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DEPARTMENT OF THE NAVY

HYDROMECHANICS

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SHIPBOARD VIBRATION AND NOISE CONSIDERATIONS IN THE DESIGN OF RIVER TOWBOATS

by

Edward F. Noonan
and
Angelos Zaloumis

STRUCTURAL MECHANICS LABORATORY
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Shipboard Vibration and Noise Considerations in the Design of River Towboats

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THE SUBJECT OF SHIPBOARD vibration has been given wide attention in recent years. Although many important papers on the subject were published prior to World War II, this field of study received its greatest impetus as a direct result of the large shipbuilding program precipitated by the war. During the war and since that time, many new types of ships and many larger ships with greatly increased powers were built to meet both commercial and naval requirements. With this large shipbuilding program many new vibration problems were encountered and much greater emphasis was naturally placed on this field of study. In the years following World War II continued technological developments in the fields of nuclear propulsion, electronics, and missile weapon systems have brought many new vibration problems to the attention of the Navy. These developments plus our new submarines, our improved sonar equipment, and such new programs as the hydrofoil craft, Ground Effects Machine, and deep-diving submarines has placed the study of shipboard vibration and noise high on the Navy's list of important research programs.

While the Navy has, directly or indirectly, contributed more to the understanding of shipboard vibration than any other segment of our scientific society, significant contributions have also been made through the research panels of professional societies and by many individual investigators. This paper will briefly review some of those problem areas which we believe to be of major importance to the naval architect in the design of river towboats and offer reasonable suggestions to be considered in the design of such vessels. Although we do not speak from broad experience in the towboat field, many shipboard problems and the approaches to their solution developed for sea going ships, will find direct application to river towboats.

BACKGROUND

Fig. 1 demonstrates "The Response of a Ship to its Environment," and, at the same time, illustrates the field of

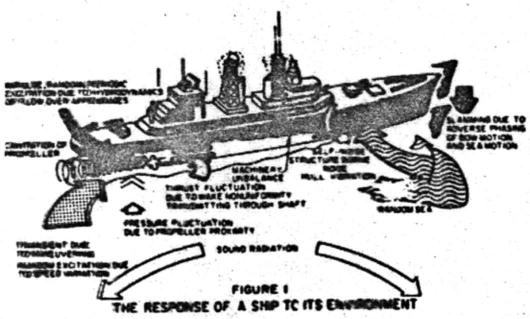

Fig. 1 - Response of ship to its environment

ABSTRACT

The problems of vibration and noise aboard ship have been given increased attention by the Navy since World War II as a result of many technological developments and tactical requirements. Many problem areas are identified and those which have particular application in the design of river towboats are reviewed.

The paper is addressed primarily to the naval architect, shipbuilder, or operator and identifies those areas to which attention should be drawn during design and development. Specific suggestions and specifications are presented, based on current naval practice.

interest of the Ship Dynamics Div of the Structural Mechanics Laboratory at the David Taylor Model Basin. Excluding the rigid body motion of a ship (roll, pitch, heave), we note the influences of random sea and maneuvering which produces a transient response of the hull's structure, the influence of slamming due to adverse phasing of bow motion and sea motion, the impulse, random and periodic excitation due to the hydrodynamics of flow over appendages, pressure fluctuations due to the propellers, thrust fluctuation, machinery unbalance and other machinery forces, noise generation and resulting sound radiation. The study of these forces and the response characteristics of ships to them is encompassed in the following programs of the Division:

1. Structural Seaworthiness -- The development of dynamic design procedures for ships' hulls

2. Slamming -- The study of the mechanism and development of rational procedures to minimize the effects of slamming.

3. Hydroelasticity -- Flutter prediction.

4. Hull vibration -- Response to steady state forces.

5. Machinery vibration.

6. Radiation studies.

7. Full-scale vibration testing and ship evaluation.

8. Development of vibration instrumentation and computer facilities.

The broad objectives of these programs include the following basic considerations:

1. Mechanical suitability (The concern for structural adequacy).

2. Habitability and operability (the concern for adverse effects on personnel and equipment)

3. Detection and detectability (the concern for efficiency in undersea warfare).

Since river towboats are not generally concerned with undersea warfare, we have limited this paper to the first two considerations. In like manner, towboat operations are not generally concerned with structural seaworthiness, slamming, or hydroelasticity problems to the same degree as sea-going vessels. Our preliminary interest, therefore, will center around hull vibration stimulated by steady state forces, either propeller or machinery excited. In addition, we have utilized some experience gained in the Navy's noise reduction program which, if applied, should be helpful in improving the general habitability and operability of river towboats.

HULL RESPONSE

A ship is, in reality, a beam designed to carry a load under varying conditions of support. When we consider the problem of ship vibration we must first consider the response of the beam or primary hull girder and then consider the response of individual items of equipment with reference to the basic girder motion at the point of attachment. As an optimum condition, any piece of shipboard equipment which is directly attached to the hull can be expected to move, as a minimum, with the same amplitude as the primary hull girder. It is important therefore, to understand and control, insofar as possible, the response of the primary hull girder.

In any mechanically vibrating system we are concerned with the natural behavior of the mass-elastic system and its response to exciting forces. A ship is such a system and studies have shown that the hull girder will approximate a free-free beam, supported at its nodal points. The particular geometry of a given ship will of course influence its response characteristics. Normally we are concerned with the flexural modes in the vertical and athwartship directions. In certain cases, such as a destroyer with long slender lines, we may encounter a torsional or twisting mode of the hull which gives trouble and in still other ships the longitudinal or accordian mode has been a cause for concern. By its geometry, however, we can probably limit our consideration in towboats to the flexural modes.

For a first approximation we can assume a ship to be a beam of uniform section and carrying a uniformly distributed load. The natural frequency is obtained from the following expression (1)*:

$$F = \frac{60}{2\pi} A\sqrt{\frac{EI}{\mu \ell^4}} \text{ cpm}$$

where:

- E = Modulus of elasticity, lb/in^2
- I = Moment of inertia of the cross-section about the neutral axis normal to the plane of vibration, $in.^4$
- μ = Mass per unit length of beam, $lb\text{-}sec^2/in^2$
- ℓ = Length of beam, in.
- A = A constant for the mode considered, 1 = 22.4, 2 = 61.7, 3 = 121, 4 = 200, 5 = 298

In applying this formulation to the actual case of a towboat, the mass per unit length μ will include the mass per unit length of the hull girder, the average mass per unit length of the machinery and equipment, and a value of mass per unit length for virtual mass of entrained water. In general, as a first approximation, we may assume the mass per unit length of the hull girder and supported equipment to equal the total mass of the ship divided by its length. The virtual mass per unit length for the flexural modes may be computed as follows:

$$\text{For athwartship vibration} = \tfrac{1}{2} JC_H \times \frac{\rho}{g} H^2$$

$$\text{For vertical vibration} = \tfrac{1}{2} JC_V \times \frac{\rho}{g} b^2$$

where:

- J = Taylor's longitudinal inertial coefficient. (To account for the three-dimensional flow effect of the virtual mass)
- C_H = Landweber's virtual mass coefficient for horizontal motion

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*Numbers in parentheses designate References at end of paper.

C_V = Landweber's virtual mass coefficient for vertical motion
ρ/g = Mass of water per cu ft
H = Draft, ft
b = Mean half-breadth of section at water line, ft

Appropriate values for the longitudinal inertia coefficient may be taken from Fig. 2. The value J is plotted against the ratio of L/B (length over beam). The values for C_H and C_V are obtained from Figs. 3 and 4 respectively. For more detailed analysis of the virtual mass problem, see Ref. 2.

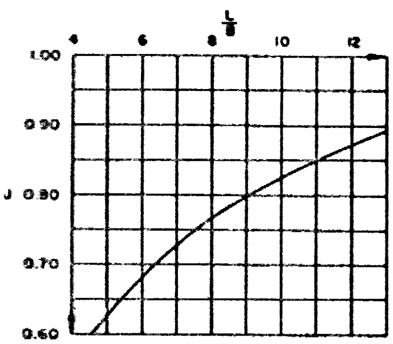

Fig. 2 - Curve for obtaining coefficient J used in virtual-mass estimate

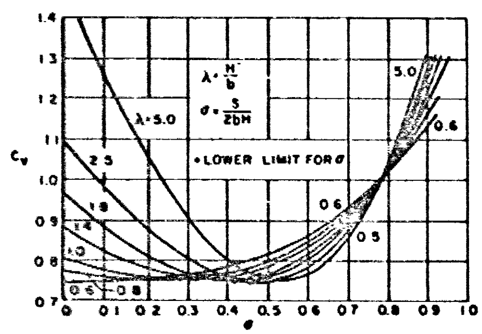

Fig. 3 - Virtual-mass coefficients for vertical motions

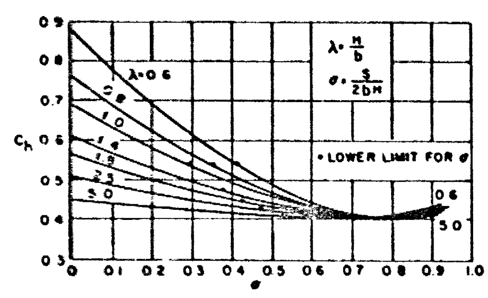

Fig. 4 - Virtual-mass coefficients for horizontal motions

The mode shapes and location of nodal points for the uniform beam and uniformly distributed load are shown in Fig. 5. These frequencies calculate in the ratio of 1.0, 2.75, 5.4, 8.9, 13.35. While the foregoing procedure may satisfactorily establish the frequencies and mode shapes of a vessel having the general geometry of a towboat, this procedure has not proved adequate for ocean going ships having finer lines at both bow and stern. Fig. 6 shows a typical weight curve developed for an ocean going vessel and Fig. 7 shows a typical area - moment of inertia curve. In each case the ship is divided into 20 stations between the forward and aft perpendiculars. An empirical form of the uniform beam formula was developed by Schlick to obtain the fundamental vertical frequency of a ship:

$$F = \phi\sqrt{\frac{I}{\Delta L^3}}$$

where:

F = Frequency, cpm
I = Moment of inertia, in.^{2} ft^2 units
Δ = Displacement, long tons

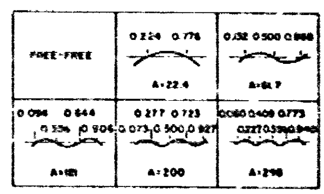

Fig. 5 - Mode shapes and location of nodal points for uniform beam with uniformly distributed load

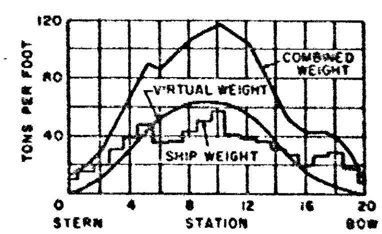

Fig. 6 - Typical weight curve for calculations of mode shapes

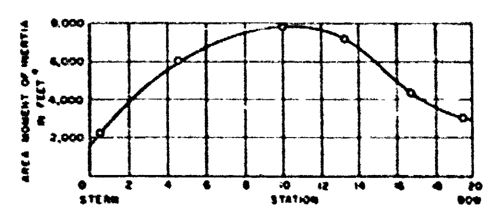

Fig. 7 - Typical area-moment inertia curve used for calculation of mode shapes

L = Length of ship, ft

ϕ = Schlick's empirical constant which includes the constant factors in the uniform beam formula and corrections for the departure of the ship girder and its loading from the basic uniform beam

Typical Schlick constants are given as follows:

For vessels of very fine lines, such as destroyers, ϕ = 156,850.

For large transatlantic passenger lines with fine lines, ϕ = 143,500.

For cargo boats, with full lines, ϕ = 127,900.

The higher modes were generally found to fall in the ratios of 1.0, 2.0, 3.0, 4.0, 5.0 for both the vertical and athwartship modes. The first two athwartship modes are generally found to equal approximately 1.5 times the corresponding vertical modes. This factor will generally increase to approximately 1.7 for the higher modes.

The frequencies for the flexural modes can be expected to vary for different load conditions and for depth of water when the distance between the keel and the bottom is less than approximately six times the draft, a condition frequently encountered in towboat operation. The light load natural frequencies in the vertical and athwartship flexural modes will vary between 1.10 times the heavy load frequency in the fundamental mode, and 1.15 times the heavy load condition for the third mode and above. If the depth of water under the keel is as little as 25% of the draft of the vessel, the hydrodynamic mass is tripled and the frequencies may be reduced to 70% of the deep water value. As a result, a ship travelling in deep water with little vibration may encounter a resonant vibration in shallow water while maintaining the same rpm. Further information on the influence of the depth of water on the virtual mass, is given in Ref. 3.

Although limited work has probably been conducted on the response of towboat hulls in the past, the increased sizes and powers of vessels of this type suggest greater attention will be given to the problem of hull vibration in the future. An understanding of the response of the hull system is a prerequisite to the solution of many such problems and to the avoidance of potential problems while the boat is still in the design stage.

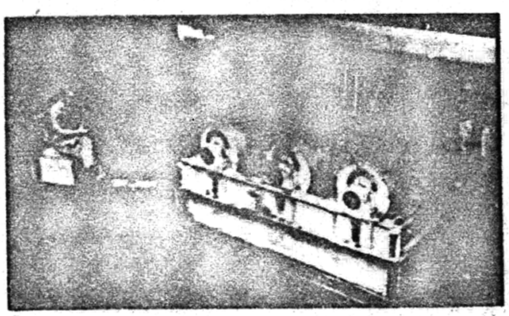

Fig. 8 - TMB 40,000 lb 3-mass vibration generator installed on NS Savannah during dockside vibration generator tests

Such studies should include both theoretical and experimental work. The foundation for such studies are the vibration generator. These generators are machines capable of setting an entire ship's hull into vibration and permit the determination of its elastic characteristics. Figs. 8 and 9 taken from Ref. 4 show two of the latest vibration generators developed by the DTMB for this purpose. The first is capable of developing a maximum force of 40,000 lb and the second is capable of developing a maximum force of 5000 lb. The vibration generator is installed at the aft perpendicular, an antinode for all frequencies, and operated at those speeds necessary to excite the normal modes of the hull. Only the first five modes are generally considered important in ocean going vessels. The magnitude and direction of the driving forces are determined by the angular position of the eccentric weights.

For recording purposes Fig. 10 shows a typical instrumentation arrangement. The 20 pickups are located at regular intervals along the length of the hull and data is obtained simultaneously from all pickups. Thus, with all pickups properly phased, the structural response and mode shapes for all critical frequencies can be obtained.

Although it may not be considered warranted in the case of towboats, the needs of the Navy in this field of study on the response of hull structures has progressed through a con-

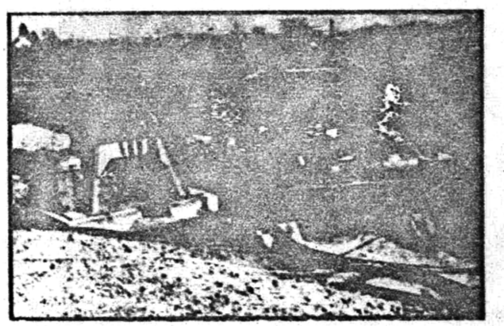

Fig. 9 - TMB 5000 lb 3-mass vibration generator installed to test radar pedestal foundations

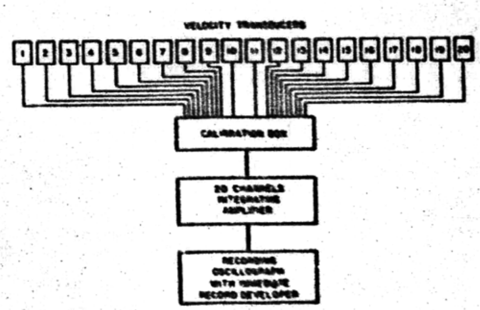

Fig. 10 - Instrumentation arrangement used for shipboard vibration generator studies

tinuous development program. In 1949 the Model Basin, in collaboration with the University of Michigan, published methods for calculating natural frequencies and normal modes of vertical vibration of a ship's hull considered as a free-free beam with bending and shearing flexibility (5). Through continuous development these methods have been extended to include flexural, torsional, torsional-bending, and longitudinal modes. By refinements in calculations of virtual masses and consideration of the effect of masses flexibly attached to the hull, better agreement of theoretical and experimental results are obtained. Calculations of the mode shape with a unit deflection on the stern of a ship, or due to a unit force applied to the propeller or stern, allow a comparison of the response of different hull designs.

Recently a comprehensive study, reviewing and extending previous work, was made on the application of equations for digital and electric-analog solution of the natural frequencies and mode shapes of a ship's hull idealized as an elastic beam (2). Effects of bending, shear, rotary inertia, coupled torsion and bending, initial curvature of the elastic axes, applied forces and torques, sprung masses, and other inertias are included. Methods for manually calculating the physical parameters of the hull from ship's plans and other sources have also been treated on the computer. The accuracy of the results obtained by these methods for uniform and non-uniform beams has been encouraging.

The validity of these theories has been verified by vibration measurements on many different types of surface ships and submarines. The most complete vibration survey ever conducted was made on SS Gopher Mariner in cooperation with the Society of Naval Architects and Marine Engineers (6). Hence, these analyses can now be used with confidence to calculate the frequencies and modes of hull vibration at the design stage so that resonance conditions can be avoided. In view of the higher frequencies of vibration, which are of interest today, and the increasing importance of the interaction of the hull with elastically attached masses and the water, the more general three-dimensional approach to ship vibration is being studied.

On certain classes of ships, flexibly mounted masses such as machinery, rudders, cargo, and superstructures affect hull vibrations (2). Therefore, to explore the possibility of a more adequate representation of a ship's hull as a mass-elastic system subject to vibration, studies have been made to investigate the characteristics of a beam with attached inertias having motion in translation, rotation or coupled translation, and rotation. Analytical, electric-analog, and digital computer methods have been devised to determine the natural frequencies and mode shapes of beam-sprung-inertia systems. Full-scale studies to verify these theories have been conducted on the NS Savannah and the USS Long Beach (CG(N)9).

For those who are interested in pursuing the subject further, the more recent works of McGoldrick (7) and Todd (8) are recommended reading. These two publications, plus Leibowitz (2), contain all significant references on the theory of hull vibration.

EXCITING FORCES

As previously noted, any vibrating system includes both a mass-elastic system and an exciting force which drives it. Earlier we considered the hull girder as the primary mass-elastic system and discussed its response characteristics. This section will deal with the exciting forces. These forces are limited to steady or periodic forces, which will produce:

1. Forced Vibration -- The response of the vibrating system to periodic forces of sufficient magnitude as to excite the system at nonresonant frequencies.

2. Resonant Vibration -- The response of the vibrating system to periodic forces tuned to a natural frequency of the system.

It is obvious then, that to minimize hull vibration, we must limit the magnitude of the exciting forces and insure resonances at important operating conditions are avoided.

The most common vibratory forces encountered may be divided into five groups:

GROUP I -- Forces caused by unbalance of rotating equipment.

GROUP II -- Vertical and horizontal forces on the ship structure caused by the passage of each propeller blade. These forces have a frequency equal to the blade frequency of the propeller, which is equal to the number of revolutions of the propeller times the number of blades. Occasionally harmonics of blade frequency are encountered.

GROUP III -- Torsional vibration of the shaft system excited by propeller blade forces or by nonuniform engine torque. The fundamental frequency of these oscillations may be propeller blade frequency or major engine harmonics.

GROUP IV -- Longitudinal vibration of the engine-shaft-propeller system in a fore and aft direction. This frequency is generally equal to the propeller blade frequency but can be associated with important torsional frequencies excited by engine harmonics, (coupled torsional and longitudinal modes).

GROUP V -- Equipment vibration. This group includes the forces generated by and peculiar to particular equipment characteristics. Included in this group would be such items as secondary engine inertia forces, harmonics of reciprocating compressors, pump vane frequency, and so forth.

Group I, Unbalanced Forces At Shaft Frequency - The unbalanced forces at shaft frequency include mass or hydrodynamic unbalance of the propeller, mass unbalance of the shafting or bull gear in a geared drive mass unbalance or primary inertia unbalance in a direct connected engine, and whipping in the shaft system. The hydrodynamic unbalance of a propeller may be caused by a damaged blade or an error in pitch. In a vessel which previously did not have troublesome vibration, its sudden appearance would throw considerable suspicion upon propeller or shafting damage. Seaweed or other fouling of a propeller may also cause a temporary vibration of this type.

To minimize mass unbalance, the Navy has established static and dynamic balance requirements for all rotating machinery. Specific requirements are given in Ref. 9 under Type II, Internally Excited Vibration. Table 1 is taken from Ref. 9 and gives the type of correction required, considering the motor dimensions and speed of rotation. From this specification we note that all propellers receive the minimum of a static balance and that all propulsion shafting receives a static balance if it rotates at 150 rpm or less and a dynamic balance if it rotates at 150 rpm. The maximum allowable residual unbalance in each plane of correction is given as:

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

or

$$U = \frac{5630W}{N^2} \text{ for speeds between 150 rpm and 1000 rpm}$$

or

$$U = 0.25W \text{ for speeds below 150 rpm}$$

where:

U = Maximum allowable residual unbalance, oz in.
W = Weight of rotating part, lb
N = Maximum operating rpm of unit

The mass unbalance or primary inertia unbalance of a direct connected engine will also provide exciting forces at shaft frequency. Since in this paper we are dealing with the application problem of the naval architect, we will limit our comments on the design aspects of the engine itself to recommendation that the choice of unit should include a consideration of its inherent unbalanced forces. Further details on the computation of the primary inertia forces in the engine are given in another paper.

Shaft whip may occur when the natural frequency of lateral vibration of a shaft system equals the rotational frequency. To avoid the possibility of shaft whip, it is recommended that the natural frequency of the shaft system be at least 115% of the operating speed. For the system shown in Fig. 11, the natural frequency is computed by the following equation taken from Ref. 10:

$$F = \frac{1}{2\pi}\sqrt{\frac{EI}{I_d\left(b + \frac{\ell}{3}\right) + \frac{W_p b^2}{g}\left(\frac{b}{2} + \frac{\ell}{3}\right) + \mu\left(\frac{b^4}{8} + \frac{\ell b^3}{9} + \frac{7\ell^4}{360}\right)}}$$

where:

E = Modulus of elasticity, lb/in.^{2}
b = Propeller overhang, in.
ℓ = Bearing span, in.
W_p = Propeller weight plus 25% for entrained water
I = Moment of inertia of shaft section about diameter, in.^{4}
μ = Shaft mass per unit length, lb-sec^2/in.^{2}
I_d = Mass moment of inertia about a diameter, lb in. sec^2
= $1/2\ I_p$ plus 60% for entrained water, where
I_p = Mass moment of inertia of propeller about axis, lb in. sec^2

All cases of first order vibration are considered deficiencies. By that is meant that if the above sources of first order vibration are carefully considered in the design and development, we should have little concern about them in the finished towboat, or conversely if first order vibration appears, it generally can be eliminated by the correction of these first order forces.

Group II, Propeller Forces - The propeller forces and moments which vibrate the ship's hull, in addition to those resulting from geometric imperfections, are those induced by:

1. The nonuniform inflow-velocity into the propeller plane which then produce forces which are transmitted to the hull through shafts, struts, bossings, or stern tube bearings.

2. Oscillating fluid pressures generated by the moving pressure fields associated with the blades of the loaded propeller when passing strut arms, bossings, or the hull.

Unlike the first order (shaft frequency) vibrations previously discussed, the propeller forces at blade frequency or harmonics of blade frequency are a function of design considerations. The nonuniform velocity field surrounding the stern of the ship and the propeller working in this medium determine the magnitude of the developed forces and are established in the design stage. Unfortunately those factors,

Table 1 - Types of Correction for Mass Unbalance

Length/Diameter*	Speed, rpm	Type of Correction
Less than 0.5	0-1000	One plane (static)
	Greater than 1000	Two plane (dynamic)
Greater than 0.5	0- 150	One plane (static)
	Greater than 150	Two plane (dynamic)

*The length and diameter refer to dimensions of rotor mass, exclusive of supporting shaft.

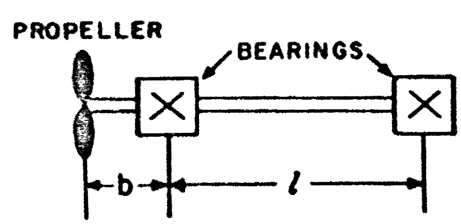

Fig. 11 - Diagram of tailshaft system

such as hull form, propeller clearances, and water depth, which greatly influence the magnitude of the forces, cannot be too well controlled in the case of towboats. However, improvements in flow have been obtained in recent years by stern modifications and the use of Kort nozzles in some instances. Force reductions have been achieved in many cases of ship vibration by the use of skewed propellers.

The ideal approach to this problem is to conduct model wake studies from which the harmonic content of the circumferential wake at various radii within the propeller plane can be determined. A harmonic analysis of this data would permit an evaluation of the relative magnitude of the propeller blade forces for propellers having various numbers of blades. This data, plus a knowledge of the hull natural frequencies and the range of shaft speeds desired will permit the selection of parameters which will minimize the forces at propeller blade frequency. Unfortunately, however, the cost of this approach generally results in its omission and we fall back on the experience of the naval architect. In this respect, shipbuilding is still considered an art.

The most important aspect of any vibration problem is that of resonance. If our propeller blade frequency is tuned to a natural frequency of the hull within an important operating speed range, we can generally expect serious vibration. Obviously the easiest way to correct this problem in a given design is to choose a propeller with the number of blades which provides the best compromise. To do this, a vibration study should be made to determine the location of the natural frequencies of the hull. This can be done ideally by a vibration generator but can generally be determined by underway tests. The location of shaft speed criticals for propellers with alternate numbers of blades can be readily predicted by dividing the observed frequencies by the number of blades. In general, the relative force inputs from alternate propellers can also be estimated at a particular critical by correcting for the change in power at the speed at which the critical occurs and by correcting for the change in intensity of the suction peak. The first correction varies as the square of the critical speed ratio and the latter varies as the cube root of the speed ratio for a free running condition. For example, if we had a hull frequency at 720 cpm and the top speed of the shaft was 180 rpm, a serious critical could occur at top speed when using a 4-bladed propeller. If we considered changing to a 5-bladed propeller, the critical would shift downward to 144 rpm, and for a free running condition the forces would be:

$$\left(\frac{144}{180}\right)^2 \cdot \sqrt[3]{\frac{144}{180}} = 0.64 \times 0.93 = 0.59\% \text{ of the original force.}$$

Of course this would be revised upward somewhat when towing and other higher modes enter into the picture. However, as in all other engineering problems, the ideal design is a compromise and can only be achieved by understanding the alternatives.

Group III. Torsional Vibration - In most installations the presence of serious criticals of torsional vibration may reflect in a torsional fatigue failure of system components or, in the case of geared drives, can produce torque reversal and hammering of gear teeth. Occasionally, however, a torsional critical will result in serious hull vibration. The most likely case of such difficulty would be that of a direct connected drive in which the engine inertia is of the same order of magnitude, or not too much larger than that of the propeller. If, in such a case, a serious first mode critical occurs in the upper operating speed and large amplitudes of torsional vibration are encountered, the amplitude at the engine may be significant when compared to the amplitude at the propeller. In this case the absolute motion of the engine masses will be large and torque reaction in the engine frame will be reflected in vibration in the hull, transmitted through the engine foundation. This source of trouble can generally be avoided, if, in the design of the propulsion system, the inertia of the engine end of the system is high compared to that of the propeller. The relative amplitude curve in this case results in greater torsional motion at the propeller and increased damping to the system introduced through the propeller.

For reasons of safety, the torsional vibration characteristics of any proposed propulsion system should be carefully checked. In the majority of cases the system evaluation is performed by the engine builder. However, in many cases the naval architect or shipbuilder will retain their own consultants to insure the adequacy of the overall system, to optimize the basic arrangement of the propulsion system, and to choose the most appropriate system components.

The design of the propulsion system requires the avoidance of serious criticals of torsional vibration within the normal operating speed. Serious criticals are defined as those having excessive torsional vibratory stresses, or excessive vibratory torque across gears (9). Within the operating speed range, excessive stress is that stress in excess of S_v, where

$$S_v = \frac{\text{Ultimate tensile strength, for steel}}{25}$$

$$S_v = \frac{\text{Torsional fatigue limit, for cast iron}}{6}$$

For average crankshaft steels, the range extends from 75,000 to 125,000 psi, and the corresponding allowable stress would run from 3000 to 5000 psi.

Excessive vibratory torque across gears for diesel installations is given as that torque greater than 75% of the driving torque at the same speed, or 25% of the full load torque, whichever is smaller.

For "pass-thru" criticals which occur below the operating speed, excessive torsional vibratory stress is that stress in excess of 1-3/4 times the value of S_v.

Normally, the naval architect will be concerned with the fundamental mode having its node between the engine and propeller. The frequency can be estimated by either of the following expressions:

When the engine inertia is large compared to the propeller inertia:

$$F = \frac{60}{2\pi}\sqrt{\frac{K}{I_p}}, \text{ cpm}$$

When engine and propeller are of the same order of magnitude:

$$F = \frac{60}{2\pi}\sqrt{\frac{K(I_e + I_p)}{I_e \times I_p}}, \text{ cpm}$$

where:

K = Torsional stiffness of connecting shafting, lb in./rad

I_e = Mass moment of inertia of the engine, lb in. sec^2

I_p = Mass moment of inertia of the propeller, lb in. sec^2 (includes 25% for entrained water)

Resonances may be excited by propeller blades or engine harmonics and critical speeds will occur at the natural frequency of the system divided by the number of propeller blades or the particular engine harmonics important to the proposed range of operations. Frequently the torsional problem will control the choice of propellers. Usually, if the system is not sufficiently close coupled to result in the major I mode critical falling above the operating speed range, the critical may be located well below the important speed range by the use of a torsionally flexible coupling.

The calculation of engine excited criticals is too complex a problem to be treated here. It will suffice to say, however, that much work has been done on the subject, both by the Torsional Vibration Committee of the Society of Automotive Engineers and the British Internal Combustion Engine Research Association. Analysis methods presently employed yield very good results, particularly in frequency determinations. As in most vibration problems, however, estimations of system damping is subject to considerable error and whenever possible, testing of the completed installation is highly recommended to confirm the original analysis. Detailed analysis procedures are well presented in Ref. 11 for engine excited criticals. The amplitude of propeller excited criticals may be estimated from Ref. 10 as:

$$\phi \text{ prop} = 0.0025\,\frac{Qp}{B}$$

where:

ϕ prop = Vibration amplitude at the propeller, radians
Qp = Alternating propeller torque, in per cent of mean propeller torque (5 to 7.5)
B = Number of propeller blades

The alternating torsional stress in the propulsion shafting, due to propeller excitation, is:

$$S_s = \frac{GD}{2\ell}\,\phi_p\left[\frac{I_p + I_e}{I_e}\right], \text{ psi}$$

where:

G = Torsional modulus, psi (11.8×10^6 for steel)
D = Diameter of shaft, in.
ℓ = Length of shaft, in.
ϕ_p = Vibration amplitude at propeller, radians
I_e = Mass moment of inertia of the engine, lb in. sec^2
I_p = Mass moment of inertia of the propeller, lb in. sec^2 (includes 25% for entrained water)

The derivation of this expression is given in Ref. 10.

Another possible source of difficulty is the coupling of torsional and longitudinal modes of vibration. Since the alternating torque produces a longitudinal vector due to the propeller oscillation, a component of thrust, at the frequency of the torsional critical, will produce an alternating force at the thrust bearing. A case has been encountered in towboat design, in which these forces at the bearing, augmented by a longitudinal resonance of the thrust bearing foundation, resulted in repetitive bearing failure in addition to producing serious hull vibration.

Group IV, Longitudinal Vibration - The nonuniform propeller forces also result in an alternating thrust load. This alternating load produces a longitudinal vibration which enters the hull at the point of attachment of the thrust bearing. If, then, the mass elastic system of the propulsion drive is tuned to the frequency of the exciting force, a condition of resonance occurs which can prove troublesome or even damaging. If resonance occurs, the only limitation to troublesome vibration of this type would be the inherent damping in the system.

The manifestation of serious hull and machinery vibration in the longitudinal direction was quite serious in some naval applications where long shafts were used. The results of studies by the Navy and the shipbuilders involved were documented in the paper by Kane and McGoldrick (12). This paper is still recommended as a basic reference on the subject of longitudinal vibration.

In applying the present understanding of the problem of longitudinal vibration to the case of the towboat, it is recommended that we strive to design the fundamental natural frequency of the drive system, including the thrust foundation flexibility, 15-20% above the maximum propeller blade frequency, and, as pointed out earlier, insure that no major orders of torsional vibration are present, which could result in serious engine excited longitudinal vibration.

A typical system, including propeller, shaft, thrust bearing, and a longitudinally flexible coupling between the main thrust and engine, may be represented diagrammatically as shown in Fig. 12. The natural frequency of this system may be estimated by the following expression:

$$F = \frac{A_f}{2\pi}\sqrt{\frac{K}{M_p + \frac{M}{3}}} \text{ cps}$$

where:

A_f = Factor for flexibility of foundation. This must be determined by test.
K = Longitudinal stiffness of shaft = EA/L, lb/in.
M = Mass of shaft, lb sec^2/in.

M_p = Mass of propeller + 60% for entrained water, lb sec^2/in.

If we were to assume the thrust is located in the engine, the system would be as shown in Fig. 13.

The natural frequencies of this system may be estimated from the roots of the equation:

$$W_n^4 - W_n^2\left(\frac{K_1}{M_1} + \frac{K_1 + K_2}{M_2}\right) + \frac{K_1 K_2}{M_1 M_2} = 0$$

where:

W_n = Natural frequency, rad/sec

K_1 = Shaft stiffness = EA/L, lb/in.

K_2 = Stiffness of engine frame and foundation, lb/in. (This is an empirical value)

M_1 = Propeller mass plus 60% for entrained water, plus 1/2 mass of the shaft, lb sec^2/in.

M_2 = Mass of engine and foundation, plus 1/2 mass of shaft, lb sec^2/in.

For other alternate arrangements, approximations must be made for the development of an equivalent mass-elastic system which lends itself to convenient analysis. Additional suggestions are given in Ref. 12.

The largest unknown in these analyses are, of course, the estimation for foundation stiffness. These values can only be obtained experimentally or by very rough calculation. Since we are interested in a high natural frequency, we should, as a rule, make the thrust foundation as rigid as possible in the longitudinal direction. This is best accomplished by spreading the foundation fore and aft, as far as possible.

Group V, Equipment Vibration - The equipment considered under Group V may be generally referred to as auxiliary equipment and is differentiated from main machinery primarily because of the size of the equipment. Ordinarily the exciting forces, although of a lower order than those produced by main engines, propellers, and shafts, have higher driving frequencies and thus introduce significant energy into the ship structure. These machines do not normally cause difficulties by exciting a discernible resonance of the hull girder (ordinarily no more than five may be noted) but rather by exciting resonances in the local structure. To minimize the adverse effects of equipment vibration, care should be exercised in the selection of equipment for minimum exciting force and in the design of foundations and local structure to avoid resonance with the known exciting forces in the equipment.

The same forces which are inherent in a machine may also set up structural resonances in the machine frame and create a serious disturbance in the form of airborne or structureborne noise (vibration). The following sections will treat this problem in greater detail and discuss the source of noise, noise paths, and noise control.

NOISE CONSIDERATIONS

The problem of noise reduction from the standpoint of habitability has been given serious attention in recent years, not only by the Navy but also the shipping industry. It is evident that a ship, which contains a multitude of noise producing machinery and where the operating personnel are confined to its boundaries for long periods of time, must have in its design certain noise control features. The control of noise is necessary for reasons such as:

1. Loss of hearing.
2. Intelligibility of speech communication.
3. Reduced alertness and undue fatigue.
4. Increased comfort.

As far as the habitability or physiological effects of noise is concerned, only the airborne component of noise need be considered. The structureborne component is of importance when dealing primarily with the waterborne sound radiated by various machinery and therefore will not be discussed to any extent except as it relates to airborne noise.

In discussing the subject of shipboard noise the following areas will be considered: sources, paths, and control.

SOURCES OF NOISE - The noise produced by a machine is generally a function of the geometry of its parts. Other

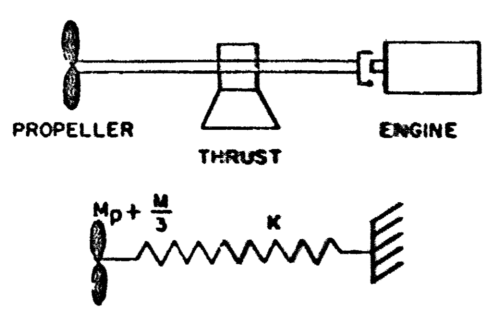

Fig. 12 - Typical drive system

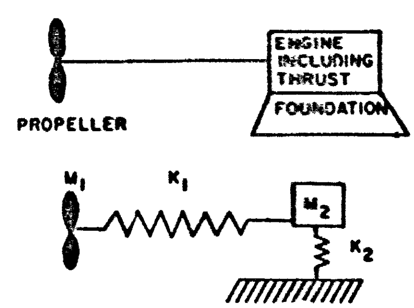

Fig. 13 - System with thrust located in engine

factors such as internal forces and individual component resonances also may contribute to the overall noise output of a machine. There are also noise sources that are common to many machines such as bearings and shaft unbalance. Among the more common types of machines that will be discussed are diesel engines, gears, pumps, fans, and electric motors.

Bearings - There are two basic types of bearings commonly used in machines, that is, rolling contact (antifriction) and sliding surface bearings. Of the two types, the sliding surface bearing is generally quieter in that it does not contain characteristic features that will generate specific or discrete frequencies except for the "stick-slip" phenomena resulting from inadequate lubrication.

Rolling contact bearings, on the other hand, are capable of producing several distinct frequencies as shown in Table 2 from Ref. 13.

Except in especially quiet machines such as motors, gyros, and the like, bearings are not the controlling source of noise unless they are damaged or their exciting frequency coincide with one or more natural frequencies of the machine or its foundation. Needless to say, damaged bearings, whether they are of the sliding surface or the rolling contact type will produce noise and, if not repaired or replaced, would cause the machine they are supporting to become inoperative.

Unbalance - Noise caused by shaft unbalance occurs at the rotational speed and associated harmonics. Generally speaking, balance limits that are specified for proper mechanical operation result in low noise levels as far as airborne noise is concerned. There have been cases, however, where the forces due to shaft unbalance have set into resonance some other connecting part such as a frame, panel, or foundation causing objectionable noise levels.

Gears - Gears, by the nature of their operation, are well known sources of noise. A gear performs its work by transmitting load and speed from one tooth on the driving gear to another tooth on the driven gear. Unless this work is accomplished at perfectly uniform velocities, such as the operation of worm type gears, impact forces will result with consequent noise. The frequencies manifested are those corresponding to tooth mesh frequencies, (number of teeth of a gear times the rotational speed). Other predominant gear frequencies are those corresponding to the rotational speeds of the individual gear elements. Here again, any natural frequencies of the gear or casing that correspond to tooth mesh or rotational speeds of the individual gears will be a source of noise.

Besides the above typical gear noises, there also exists a source of noise peculiar to helical gears known as hobbing frequency noise which is a result of undulations along a gear tooth produced by the hobbing machine (14). This noise is produced at a frequency equal to the product of the rotational speed of the gear and the number of teeth of the hobbing machine worm wheel. Under normal circumstances the level of the hobbing frequency is usually well below that of the mesh frequencies.

Diesel Engines - The diesel engine, together with its principal moving parts and accessories, contains numerous sources of noise. The individual rotating and reciprocating components as well as the combustion explosions give rise to many frequencies, all of which combine to form the overall noise output.

In an attempt to understand the complex noise output of diesel engines the Bureau of Ships undertook a program to determine the basic sources of noise of a typical diesel engine (15).

Some characteristics of the engine tested are: Number of cylinders - 8; Type - 2 stroke cycle; Bhp - 425 at 1200 rpm.

The engine was fitted with a scavenging air blower and other accessories such as fresh water and sea water pumps and lube oil pumps. The engine was connected to a generator and both mounted on a concrete slab. The slab weighed approximately 23 tons and was isolated from a concrete floor by cork pads. Results of the test showed the following sources of airborne noise, in decreasing order of importance, to be:

1. Scavenging air blower.
2. Piston and connecting rods.
3. Exhaust valve mechanism.
4. Fuel injectors.
5. Fuel combustion.
6. Timing gears.

A representation of the acoustical power contribution of the various engine elements are shown on Fig. 14.

Table 2 - Bearing Frequencies

Origin	Frequency, cps
Unbalance or Eccentricity	N
Irregularity of Housing	$\dfrac{NR_1}{R_1 + R_2}$
Irregularity of Rolling Element	$\dfrac{NR_1R_2}{R_3\,(R_1 + R_2)}$
Irregularity of Inner Raceway	$\dfrac{nNR_2}{R_1 + R_2}$
Irregularity of Outer Raceway	$\dfrac{nNR_1}{R_1 + R_2}$

where:

N = Shaft rpm divided by 60
n = Number of balls or rollers
R_1 = Radius of inner raceway
R_2 = Radius of outer raceway
R_3 = Radius of ball or roller

The scavenging air blower radiated direct airborne noise. The engine proper radiated the airborne noise through structureborne vibrations excited by the moving parts of the engine.

Although these results represent the findings from only one engine it is considered that it generally indicates the basic noise sources of all diesel engines.

In addition to the above findings, a large number of natural frequencies were discovered on the engine. A total of 31 natural frequencies ranging from 180 - 3400 cps were excited by "bump" tests.

The noise level of any engine element would be amplified if its frequency of excitation corresponds to any of these natural frequencies.

Pumps - Pump noises are usually of concern in the structureborne or fluidborne sense rather than airborne since most of the acoustic energy developed by a pump is contained within the attached piping system. As such, the airborne noise contribution of pumps relative to other machinery is low.

It should also be pointed out that the noise produced by pumps depends on two factors. One is the type of pump and the other is the system to which the pump is connected. That is, the noise characteristics of a pump, considered as a component, will vary depending on the attached system.

Fans - Fan noise, similar to pump noise, depends both on the fan itself as well as the attached connections such as ducts. However, unlike pumps which operate in closed piping systems, fans radiate noise directly to open airborne paths. Since the principal use of fans aboard ship are in the ventilating and air conditioning systems their acoustical contribution regarding habitability must be seriously considered.

The aerodynamic noise produced by fans consists of a rotational component and a vortex component. The rotational component exhibits frequencies which correspond to the number of blades times the rotational speed and associated harmonics. The vortex component results from flow separation causing eddy currents to form in a random fashion and, as such, no particular frequencies are manifested.

Electrical Machinery - The major noise sources of motors and generators, other than that due to bearings and unbalance, is magnetic noise, brush noise, and airflow noise. Magnetic noise is a result of eccentricity between the rotor and stator causing a fluctuation in the air gap. This sets up periodic radial forces on both the rotor and stator causing them to vibrate at frequencies which are a function of the number of poles and slots and the operating speed. Brush noises are mainly due to poor mating of the brushes and commutator bars. There is no particular frequency associated with this noise but it is usually of a high frequency nature. Brush assemblies are also a source of noise if their natural frequencies coincide with the number of commutator bars times the rotational speed.

Airflow noises are caused by either the rotor bars chopping the airstream or the ventilating fan blades passing areas of obstructed airflow. These airflow noises may be further amplified if their frequency of excitation corresponds to the resonance of the air cavity confined within the motor casing.

NOISE PATHS - Noise, depending on the path it takes, may be classified as either airborne, structureborne, or waterborne. Airborne noise, as detected by the human ear or microphone, is the measure of pressure fluctuations in the air path caused by a vibrating object. The vibrating object may be a loudspeaker, fan blade, bulkhead panel, machine foundation, or the like.

Structureborne noise is a measure of vibrations which are transmitted through solids such as machinery foundations, hull, and deck plating and is ultimately radiated to the air or water.

Waterborne noise is the direct result of pressure fluctuations set up in the water by structureborne vibrations. To a lesser degree airborne noise may also contribute to the overall waterborne noise level. Fig. 15 illustrates the various transmission paths noise may follow.

NOISE CONTROL -

Objectives - As previously stated, the control of noise is necessary for habitability and physiological reasons. The extent to which noise is to be controlled is a function of the various requirements that are specified to insure certain desired conditions.

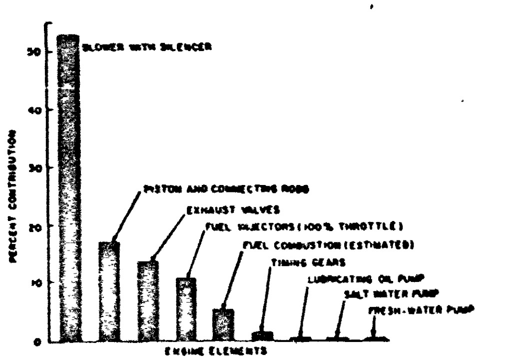

Fig. 14 - Acoustical power distribution of various engine elements on percentage basis

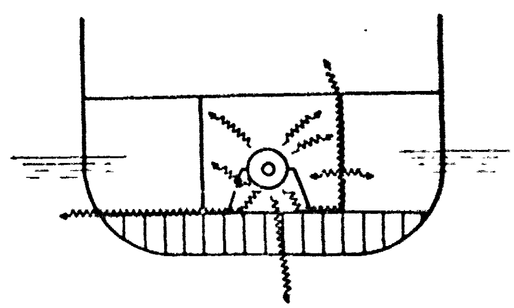

Fig. 15 - Transmission paths

Specifications regarding noise on Naval ships define five noise categories and specifies airborne noise criteria for each. The five categories are as follows:

Category A - 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 within category D spaces where intelligible speech communication is necessary.

The speech interference level (SIL) is a measure of the background noise on intelligible speech communication. Numerically it is the arithmetical average of the sound pressure level, in decibels, in the octave bands: (300-600), (600-1200), (1200-2400), and (2400-4800) cps. For noise categories A and E Table 3 shows the SIL values that are not to be exceeded.

The noise levels for categories B, C, and D, cover the entire octave range and are shown in Table 4.

Table 3 - Speech Interference Levels

Noise Category	Volume, cu ft	Speech Interference Level
A	500-1999	60
	2000-7999	55
	8000 and larger	50
E	Any	72

The category assigned to a particular space depends on the objectionable noise effects that will be encountered. For example, in an engine room comfort and speech intelligibility are usually disregarded and deafness avoidance criteria will prevail.

Methods - Noise control must be considered in a system sense since there are three aspects involved: the source, the path, and the receiver. Assuming that nothing can be done with the receiver leaves only the sources and path. Experience indicates that to achieve significant reductions of noise at the source generally involves significant time and cost expenditures or, in other words, the "db per dollar" costs may be prohibitive. The path then remains as the most practical portion of the system that can be approached for noise reduction. Noise reduction in the path may be accomplished by one or more of the following methods.

1. Isolation.
2. Damping.
3. Absorption.

Isolation - Consider a machine solidly connected to a ship's hull as shown on Fig. 15. As a result, any exciting frequencies will be transmitted to the surrounding structure which can act as a sounding board and possibly further amplify, due to local resonances, some of the frequencies. Any complex structure such as a ship's hull and connecting structure theoretically possesses an infinite number of natural frequencies which can be excited if the proper exciting frequencies exist. An ideal isolation system would attenuate all of the exciting frequencies and minimize their effects on the surrounding structure; however, from a practical standpoint the ideal system rarely exists.

The most common method used to isolate a machine from the surrounding structure is to support it by resilient mounts. These mounts may take the form of steel springs, rubber, or other resilient materials whose stiffness characteristics can be controlled to obtain a desired mass-elastic system.

Table 4 - Noise Level Requirements

Noise Category	Space Volume, cu ft	Octave Band, cps 37.5-75	75-150	150-300	300-600	600-1200	1200-2400	2400-4800	4800-9600
B Comfort	500-1999	86	82	78	74	70	66	62	58
	2000-7999	82	78	74	70	66	62	58	54
	8000 and larger	78	74	70	66	62	58	54	50
C Quiet	500-1999	80	76	72	68	64	60	56	52
	2000-7999	76	72	68	64	60	56	52	48
	8000 and larger	72	68	64	60	56	52	48	44
D Injury	Any	110	105	100	90	90	85	85	85

An isolated machine represented as a single-degree of freedom system is shown in Fig. 16 where M is the mass of the machine (lb-sec^2/in.), K is the stiffness of the spring or isolator (lb/in), and the foundation is assumed rigid. The mass-elastic characteristics of the system will define the natural frequency as:

$$F_n = \frac{1}{2\pi}\sqrt{\frac{K}{M}}$$

The effectiveness of the system, regarding isolation, is expressed as transmissibility which is the ratio of the force transmitted to the foundation to the exciting force of the machine.

For an undamped system the transmissibility (T) may be calculated by: $T = 1/1-(F/F_n)^2$ where (F/F_n) is the ratio of the exciting frequency of the machine to the natural frequency of the system. Fig. 17 is a plot of the transmissibility curve for an undamped single degree of freedom system. At the point where $(F/F_n) = 1$, a resonant condition exists and for an undamped system the transmissibility is infinite.

Isolation of the forcing frequency is effected at values of (F/F_n) greater than $\sqrt{2}$. In practice, the ratio of the lowest exciting frequency to the natural frequency is generally kept above 3.

The following example illustrates some of the steps to be taken for isolating a machine on a single-degree of freedom basis:

Unit to be resiliently mounted - Diesel Generator Set; rpm - 1200; weight - 40,000 lb; No. of cylinders - 8 (2 cycle).

1. At a constant speed operation of 1200 rpm the lowest exciting frequency resulting from unbalance is 1200/60 or 20 cps.

2. Assume a minimum value of $F/F_n = 3$. This results in a desired natural frequency of 20/3 or about 6.7 cps.

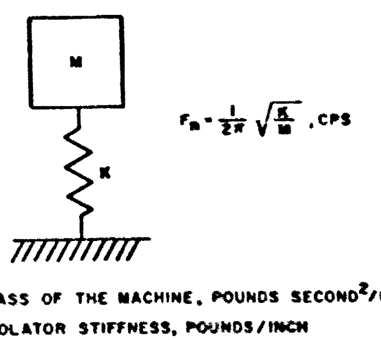

Fig. 16 - Isolated mass

3. Using the formula, $Fn = \frac{1}{2\pi}\sqrt{\frac{K}{M}}$ solve for K (isolator stiffness).

$$K = (2\pi \times 6.7)^2 \left(\frac{40000}{386}\right) = 183{,}580 \text{ lb/in}$$

This stiffness represents the total isolator stiffness. If (n) isolators are to be used, then each isolator will have a K of (183,580/n).

From Fig. 17 at $F/F_n = 3$, it can be seen that the unbalanced force transmitted to the foundation is approximately 12% of the force existing above the isolators. At the firing frequency (20 X 8 = 160 cps) the value of F/F_n is approximately 23 which results in less than 1% of transmitted force.

Although damping has been neglected here, in an actual system some finite amount of damping does exist. The effects of damping result in a reduction of the transmitted force at resonance but at higher frequencies it increases the transmitted force over that of an undamped system. Therefore, a large amount of damping is not always considered desirable.

It should be noted that the value of K is the 'dynamic' stiffness at any particular loading of the isolator. For isolators that possess linear load-deflection characteristics the dynamic stiffness value is equal to the static value. However, for certain isolators such as the more commonly used rubber types, this is not true. Experience indicates that rubber isolators have dynamic stiffness characteristics approximately 2-3 times higher than the static values obtained by load-deflection curves. This variance depends on the rubber characteristics and configuration of the isolator. In practice, however, the designer need only concern himself with the desired natural frequency since the manufacturers of the various isolators catalogue them as a function of load carrying capacity and natural frequency.

The above example dealt only with an idealized single-degree of freedom system. In a practical system there are actually six degrees of freedom, three rotational and three translational, which are basically a function of the location of the center of gravity and the three principal moments of inertia.

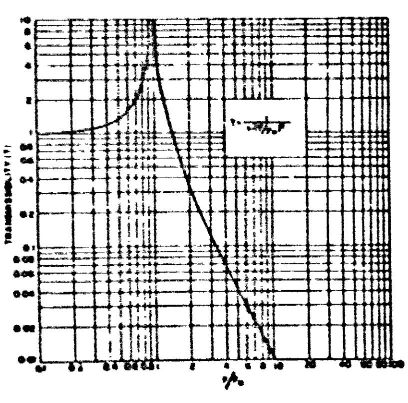

Fig. 17 - Plot of transmissibility curve for undamped single degree of freedom system

Fig. 18 illustrates the six different modes of vibration of a mounted system. Three modes are translational in the direction of the X, Y, and Z axes and the other three are rotational about the axes.

To assure a satisfactory mounting system, it is good practice to calculate all six natural frequencies. Ref. 16 contains a complete discussion on this subject.

The excitation of a mounted system may be not only that of the isolated machine but also from hull excited frequencies. Thus, it is important that the mounted system natural frequencies do not coincide with any steady state excitations either from the mounted machine or the hull.

When a machine is isolated, considerations must also be made pertaining to the movement of the machine during conditions of shock. This shock may be a result of the ship striking a pier during berthing, running aground, or any other situation that will cause an abrupt loading change of the isolated system. As a result, sufficient clearance must be provided around the isolated machine to prevent it from striking nearby objects. For equipment supported by isolators in the same horizontal plane the clearance required may be estimated by the following formula:

$$C = \frac{2\,DH}{W} + E$$

where:

- C = Motion at point in question
- D = Maximum deflection of the isolators in the vertical direction
- E = Maximum deflection of the isolators in the horizontal direction
- H = Perpendicular distance from plane of isolators to point in question
- W = Distance between centers of most widely spaced isolators in the direction being considered

All external connections to the isolated machine must be capable of absorbing the motions as determined above. These connections may be electrical cable, piping, input or output shafting, ductwork, and similar items. The connections must not only be capable of absorbing motion but must also be flexible enough and properly installed so as not to alter the isolator characteristics. In the case of flexible pipe connections Fig. 19 illustrates some examples of good and bad installations with respect to the service life of the connection.

Damping - As previously mentioned, a ship's hull and associated structure theoretically contains an infinite number of natural frequencies. Any number of these frequencies may be excited by the various machinery, either by structure-borne or airborne noise, resulting in amplification of the exciting frequencies. The most effective method of minimizing resonant effects is to provide damping treatment which dissipates some of the vibratory energy. Most damping materials dissipate this energy by converting it to heat as a result of internal friction. Therefore, any materials that have high internal losses are suitable for damping.

Damping treatments should be applied as close as possible to the noise source. This includes machinery casings, foundations, and portions of the connecting hull structure.

One particular damping treatment extensively used by the Navy consists of zinc-chromate impregnated wool felt (17), which is actually a gasket material, in combination with a steel septum loading plate. Fig. 20 shows a typical installation. The septum plate is approximately 25% as thick

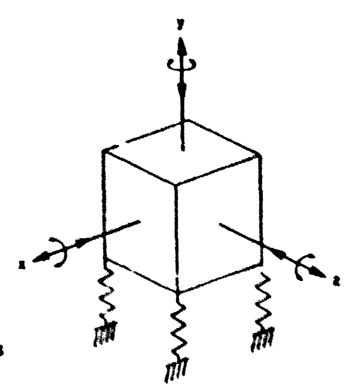

Fig. 18 - Six degrees of freedom

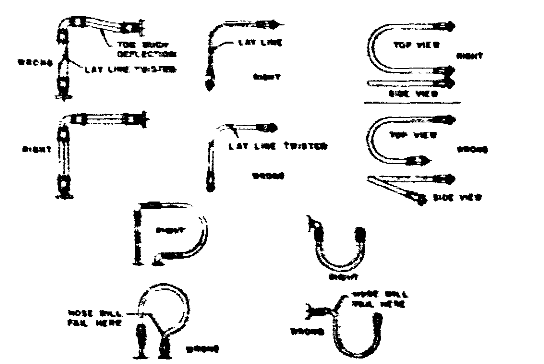

Fig. 19 - Flexible hose connections

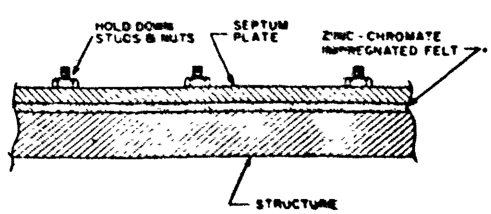

Fig. 20 - Felt-septum damping treatment

as the damped structure. The thickness of the felt sheet may range from 1/16 to 1/4 in. depending on the thickness of the plate or structure to be damped. The septum is secured by hold-down nuts fitted to studs which are welded to the damped plate on approximately 12 in. centers. The constraining pressure obtained by torquing of the hold-down nuts is usually about 40 psi. The following formula may be used to obtain the constraining pressure (18):

$$P = \frac{T}{0.2 \times D \times A}$$

where:

P = Constraining pressure, psi
T = Applied torque, in.-lb
D = Diameter of stud, in.
A = Area of constraining layer per studs, in.^2

(For studs spaced on 12 in. centers, A = 144 in.^2)

Damping values that can be expected by use of chromated felt range from 2-5% of critical damping in a frequency range of 50-2000 cps.

Another type of damping material suitable for damping higher frequencies (2000-11,000 cps) is a sand or aluminum oxide filled plastic tile (19). The material is in the form of 1 ft square tile and is about 1/2 in. thick. It is applied by cementing it to the structure. The damping characteristics of this material range from 3-5.5% of critical damping.

Absorption - The isolation and damping methods previously discussed serve to reduce airborne noise only by minimizing the contributing effects of structureborne noise. Any airborne noise that is not attenuated by these methods must be reduced by absorption.

Consider a noise source in a compartment which is made up of hard surfaced bulkheads, overhead, and deck. The sound pressure waves radiating toward the hard surfaces will be reflected back and forth unitl a reverberant noise level is greater than that which the noise source would produce in a free field.

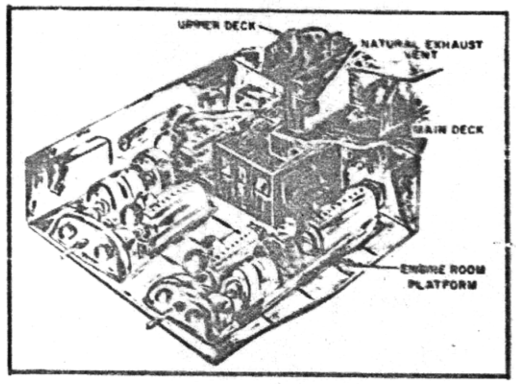

Fig. 22 - Minesweeper main propulsion engine room with enclosed control room

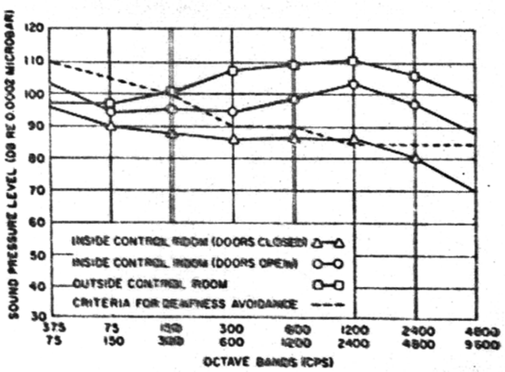

Fig. 23 - Minesweeper engine room airborne noise levels

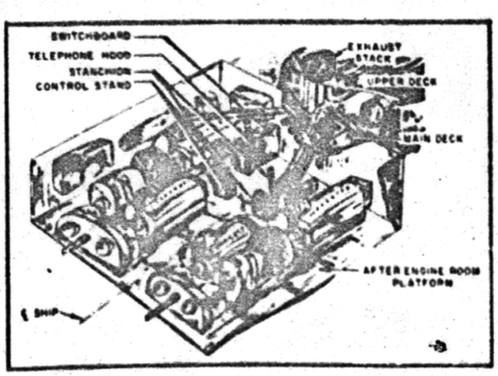

Fig. 21 - Minesweeper main propulsion engine room as originally designed

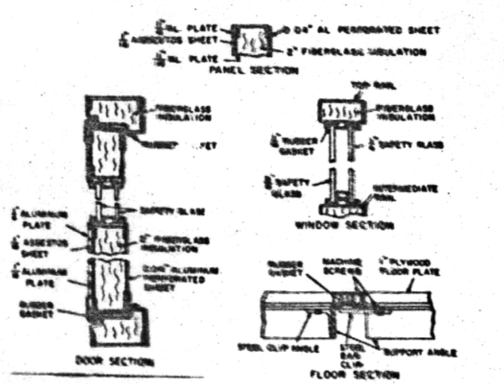

Fig. 24 - Construction details of control room

To attenuate such noise one or more of the following measures must be taken:

1. Acoustically treat the compartment with sound absorbing material.
2. Provide an enclosure around the personnel.
3. Provide an enclosure around the machine.

Acoustical Treatment of Compartment - Acoustically treating the surfaces of the room with sound absorbing material at best would only reduce the noise level to that which would exist in a free field. This is assuming 100% absorption of sound by the acoustical treatment to eliminate any of the reverberant effects but the direct path of airborne noise from the source will still exist. Depending on the compartment involved this type of treatment may be sufficient. However, take the case of a diesel engine room which contains extremely high noise levels. Acoustical treatment of such a compartment would probably, in itself, not provide sufficient noise reduction and would also be impractical and uneconomical. In this case, the only solution would be to either enclose the operating personnel in an acoustically treated operating booth to provide an enclosure around the diesel engines.

Enclosures Around Personnel - Several years ago the Navy was faced with a problem of excessive noise in the engine rooms of a new class of minesweepers. The noise levels that existed in the engine room spaces were found to exceed considerably the deafness avoidance criteria. The solution to the problem was approached by designing and installing an enclosed control room in each of the two engine rooms (20). Fig. 21 and 22 illustrate the main propulsion engine room arrangement before and after the installation of the enclosed control room. The auxiliary machinery space was also fitted with a similar control room.

Fig. 23 shows the octave band noise levels measured in the main propulsion engine room. Outside the control room the noise levels measured exceeded by a considerable amount the deafness avoidance criteria in the upper five octave bands. The noise levels inside the control room demonstrated that a significant reduction took place in all eight octave bands and that the deafness avoidance criteria was adequately met. The construction details of the control room are shown in Fig. 24. It should be noted that the control room is not solidly connected to the hull framing, instead it is resiliently supported by strips of rubber. This feature minimizes structureborne vibrations and associated airborne noise from entering the control room.

Enclosures Around Noise Source - There are circumstances when enclosures for personnel are not possible, in which case an enclosure around the noise source should be considered. A particular investigation to determine the effectiveness of such an installation was conducted by the Navy on a diesel-generator set installed in the machinery space of an 80 ft aluminum PT boat. The diesel generator set consisted of a 4 cyl 2 cycle diesel directly connected to a 60 kw a-c generator operating at a governed speed of 1800 rpm. The noise reduction measures taken were isolation of the diesel generator for structureborne noise attentuation and an acoustical hood around the entire unit. Fig. 25 schematically illustrates the installation. Measurements of the airborne noise showing the effectiveness of the installation are shown on Fig. 26. Without the enclosure the airborne noise levels exceeded the deafness avoidance criteria in four of the eight octave bands. With the engine enclosure in place,

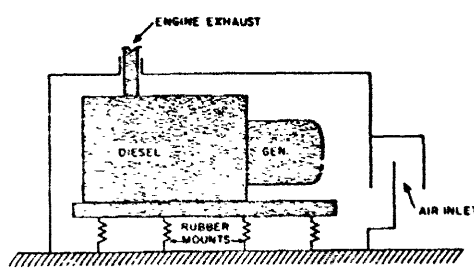

Fig. 25 - Schematic of enclosed generator set

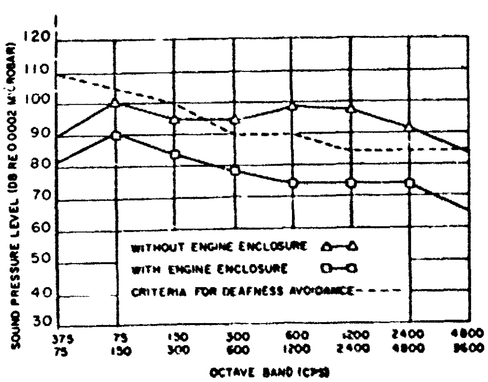

Fig. 26 - PT boat machinery space airborne noise levels

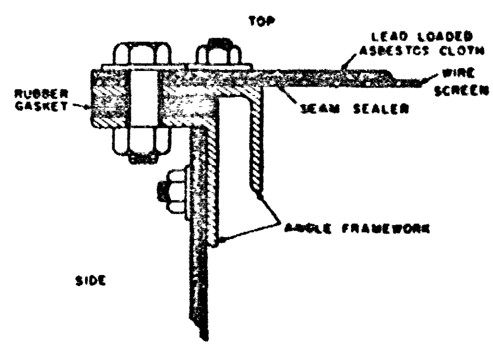

Fig. 27 - Typical corner section of engine enclosure

the airborne noise levels were markedly reduced across the octave band range and were well below the deafness avoidance criteria.

The construction of the enclosure basically consisted of lead-loaded asbestos cloth supported by wire screen and an angle iron framework. Fig. 27 shows a typical section of the enclosure.

Although the above represents the findings of a single investigation it should be noted that other similar studies on various installations were also performed that gave equally good results.

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The paper is addressed primarily to the naval architect, shipbuilder, or operator and identifies those areas to which attention should be drawn during design and development. Specific suggestions and specifications are presented, based on current naval practice.

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2. Tugs--Vibration
3. Tugs--Noise characteristics

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