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LONGITUDINAL VIBRATION OF PROPULSION SYSTEM ON USS CANOPUS (AS-34)

by

Gary P. Antonides and Paul M. Honke

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ABSTRACT

As part of an effort to increase the adequacy of prediction methods used in shaft design, a study was made of the longitudinal vibration characteristics of the propulsion system on the USS CANOPUS (AS-34). The lever effect, previously detected on the USS SIMON LAKE (AS-33) was confirmed. This means that the amplitudes of the gear case, turbines, and condenser are roughly dependent on the relative heights of these components to the level of the shaft. The effective foundation mass, foundation stiffness, and thrust-bearing stiffness are computed from the measured response of the system. A resonance was found to exist in the operating range, but it was not considered detrimental. The exciting forces are lower than usual, except at or near full power.

ADMINISTRATIVE INFORMATION

The work described in this report was authorized by Bureau of Ships letter Serial 436-346 of 3 November 1965, and was funded under P.O. 60041/SP.

INTRODUCTION

The irregular wake created by the hull causes the ship propeller to generate alternating thrust. This in turn produces longitudinal vibration of the propeller shaft and propulsion system. The predominant excitation occurs at blade frequency. When the critical speed falls within the range of operating speed of the ship, the amplitudes are often magnified by the dynamics of the system. Occasionally the alternating thrust is large enough to cause thrust reversal and pounding of the thrust bearing, excessive wear on gears and couplings, or undesirable vibration of pipes or other parts of the propulsion machinery. Unfortunately, with the present procedures for predicting the vibratory behavior, it is difficult, if not impossible, to predict these conditions in the design stage.

As part of an effort to improve these prediction techniques, full-scale trials were conducted on the Polaris submarine tenders USS SIMON LAKE (AS-33) and USS CANOPUS (AS-34). A report[1] on SIMON LAKE has already been

[1]References are listed on page 35.

issued. The objectives of the trial on the CANOPUS (same class of ship) were (1) to confirm longitudinal vibration characteristics of this class of ship; (2) to study how the gear case, turbines, condenser, and machinery foundation affect longitudinal vibration; (3) to determine the exciting forces and damping associated with longitudinal vibration; and (4) to determine if a coupling exists between longitudinal and torsional shaft vibration.

SHIP CHARACTERISTICS

The ship characteristics are given in Table 1, and the propeller arrangement is shown in Figure 1.

TABLE 1

Ship Characteristics

Length:	
Overall	638 feet
Between perpendiculars (LWL) .	620 feet
Breadth	85 feet
Depth (to main deck)	57 feet
Draft (DWL)	24 feet
Normal displacement	22,000 tons
Maximum propeller speed	150 rpm
Maximum shaft horsepower	22,500 hp
Trial conditions:	
Draft:	
Forward	18.3 feet
Aft	21.6 feet
Displacement	17,257 tons

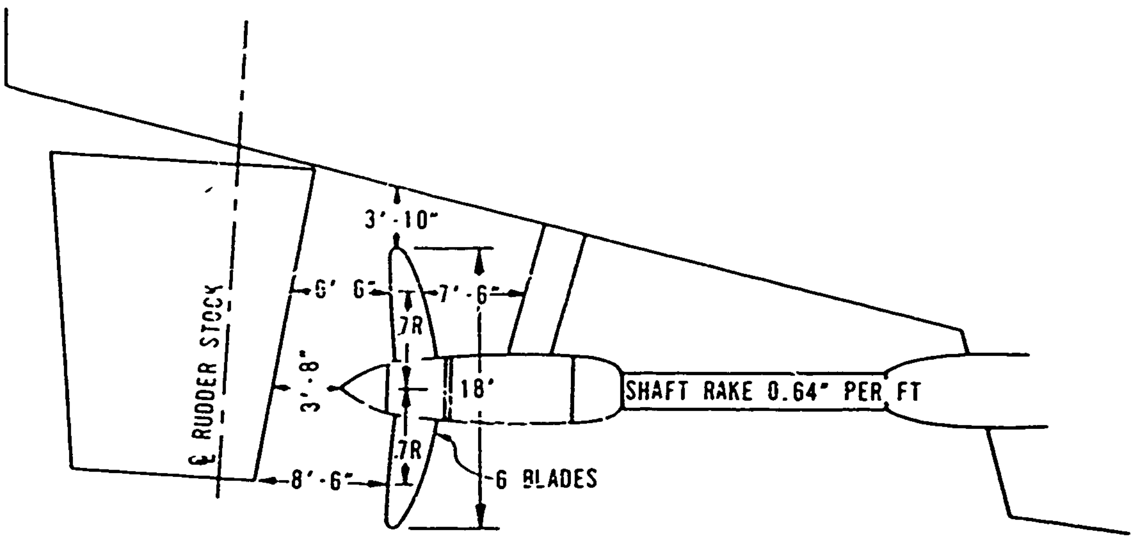


Figure 1 - Propeller Arrangement of CANOPUS

FULL SCALE TRIALS

The instruments used in recording are:

7 CEC-Type 4-102A vibratory velocity gages.

8 Kulite-Bytrex Corp., semiconductor strain gages.

1 AVL/DTMB rpm indicator.

1 Ampex FR-1300 tape recorder.

1 Monitoring oscilloscope.

7 CEC-System-D linear/integrating amplifiers.

3 Shaft-strain telemetering systems.

1 TMB calibration source and switch box.

Figure 2 indicates the locations of the velocity gages and the strain gages. Figure 3(a) shows the orientation of the strain gages for both thrust and torque measurement; Figure 3(b) shows the wiring arrangement of a typical strain gage bridge; and Figure 3(c) shows the general wiring arrangement of all instrumentation.

Measurements were taken at all gage locations while the ship was performing the following maneuvers: acceleration over operating range; left and right full rudder turns at full power; crashback; and steady speeds of 50, 60, 70, 80, 90, 100, 105, 110, 115, 120, 125, 130, 135, 140, 145, and 150 rpm. Before the steady speed runs the velocity gage on the condenser was accidentally knocked loose. This condition was discovered and corrected before the 130 rpm run.

All trials were conducted in water which had a depth greater than six times the draft and which was calm throughout the trials.

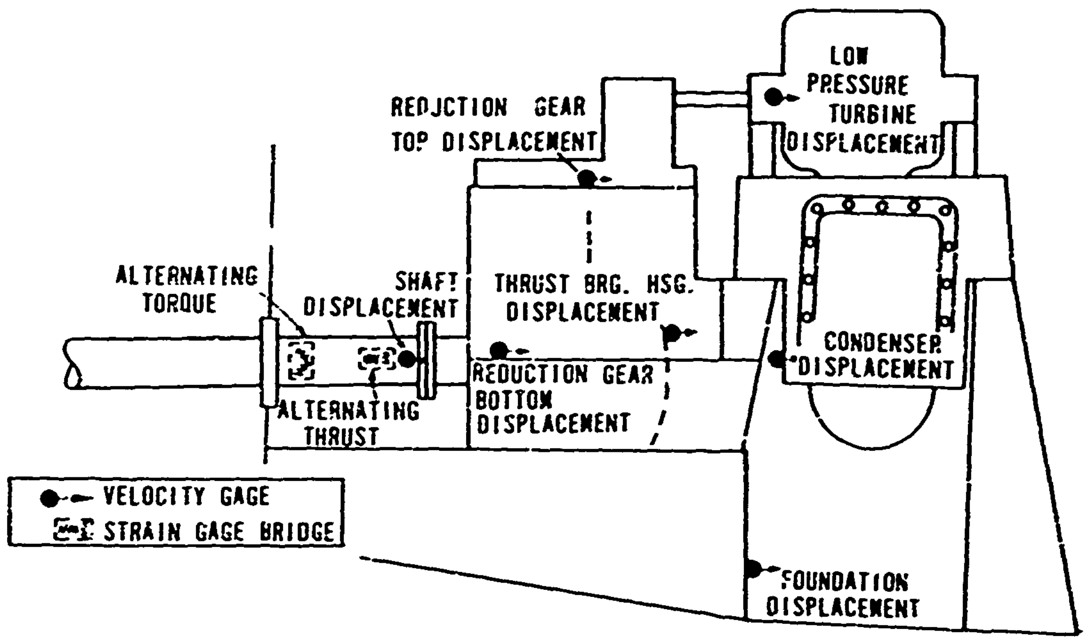


Figure 2 - Main Propulsion Plant of CANOPUS Showing Location and Orientation of Gages

TEST RESULTS

Measurements from all stations at all speeds were analyzed by filtering the signal electronically to isolate blade frequency. The signals were electronically averaged so that they could be used to find the shape of amplitude versus rpm curves. Maximum values of blade frequency were also analyzed. It is possible that sharp increases in amplitudes--due to waves, ship motion, or some other factor--will give a false indication of the resonant frequency or damping. However, the maximum values must be considered when designing a system for vibratory forces.

Figures 4 through 12 show the plots of average and peak blade-frequency alternating thrust, torque, and displacements with respect to rpm for the steady speed runs. All blade-frequency vibration was in phase throughout the speed range. The plot of shaft displacement shown in

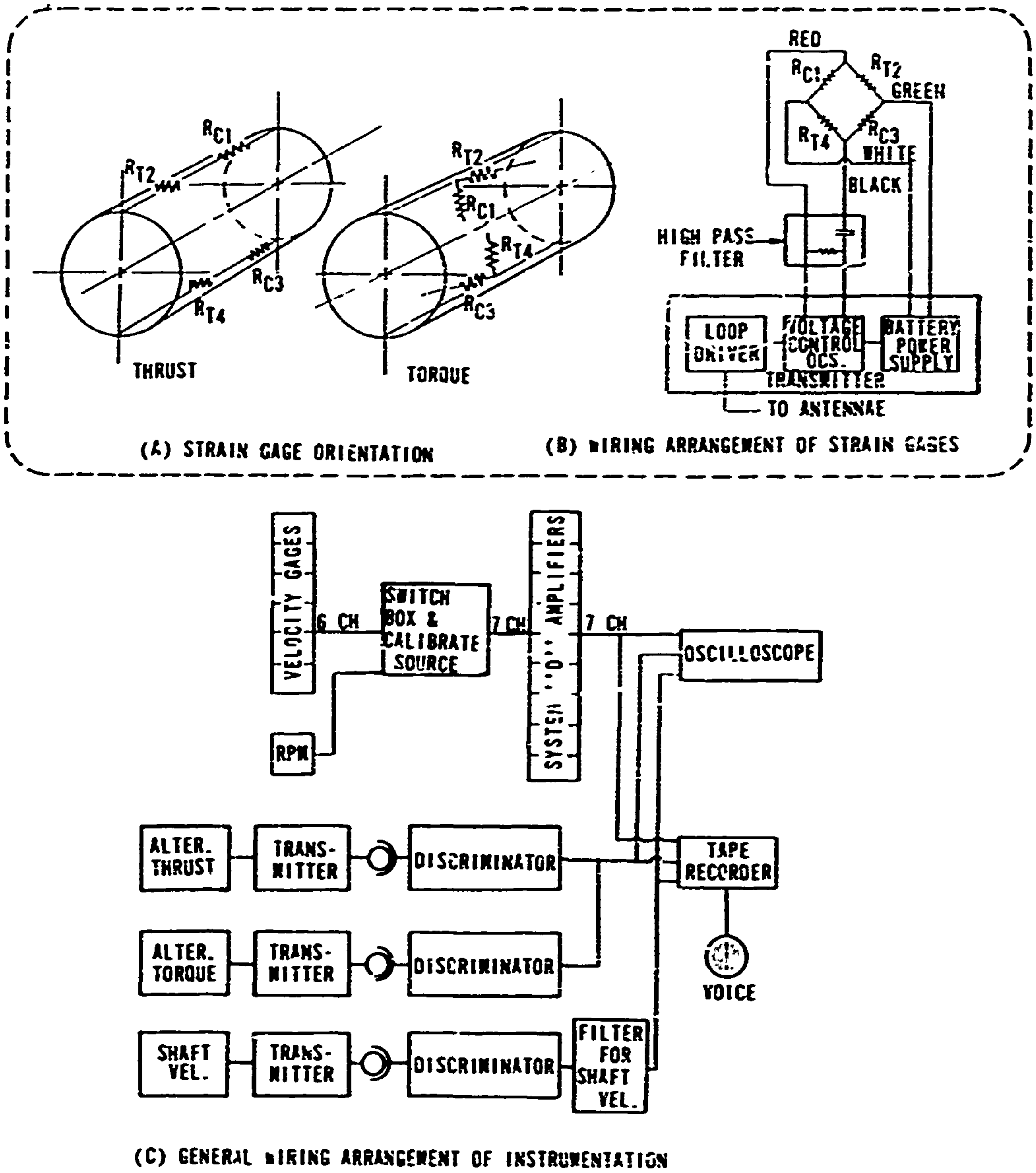


Figure 3 - Schematic of Instrumentation Aboard CANOPUS

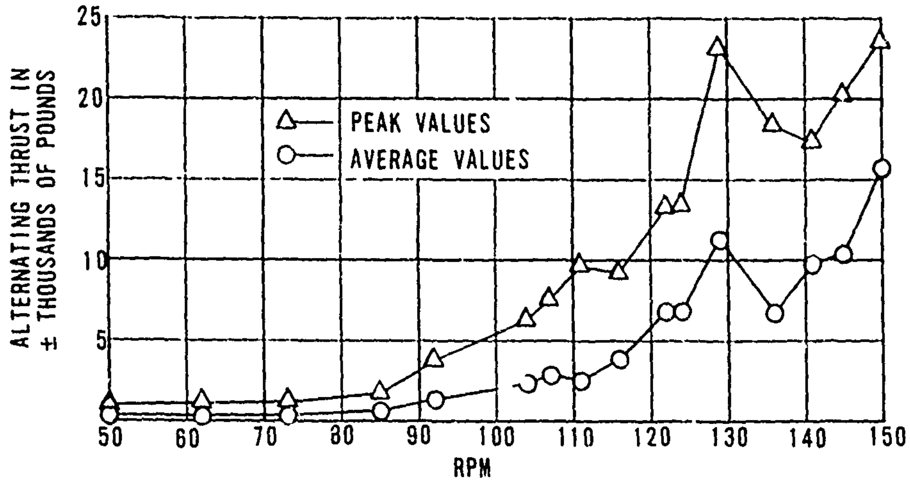


Figure 4 - Blade-Frequency Alternating Thrust in Shaft versus RPM

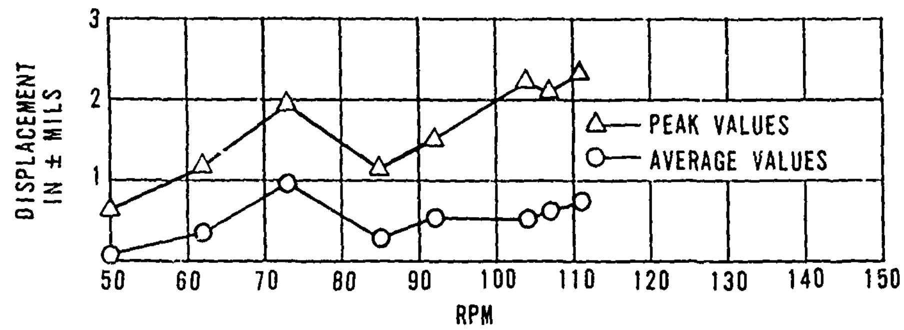


Figure 5 - Longitudinal Blade-Frequency Displacement of Shaft versus RPM

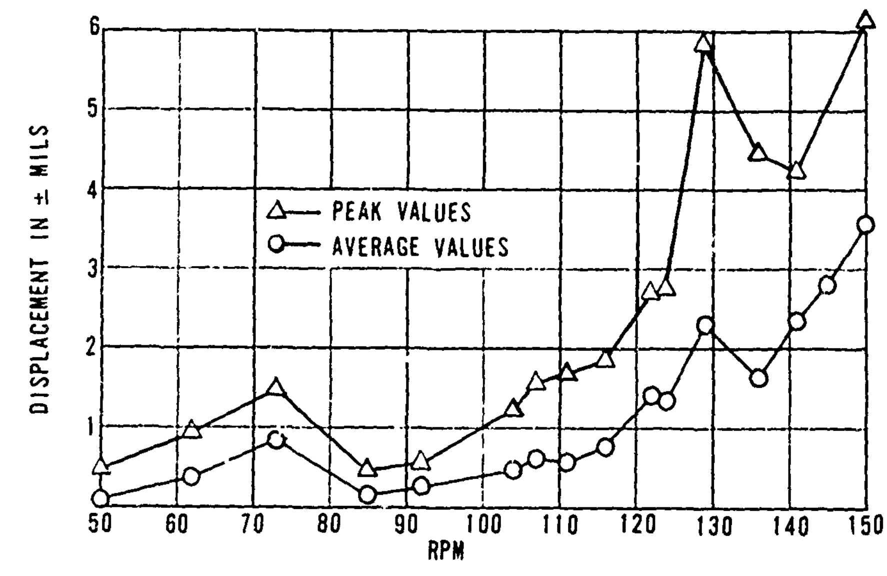


Figure 6 - Longitudinal Blade-Frequency Displacement of Thrust-Bearing Housing versus RPM

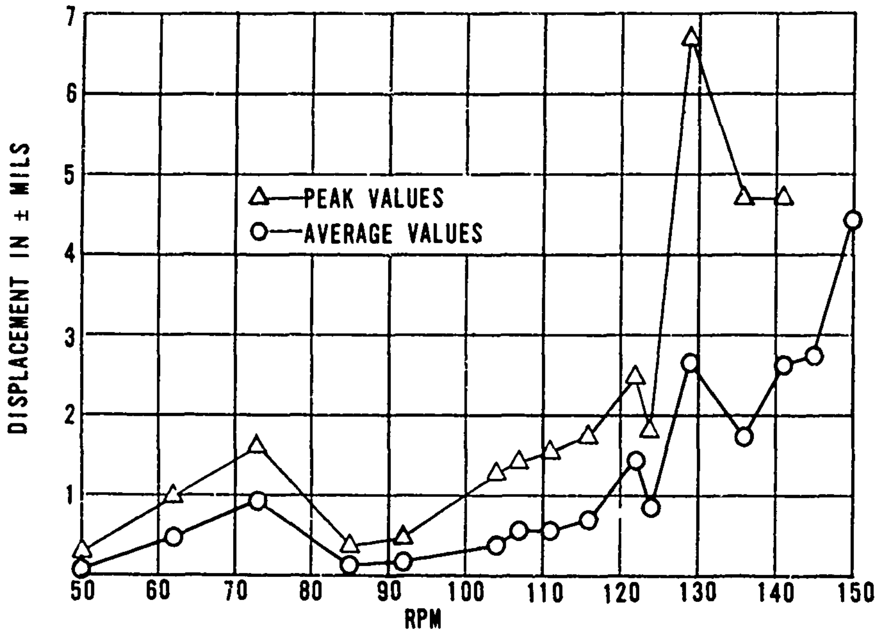


Figure 7 - Longitudinal Blade-Frequency Displacement of Gear-Case Top versus RPM

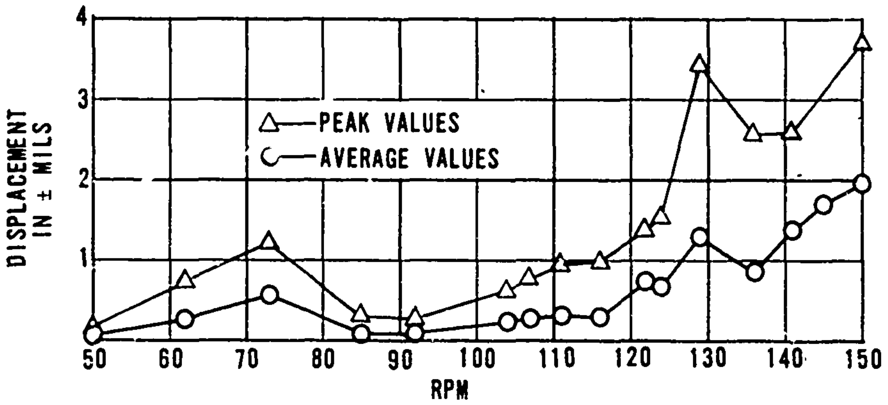


Figure 8 - Longitudinal Blade-Frequency Displacement of Gear-Case Bottom versus RPM

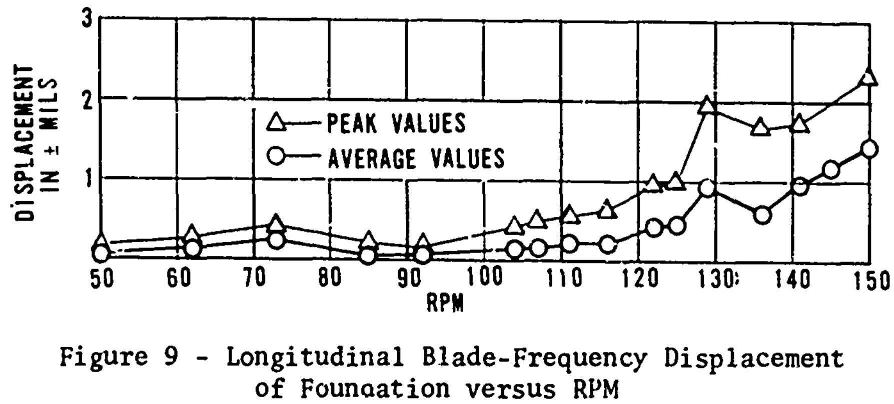


Figure 9 - Longitudinal Blade-Frequency Displacement of Founaation versus RPM

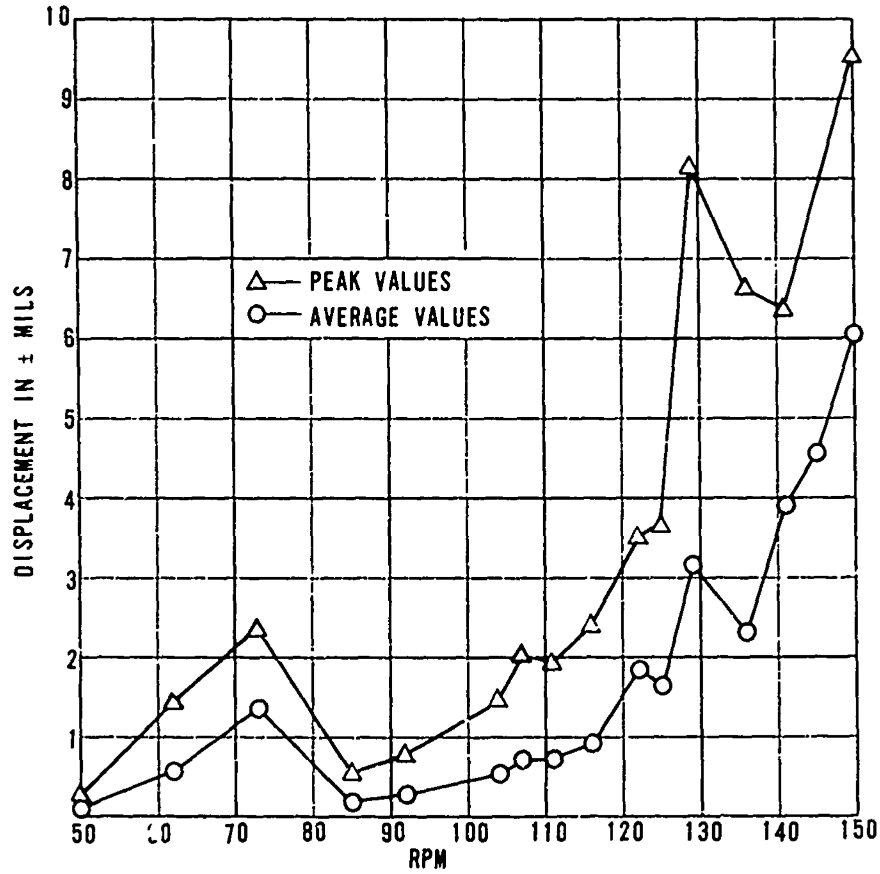


Figure 10 - Longitudinal Blade-Frequency Displacement of Low-Pressure Turbine versus RPM

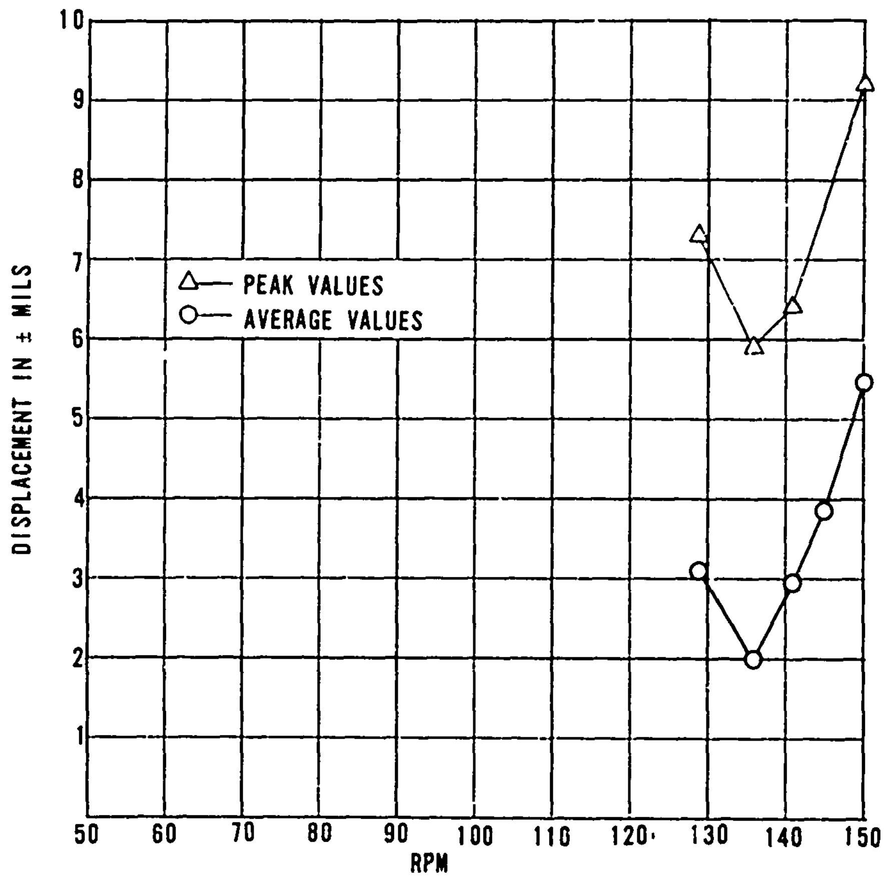


Figure 11 - Longitudinal Blade-Frequency Displacement of Condenser versus RPM

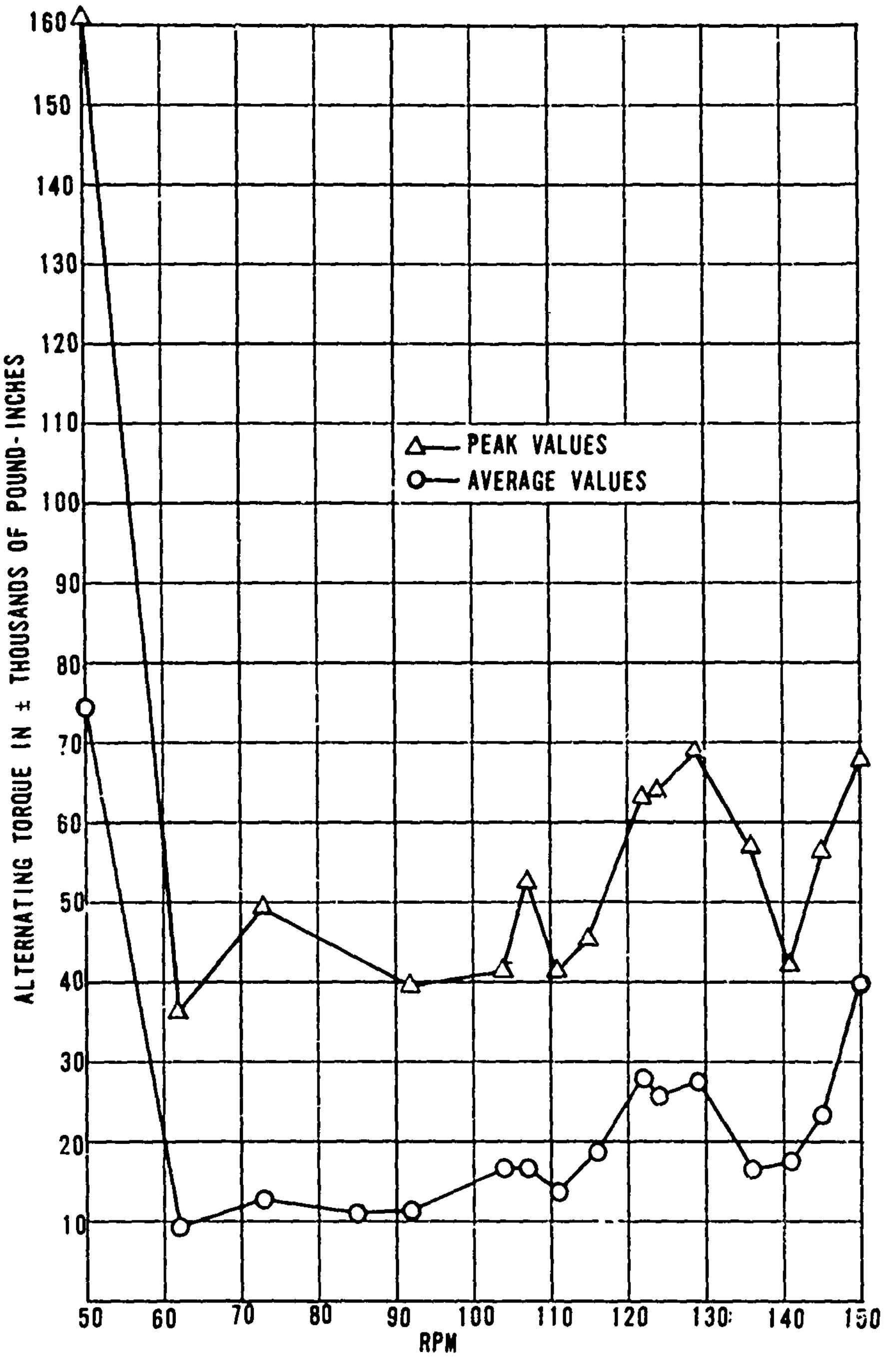


Figure 12 - Blade-Frequency Alternating Torque in Shaft versus RPM

Figure 5 was terminated at 110 rpm because the signals from the velocity gage were too large for the telemetering unit. The fundamental longitudinal shaft resonance occurs at about 129 rpm and is apparent at all stations. An unanticipated peak at 73 rpm is also reflected in all machinery plots.

The alternating torque (Figure 12) peaks at 50 rpm due to the fundamental torsional resonance of the shafting and peaks again in the area of the longitudinal resonance.

Figure 13 shows the double blade-frequency amplitudes for alternating thrust at all steady speeds. Peaks are evident at 18 and 22 cps. The latter is taken as the second mode frequency since it corresponds more closely with data taken on SIMON LAKE.[1]

The maximum vibration during maneuvers occurred at blade frequency, except for alternating torque. These amplitudes, obtained from oscillograph records, are given in Table 2.

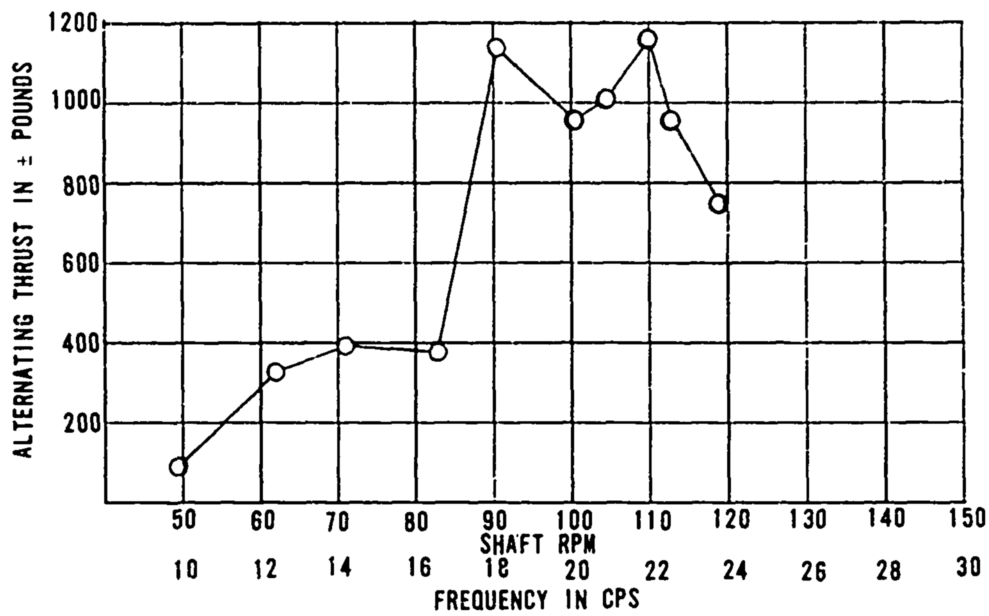


Figure 13 - Average Double Blade-Frequency Alternating Thrust in Shaft versus RPM

TABLE 2

Maximum Amplitudes of Vibration, Thrust, and Torque During Maneuvers*

Station	Maneuver			
	Acceleration (at 135 rpm)	Left Turn (at 136 rpm)	Right Turn (at 136 rpm)	Crashback (Peak at -85 rpm)
Thrust Bearing Housing---mils	± 3.9	± 7.0	± 9.0	± 15.0
Reduction Gear Top-------mils	± 4.4	± 7.5	± 11.0	± 16.0
Reduction Gear Bottom----mils	± 2.1	± 3.9	± 4.8	± 8.5
Foundation---------------mils	± 2.3	± 3.6	± 3.6	± 6.6
Turbine------------------mils	± 5.1	± 8.0	± 11.0	± 18.0
Condenser----------------mils	± 4.8	± 4.5	± 6.0	± 17.0
Alternating Thrust-------lb	± 17,200	± 38,400	± 30,300	± 24,800
Alternating Torque Frequency------ -------lb-in.	±137,000 (1.75 X shaft)	±167,000 (1.9 X shaft)	±269,000 (1.5 X shaft)	±548,000 Peak at -54 rpm (5 X shaft)

*All values shown are at blade frequency (6 X shaft) except where otherwise shown.

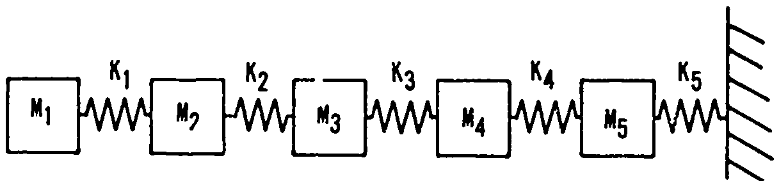
ANALYSIS

This analysis has been performed to define, as nearly as possible from the trial data, an equivalent mass-elastic system, the exciting forces, and damping associated with the CANOPUS propulsion system.

MASS-ELASTIC SYSTEM

The structure of the propulsion system from the propeller to the thrust bearing is comparitively simple. The values of mass and stiffness associated with this portion of the system can be adequately defined in accordance with established procedure. Therefore, the values of the parameters of the mass-elastic system used in the Model Basin prediction* and shown in Figure 14 will be used, except for thrust-bearing stiffness, foundation stiffness, and foundation mass. These three parameters are difficult to estimate and are later derived from the measured data.

*In an unpublished TMB letter report.



Parameter	Description of Parameter	Value of Parameter
M_1	Mass of propeller, including 60 percent for virtual mass, plus 1/2 propeller shaft (lb-sec^2/in.).	183
M_2	1/2 mass of prop.-shaft, plus 1/2 mass of stern tube shaft	90
M_3	1/2 mass of stern tube shaft, plus 1/2 mass of line shaft	70
M_4	1/2 mass of line shaft, plus mass of Bull gear and second reduction pinions	128
M_5	Foundation mass (condenser, turbines, gear case, first reduction gears and pinions, and part of foundation)	525*
K_1	Stiffness of prop. shaft (lb/in.)	18.1×10^6
K_2	Stiffness of stern tube shaft	11.1×10^6
K_3	Stiffness of line shaft	14.7×10^6
K_4	Thrust bearing stiffness	Unknown
K_5	Foundation stiffness	Unknown
*Modified later to account for a "lever effect"		

Figure 14 - Conventional Mass-Elastic System of CANOPUS Propulsion Plant

The thrust-bearing stiffness includes the stiffness of the elements, collar, and housing acting in series. The collar is stiff enough so that its effect can be ignored. CANOPUS has a six shoe thrust bearing, 43 in. in diameter, whose elements have a stiffness of approximately 18×10^6 lb/in.[2] The stiffness of the thrust-bearing housing is difficult to estimate, as it represents the stiffness of the housing and part of its supporting structure. In this analysis, the relative motion between the shaft at the collar and the gear-case mass is considered to be the displacement across the thrust-bearing spring.

The estimation of foundation stiffness is normally based on the results of experimental measurements on similar installations. Calculations to determine this parameter are complex and have not matched experimentally determined values closely enough to be useful in design prediction.

The foundation mass is normally assumed to be the actual mass of the reduction gear case, first reduction gears and pinions, low-pressure and high-pressure turbines, condenser, and a portion of the supporting structure. There are two considerations which may complicate this otherwise straightforward procedure.

First, the condenser and turbines are mounted on some ships with enough compliance to treat them as a mass separate from the other machinery. However, on CANOPUS, the measurements indicate that the single mass approach may be taken.

Second, since the greater part of the foundation mass is above shaft level and is structurally attached to the ship bottom, a "lever effect" may tend to increase the amplitudes of most of the foundation mass, which is above shaft level. This effect was detected in the longitudinal vibration study on SIMON LAKE[1] and is substantiated by CANOPUS measurements as shown in Figure 15. The level of each gage is projected to a vertical scale, and the displacements are plotted for eight different speeds. The points considered to be the most reliable indicators of the lever motion are indicated with solid dots. The thrust-bearing housing, condenser, and turbine displacements may be unreliable for determining lever motion because (1) local deformation of the thrust-bearing housing makes that displacement larger than other parts of the gear case at the same level, and (2) the turbine and condenser are supported only at one point on the gear case; they probably move longitudinally but do not rotate about an athwartship axis.

For calculations, it seems reasonable to assume that the inner bottom of the ship does not vibrate longitudinally and that the longitudinal vibration of any part of the foundation mass is proportional to the height above this point of rotation, as indicated by the straight lines of Figure 15. To account for the lever effect, the mass-elastic system of Figure 16 might be used. In this simplified representation, the parameters are the same as previously with K_4 and K_5 unknown, and M_5 equal to the actual foundation mass 525 $\frac{\text{lb-sec}^2}{\text{in.}}$

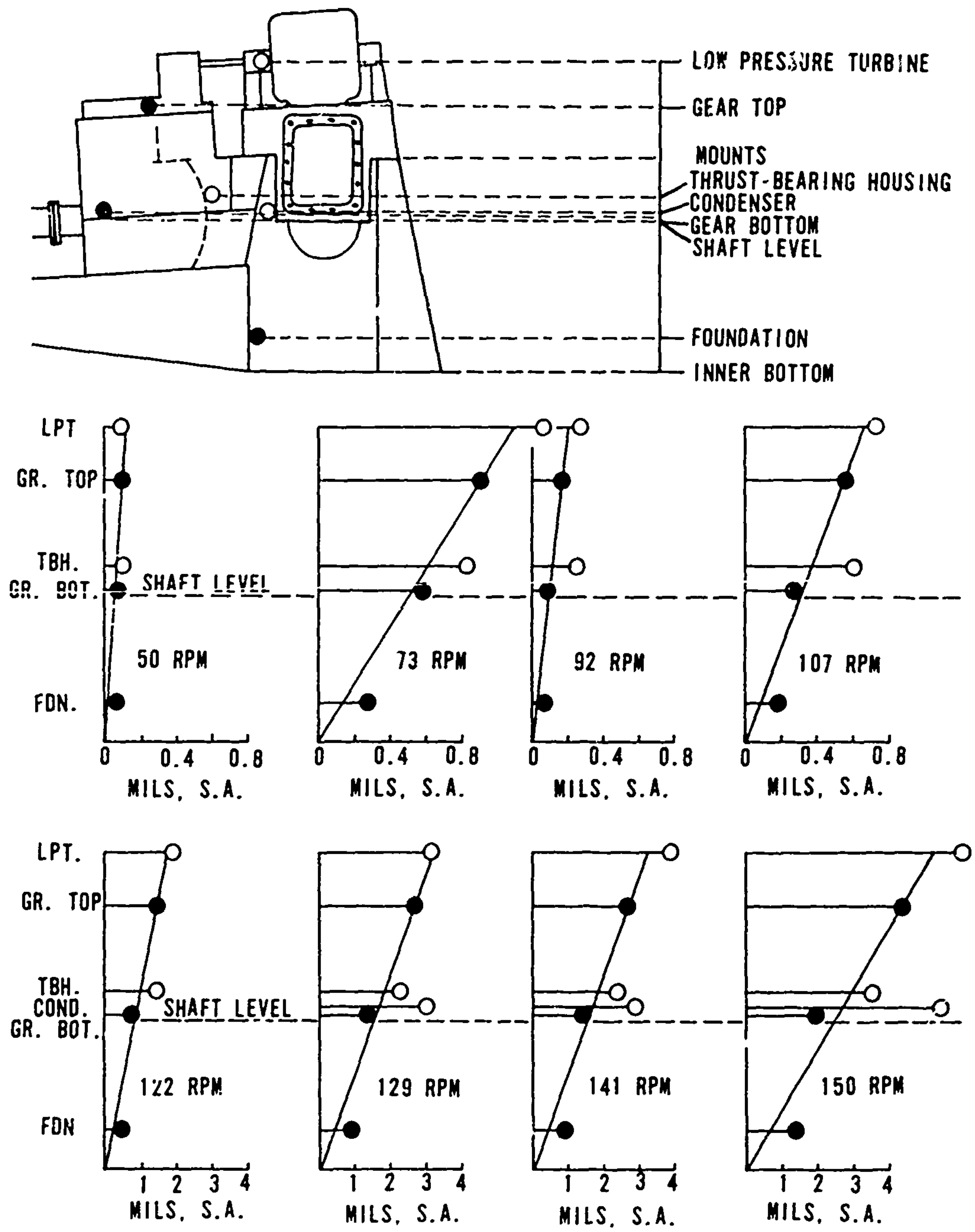


Figure 15 - Profile of Propulsion-System Vibration Showing Lever Effect

Additionally

a is the height of the shaft ℄ above the inner bottom of the ship,

b is the "effective height" of the foundation mass M_5 above the inner bottom (found later from measured data),

x_b is the longitudinal displacement of M_5, and

x_a is the longitudinal displacement of the point on the lever at shaft level.

The last two quantities are used in place of x_5.

In the equations of motion, the inertial force of M_5 is referred to shaft level and is equated to the applied spring forces. Assuming a small angle of rotation of the lever, the inertial force of M_5 at a height of b is $M_5\ddot{x}_b$. Because of the lever, when referred to shaft level this force is $\left(\frac{b}{a}\right) M_5\ddot{x}_b$. Since $\ddot{x}_b = \left(\frac{b}{a}\right)\ddot{x}_a$, substitution results in an inertial force of $\left(\frac{b}{a}\right)^2 M_5\ddot{x}_a$. The only difference between the inertial force of M_5 of the inline system (Figure 14) and that of the lever system is the factor of $\left(\frac{b}{a}\right)^2$.

To determine the corresponding inertial force in an actual propulsion plant, it is necessary to break down the foundation mass into several masses M_i, to determine the height b_i of each, and to sum the inertial forces of each mass to get the total inertial effect of the foundation mass F_{M_5}

$$F_{M_5} = \sum_i \left(\frac{b_i}{a}\right)^2 M_i \ddot{x}_a$$

At least two complications to this procedure are as follows:

1. The turbine and condenser are supported at only one point on the gear case. Consequently, the lever rotation is probably not effectively transmitted from the gear case to the turbine and condenser. If this is so, b_i for the turbine and condenser should be the height of the mounts.

2. The turbine rotors and the water in the condenser may not move with the rest of the foundation mass.

Because of these and possibly other complications, an empirical approach may be more appropriate. The ratio of the "effective height" of the foundation mass to the shaft height $\frac{b}{a}$ may be nearly the same for many turbine-double reduction gear plants if their arrangement and mass distribution are similar. In the analysis of this system the "effective foundation mass" is taken as $\left(\frac{b}{a}\right)^2 M_5$, and the ratio $\frac{b}{a}$ is found from measured data.

Since there are three unknown quantities--K_4, K_5, and $\left(\frac{b}{a}\right)^2 M_5$--there must be three suitable conditions or facts about the response of the system in solving for the unknowns. The first two natural frequencies of 12.9 and 22.0 cps will be used.

The selection of the third condition is prompted by the results obtained on SIMON LAKE.[1] In the analysis of the propulsion system of SIMON LAKE the exciting force was derived from both measured alternating thrust and a mass-elastic system, which was partially derived from measured data. For a check, the derived forces were applied in a computer problem to the partially derived system to see if the displacement of the foundation mass as given by the computer agreed with the measured displacements. They agreed up to and including the first mode resonance which occurred at 122 rpm. However, the computed displacements near full power (150 rpm) were about 50 percent high. (Figure 19 of Reference 1). In other words, there is an inaccuracy in the mass-elastic representation that causes a discrepancy in the proportion between alternating thrust in the shaft and foundation displacement near 150 rpm. In that analysis, the thrust-bearing housing was estimated to be 6×10^6 lb/in. It is felt that this is by far the most likely of the estimated parameters to be in error. Therefore, in the CANOPUS analysis, the thrust-bearing stiffness will be derived from the measured data rather than estimated from its physical configuration. Specifically, the third condition imposed on the mass-elastic system will be that the ratio of "alternating thrust in the shaft" (Force in spring K_3) to foundation-mass displacement at shaft level x_a at 15.0 cps (150 rpm) must be the same as measured during the trials. These measured quantities are taken from Figures 4 and 15.

The mathematics of the problem is as follows: in the mass-elastic system of Figure 16, the values of $\frac{b}{a}$, K_4, and K_5 must be determined so that the first two resonant frequencies are 12.9 and 22.0 cps and so that the ratio of force in spring K_3 to the displacement x_a at 15.0 cps is 6600 $\frac{lb}{mil}$. This mass-elastic system must have certain values of $\left(\frac{b}{a}\right)$ and K_5 for any particular value of K_4 to show resonances at 12.9 and 22.0 cps. Therefore, the first step is to find the values of $\left(\frac{b}{a}\right)$ and K_5 that correspond to several arbitrary values of K_4. The second step is to find, for each of the resulting mass-elastic systems, the ratio of force in K_3 to x_a at 15.0 cps. This ratio is plotted as a function of K_4, and the value of K_4 corresponding to the measured ratio is taken to be the actual K_4.

For the first step, values of 6, 8, 10, and $12X10^6$ lb/in. are arbitrarily chosen for K_4. The corresponding values of K_5 and $\left(\frac{b}{a}\right)$ can be determined from two Holzer tables, one for the fundamental frequency and one for the second natural frequency. These tables for $K_4 = 6.0X10^6$ are shown in Table 3. The last line of each table is written in terms of the unknown parameters and equations obtained from the fact that for the last mass $x = \Delta x$. Solutions of these equations and similar equations for the other three values of K_4 yield the results shown in the first two columns of Table 4.

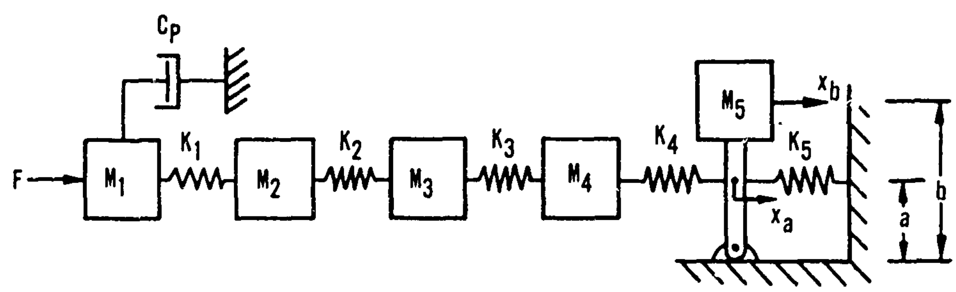


Figure 16 - Mass-Elastic System Accounting for Lever Effect

TABLE 3

Sample Holzer Solution for CANOPUS Mass-Elastic System of Figure 16

1st Mode: f_n = 12.9 cps ω_n = 81.0 rad/sec ω_n^2 = 6561						
M, $\left(\frac{\text{lb-sec}^2}{\text{in.}}\right)$	$M\omega^2/10^6$	x, inch	$M\omega^2x/10^6$	$\Sigma M\omega^2x/10^6$	$K/10^6$, $\left(\frac{\text{lb}}{\text{in.}}\right)$	Δx, inch
183	1.201	1.000	1.201	1.201	18.1	0.066
90	0.590	0.934	0.551	1.752	11.1	0.158
70	0.459	0.776	0.356	2.108	14.7	0.143
128	0.840	0.633	0.532	2.640	6.0	0.440
				2.640		
$525\left(\frac{b}{a}\right)^2$	$3.44\left(\frac{b}{a}\right)^2$	0.193	$0.662\left(\frac{b}{a}\right)^2$	$+0.662\left(\frac{b}{a}\right)^2$	$K_5/10^6$	$\frac{2.640 + 0.662(b/a)^2}{K_5/10^6}$

2nd Mode: f_n = 22.0 cps ω_n = 138.0 rad/sec ω_n^2 = 19044						
M, $\left(\frac{\text{lb-sec}^2}{\text{in.}}\right)$	$M\omega^2/10^6$	x, inch	$M\omega^2x/10^6$	$\Sigma M\omega^2x/10^6$	$K/10^6$, $\left(\frac{\text{lb}}{\text{in.}}\right)$	Δx, inch
183	3.485	1.000	3.485	3.485	18.1	0.192
90	1.715	0.808	1.383	4.868	11.1	0.438
70	1.333	0.370	0.493	5.361	14.7	0.365
128	2.440	0.005	0.012	5.373	6.0	0.893
				5.373		
$525\left(\frac{b}{a}\right)^2$	$10.00\left(\frac{b}{a}\right)^2$	-0.888	$-8.88\left(\frac{b}{a}\right)^2$	$-8.88\left(\frac{b}{a}\right)^2$	$K_5/10^6$	$\frac{5.373 - 8.88(b/a)^2}{K_5/10^6}$

From the first mode:

$$0.193 = \frac{2.640 + 0.662\left(\frac{b}{a}\right)^2}{K_5/10^6}$$

From the second mode:

$$-0.888 = \frac{5.373 - 8.88\left(\frac{b}{a}\right)^2}{K_5/10^6}$$

Rearranging and solving:

$$0.193\ K_5/10^6 - 0.662\left(\frac{b}{a}\right)^2 = 2.640$$

$$-0.888\ K_5/10^6 + 8.88\left(\frac{b}{a}\right)^2 = 5.373$$

$$K_5/10^6 = \frac{\begin{vmatrix} 2.64 & -0.662 \\ 5.373 & +8.88 \end{vmatrix}}{\begin{vmatrix} 0.193 & -0.662 \\ -0.888 & +8.88 \end{vmatrix}} = 23.9$$

$$\left(\frac{b}{a}\right)^2 = \frac{\begin{vmatrix} 0.193 & 2.64 \\ -0.888 & 5.373 \end{vmatrix}}{\begin{vmatrix} 0.193 & -0.662 \\ -0.888 & +8.88 \end{vmatrix}} = 2.98$$

$$K_5 = 23.9 \times 10^6 \text{ lb/in.} \qquad \left(\frac{b}{a}\right) = 1.72$$

TABLE 4

Results of Holzer Table and Computer Calculations

Assigned Value of K_4	Corresponding Value of:				
	b/a	K_5	Force in K_3	x_a (mils)	$\frac{\text{Force in } K_3}{x_a}$
6.0×10^6	1.72	23.9×10^6	±1690 lb	0.2420	6980
8.0×10^6	1.6	17.4×10^6	±1193 lb	0.3336	3570
10.0×10^6	1.62	16.1×10^6	± 817 lb	0.3603	2265
12.0×10^6	1.68	16.2×10^6	± 566 lb	0.3650	1550
	From Holzer Tables of 1st & 2nd Modes		From Computer Solution at 15.0 cps		

For the second step, a digital computer was used to find the displacement of all five masses in each of the four systems when excited at the propeller mass by an arbitrary $\pm$1000-lb sinusoidal force at 15.0 cps. From the computer output, the force in K_3 at 15.0 cps is found by multiplying the stiffness of the spring K_3 by the relative displacement across it $(x_3 - x_4)$ at 15.0 cps. This force and x_a for 15.0 cps are given for each of the four systems in Table 4. The ratio $\frac{\text{force in } K_3}{x_a}$ is shown as a function of K_4 in Figure 17. The actual ratio from measurements (about 6600) corresponds to a value of K_4 of about 6.1. The Holzer table for $K_4 = 6.0 \times 10^6$ is considered accurate enough for this analysis.* The mass-elastic system considered to be equivalent to the actual mechanical system is that shown in Figure 18.

The value obtained for K_5 is close to the value calculated by Puget Sound Naval Shipyard (20×10^6). As is often the case, the calculated value was mistrusted, and a smaller value (10×10^6) was used for calculations. Past discrepancies between calculated and experimental values of foundation stiffness may be due largely to not accounting for a lever effect.

*This happens to be the value that was selected for SIMON LAKE calculations. As will be seen later, however, this procedure yields a value of $K_4 = 5.0 \times 10^6$ for SIMON LAKE.

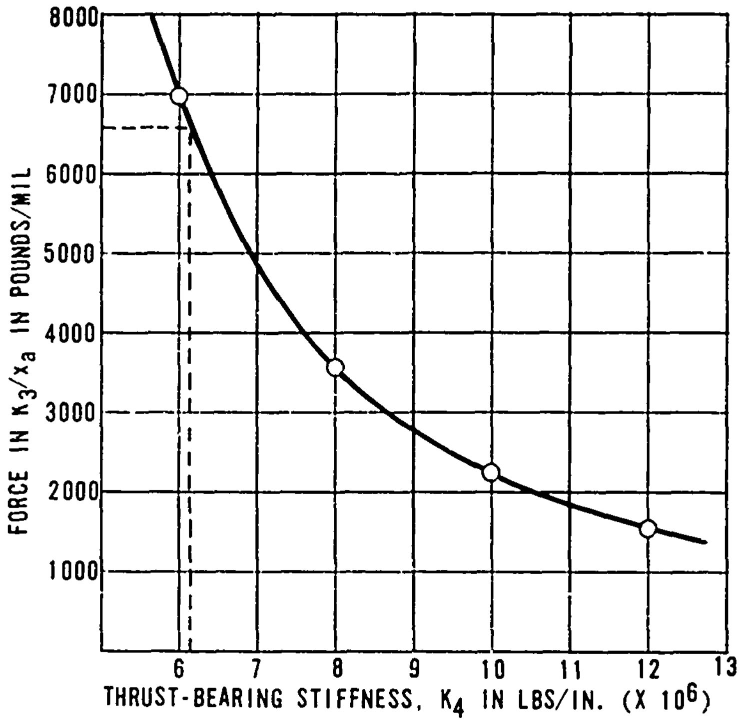


Figure 17 - Ratio of Force in Spring K_3 to x_a at 150 rpm for Various Values of K_4 (Refer to Figure 16)

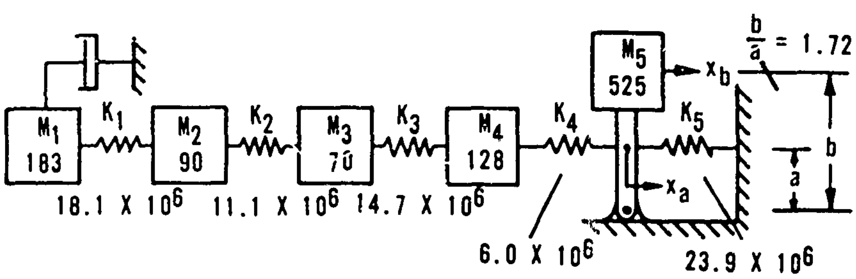


Figure 18 - CANOPUS Equivalent Mass-Elastic System

EXCITING FORCES AND DAMPING

A digital computer was used to obtain amplitude versus rpm curves of the mass-elastic system described for Figure 18. These curves are compared with the measured response to obtain the exciting forces and damping. The system included a single value of damping at the propeller, estimated to be 3000 $\frac{\text{lb-sec}}{\text{in.}}$ for the computer problem and determined more accurately later.

Exciting Forces

An exciting force of $\pm$1000 lb was used in the computer problem throughout the frequency range. The resulting response curves are given in Figure 19.

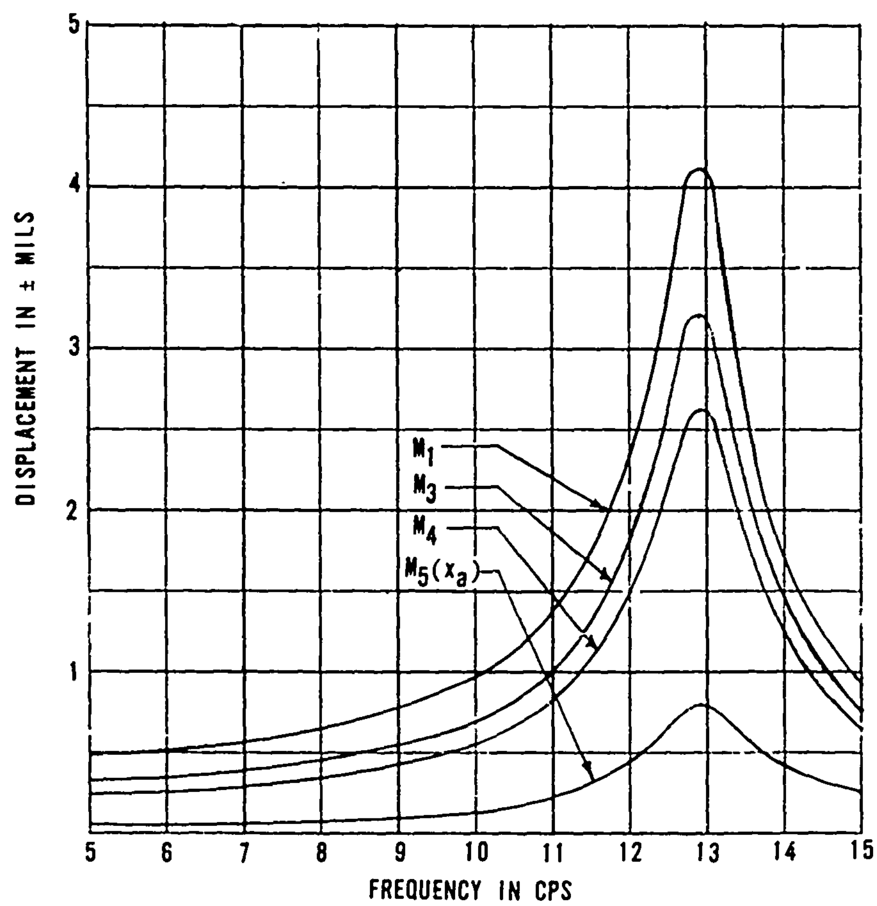


Figure 19 - Computed Response of System Shown in Figure 18 with C_p = 3000 lb-sec/in. and an Exciting Force of $\pm$1000 Pounds

In the mass-elastic system, K_3 corresponds to the section of shaft where the strain gages were installed. The force in K_3 can be determined by multiplying the stiffness of K_3 by the difference in the displacements of masses M_3 and M_4. That force, as a function of frequency for the $\pm$1000-lb exciting force, is given as Curve (a) in Figure 20. The difference between the force in K_3 and the exciting force at the propeller is due to the dynamic effects of the system; the ratio of the two will be called the "dynamic magnifier." On this basis the actual exciting force at the propeller can be found since the measured alternating thrust at K_3 (Curve (b) of Figure 20) differs from the exciting force by the same factor. Dividing the ordinates of Curve (b) by the dynamic magnifier for each frequency yields the exciting force at the propeller (Curve (c)). The portion of Curve (c) near resonance is not reliable since it is controlled by damping, which has only been roughly estimated. Instead the curve is faired through this range (dotted line). Note that average measured values of thrust were used. If peak measured values were used, the derived exciting force would be about three times as large.

A calculation of alternating thrust was made on the basis of a model wake survey at a speed corresponding to 130 rpm.* Calculations were made using the Burrill method, considering axial and tangential velocity components over the entire blade. The result was an alternating thrust of $\pm$1700 lb, as shown by the solid circle in Figure 20. If the assumption is made that alternating thrust varies as the square of rpm, as is often done, the resulting prediction (Curve (d) of Figure 20) is close to the derived curve below 130 rpm. Above 130 rpm agreement is obviously poor.

Damping

An accurate value for damping at resonance can be found from the formula:

$$C_p = \frac{F}{\omega x_p}$$

*In an unpublished TMB letter report.

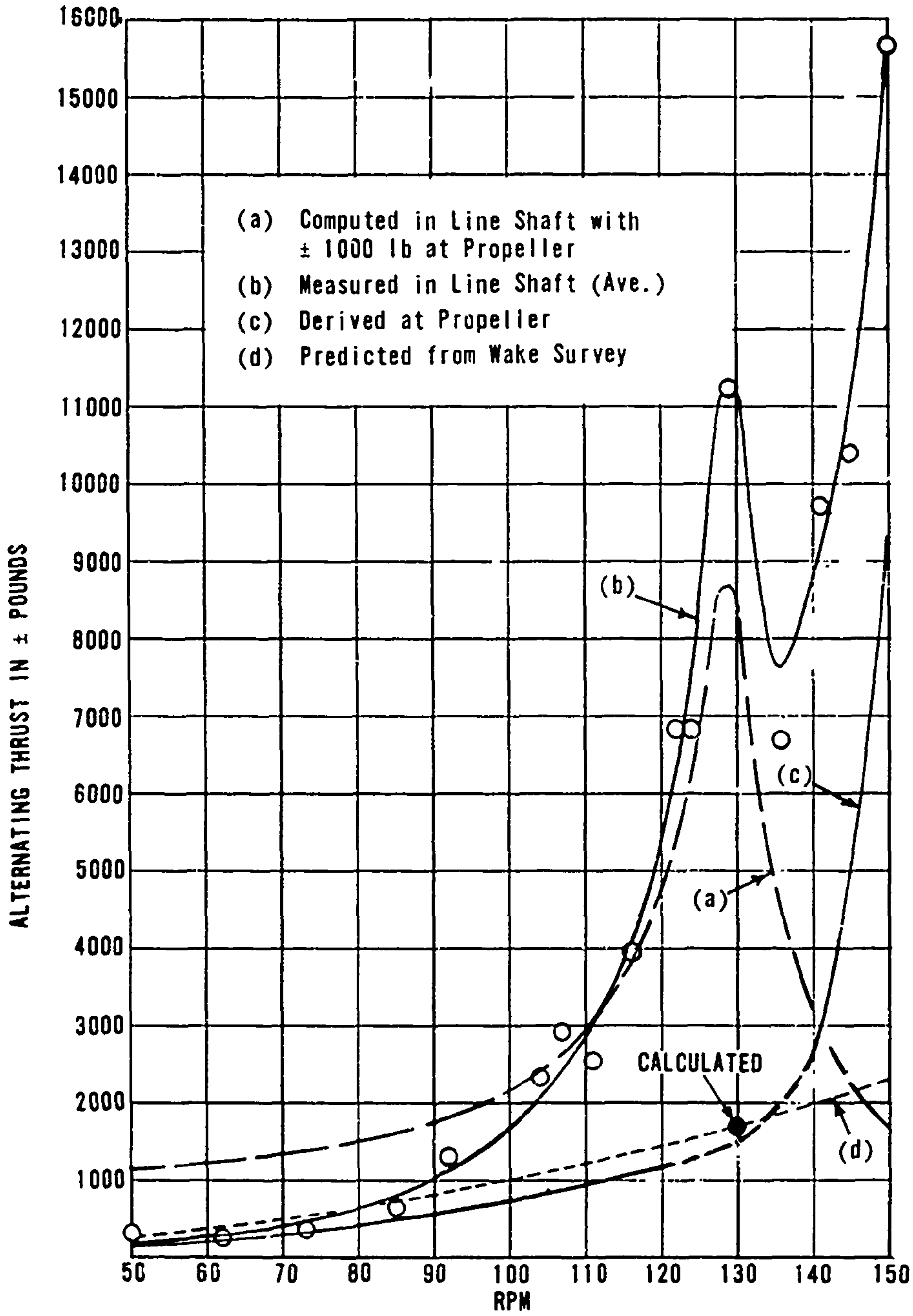


Figure 20 - Alternating Thrust at Propeller and in Line Shaft

where x_p is amplitude of propeller in inches

F is amplitude of exciting force in pounds

ω is circular frequency

C_p is equivalent viscous damping at the propeller in $\frac{\text{lb-sec}}{\text{in.}}$

This relationship is true for a system with propeller damping only and only at resonance where the input energy is equal to the damping energy. The exciting force at resonance from Curve (c) of Figure 20 is estimated to be ± 1500 lb. From the Holzer table mode shape and the measured amplitude of x_a (± 1.6 mils), $x_p = 8.3$ mils. Therefore:

$$C_p = \frac{1500}{2\pi \times 12.9 \times 0.0083} = 2240 \ \frac{\text{lb-sec}}{\text{in.}}$$

CHECKING VALIDITY OF ANALYSIS

If the derived mass-elastic system, damping constant, and exciting forces are accurate; a computer, using these as inputs, should calculate displacements comparable to measured data. The only measured displacements that can be used as a comparison are those of the foundation mass at shaft level, taken from displacement profiles such as shown in Figure 15. The results of this comparison (Figure 21) show excellent agreement throughout the operating range.

CONCLUSIONS

1. A longitudinal resonance exists in the operating range of the shaft about 86 percent of full power rpm.
2. This resonance is not detrimental, and the amplitudes of vibration even at resonance are considered acceptable.
3. The major reason for low amplitudes is that the combination of desirable stern configuration and a six blade propeller result in low exciting forces.
4. The equivalent mass-elastic system that was derived from measured data is considered reasonably accurate. This is verified by a good agreement between the calculated amplitude response of the mass-elastic system and the measured response.

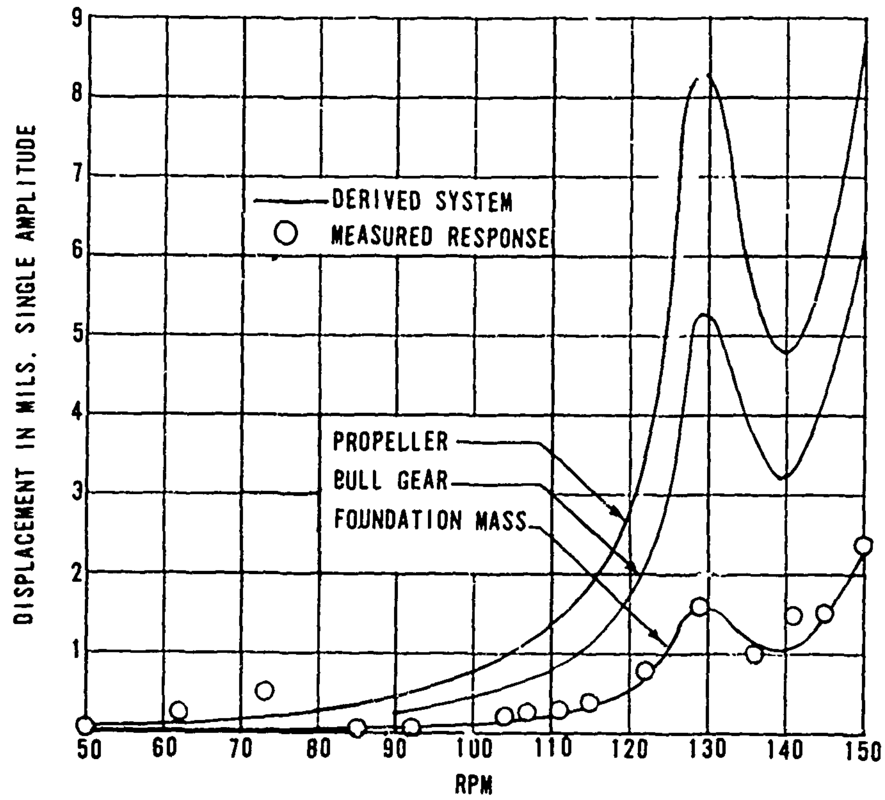


Figure 21 - Amplitude Response of Derived-Equivalent System to the Derived Forces Compared to the Measured Response

5. Alternating thrust, as calculated from a model wake survey, is ± 1700 lb at 130 rpm. Trial data at 130 rpm indicate a value of ± 1500 lb for average alternating thrust, which is in excellent agreement considering the complexity of the task of predicting this quantity.

6. Alternating thrust is normally assumed to vary as the square of rpm. Trial data show a distinct departure from this relationship above 130 rpm. This should be investigated on other types of ships.

7. There are two characteristics (long shafts and more blades on the propellers) of modern Navy ships which can bring two modes of longitudinal vibration within the operating range of blade frequency. At present, prediction techniques are only good enough to get a rough idea of what happens in the first mode. A special effort must be made to improve our methods of prediction above the first mode. The discussion of the lever effect is a step in this direction.

8. The traditional disagreement between experimentally determined and calculated values of foundation stiffness does not exist on this class ship when the "lever effect" is considered. Reliable values of this parameter for design predictions may not be as elusive as they once appeared. This is another area that needs work.

9. When the first longitudinal resonance is excited (129 rpm), the level of the torsional vibration increases. However, when the first mode in torsion is excited (48 rpm), no effect is apparent on the longitudinal vibration. Although it is expected that there is an exchange of energy between longitudinal and torsional vibration, the data from this trial are inconclusive in this respect.

FUTURE PLANS

Reference 3 requires that Navy ships have no longitudinal critical speeds from 50 to 115 percent of full power rpm. The fact that CANOPUS has a critical speed in the middle of this range that is not detrimental indicates that this requirement is not always realistic. The Model Basin has recently prepared recommendations, now being reviewed by Naval Ship Systems Command, for an interim revision of Military Standard 167.[3] It is expected that as more ships are studied, it will be possible to make a more complete revision as well as improve techniques of predicting longitudinal shaft vibration.

Trials have been conducted on the USS BELKNAP (DLG-26) and the USS AUSTIN (LPD 4); the significance of data taken will be investigated shortly. Trials are planned for the USS BRUMBY (DE 1044). On these trials the character of the investigation will be much the same as for CANOPUS--concerning lever effect, foundation stiffness, exciting forces, damping, and effects of longitudinal torsional coupling. In addition, more consideration will be given to the turbine rotor and its longitudinal motion inside the casing. Presently there are no requirements for this, even though it has caused problems in the past.

Also, since the measured exciting forces are not as expected, the alternating thrust will be calculated from the model wake survey at 3 speeds, 60, 80, and 100 percent of full power, for BRUMBY and AUSTIN.

ACKNOWLEDGMENTS

The officers and crew of CANOPUS were most cooperative while the trials were being conducted. The authors are indebted to Mr. R. Farmer for recording the data and to Mr. W. Hinterthan for his estimate of alternating thrust.

APPENDIX A
MODIFIED SIMON LAKE ANALYSIS

Since an improved technique was used for CANOPUS, the analysis of SIMON LAKE should be revised accordingly. The logic and the steps taken in deriving the parameters were the same for both ships, so it should suffice to include only the tables, calculations, and figures for SIMON LAKE. These are directly comparable to data from CANOPUS. See Tables A1 and A2, and Figures A1 through A5.

TABLE A1

Sample Holzer Solution for SIMON LAKE Mass-Elastic System
(Comparable to Table 3 for CANOPUS)

1st Mode: $f_n = 12.2$ $\omega_n = 76.7$ $\omega_n^2 = 5870$						
M	$M\omega^2/10^6$	x	$M\omega^2 x/10^6$	$\Sigma M\omega^2 x/10^6$	$K/10^6$	Δx
183	1.072	1.000	1.072	1.072	18.1	0.059
90	0.528	0.941	0.497	1.569	11.1	0.142
70	0.410	0.799	0.327	1.896	14.7	0.129
128	0.750	0.670	0.502	2.398	5.0	0.48
$525\left(\frac{b}{a}\right)^2$	$3.070\left(\frac{b}{a}\right)^2$	0.190	$0.583\left(\frac{b}{a}\right)^2$	$2.398 + 0.583\left(\frac{b}{a}\right)^2$	$K_5/10^6$	$\dfrac{2.398 + 0.583\left(\frac{b}{a}\right)^2}{K_5/10^6}$
2nd Mode: $f_n = 21.0$ $\omega_n = 132$ $\omega_n^2 = 17400$						
M	$M\omega^2/10^6$	x	$M\omega^2 x/10^6$	$\Sigma M\omega^2 x/10^6$	$K/10^6$	Δx
183	3.18	1.000	3.18	3.18	18.1	0.176
90	1.57	0.824	1.30	4.48	11.1	0.403
70	1.22	0.421	0.513	4.993	14.7	0.340
128	2.23	0.081	0.181	5.174	5.0	1.035
$525\left(\frac{b}{a}\right)^2$	$9.13\left(\frac{b}{a}\right)^2$	-0.954	$-8.72\left(\frac{b}{a}\right)^2$	$5.174 - 8.72\left(\frac{b}{a}\right)^2$	$K_5/10^6$	$\dfrac{5.174 - 8.72\left(\frac{b}{a}\right)^2}{K_5/10^6}$
$0.190\ K_5/10^6 - 0.583\left(\frac{b}{a}\right)^2 = 2.398$						
$-0.954\ K_5/10^6 + 8.72\left(\frac{b}{a}\right)^2 = 5.174$						

$$K_5/10^6 = \frac{\begin{vmatrix} 2.398 & -0.583 \\ 5.174 & 8.72 \end{vmatrix}}{\begin{vmatrix} 0.190 & -0.583 \\ -0.954 & 8.72 \end{vmatrix}} = 21.7 \qquad \boxed{K_5 = 21.7 \times 10^6}$$

$$\left(\frac{b}{a}\right)^2 = \frac{\begin{vmatrix} 0.190 & 2.398 \\ -0.954 & 5.174 \end{vmatrix}}{1.102} = 2.97 \qquad \boxed{\frac{b}{a} = 1.72}$$

TABLE A2

Results of Holzer Table and Computer Calculations for SIMON LAKE (Comparable to Table 4 for CANOPUS)

Assigned Value of K_4	Corresponding Value of:				
	b/a	K_5	Force in K_3 (lb)	x_a (mils)	$\frac{\text{Force in } K_3}{x_a}$
5.0×10^6	1.72	21.7×10^6	±1590	±0.323	4920
6.0×10^6	1.60	16.6×10^6	±1080	±0.396	2730
7.0×10^6	1.58	15.0×10^6	± 672	±0.419	1600
	From Holzer Table		From Computer		

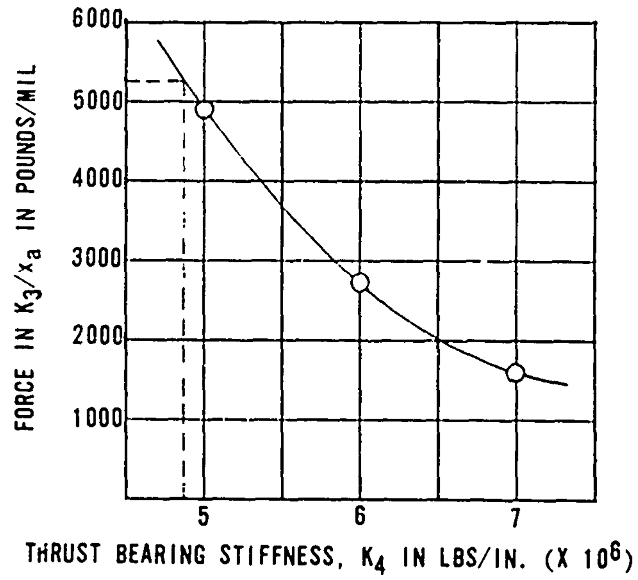


Figure A1 - Ratio of Force in Spring K_3 to x_a for Various Values of K_4 on SIMON LAKE (Comparable to Figure 17 for CANOPUS)

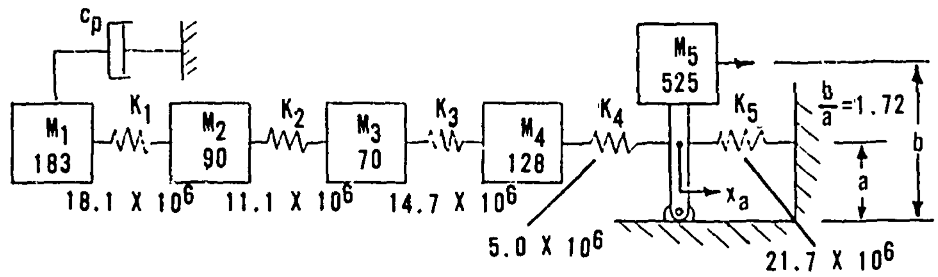


Figure A2 - SIMON LAKE Equivalent Mass-Elastic System (Comparable to Figure 18 for CANOPUS)

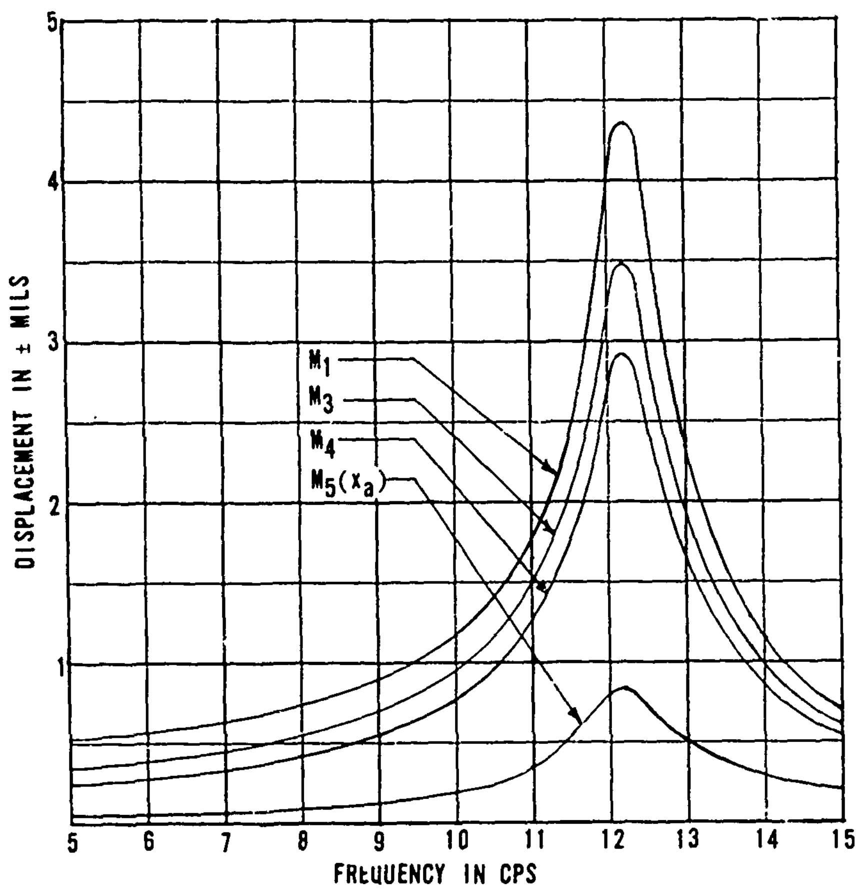


Figure A3 - Computed Response of SIMON LAKE System Shown in Figure 23 with C_p = 3000 lb-sec/in. and an Exciting Force of $\pm$1000 Pound (Comparable to Figure 19 for CANOPUS)

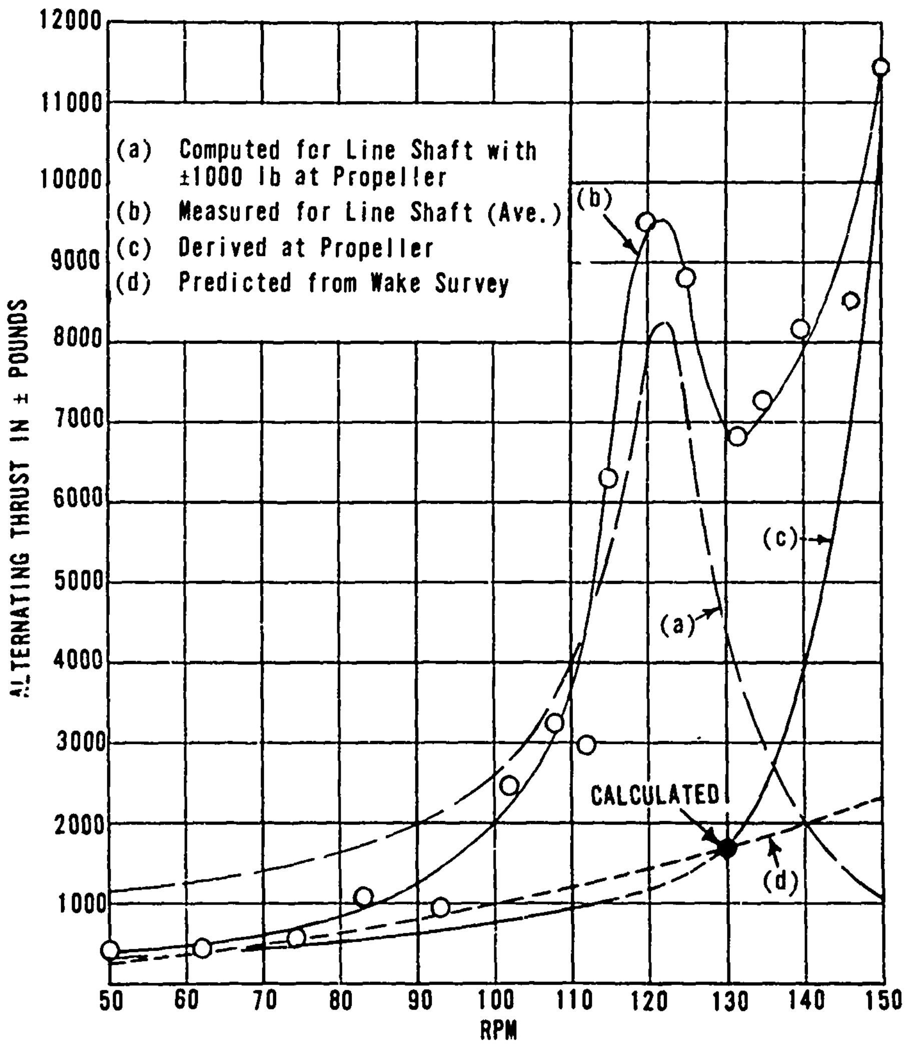


Figure A4 - Alternating Thrust at Propeller and in Line Shaft of SIMON LAKE (Comparable to Figure 20 for CANOPUS)

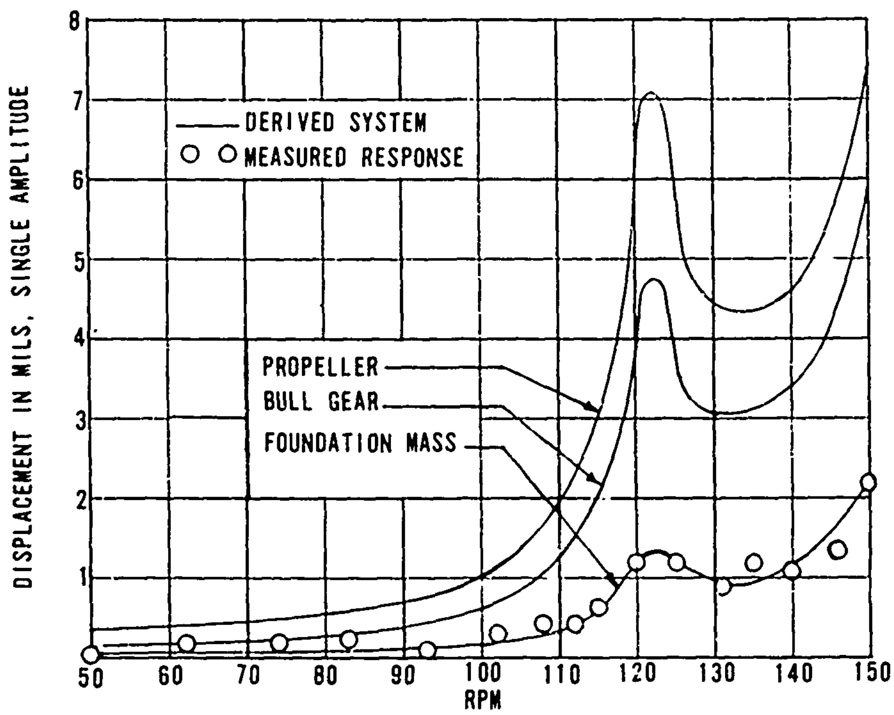


Figure A5 - Amplitude Response of Derived-Equivalent System to the Derived Forces Compared to the Measured Response for SIMON LAKE (Comparable to Figure 21 for CANOPUS)

Note: Damping for the SIMON LAKE Equivalent System was obtained as follows:

x_a = 1.35 mils, s.a. at resonance (from measurements)

$\therefore x_p$ = 7.1 mils, s.a. at resonance (from Holzer mode shape)

F = amplitude of exciting force at resonance = $\pm$1240 lb (from Figure A4)

$$C_p = \frac{F}{2\pi f x_p} = \frac{1240}{2\pi x 12.2 x 0.0071} = 2280 \frac{\text{lb-sec}}{\text{in.}}$$

REFERENCES

1. Antonides, G.P., "Longitudinal Vibration of Propulsion System on USS SIMON LAKE (AS-33)," David Taylor Model Basin Report 2147 (Jan 1966).

2. Kane, J.R. and McGoldrick, R.T., "Longitudinal Vibrations of Marine Propulsion - Shafting Systems," Transactions, Society of Naval and Marine Engineers, Vol. 57; also David Taylor Model Basin Report 1088 (Nov 1956).

3. US Navy Bureau of Ships Military Standards, "Military Standard - Mechanical Vibrations of Shipboard Equipment," Military Standard 167 (SHIPS) (20 Dec 1954).

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13 ABSTRACT

As part of an effort to increase the adequacy of prediction methods used in shaft design, a study was made of the longitudinal vibration characteristics of the propulsion system on the USS CANOPUS (AS-34). The lever effect, previously detected on the USS SIMON LAKE (AS-33) was confirmed. This means that the amplitudes of the gear case, turbines, and condenser are roughly dependent on the relative heights of these components to the level of the shaft. The effective foundation mass, foundation stiffness, and thrust-bearing stiffness are computed from the measured response of the system. A resonance was found to exist in the operating range, but it was not considered detrimental. The exciting forces are lower than usual, except at or near full power.

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