2 d^4$
   (e) Transverse bending moment/$\rho n^2 d^5$
   (f) Vertical bending moment/$\rho n^2 d^5$

where

ρ = fluid density
n = propeller rpm
d = propeller diameter

3. Blade bending moments about the pitch line at various radial positions and for various orders of excitation.
4. Information for the study of cavitation inception.
5. Information for the study of blade stress analysis which is performed by utilizing the STARDYNE-CDC finite element computer program.

Input Information

1. The propeller blade geometry.
2. The Fourier components of the spatial variation of the axial and tangential components of the wake.

Basis for Calculation

The program uses unsteady lifting-surface theory and takes into consideration all the relevant propeller geometry and the spatial nonuniformity of the inflow field.

The program is available through Davidson Laboratory for $6,000. See Reference (14).

APPENDIX 6-2. HARMONIC FORCES AND MOMENTS GENERATED BY A PROPELLER IN NON-UNIFORM FLOW

Program Developed and Used by Littleton Research and Engineering Corporation

Output Information

1. Magnitude and phase of the three components of harmonic propeller force and the three components of harmonic propeller moment.
2. The steady vertical and horizontal forces and moments arising from first-order wake action (thrust offset).

Input Information

1. Propeller drawing.

   The propeller drawing should show the following information: propeller diameter, hub diameter, rake, number of blades and propeller material; the variation with radius of chord, skewback, and pitch; propeller sections at several radii showing the variation of thickness along the chord. For propellers designed in Europe, the variation with radius of the distance from the reference line to the leading edge, trailing edge, and point of maximum thickness is acceptable in place of the variation of chord and skewback.

2. Ship speed and corresponding shaft rpm.

3. Wake as measured in a model test.

   The results of a harmonic analysis of the measured wake are required. If the harmonic analysis results are not available, the measured inflow velocities specified at several points along the radius and at frequent points around the circumference are acceptable, and a harmonic analysis will be performed.

   If a measured wake is not available, it can be inferred from the available wakes of other ships.

Basis for Calculations

Propeller forces are determined by lifting line theory. This is much less complex than the Davidson Laboratory lifting surface theory, but is considered adequate in view of uncertainties in the wake and the wide variation in service wake due to ship motions and sea action.

The main reason for continuing to use the lifting line theory calculation is that it is the basis for the predictions of hull pressure and hull forces (see Appendix 7-1).

This is a proprietary program not developed for general distribution.

APPENDIX 6-3. CALCULATION OF STEADY AND HARMONIC PROPELLER FORCES

Computer Program Used by the American Bureau of Shipping

Output Information

1. Mean and blade-frequency components of the three forces and three moments acting on the propeller.

2. Time-varying blade pressure distribution at each wake harmonic.

Input Information

1. Propeller blade geometry.

2. Fourier coefficients of the spatial variation of the axial and tangential components of wake.

Basis of Calculation

The program employs an extended version of unsteady lifting line theory as developed by Dr. Neal A. Brown at MIT (15). The extension includes the effects of propeller skew, which were not treated in the original theory. The results of this program are used as partial input to the "Surface Force" program described in Appendix 7-3.

APPENDIX 6-4. HARMONIC FORCES AND MOMENTS GENERATED BY A PROPELLER IN NON-UNIFORM FLOW

A Computer Program Under Development at Massachusetts Institute of Technology by Professor Justin E. Kerwin, Department of Ocean Engineering

The program represents the propeller blade by grid points distributed over the surface and the wake spatially defined (cylindrical coordinates) in three directions: longitudinal, tangential, and radial. A distribution of vorticity is assumed over the surface, and by successive iteration is refined to be compatible with the boundary of the propeller surface and the laws of hydrodynamics, Kelvin's theorem, and the Kutta requirement for flow continuity at the trailing edge.

This discrete element approach appears to offer a number of advantages as a starting point for the computation of unsteady, partially cavitating flows:

(a) It is capable of yielding accurate predictions of mean loading, both at design and off-design conditions.

(b) Being a numerical procedure, blade geometry can be incorporated exactly so that propellers with large skew, rake, and varying pitch distribution can be accommodated. This is considered essential, since it is through the variation of these parameters that optimum propeller designs can be evolved.

(c) Since the procedure includes all three components of induced velocity, there is no particular problem in including tangential and radial wake field components.

(d) Since no loading mode functions are employed, the modifications ultimately required to include the cavities would appear to be feasible. Source elements presently included to represent blade thickness can assume the further role of representing the cavity volume.

(e) A discrete element method lends itself naturally to a step-by-step domain solution, which is also essential for the subsequent inclusion of unsteady cavitation.

The procedures are still under development, but have been applied to specific cases with good results. See References (16) and (30).

APPENDIX 6-5. HARMONIC FORCES AND MOMENTS GENERATED BY A PROPELLER IN NON-UNIFORM FLOW

Computer Program Developed and Used by the Admiralty Research Laboratory, Teddington, England

Output Information

1. The input data.

2. If wakes are given as velocity measurements, the harmonic values are printed (to the 71 harmonic). If given as Fourier components, these are listed.

3. The contribution to thrust, torque, vertical and horizontal forces, and moments from each specified radial section.

4. The integrated thrust, torque, horizontal and vertical forces, and moments for multiples of blade rate harmonics.

Input Information

1. Shaft speed.

2. Propeller geometry, including skew, chord length, blade pitch angle at specific radii.

3. Wake, either in Fourier Series, amplitude-plus-phase form, or as equally spaced measurements of wake at the radii where the propeller geometry information is given. Only axial or both axial and tangential wakes may be specified.

4. Calculations can be run for successive skew values.

5. Input radii may vary from 4 to 14.

6. As many as 20 skew configurations may be determined.

7. As many as 140 harmonics of the blade frequency forces may be calculated, but generally the number is limited to 10.

8. As many as 100 wake harmonics and 200 wake measurements per radius may be input.

Basis for Calculation

The calculation of the fluctuating forces on a propeller falls into three parts. The first part is the calculation of the variation of the inflow velocity to the blades; the next stage involves the calculation of the fluctuating lift-distribution on a section of blade associated with this fluctuating inflow; the final stage is the calculation of the propeller forces and moments. The calculation of the fluctuating lift is based on two-dimensional unsteady airfoil theory. It ignores blade-to-blade interaction and the variation with radius of the various significant parameters. These approximations would be unacceptable for predicting the steady lift, but are acceptable for the unsteady lift, probably overestimating the lift. See Reference (17).

APPENDIX 7-1. CALCULATION OF HARMONIC FORCES AND MOMENTS ON THE HULL GENERATED BY PROPELLER ACTION

Program Developed and Used by Littleton Research and Engineering Corporation

Output Information

1. The harmonic hull surface pressure at blade beat frequency generated by the loaded, noncavitating propeller in the region of the propeller (generally at grid points corresponding to underwater intersections of buttocks and frames within 4 diameters of the propeller).

2. By integration of the above, the blade frequency harmonic hull forces and moments acting on the hull because of noncavitating propeller action.

Input Information

1. The computed propeller lift distribution along the propeller blade (see Appendix 6-2).

2. The geometry of the propeller.

3. The hull coordinates at the points of pressure determination.

Basis for Calculation

The free-field pressures (i.e., the pressures that would exist in open water if the hull were not present) are calculated at each hull grid point due to (1) the loading on the propeller blades (assumed to be concentrated at the forward quarter point of the blade chord), (2) the thickness of the propeller blade. The sum of these two pressures, in their proper phase, is multiplied by 2 to give the reported pressure on the hull surface. The pressure from a harmonically varying force having x, y, and z components involves the distance from the point to the location of the force. Substituting steady and harmonic forces and distances as a function of shaft angle yields values of the pressure. The resulting equations involve a series which under certain conditions converges slowly. Originally only a few terms were developed. More recently the general term has been developed, allowing sufficient terms to assure convergence. This results in pressures that correspond to measured values. The integration for blade thickness is similar. If the cavitation volume on the blade could be defined by a Fourier series, the same process could be applied. This has not yet been done.

This is a proprietary program not developed for general distribution. See Reference (13).

APPENDIX 7-2. CALCULATION OF STEADY AND HARMONIC PRESSURE FIELDS GENERATED BY A NONCAVITATING PROPELLER

Program Developed by Davidson Laboratory, Stevens Institute of Technology, Under U.S. Navy Contracts

Output Information

This program furnishes the steady and harmonic components of the pressure field generated by a noncavitating ship propeller operating in a spatially variable inflow.

Input Information

1. The propeller blade geometry.

2. The Fourier components of the spatial variation of the axial and tangential components of the wake.

3. The spatial location of the points where the pressures are desired.

4. The steady and time-dependent blade loading distribution at multiples of any shaft frequency as produced by the program described in Appendix 6-1.

Basis for Calculation

This program is a continuation of the one described in Appendix 6-1 and requires data generated in that program. It is available through Davidson Laboratory for $5,000. See Reference (31).

APPENDIX 7-3. CALCULATION OF PROPELLER-INDUCED HULL SURFACE FORCES

Program Developed and Used by Professor William S. Vorus (University of Michigan) and the American Bureau of Shipping

Output Information

This program computes all components of the hull force and moment at multiples of the propeller blade rates. (In general, the vertical force component is the only one desired.)

Input Information

1. Propeller geometry.

2. Wake distribution.

3. Stern lines and coordinates describing the sectional geometry of approximately the aft one-third of the ship.

4. Time-dependent geometry of propeller cavitation effects (optional).

5. Time-varying blade pressure distribution at each wake harmonic (output from program described in Appendix 6-3).

Basis of Calculation

This program employs the method presented by Professor William S. Vorus in Reference (19). The conventional procedure of evaluating the hull forces is to integrate the propeller-generated pressures over the hull surface. These pressures are due to diffraction of the propeller-induced water flow by the hull. The diffraction problem and hence the pressure integration difficulties are avoided in the analysis and computer program by utilizing a special application of Green's Theorem.

APPENDIX 14-1. LONGITUDINAL AND TORSIONAL SHAFTING VIBRATIONS

Program Used by Maritime Administration

Number and/or Name

**C-9-002

Category(s)

**Hull
**Shafting Calculations

Descriptive Program Title

**Shaft Vibrational Analysis
**Using Holzer Method

Source Activity

**Office of Ship Construction
**Maritime Administration
**Washington, D.C.

Engineer(s) Name-Code-Phone

**Richard Siebert, 721.21, 254-7048

Programmer Name-Code-Phone

**NAVSEC

Program Status

**Production

Classification (Security)

**Restricted - NAVSEC Program

Programming Language

**FORTRAN IV

Computer Type Used

**Control Data 6600

Special Hardware

**None

Special Software/Operation

**None

Program Size-Source Deck Cards

**258

Program Size-Object Core Words

**CM50000 Octal Words

Average Running Time (Min)

**2.57

Program Availability

**September 1970

Documentation Status

**Informal - Complete (15 pages)

Program Abstract

This program calculates torsional and longitudinal critical vibration frequencies using the Holzer Method. It was originally developed by NAVSEC for the IBM-7090 and subsequently converted to the CDC-6600 by the Maritime Administration. Double precision requirements were eliminated. Input requires hand calculation of all masses, inertias, and stiffness factors for each component in the turbine-gear-shaft-propeller system. Damping factors are not included in the calculation. Output consists of critical frequencies in CPS and RPM for various numbers of blades.

APPENDIX 14-2. LONGITUDINAL SHAFTING VIBRATIONS

Program Used by J. J. McMullen Associates, Inc.

Number and/or Name

**F-8-008

Category(s)

**Machinery
**Shafting and Bearing Calculations

Source Activity

**John J. McMullen Associates, Inc.
**One World Trade Center-Suite 3047
**New York, New York 10048

Engineer(s) Name-Code-Phone

**Engineering Division

Program Status

**Production

Classification (Security)

**Unclassified

Programming Language

**FORTRAN IV

Computer Type Used

**IBM 360/40 and IBM 1130

Documentation Status

**Informal - User's Guide

Program Abstract

Lumped mass system, using "level" effect. Text description found in NSRDC Report 3358, September 1970. Computes frequencies up to four modes.

APPENDIX 14-3. LONGITUDINAL AND TORSIONAL SHAFTING VIBRATIONS

Program Used by Newport News Shipbuilding and Drydock Company

Number and/or Name

**F-0-016 9.5.0251 FORCE VIB

Category(s)

**Machinery
**Shafting and Bearing Calculations

Descriptive Program Title

**Longitudinal and Torsional Vibration in Propulsion Shafting Systems

Source Activity

**Newport News Shipbuilding and Drydock Company
**Technical Systems Division
**4101 Washington Avenue, Newport News, Virginia 23607
(804) 247-7500

Engineer(s) Name-Code-Phone

**A. S. Pototzky

Programmer Name-Code-Phone

**F. E. Siegel

Program Status

**Active Production

Classification (Security)

**Unclassified

Programming Language

**FORTRAN IV

Computer Type Used

**Honeywell 6080

Program Availability

**Not Available for General Distribution

Documentation Status

**Incomplete

Program Abstract

FORCE VIB is a computer program to calculate the steady state longitudinal or torsional vibratory response of branched shafting systems, such as propulsion systems. The system may have a maximum of 35 elements consisting of masses, dampers, and springs, all with only one degree of freedom. The masses and springs may be lumped or distributed, and the dampers may be viscous or solid. The program uses the mechanical impedance method to calculate displacements, forces, and phase angles, which may all be frequently dependent. The program also allows the varying of values to conduct parametric studies.

APPENDIX 14-4. LONGITUDINAL VIBRATION OF SHAFTING, II

Program Developed and Used by Littleton Research and Engineering Corporation

Output Information

1. A plot of the blade order harmonic force at the thrust bearing as a function of rpm.

2. A plot of the amplitudes of axial motion at the propeller and at the thrust bearing as a function of rpm.

3. Tabular data for above.

Input Information

1. Shafting arrangement, diameters, and lengths.

2. Propeller mass, diameter, number of blades, pitch, and developed area ratio (or mean width ratio).

3. Harmonic thrust (can be determined by program described in Appendix 6-2).

4. The stiffness of the thrust bearing and its foundation.

5. Reduction gear weight.

Basis for Calculation

The propeller is represented by its mass plus entrained water and damping, estimated by Lewis and Auslaender's recommendations.

The shaft is represented by a distributed mass and elasticity and is assumed to have a hysteretic damping (nominally 4%).

The thrust bearing is represented as a concentrated mass elastically connected to a rigid hull.

This is a proprietary program not developed for general distribution.

APPENDIX 17-1. TRANSVERSE RESPONSE OF A BEAM

Program Used by Newport News Shipbuilding and Drydock Company

Number and/or Name

**A-12-00 9.5.0301 Beam Vibration

Category(s)

**Conceptual Design
**Ship Vibrations

Descriptive Program Title

**Vibration Analysis of Beams

Source Activity

**Newport News Shipbuilding and Drydock Company
**Production Computer Systems Division
**4101 Washington Avenue, Newport News, Virginia 23607
(804) 247-7500

Program Status

**Production Use

Classification (Security)

**Unclassified

Programming Language

**FORTRAN V

Computer Type Used

**Honeywell 6080

Special Hardware

**None

Program Availability

**Not Available for General Distribution

Program Abstract

Computes the steady-state transverse vibratory response of a beam with any number of intermediate flexible supports, with generalized end conditions, section properties, and loading.

APPENDIX 17-2. TRANSVERSE VIBRATION OF SHAFTING AND PROPELLER

Program Developed and Used by Littleton Research and Engineering Corporation

Output Information

1. Plots of bearing forces, in two normal directions, as a function of frequency.

2. Plots of shaft motions, in two normal directions, as a function of frequency, at the propeller and at other critical locations.

3. Plots of the shaft deflection curves at each natural frequency within the operating speed.

4. Plots of the steady plus harmonic bending moments in the shaft of the aftermost bearing.

5. Computer tables for above.

Input Information

1. Shafting arrangement, diameters, and lengths.

2. Propeller weight and moment of inertia about its rotation axis, diameter, number of blades, pitch, and developed area ratio.

3. Stiffness or flexibility matrices for each bearing about axes perpendicular to the axis of rotation (force and rotation).

4. Horizontal and vertical harmonic forces and moments, and the steady thrust (can be determined by the program described in Appendix 6-2).

Basis for Calculations

The propeller is represented by its mass, its entrained water, its moment of inertia about the rotational axis, its moment of inertia about an axis perpendicular to the rotational axis, the moment of inertia of its entrained water, and the hydrodynamic damping in its several modes of motion.

The shaft is represented as a series of uniform beams having distributed mass and bending stiffness and hysteretic damping.

The bearings are represented by their stiffness in translation in two directions mutually perpendicular to the shaft axis and by their stiffness in rotation about the same two axes. The bearings are assumed to bend with the shaft; however, where there is flexibility between shaft and bearing, e.g., rubber staves, this flexibility, lateral and angular, is incorporated in the strut matrix. It is generally acceptable to terminate the shaft at the after inboard lineshaft bearing.

The program computes the vibration in terms of coupled properties in the horizontal and vertical directions. It includes the influence of the steady thrust (small effect), but not that of the steady torque (very small).

This is a proprietary program not developed for general distribution.

APPENDIX 20-1. GENERAL BENDING RESPONSE PROGRAM

Program Developed and Used by Naval Ship Research and Development Center

Name

**GBRP

Category(s)

**Lateral, Longitudinal, and Torsional Beam Vibrations
**Bending Coupled with Torsional Beam Vibrations
**Whirling Vibrations of Propeller Shafts

Descriptive Program Title

**General Bending Response Program

Source Activity

**Naval Ship Research and Development Center
**Bethesda, Maryland 20084

Engineer(s)

**Michael E. Golden
**Francis M. Henderson

Program Status

**Active Production

Classification (Security)

**Unclassified

Programming Language

**FORTRAN IV

Computer Type Used

**CDC 6000 Series

Output Plotting

**SC 4020 Plots

Special Software/Operation

**Overlays for Program Subroutines
**Open Core to Optimize Storage

Program Availability

**Availability for General Distribution through David Taylor Naval Ship Research and Development Center

Documentation

**Complete

Program Abstract

The General Bending Response Program (GBRP) consists of the union of three programs: General Bending Response Code 1 (GBRC1) for lateral, longitudinal, and torsional vibrations; GBRC2 for vibrations involving bending coupled with torsion; and GBRC3 for whirling vibrations of propeller shafts. The latter two codes resulted from an extended application of the mathematical model used in the first code. The program formulates the finite difference equations which approximate the boundary-value problem representing the steady-state motion of a vibrating nonuniform mass-spring system such as a ship hull or shafting in bending. The program calculates natural frequencies and mode shapes and the response to specified harmonic driving forces and moments. The program can represent a ship hull connected elastically to other systems such as the propulsion system and to sprung masses. Longitudinal or torsional vibration problems can also be solved by dividing each beam into sections connected by springs, thus reducing the model to a mass-spring system. See Reference (27).

APPENDIX 20-2. SHIP HULL VIBRATORY RESPONSE

Program Used by USS Engineers and Consultants, Inc.

Number and/or Name

**A-12-006 SHRVS

Category(s)

**Conceptual Design
**Ship Vibrations

Descriptive Program Title

**Simulated Ship-Hull Vibration

Source Activity

**USS Engineers and Consultants, Inc.
**600 Grant Street
**Pittsburgh, Pennsylvania 15230

Engineer(s) Name-Code-Phone

**F. Ronald Griffith
(412) 433-6517

Program Status

**Production

Classification (Security)

**Unclassified

Programming Language

**FORTRAN IV

Computer Type Used

**CDC 6500

Special Hardware

**None

Special Software/Operation

**None

Program Size-Source Deck Cards

**4,000 cards

Program Size-Object Core Words

**160K Octal Words

Average Running Time (Min)

**3 Minutes

Program Availability

**Time Sharing Service
Call F. R. Griffith

Cost

**Negotiable

Documentation Status

**Informal Complete

Program Abstract

The purpose of SHRVS is the accurate prediction of the vibratory response of a ship hull to either steady-state or transient loads applied in the vertical centerline plane of the hull. Factors considered include cargo distribution, bulkhead location, machinery space location, as well as the flexural and shear stiffness of the main-hull girder and of the double-bottom structure. See Reference (32).

APPENDIX 20-3

COMPUTER PROGRAMS AVAILABLE FOR COMPLETE HULL ANALYSIS

	Lloyd's Register of Shipping	Netherlands Ship Model Basin	Institut für Schiffstechnik Technische Universität Berlin	Electric Boat Division, General Dynamics Corp.	Computas, Subsidiary of Det norske Veritas
NAME OF PROGRAM	Hull Vibration LR57P	DASH	FREIS, ERZS	GENSAM	SESAM-69
PERSON TO CONTACT	Geoffrey H. Sole	Dr. S. Hylarides	Dr. E. Metzmeier	Dr. Henna Allik	B. Aamodt
TYPE OF ANALYSES	Eigenvalues, Eigenvectors	Eigenvalues, Eigenvectors, Steady-State, and Transient Response	Eigenvalues, Eigenvectors, Steady-State, and Transient Response	Eigenvalues, Eigenvectors, Steady-State, Transient, and Random Response	Forced and Free Vibration Analyses of Steady-State or Transient Vibrations
DATE OF INITIAL COMPLETION/LATEST REVISION	U/1976	1970/1975	1974/U	1967/1976	1968/1975
AVAILABILITY/ PRICE	Service from Lloyd's Register	Service from NSMB	U	Proprietary - Available on a case basis	Service use available for rent or sale
LANGUAGE	FORTRAN IV	ALGOL	FORTRAN	FORTRAN V	USASI FORTRAN
COMPUTER SYSTEM	IBM 370/158	Irrelevant	CDC 6500/ Scope 3.4	UNIVAC 1106, 1108, 1110 EXEC 8	UNIVAC 1108, Other Computers
BASIS OF PROGRAM	U	Finite Element	Finite Difference	Finite Element	Finite Element
TYPE OF STRUCTURES	Represented by a Free-Free Beam	Complete or Partial Ship	Ship Hull	General Structures	General Structures
MANNER OF REPRESENTATION	Beams	Beams, Plates	Beams	Beams, Plates, Shells, 2- and 3-D Continua	Beams, Plates, Shells, 2-D and 3-D Continua
CONCENTRATED OR DISTRIBUTED MASS	Concentrated	Concentrated	Concentrated	Both	Concentrated
SUBSTRUCTURING	No	U	U	Yes	Yes
COMPRESSION OF DYNAMIC MATRIX	No	No	U	Yes	Yes
TREATMENT OF WATER INERTIA	Direct Input or Internal Computation	Direct Input	According to Lewis and Kumai	U	Direct Calculation
TREATMENT OF DAMPING	No Damping	Viscous	Viscous and Hysteretic	User Supplied Damping Arrays Except Viscous	Modal Damping
PROGRAM CAPACITY	121 Nodes 240 DOF	Unlimited	41 DOF Long. 82 DOF Vert. 164 DOF Coup. Hori.-Torsion	Unlimited	Unlimited
REFERENCES	(33)	(34)	None given	Proprietary	(29)

U = undefined