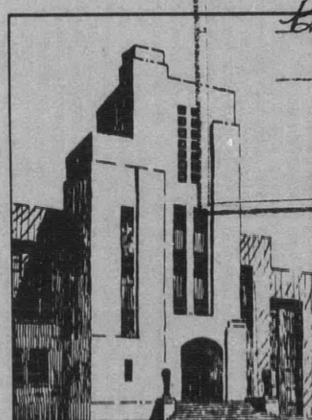
THE DAVID W. TAYLOR MODEL BASIN

UNITED STATES NAVY

AXIAL VIBRATION OF PROPULSION SYSTEMS OF BATTLESHIPS OF THE BB61 THROUGH 66 CLASS

BY R.T. McGOLDRICK



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PERSONNEL

This report is based on measurements of vibration and alternating components of thrust made by the David Taylor Model Basin and the Vibration Section of the New York Naval Shipyard during the initial sea trials of the USS IOWA (BB61) and USS NEW JERSEY (BB62). In working out the procedures to be followed in studying the vibration of these vessels, the Taylor Model Basin was represented in the conferences on board ship by Capt. J. Ormondroyd, USNR. The members of the Model Basin staff chiefly concerned with the experimental and theoretical work discussed were F.F. Vane, F. Mintz, E.O. Berdahl, R.B. Allnutt, C.E. Bowman, V.S. Hardy, E.J. Adams, and R.T. McGoldrick.

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AXIAL VIBRATION OF PROPULSION SYSTEMS OF BATTLESHIPS OF THE BB61 THROUGH 66 CLASS

ABSTRACT

The axial vibration of the propulsion systems and the alternating components of thrust measured during the sea trials of the IOWA (BB61) and NEW JERSEY (BB62) are discussed. The data obtained are compared with the values predicted on the basis of experience with battleships of the NORTH CAROLINA (BB55) and SOUTH DAKOTA (BB57) Classes. Comparison of wake variation behind the skegs is given for the three recent classes of battleships, BB55 and 56, BB57 through 60, and BB61 through 66, on the basis of experiments made in the model basin.

An explanation is attempted of the contribution of axial vibration to the wear in the turbine couplings which has been observed in the IOWA and SOUTH DAKOTA Classes. Finally, the general problem of axial vibration on recent battleships is discussed, and comparative data on the constants of the vibratory systems of the three classes are given. These data are recommended for making estimates of resonance frequencies for future designs.

INTRODUCTION

Although axial vibration of propulsion systems had occurred previously, it had not, in general, been considered a serious problem in naval design prior to its occurrence on the NORTH CAROLINA (BB55) and the WASHINGTON (BB56), which comprised the first class of the three recent classes of battleships built in the United States. Axial vibration and the means adopted to reduce it on these two vessels are discussed in an earlier report (1).* In that report it is pointed out that the severe vibration existing during the initial trials of the vessels was due chiefly to the combination of high wake variation in the propeller races with resonance of the vibratory systems. The resonance resulted from the long shafts and large propeller masses so reducing the natural frequencies that they fell within the operating range of blade frequencies.

In view of the effectiveness of 5-bladed propellers in reducing the axial vibration of the shafts enclosed by skegs on the NORTH CAROLINA, the Bureau of Ships decided to try a combination of 4- and 5-bladed propellers on vessels of the SOUTH DAKOTA Class (BB57 through 60). Three of these vessels were fitted with 4-bladed propellers on the inboard shafts and 5-bladed propellers on the outboard shafts, whereas the SOUTH DAKOTA (BB57) retained 4-bladed propellers on all shafts.

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* Numbers in parentheses indicate references on page 37 of this report.

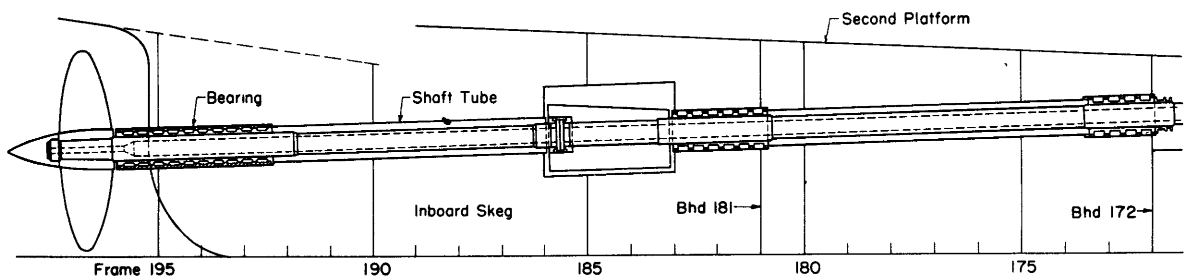

Figure 1 - Elevation of Port Inboard Propulsion System of the USS IOWA (BB61)

In the IOWA (BB61)and NEW JERSEY (BB62), which, like the NORTH CAROLINA and WASHINGTON, had skegs enclosing the inboard shafts, thrust bearings were installed in the shaft alleys and sufficient axial movement was provided in the turbine couplings to take care of any fore-and-aft movement of the reduction gears due to expansion of the shaft forward of the thrust bearing. This expedient was expected to isolate the gear case, turbines, and condenser from any fore-and-aft vibration that might exist in the shaft and gears. Thrust bearings were also provided at the forward end of the gear case on all vessels of the IOWA Class (BB61 through 66) as on the NORTH CAROLINA as a precaution in case the after thrust bearings should prove inadequate. The thrust shoes were not installed on the forward thrust bearings but were kept on hand in case they should be required.

An elevation of the port inboard propulsion system of the battleship IOWA is shown in Figure 1, which also shows the form of the skeg and the location of the after thrust bearing with relation to the propeller and the machinery. The afterbody of the ship is further illustrated in Figures 2, 3, and 4.

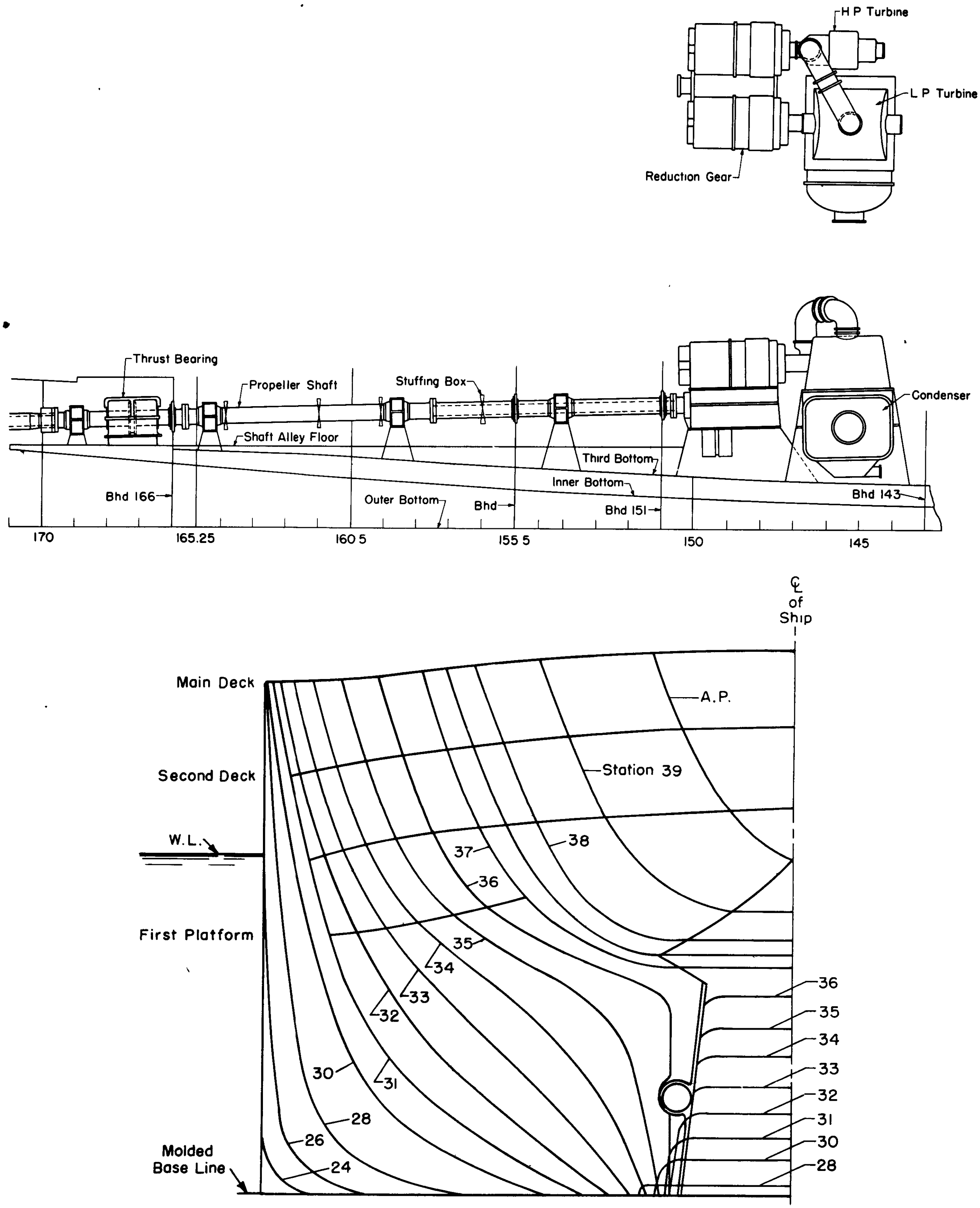

Figure 2 - Plan of Afterbody of USS IOWA (BB61), Showing Shape of Skeg and Tunnel between Skegs

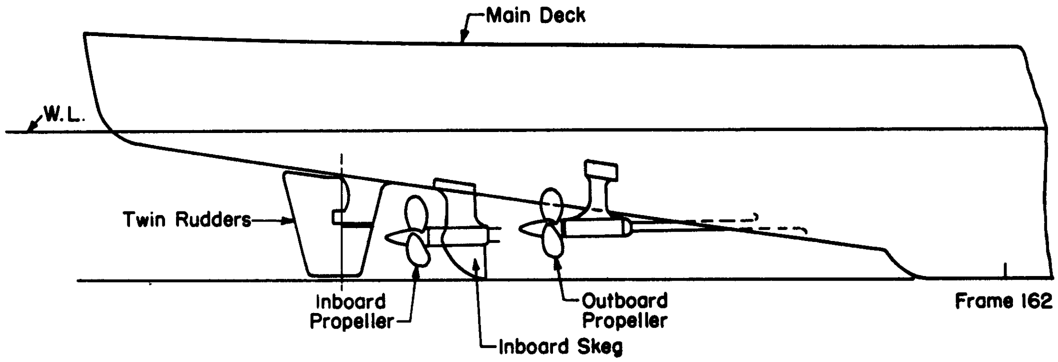

Figure 3 - Outboard Profile of Afterbody of USS IOWA (BB61)

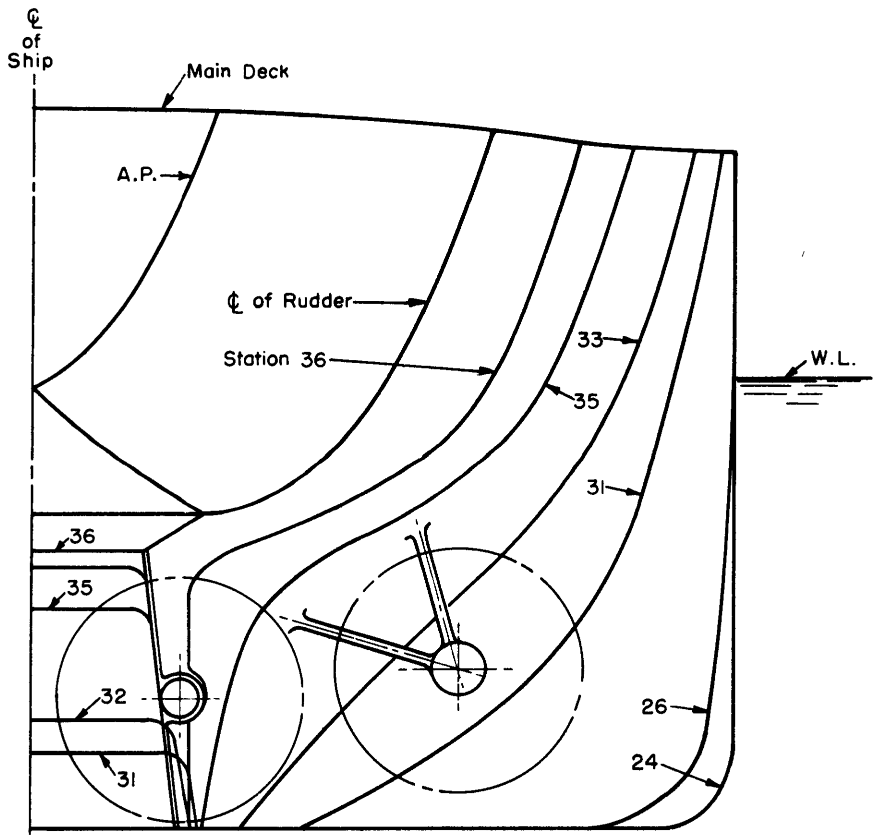

Figure 4 - Partial Body Plan of USS IOWA (BB61), Showing Position of Propellers with Reference to Stern of Ship

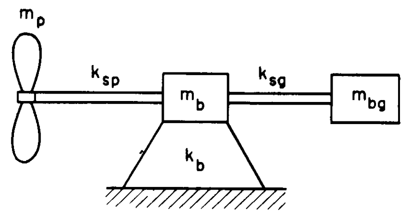

Figure 5 - Ideal 3-Body Vibratory System Used in Calculating Natural Frequencies of Axial Vibration of the Propulsion Systems on the BB61 through 66 Class

ESTIMATES OF THE RESONANCE FREQUENCIES

When the thrust bearings are located aft and the gears are free to move axially, the only connection between the gears and the rest of the machinery and foundations resides in the friction in the bearings and in the turbine couplings. It therefore seems reasonable, for the purposes of computation, to represent the vibratory system by the ideal system shown schematically in Figure 5. In the ideal system the gear case and turbine are eliminated entirely from the vibratory system, and only mass and elasticity are represented, friction being assumed to have a negligible effect on the natural frequencies. In Figure 5, m_p equals the mass of the propeller, plus 60 per cent to allow for the virtual mass of the entrained water, plus one-half the mass of the shaft between the propeller and the thrust bearing; m_b equals the mass of the thrust bearing, plus one-half the mass of the shaft between the thrust bearing and the propeller, plus one-half the mass of the shaft between the thrust bearing and the reduction gear, plus one-fourth the mass of the thrust-bearing foundation; m_{bg} equals the mass of the entire gear train and the parts of the turbine couplings that move with the gears, plus one-half the mass of the shaft between the gears and the thrust bearings; k_{sp} is the fore-and-aft spring constant of the part of the shaft between the thrust bearing and the propeller; k_{sg} is the fore-and-aft spring constant of the part of the shaft between the thrust bearing and the gears; and k_b is the effective fore-and-aft spring constant of the thrust-bearing foundation.

Of the constants representing the system of Figure 5 all but k_b can be obtained fairly readily from design data, but because of the flexibility of the surrounding hull structure the spring constant of the thrust-bearing foundation is difficult to estimate. There is some question as to the proper value of m_{bg} for the lock-train gear system where axial flexibility exists within the gear train itself. So far no allowance for this condition has been devised.

The "frequency equation" for this system, derived by setting up the differential equations for the motion of each of the masses during free vibration, is

$$-m_{bg}m_p m_b\,\omega^6 + (k_{sg}m_p m_b + k_{sp}m_b m_{bg} + k_{sg}m_p m_{bg} + k_{sp}m_p m_{bg} + k_b m_{bg} m_p)\,\omega^4$$

$$-(k_{sp}k_{sg}m_b + k_{sg}k_b m_p + k_{sg}k_{sp}m_p + k_{sp}k_b m_{bg} + k_{sp}k_{sg}m_{bg})\,\omega^2 + k_{sp}k_{sg}k_b = 0$$

Here ω is the circular frequency, or 2π times the natural frequency. This is a cubic equation in ω^2, which accordingly has three roots. The three positive values of ω give the resonance circular frequencies. The system of units adopted is the inch-pound-second system, that is, the unit of mass is pound-seconds^2 per inch; the unit of the spring constant is pounds per inch, the unit of frequency is cycles per second, and the unit of circular frequency is radians per second.

The natural frequencies of such a system can also be found by the Holzer method, in which the graphical representation is of great assistance, as shown in Reference (2).

The constants derived for the propulsion systems of the IOWA Class are given in Table 1. All the data given here were derived directly from design data, except k_b, which was based on dial-gage measurements of steady deflections of the after thrust bearing under load taken during the engineering trials.

TABLE 1

Constants of the Vibratory Systems Representing the Propulsion Systems of BB61 through 66 Considered As 3-Body Systems Such As Illustrated in Figure 5

Propulsion Unit	Constants of 3-Body System					
	m_p lb-sec^2/in	m_b lb-sec^2/in	m_{bg} lb-sec^2/in	k_{sp} lb/in	k_{sg} lb/in	k_b lb/in
Starboard Outboard	361	410	495	7.4×10^6	3.02×10^6	13.5×10^6
Starboard Inboard	370	310	392	7.3×10^6	5.89×10^6	13.5×10^6
Port Inboard	370	246	326	7.3×10^6	12.8×10^6	13.5×10^6
Port Outboard	361	344	431	7.4×10^6	3.87×10^6	13.5×10^6

Table 2 gives the calculated resonance frequencies of the first and second modes, together with the calculated relative amplitudes at the propeller, at the thrust bearing, and at the reduction gear, the value at the thrust bearing being taken as unity. It should be noted that in the first mode all members move in phase, whereas in the second mode one member moves out of phase with the other two, the nodal point being either forward or aft of the thrust bearing, depending on the numerical values of the constants of the vibratory system.

TABLE 2

Calculated Frequencies and Relative Amplitudes of the First Two Modes of Axial Vibration of the Propulsion Systems of BB61 through 66, Based on the Values Given in Table 1

Propulsion Unit	Natural Frequency, CPM		Amplitude*					
	First Mode	Second Mode	A_1	B_1	C_1	A_2	B_2	C_2
Starboard Outboard	648	1146	1.3	1	4.1	3.3	1	-0.74
Starboard Inboard	846	1260	1.7	1	2.1	8.1	1	-2.0
Port Inboard	939	1602	1.7	1	1.3	-4.1	1	3.4
Port Outboard	747	1158	1.4	1	3.1	3.5	1	-1.6

* A_1 is the amplitude at the propeller in the first mode.
B_1 is the amplitude at the thrust bearing in the first mode.
C_1 is the amplitude at the reduction gear in the first mode.
A_2 is the amplitude at the propeller in the second mode.
B_2 is the amplitude at the thrust bearing in the second mode.
C_2 is the amplitude at the reduction gear in the second mode.

WAKE ANALYSIS

In the course of the extensive investigation of axial vibration on the NORTH CAROLINA and WASHINGTON a technique was developed for exploring the wake in the propeller races in the towing basin. The data thus obtained showed large variations in the fore-and-aft component of the wake fraction* in the races of the propellers abaft the skegs, as had been expected. This technique was later applied to models of the SOUTH DAKOTA and IOWA Classes.

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* The wake fraction is equal to $1 - \dfrac{\text{velocity of water relative to ship}}{\text{velocity of ship}}$.

Figures 6, 7, and 8 show the variation in the fore-and-aft component of the wake fraction at the propeller-tip radius for the propeller races abaft the skegs on the three classes of vessels in question. Complete wake patterns for the IOWA are given in Figure 9. It will be observed from these figures that the wake variation abaft the skegs is greatest for the IOWA Class and least for the SOUTH DAKOTA Class. Further details of the method and definitions of terms are given in Reference (1).

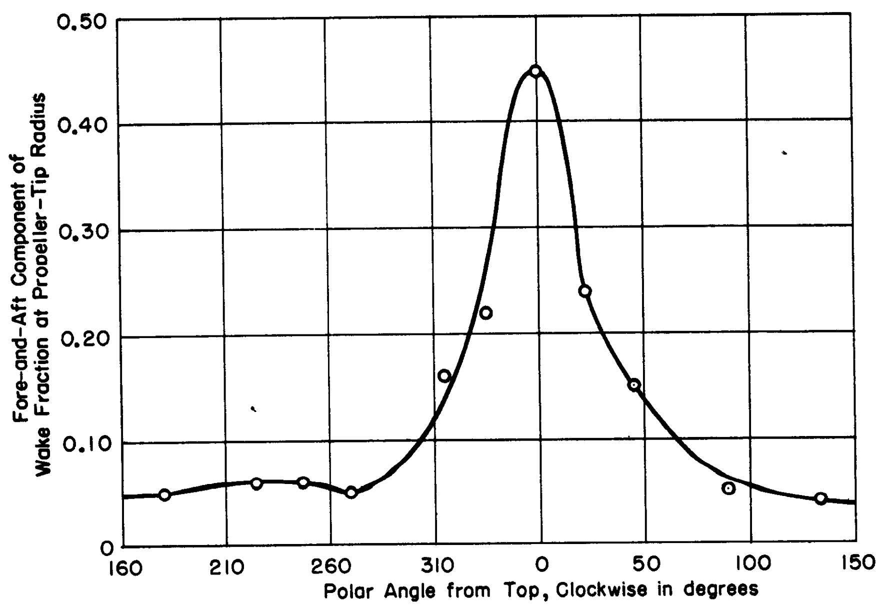


Figure 6 - Wake in Way of Inboard Propeller with Outboard Propeller Working, for Battleships of the NORTH CAROLINA Class (BB55 and 56)

These results were obtained in model tests.

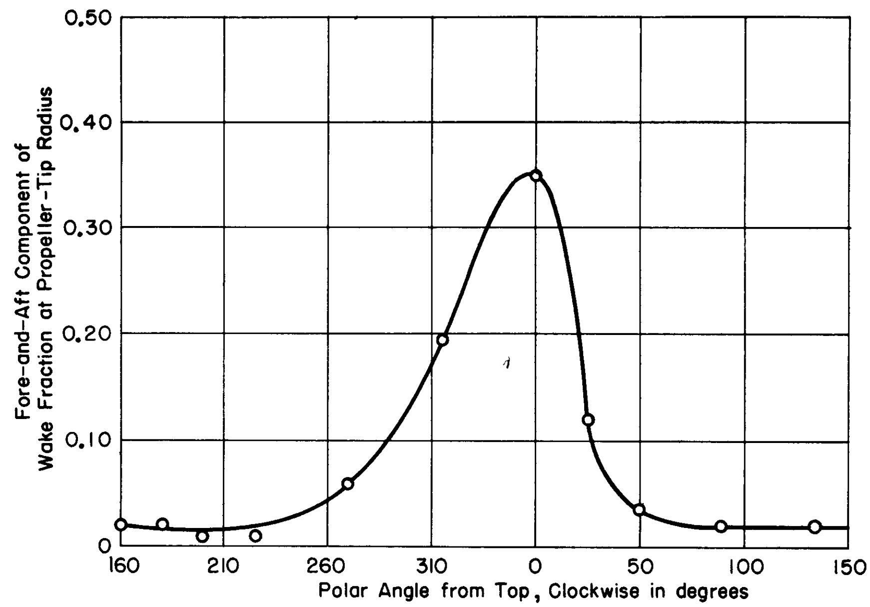


Figure 7 - Wake in Way of Outboard Propeller with Inboard Propeller Working, for Battleships of the SOUTH DAKOTA Class (BB57 through 60)

These results were obtained in model tests.

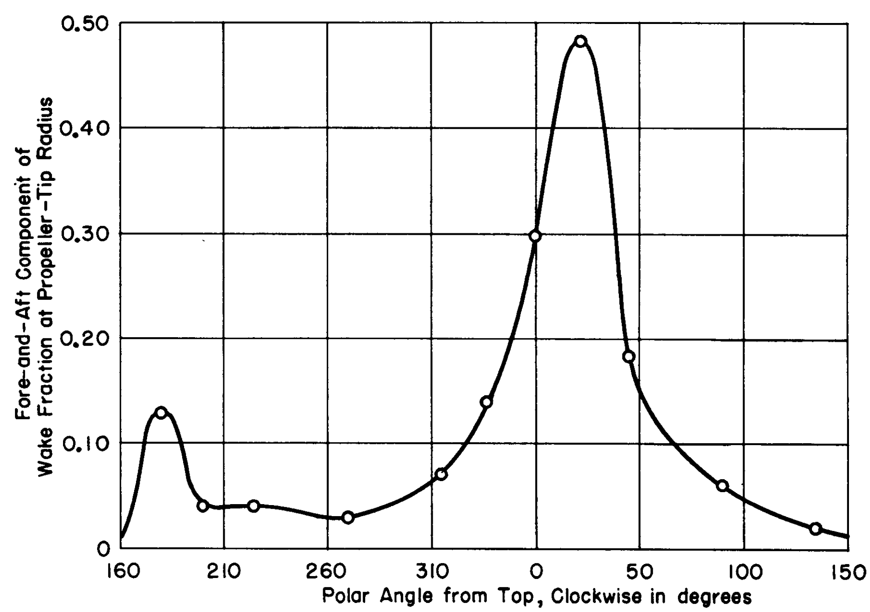


Figure 8 - Wake in Way of Inboard Propeller with Outboard Propeller Working, for Battleships of the IOWA Class (BB61 through 66)

These results were obtained in model tests.

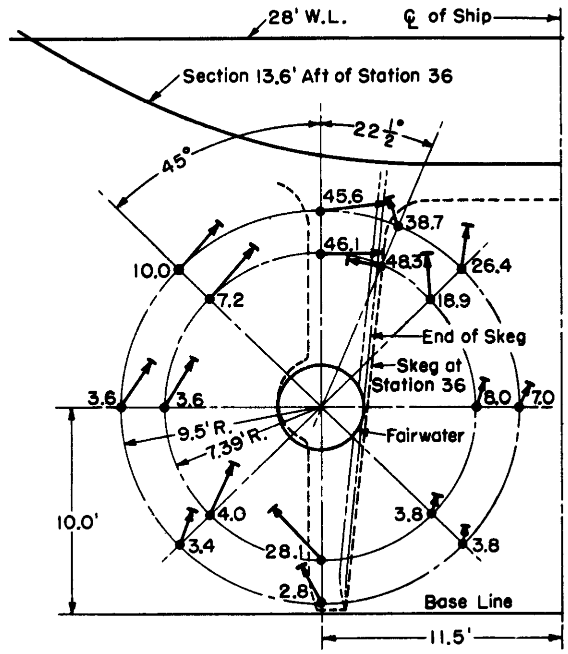


Figure 9a - Wake in Way of Inboard Propeller with Outboard Propeller Working

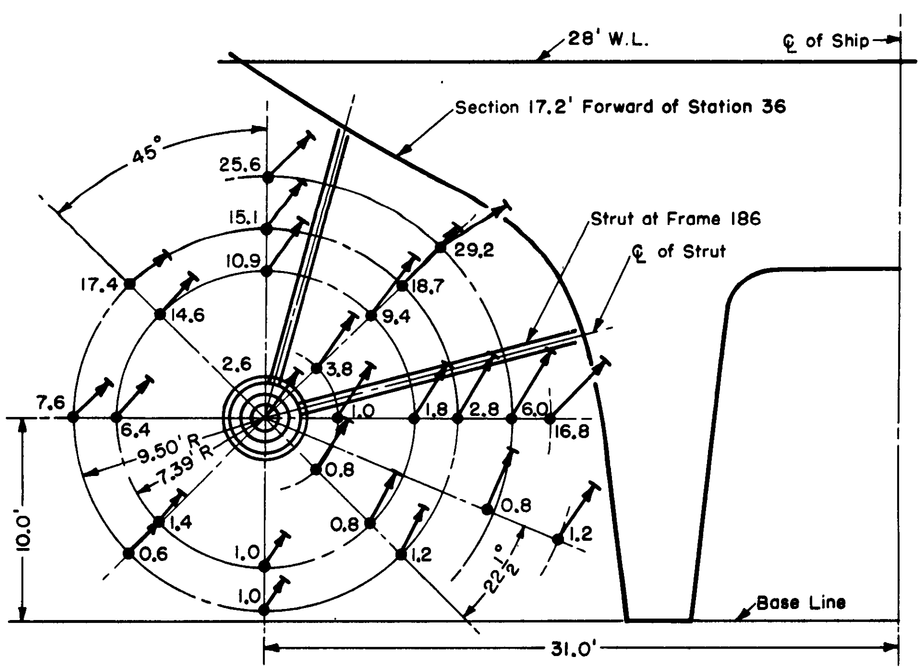


Figure 9b - Wake in Way of Outboard Propeller with Inboard Propeller Working

Figure 9 - Wake Pattersn Obtained in Model Tests of the IOWA Class (BB61 through 66)

The numbers indicate the magnitudes of the fore-and-aft components of the wake fraction. The vectors show the transverse component.

PROCEDURE IN MAKING VIBRATION MEASUREMENTS DURING SEA TRIALS

The investigation of axial vibration of the propulsion systems during the sea trials of the IOWA and NEW JERSEY followed along the general lines of that on the SOUTH DAKOTA Class. However, a change in procedure was necessitated by the fact that the vessels of the IOWA Class were equipped with thrust bearings well aft in the shaft alleys. Thrust bearings were also installed at the front of the gear cases as in the NORTH CAROLINA and SOUTH DAKOTA, but the shoes were omitted from these bearings, and on all trials mentioned in this report only the after thrust bearings were in use.

The technique in measuring the alternating thrust and vibration was the same as that used on the BB57 through 60 Class (3) except that the measurements were made both forward and aft of the thrust bearings so that a total of eight strain-gage channels was required on the oscillograph.

A Geiger vibrograph was installed on each of the after thrust bearings for recording axial vibration, and in addition dial micrometers were installed for indicating the relative motion between the thrust bearings and the hull structure. During the trials, observers recorded not only the double

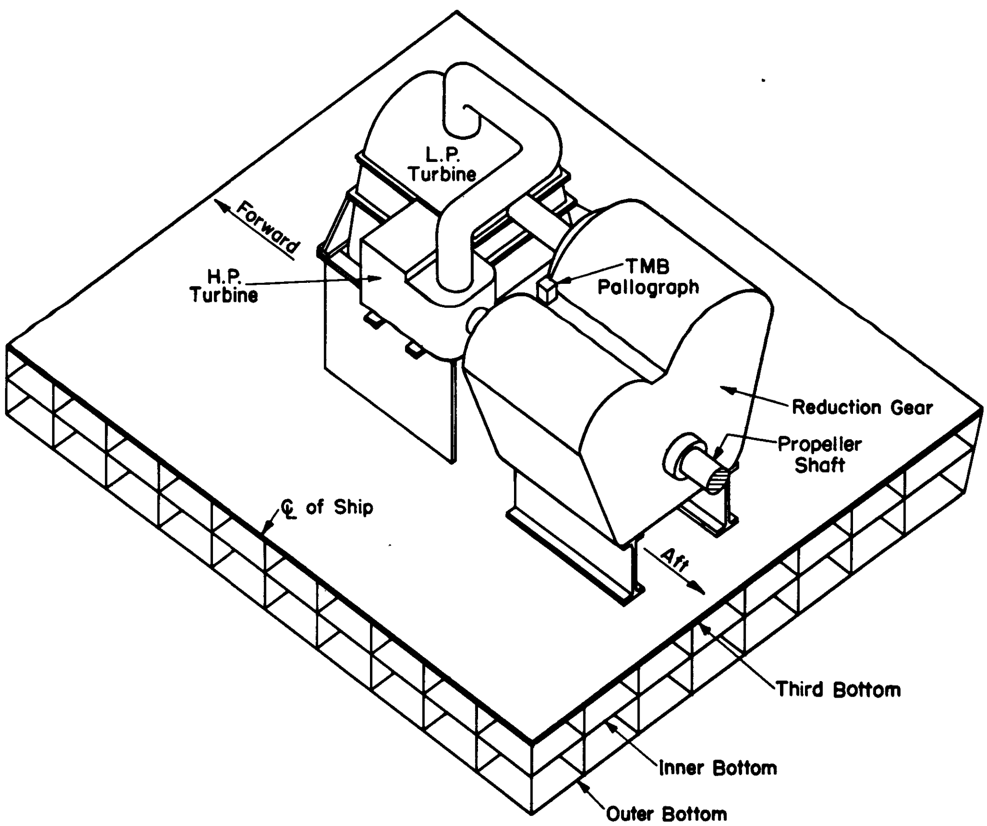


Figure 10 - Diagram Showing Location of TMB Pallograph on Port Inboard Engine of the USS IOWA (BB61)

TABLE 3

Schedule of Trial Runs on BB61 and BB62 during Which Vibration Data Were Taken

Vessel	Date	Number of Propeller Blades		Measurements Made
		Inboard	Outboard	
IOWA (BB61)	13 Apr 43	5	4	Alternating thrust in propeller shafts both forward and aft of thrust bearings; fore-and-aft movement of thrust bearings, both steady and vibratory; fore-and-aft vibration of starboard outboard and port inboard low-pressure turbines.
IOWA (BB61)	1 Jun 43	3	3	Alternating thrust in propeller shafts both forward and aft of thrust bearings; fore-and-aft movement of thrust bearings, both steady and vibratory; fore-and-aft vibration of starboard outboard and port inboard low-pressure turbines.
IOWA (BB61)	15 Jul 43	5	4	Fore-and-aft vibration of main thrust bearings and of starboard outboard and port inboard low-pressure turbines.
NEW JERSEY (BB62)	13 Oct 43	5	4	Alternating thrust in propeller shafts both forward and aft of thrust bearings; fore-and-aft movement of thrust bearings, both steady and vibratory.
IOWA (BB61)	8 Mar 45	5	4	Fore-and-aft vibratory motion of after main thrust bearings; fore-and-aft vibration of reduction-gear cases.

amplitude of the motion of the dial-micrometer needles but also the average reading, so that a curve of steady deflection plotted against propeller thrust could be obtained. Vibrographs were also installed on certain of the low-pressure turbines for the purpose of checking the vibration transmitted to the machinery as shown in Figure 10.

The schedule of trial runs on which the data given in this report were taken is summarized in Table 3.

RESULTS OF MEASUREMENTS DURING SEA TRIALS

The curves showing amplitudes of fore-and-aft vibration of the thrust bearings, plotted against shaft RPM, are given in Figures 11 through 16. These are based on Geiger vibrograph records. The dial-gage readings of amplitude were considered unreliable, owing to the flexibility of the frames supporting the gages. It was possible, however, to make an estimate of the

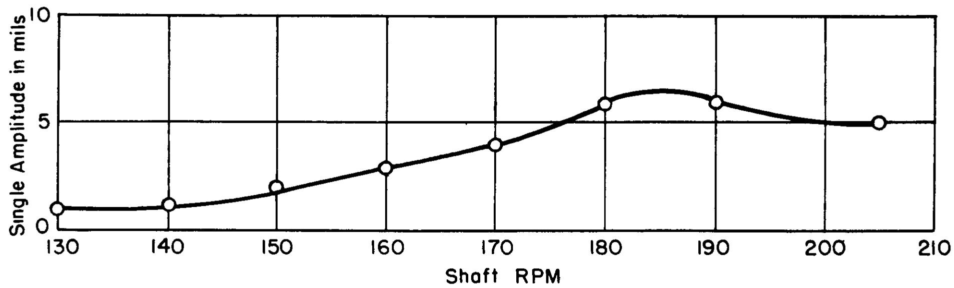


Figure 11a - Starboard Outboard Shaft, 4-Bladed Propeller

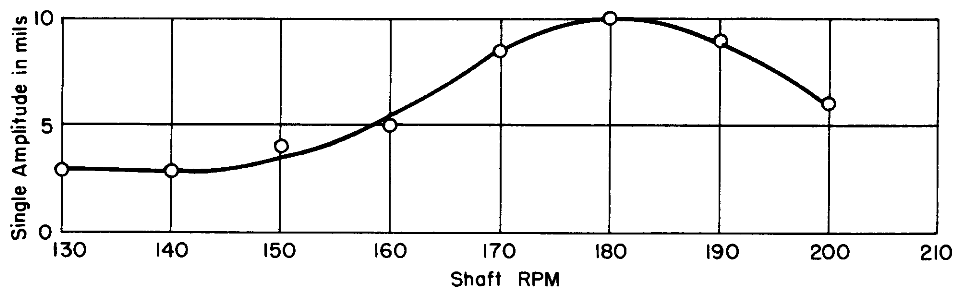


Figure 11b - Starboard Inboard Shaft, 5-Bladed Propeller

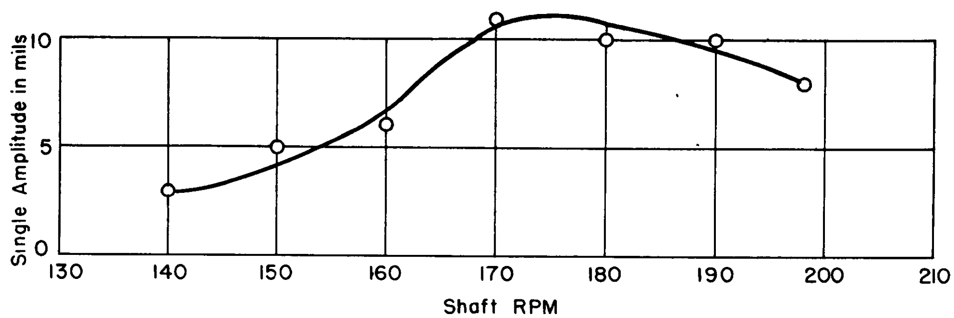


Figure 11c - Port Inboard Shaft, 5-Bladed Propeller

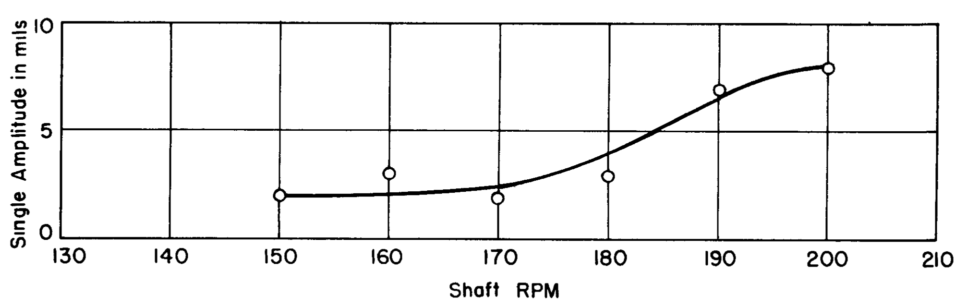


Figure 11d - Port Outboard Shaft, 4-Bladed Propeller

Figure 11 - USS NEW JERSEY (BB62) - Blade-Frequency Component of Fore-and-Aft Vibration of After Main Thrust Bearing Measured during Trial of 13 October 1943

These measurements were made with a Geiger vibrograph.

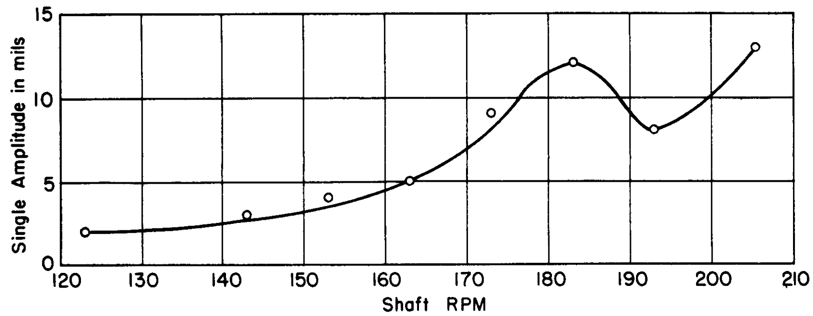


Figure 12a - Starboard Outboard Shaft, 4-Bladed Propeller

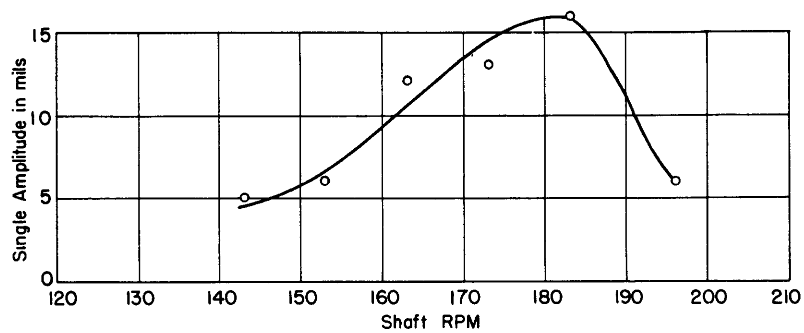


Figure 12b - Starboard Inboard Shaft, 5-Bladed Propeller

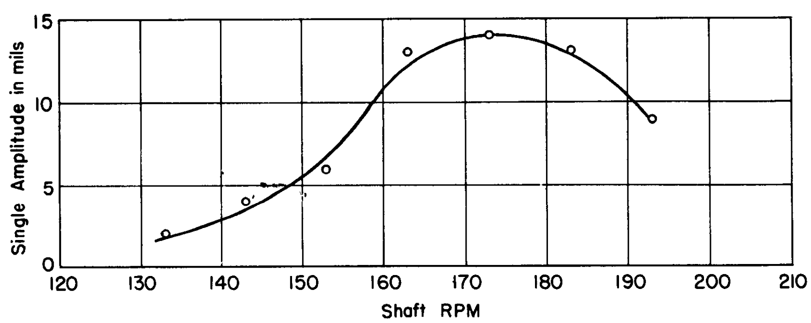


Figure 12c - Port Inboard Shaft, 5-Bladed Propeller

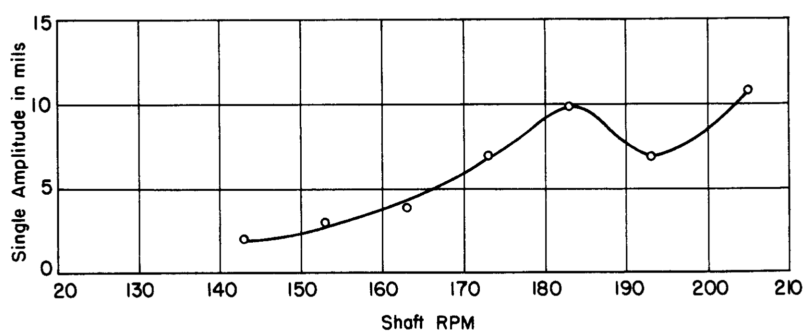


Figure 12d - Port Outboard Shaft, 4-Bladed Propeller

Figure 12 - USS IOWA (BB61) - Blade-Frequency Component of Fore-and-Aft Vibration of After Main Thrust Bearing Measured during Trial of 13 April 1943

These measurements were made with a Geiger vibrograph.

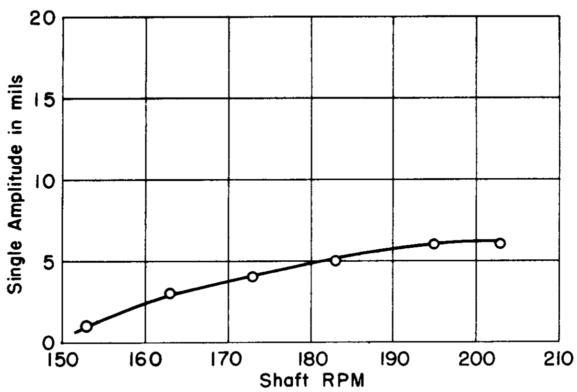


Figure 13a - Starboard Outboard Shaft,
3-Bladed Propeller

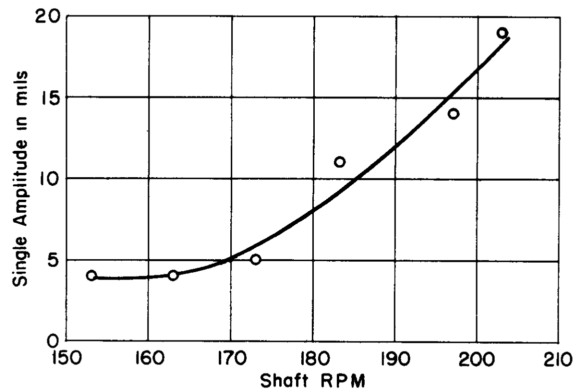


Figure 13b - Starboard Inboard Shaft,
3-Bladed Propeller

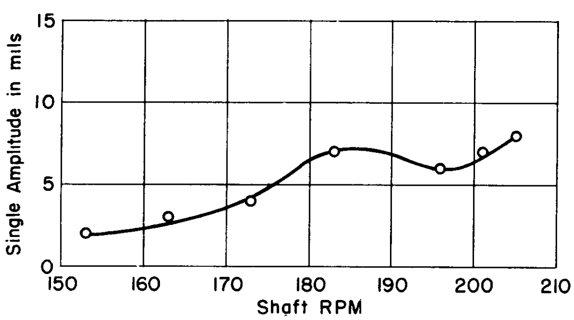


Figure 13c - Port Inboard Shaft,
3-Bladed Propeller

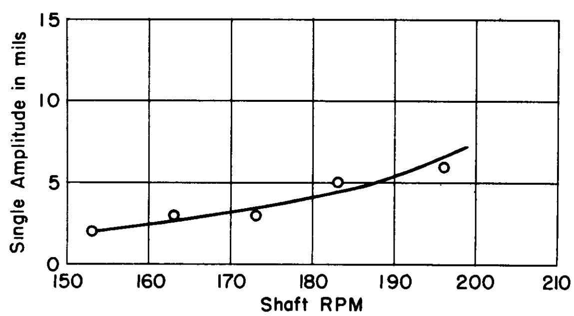


Figure 13d - Port Outboard Shaft,
3-Bladed Propeller

Figure 13 - USS IOWA (BB61) - Blade Frequency Component of Fore-and-Aft Vibration of After Main Thrust Bearing Measured during Trial of 1 June 1943

These measurements were made with a Geiger vibrograph.

spring constant of the thrust-bearing foundation from the steady deflection shown by the dial gages at known shaft speeds and thrusts. In this estimate the curves of steady thrust plotted against shaft RPM in Figure 20 were used. The average of the fore-and-aft spring constants of the thrust-bearing foundations derived from data on the four units was 13.5×10^6 pounds per inch.

The amplitudes of vibration of the thrust bearings on the IOWA were substantially the same as those on the NEW JERSEY when both vessels were fitted with 5-bladed propellers on the inboard skeg-enclosed shafts and 4-bladed propellers on the outboard shafts. The first-mode resonances occurred at about the same frequency on both vessels, with the exception of the port outboard shaft; this shaft showed a resonance at 183 RPM on the IOWA, Figure 12d, whereas on the NEW JERSEY a definite peak did not appear, Figure 11d. The maximum single amplitude observed on all thrust bearings with the 4- and 5-bladed propeller arrangement was 16 mils, which was found at resonance on the starboard inboard thrust bearing of the IOWA; see Figure 12b.

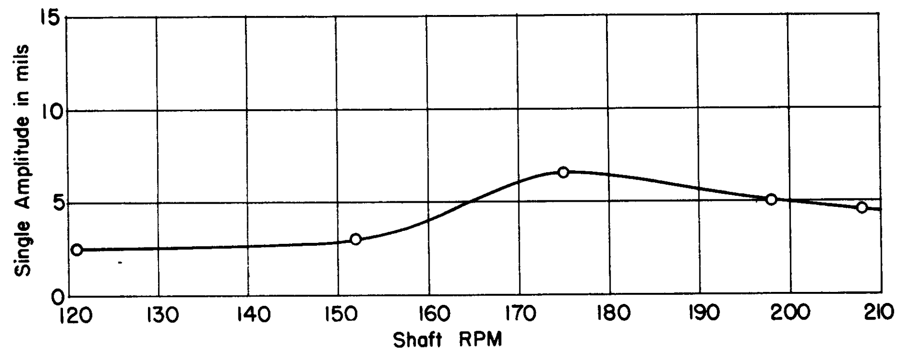


Figure 14a - Starboard Outboard Shaft, 4-Bladed Propeller

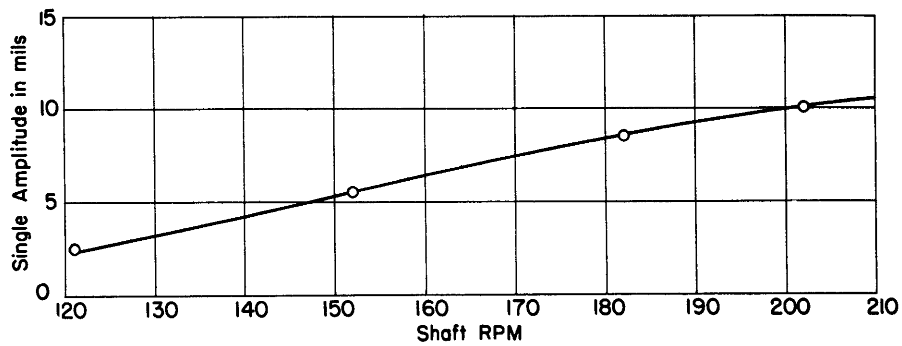


Figure 14b - Starboard Inboard Shaft, 5-Bladed Propeller

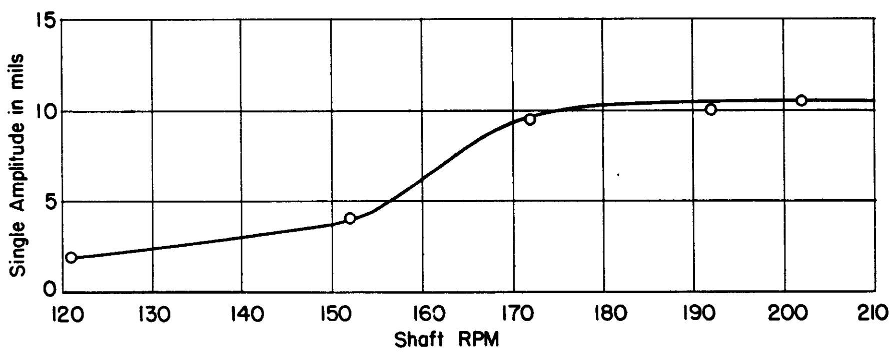


Figure 14c - Port Inboard Shaft, 5-Bladed Propeller

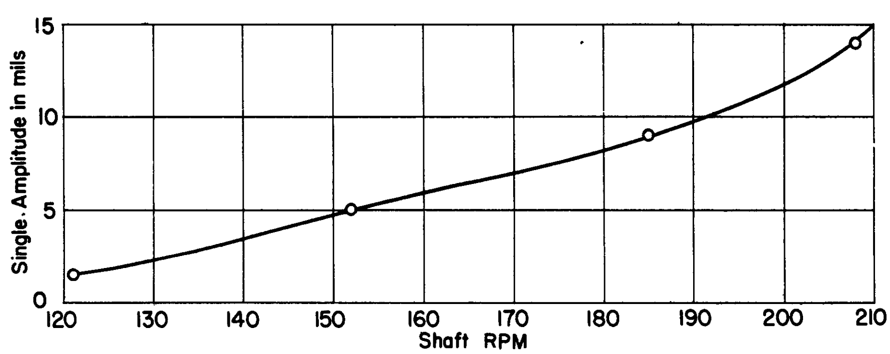


Figure 14d - Port Outboard Shaft, 4-Bladed Propeller

Figure 14 - USS IOWA (BB60) - Fore-and-Aft Vibration of After Main Thrust Bearing Measured during Post-Repair Trial of 8 March 1945

These measurements were made with Sperry pickups.

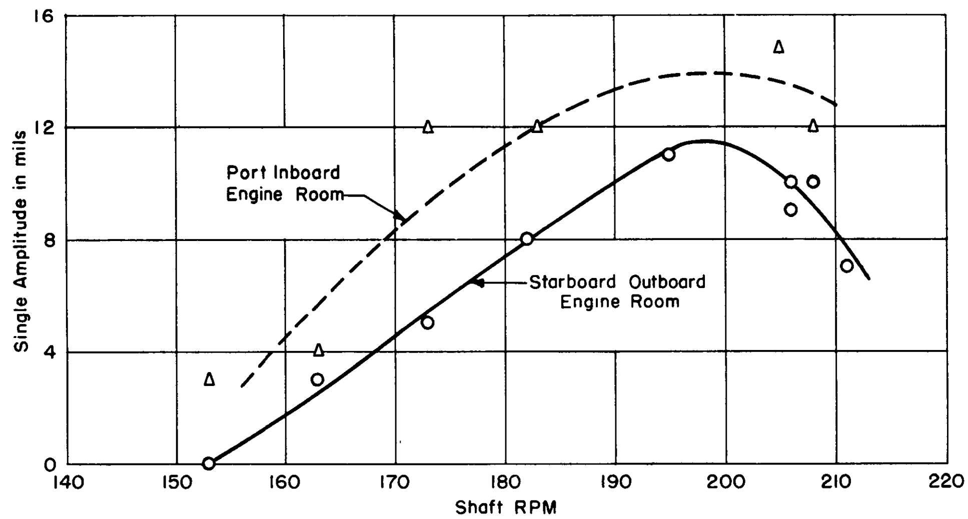


Figure 15 - USS IOWA (BB61) - Fore-and-Aft Vibration of Forward Bearing of Low-Pressure Turbines Measured during Trial of 1 June 1943

These measurements were made with a Geiger vibrograph. Three-bladed propellers were mounted on all shafts.

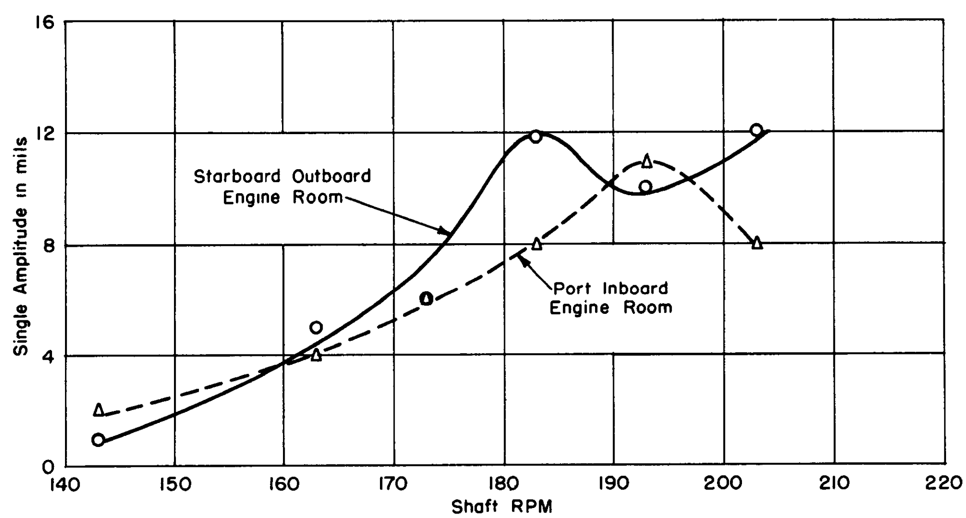


Figure 16 - USS IOWA (BB61) - Fore-and-Aft Vibration of Forward Bearing of Low-Pressure Turbines Measured during Trial of 15 July 1943

These measurements were made with a Geiger vibrograph. Four-bladed propellers were mounted on the outboard shafts, and three-bladed propellers on the inboard shafts.

With 3-bladed propellers on the IOWA the lowest-mode resonance was not reached, and consequently the amplitudes of vibration were generally lower, Figure 13, except for the starboard inboard unit, where a single amplitude of 19 mils was reached on the thrust bearing at 203 RPM.

The amplitude measurements made on the main thrust bearings during the post-repair trials of the IOWA on 8 March 1945, Figure 14, failed to reveal distinct resonance points, and the amplitudes of vibration of the main thrust bearings were much smaller than those on the initial trials. This may have been due to the increased friction in the stern-tube bearings, the rubber bearings having been replaced by lignum vitae during the repairs, but confirmation of this explanation is lacking. If the small amplitudes were due to stern-tube bearing friction, they should be expected to increase as the bearings wear in since at resonance the amplitude is limited only by the damping.

In addition to the vibration instruments installed on the after thrust bearings, vibrographs were also installed on the forward bearings of the starboard outboard and port inboard low-pressure turbines during certain trial runs. The results on the IOWA with 3-bladed propellers are shown in Figure 15 and with the combination of 4- and 5-bladed propellers in Figure 16. Neither of these figures indicates excessive vibration. The frequencies at which the peak values occur agree with the resonance frequencies found on the main thrust bearings only for the starboard outboard unit for the trial of 15 July 1943. However, as shown theoretically by means of the electrical analogy, see page 27 and Figure 29, the turbines should vibrate only a small amount at the first-mode resonance if there is only friction coupling between them and the gears. The peaks observed may be due to local resonance at the point of measurement on the turbines, but no further data were obtained to confirm this.

Figures 17 and 18 give the alternating component of thrust in per cent of steady thrust for the IOWA, fitted with 4- and 5-bladed propellers. These curves give the thrust variation, not at the propeller, but in the shaft itself at the point where metalectric strain gages were installed, namely, near the thrust bearing. The steady-thrust values as derived from the model tests are given in Figure 20. At frequencies near resonance the thrust variations in the shaft are greater than at the propeller. In Figure 17, which shows plotted data for the outboard shafts, resonance peaks are not clearly defined. In Figure 18, which shows plotted data for the inboard shafts, a resonance peak occurs at about 175 RPM on the port-inboard shaft and at about 163 RPM on the starboard-inboard shaft, with a second peak at 183 RPM. It should be noted that, although the alternating thrust forward of the thrust bearing is less than that aft of the bearing, it is by no means negligible and has peak values which are reached at approximately the same frequency as for the after section. This condition is understandable if the vibratory system is assumed to be represented by that shown in Figure 5. When this

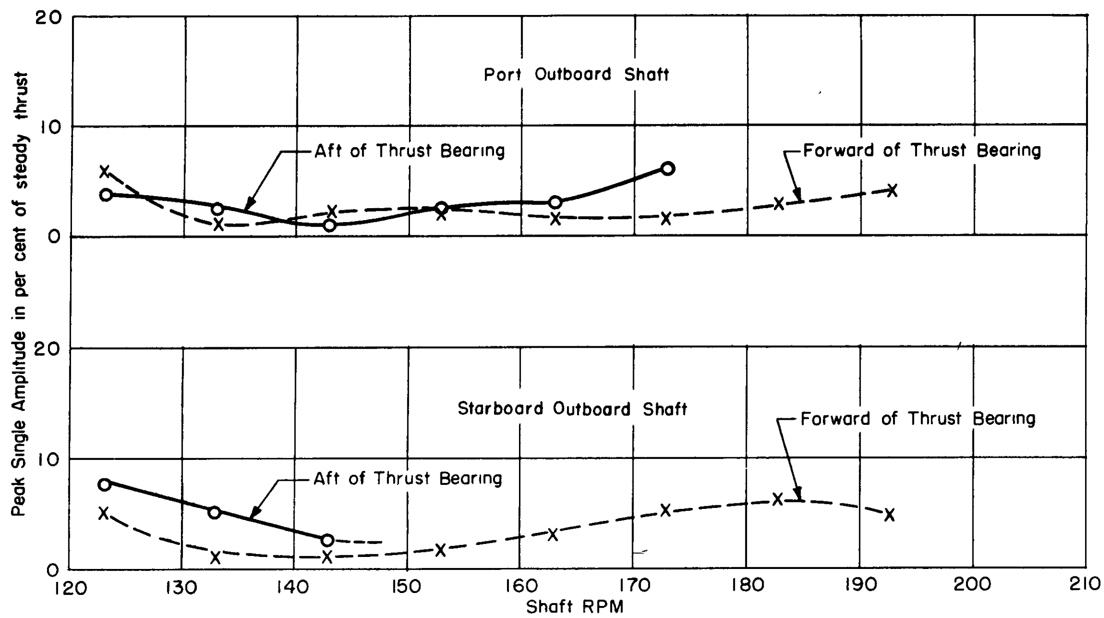


Figure 17 - USS IOWA (BB61) - Blade-Frequency Component of Alternating Thrust in Outboard Shafts, in Per Cent of Steady Thrust, Observed during Trial of 13 April 1943

Four-bladed propellers were mounted on the outboard shafts.

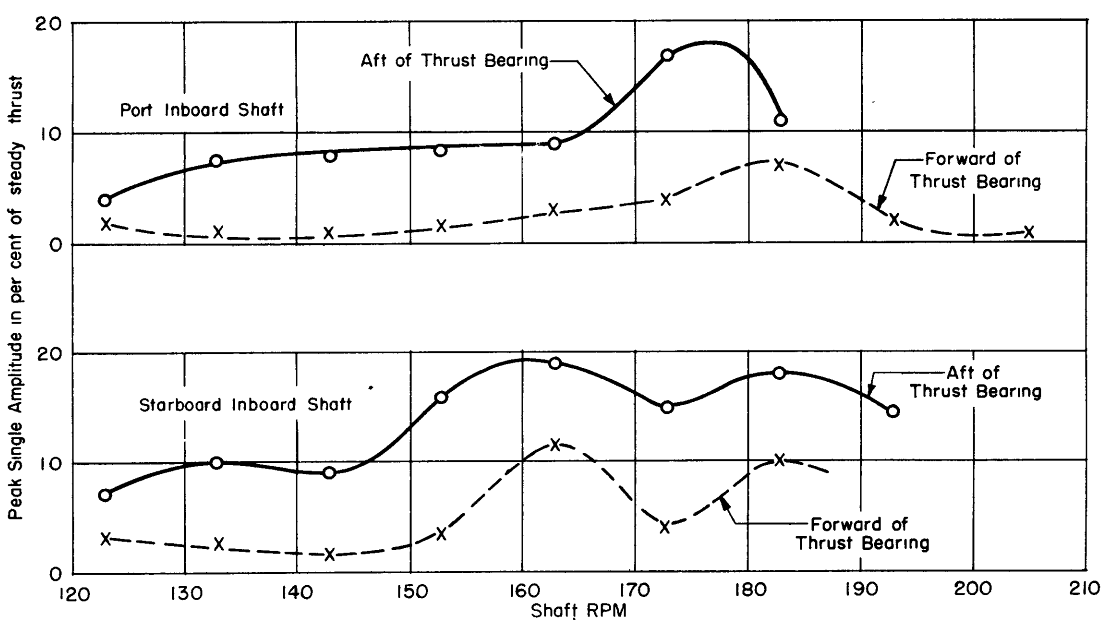


Figure 18 - USS IOWA (BB61) - Blade-Frequency Component of Alternating Thrust in Inboard Shafts, in Per Cent of Steady Thrust, Observed during Trial of 13 April 1943

Five-bladed propellers were mounted on the inboard shafts.

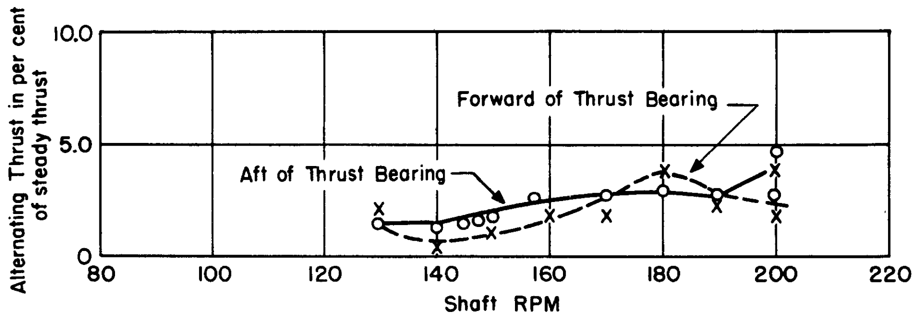


Figure 19a - Starboard Outboard Shaft, 4-Bladed Propeller

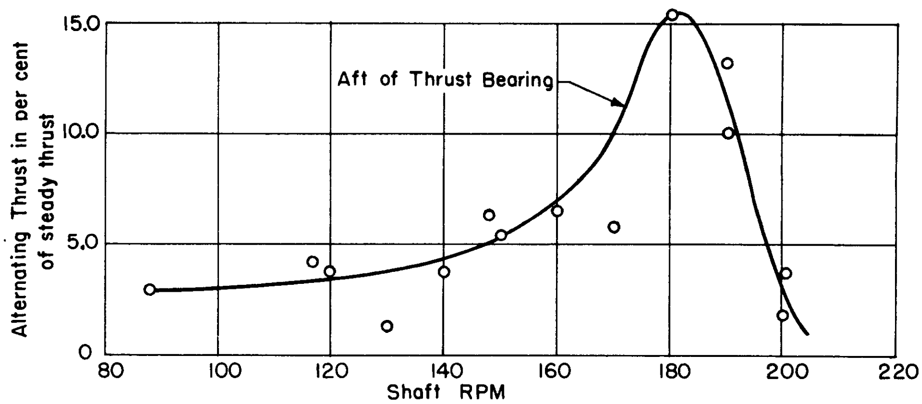


Figure 19b - Starboard Inboard Shaft, 5-Bladed Propeller

No data were taken forward of the thrust bearing on this shaft.

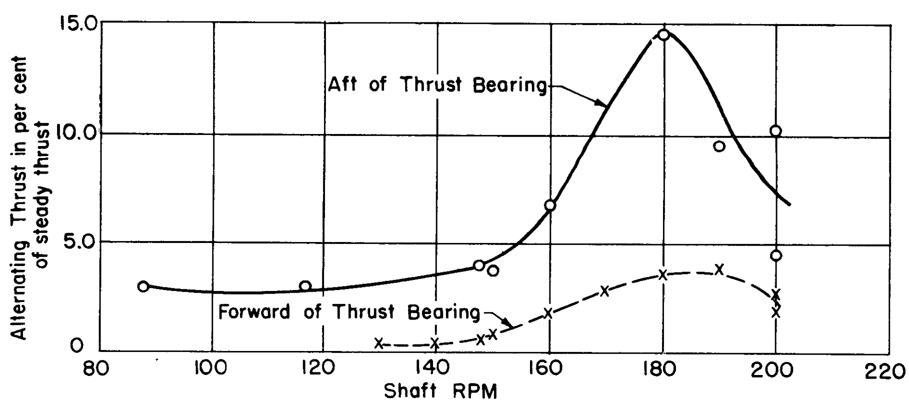


Figure 19c - Port Inboard Shaft, 5-Bladed Propeller

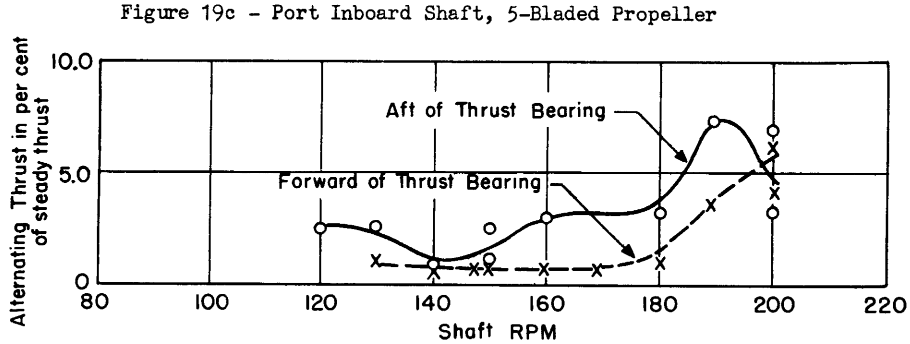


Figure 19d - Port Outboard Shaft, 4-Bladed Propeller

Figure 19 - USS NEW JERSEY (BB62) - Alternating Thrust, in Per Cent of Steady Thrust, Observed during Trials of 13 October 1943

system vibrates in one of its normal modes, the stresses on either side of the bearing will depend on the amplitudes of m_p and m_{b_g} relative to m_b, as well as on the lengths of the respective shafts. As shown in Table 2, the theoretical amplitudes at the reduction gear for the undamped system may be as high as four times the amplitude at the thrust bearing in the first mode vibration. Hence the alternating stress forward of the thrust bearing can be several times as great as that aft of the thrust bearing when resonance is reached. In the actual case, as contrasted with the ideal system represented in Figure 5, the stresses will depend on the driving force and the damping, that is, on the thrust variation at the propeller and the friction in the bearings and couplings.

Figure 19 shows the alternating thrust as a percentage of steady thrust measured on the NEW JERSEY during the trial of 13 October 1943. At this time the NEW JERSEY was fitted with 4- and 5-bladed propellers so that conditions were the same as on the IOWA trial of 13 April 1943. The results as shown by comparison of Figures 17 and 18 with Figure 19 are generally similar except that the double peak shown for the starboard inboard shaft on the IOWA does not appear on the curve for the NEW JERSEY. As the single peak for the NEW JERSEY is in agreement with the vibrograph data taken on the thrust bearing, more credence is given to the NEW JERSEY data.

Measurements of alternating thrust were also made during the trial of the IOWA on 1 June 1943, at which time all shafts were fitted with 3-bladed propellers. Because of the lowering of the blade frequencies by the use of 3-bladed propellers, resonance was not reached on any of the propulsion systems. The records were therefore analyzed also for the double-blade-frequency component, but the results were inconclusive. The highest blade-frequency component of thrust variation observed was about 11 per cent of the steady thrust at 203 RPM on the starboard-inboard shaft.

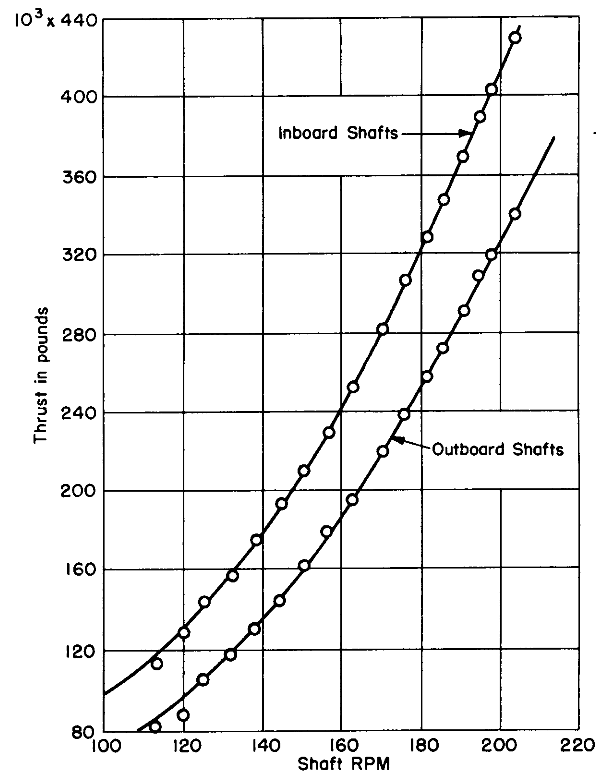


Figure 20 - USS IOWA (BB61) - Thrust per Shaft As Derived from Tests on Model 3738

Alternating-thrust measurements were made also on the model of the BB61 through 66 Class by the same technique as that used for the BB55 and 56 Class; see Figure 20. The average values from all tests indicated that the thrust variation at the propeller was 12.4 per cent of the steady thrust for the inboard shafts with 2-bladed propellers, 8.5 per cent inboard and 3.9 per cent outboard with 3-bladed propellers, 4.8 per cent for all shafts with 4-bladed propellers, and 6.8 per cent inboard and 3.9 per cent outboard with 5-bladed propellers.

GENERAL CONSIDERATIONS REGARDING AXIAL VIBRATION ON RECENT BATTLESHIPS

In commenting on the axial vibration problem on all three recent classes of battleships, namely, the NORTH CAROLINA, the SOUTH DAKOTA, and the IOWA Classes, it seems desirable first to point out that although corrective measures have been found for reducing the vibration below dangerous levels on all ships, the vibration has not been eliminated and if further construction of large naval vessels were under consideration this type of vibration would have to be taken into account. The tests described in this report show the value of determining the elastic constants of the vibratory systems and of using an electrical analogy to determine the steady-state response of such vibratory systems at an early stage in the design.

Axial vibration exists on all recent battleships, and on the SOUTH DAKOTA and IOWA Classes it is believed to have been the cause of excessive wear in the turbine couplings. The adoption of 5-bladed propellers behind the skegs has resulted in great improvement in engine-room conditions on all vessels tested so far.

The installation of thrust bearings aft in the shaft alleys has given satisfactory results on the IOWA as far as externally observable vibrations are concerned. No data are available to show what the vibratory conditions would be if the thrust bearings located at the forward end of the bull gear in the IOWA Class were in use. It was not found necessary to use these bearings in view of the satisfactory performance of the after bearings.

Subsequently, however, there has been evidence of wear in the turbine couplings of the IOWA and NEW JERSEY, indicating considerable fore-and-aft vibration of the gears and pinions. The presence of such vibration is also supported by theoretical calculations. The only practicable remedy for this condition at the present time on these completed vessels is to replace the worn turbine couplings by couplings specially designed to resist wear, as adopted for the SOUTH DAKOTA Class. This replacement program is underway, and with these modifications it is expected that wear of the couplings will be reduced to negligible proportions.

VALUE OF THE CONSTANTS OF THE PROPULSION SYSTEMS

Experience to data with axial vibration of propulsion systems on recent battleships has shown the value of determining the elastic constants of the vibratory systems and of making use of the data already on hand to forecast the critical speeds and frequencies of new designs which are under consideration.

It is believed that practically all axial vibration of geared-turbine propulsion systems can be adequately treated mathematically by making use of one of the six model systems illustrated in Figure 21. More elaborate systems have been proposed in which the machinery is broken down into more elements than in any of these models, but at least for the first two modes of axial vibration the simpler systems here illustrated appear to be adequate inasmuch as the higher modes have not so far been noticeable.

Mathematically Model 2 and Model 4 are identical, but separate models and separate symbols are used to make clear that for Model 2 the main thrust bearing is at the forward end of the shaft and for Model 4 it divides the shaft into two parts. The difference between Model 4 and Model 3 is that in Model 4 the machinery is assumed to move as a unit, whereas in Model 3 the gears are assumed not to be coupled axially to the rest of the machinery.

Following are the definitions of symbols used in Figure 21:

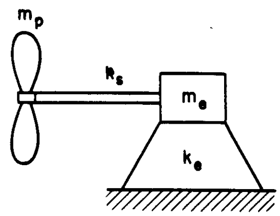
m_p is the mass of the propeller, plus 60 per cent co allow for the virtual mass of the entrained water, plus one-half the mass of the shaft between the propeller and the thrust bearing.

m_e is the effective mass of the entire propulsion machinery at the level of the shaft, including mass of gears, turbines, condenser, one-fourth the mass of the machinery foundations, and one-half the mass of the entire shaft when the thrust bearing is in the gear case, or one-half the mass of the shaft between the gear case and the after thrust bearing when that bearing is located in the shaft alley.

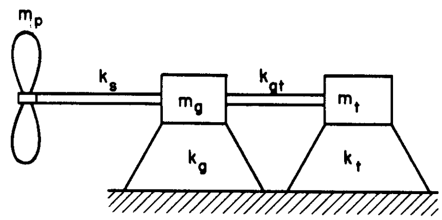
m_g is the mass of the entire gear unit, plus one-fourth the mass of the gear foundation, plus one-half the mass of the shaft.

m_t is the mass of both turbines, plus the mass of the wet condenser, plus one-fourth the mass of the turbine foundations.

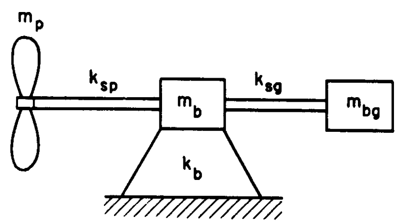
m_b is the mass of the after thrust bearing, plus one-fourth the mass of the after thrust-bearing foundation, plus one-half the mass of the shaft aft of the thrust bearing, plus one-half the mass of the shaft forward of the thrust bearing.



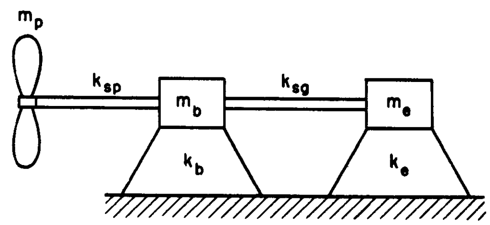
Model 1



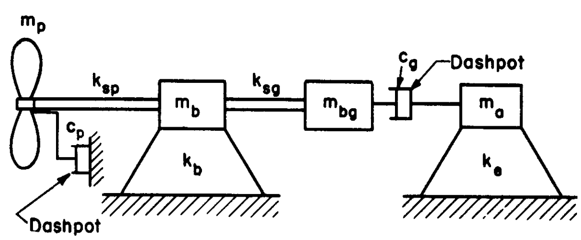
Model 2



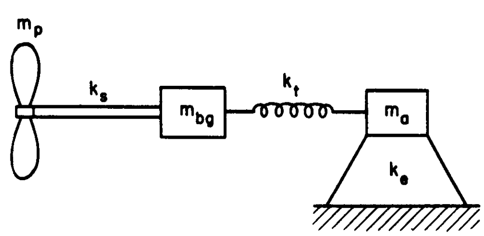
Model 3



Model 4



Model 5



Model 6

Figure 21 - Simple Model Systems Useful for Computing Natural Frequencies of Axial Vibration of Geared-Turbine Propulsion Systems

m_{bg} is the mass of the bull gear and all pinions and gears constrained to follow it in its fore-and-aft motion, plus one-half the mass of the shaft between the gear and the after thrust bearing.

m_a is the effective mass of the entire propulsion machinery and housings at the level of the shaft, less the mass of all gears and pinions that are free to move axially together, plus one-fourth the mass of the machinery foundations.

k_s is the axial spring constant of the entire propeller shaft.

k_e is the effective fore-and-aft spring constant of the combined turbine and gear foundations referred to the shaft level.

k_g is the fore-and-aft spring constant of the gear foundations referred to the shaft level.

k_t is the fore-and-aft spring constant of the turbine foundations referred to the shaft level.

k_{gt} is the fore-and-aft spring constant of the elastic connection between the turbines and the gear case.

k_b is the fore-and-aft spring constant of the after thrust-bearing foundation referred to the level of the shaft centerline.

k_{sp} is the axial spring constant of the part of the shaft between the after thrust bearing and the propeller.

k_{sg} is the axial spring constant of the part of the shaft between the after thrust bearing and the gears.

c_p is the constant of equivalent viscous damping due to the friction at the propeller and in the stern-tube bearing.

c_g is the constant of equivalent viscous damping due to the frictional resistance to fore-and-aft motion in the bearings and reduction gears.

The natural frequencies of any of these models may be determined from algebraic equations derived from the differential equations for free vibration, or by Holzer's method (2), or by means of the electrical analogy. However, in making estimates of the amplitudes of the system at the resonance frequencies it is necessary to assume values of friction, and few experimental data are available on which to base such assumptions. Hence any experimental data on the amount of damping are of great value.

The estimated values of the constants of the propulsion systems for the recent classes of naval vessels on which this type of vibration has been considered are summarized in Table 4. The experimental and the computed

TABLE 4

System Constants and Calculated and Measured Frequencies of Axial Vibration of Propulsion Systems of Recent Classes of Naval Vessels

Vessel	Propulsion Unit	Model System	N_1 CPM		N_2 CPM		m_p	m_e	m_g	m_t	m_b	m_{bg}	m_a	k_e $\times 10^{-6}$
			Experimental	Theoretical	Experimental	Theoretical								
BB55 and 56	Starboard Outboard	1	510	515	970	980	364	1025						1.3
	Starboard Inboard	1	600	600	1020	1030	315	1025						1.68
	Port Inboard	1	680	695	1030	1180	280	1243						2.17
	Port Outboard	1	600	550	1090	1080	329	1025						1.6
BB57 through 60	Starboard Outboard	1	520	530	900	900	419	1200						1.83
	Starboard Inboard	1	650	620	1100	1100	350	1130						2.75
	Port Inboard	1	700	660	1240	1210	318	1100						3.39
	Port Outboard	1	600	565	980	980	390	1180						2.11
BB57 through 60	Starboard Outboard	6	520	483	900	928	419					454	750	1.83
	Starboard Inboard	6	650	565	1100	1090	350					380	750	2.75
	Port Inboard	6	700	586	1240	1184	318					353	750	3.39
	Port Outboard	6	600	509	980	978	390					426	750	2.11
BB61 through 66	Starboard Outboard	3	732	648		1146	361				410	495		
	Starboard Inboard	3	900	846		1260	370				310	392		
	Port Inboard	3	865	939		1602	370				246	326		
	Port Outboard	3	732	747		1158	361				344	431		
CV9	Starboard Outboard	2	560	645		1630	310		350	205				
	Starboard Inboard	2	720	850		1422	254		294	205				
	Port Inboard	2	720	850		1422	254		294	205				
	Port Outboard	2	560	645		1630	310		350	205				
CVB41	Starboard Outboard	3		812		1220	327				615	327		
	Starboard Inboard	3		540		834	367				615	514		
	Port Inboard	3		673		896	367				615	422		
	Port Outboard	3		812		1220	327				615	327		
CB1	Starboard Outboard	1	640	776		1710	244	545						1.87
	Starboard Inboard	1	704	840		2100	210	506						2.32
	Port Inboard	1	684	840		2100	210	506						2.32
	Port Outboard	1	640	776		1710	244	545						1.87

TABLE 4 (Continued)

$k_s \times 10^{-6}$	$k_t \times 10^{-6}$	$k_{gt} \times 10^{-6}$	$k_b \times 10^{-6}$	$k_{sp} \times 10^{-6}$	$k_{sg} \times 10^{-6}$	Number of Blades	Maximum RPM
8.8						4	200
8.8						5	200
12.6						5	200
8.8						4	200
8.0						5	200
8.0						4	200
8.0						4	200
8.0						5	200
8.0	10.0					5	200
8.0	10.0					4	200
8.0	10.0					4	200
8.0	10.0					5	200
			13.5	7.4	3.02	4	200
			13.5	7.3	5.89	5	200
			13.5	7.3	12.8	5	200
			13.5	7.4	3.87	4	200
	5.0	10.0	15.0			4	270
	5.0	10.0	15.0			4	270
	5.0	10.0	15.0			4	270
	5.0	10.0	15.0			4	270
			13.0	4.02	7.42		200
			13.0	3.58	2.03		200
			13.0	3.58	3.17		200
			13.0	4.02	7.42		200
15.0						4	265
15.0						4	265
15.0						4	265
15.0						4	265

values of the natural frequencies are also given in that table. The model system used for each calculation is indicated in accordance with the numbering of the systems as shown in Figure 21.

By judicious use of this information it may be possible to forecast critical frequencies for proposed new designs with a fair degree of accuracy.

USE OF ELECTRICAL ANALOGY

The electrical analogy has proved a great aid in determining the steady-state response of vibratory systems of the type considered in this report. The analogy is especially helpful when damping resistance is to be taken into account, in which case the mathematical solution becomes much more complicated. However, it is only when damping is considered that any estimate can be made of the amplitude produced at resonance by a given driving force.

Although the IOWA was fitted with thrust bearings both at the reduction gear and in the shaft alley, no information is available as to the performance of the ship with the forward thrust bearings in use. As previously noted, when the thrust bearing is located astern, the gears and pinions are free to move axially in their bearings, and this motion is taken up in the flexible couplings between the turbines and the gears. If the frictional resistance to the fore-and-aft motion of the gears is neglected altogether, the system is represented by Model 3, Figure 21, whereas if a friction coupling is assumed to act between the gears and the gear case the system is represented by Model 5, provided the rest of the machinery, i.e., turbines and condenser, is assumed to move with the gear case. The effect of various amounts of friction coupling is readily demonstrated by means of the electrical analogy, for the equivalent resistance can be varied from zero to infinity by means of a decade resistance box.

The rules laid down in Reference (5) are satisfied by the mechanical and electrical analogies for Model 5 given in Figure 22. Regardless of the procedure by which the electrical analogy is set up, it is necessary only to write down the differential equations for the displacements of the masses in the mechanical system and for the charges moving through the elements in the electrical circuit to prove the equivalence.

In the true electrical analogy the value of any electrical quantity in terms of its fundamental units would be the same numerically as the value of the corresponding mechanical quantity in terms of its fundamental units. When the values for the true analogy are tabulated, it is seen that the required inductances in the electrical circuit are too large for convenience.

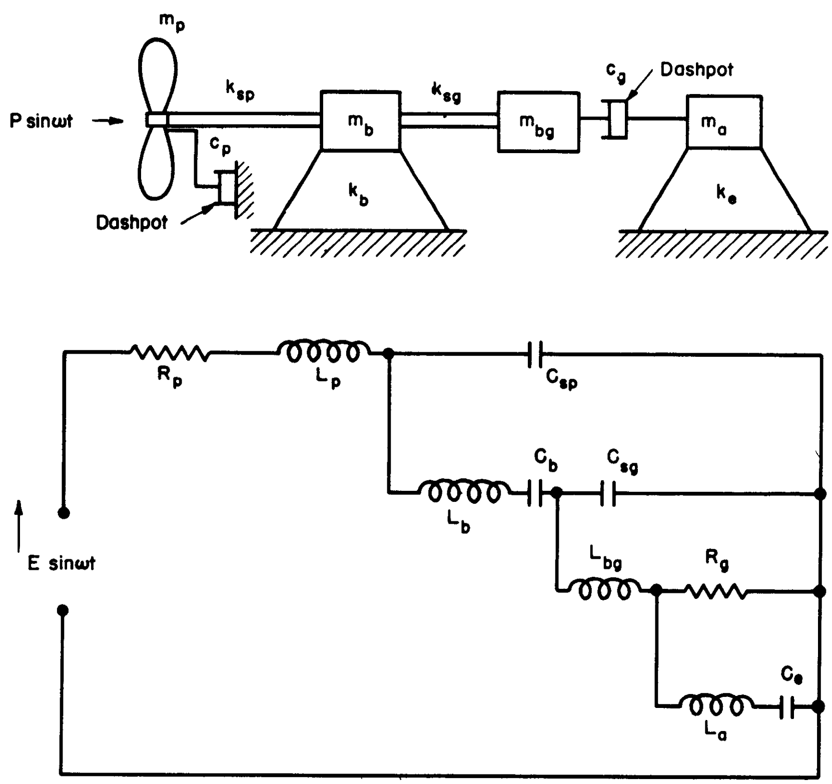


Figure 22 - Model 5 and its Electrical Analogy

R is resistance, *L* is inductance, and *C* is capacitance.

Furthermore, the frequencies to be explored are too low for satisfactory electrical measurements. The desired frequency range for the problem in question would normally be 300 to 3000 CPM or 5 to 50 CPS. Electrically it has proved much more convenient to work with the frequency range of 50 to 500 CPS.

Fortunately, the electrical analogy can be modified by various transformations without impairing its usefulness. Since the impedance of the network, which in this case is made up of lumped elements, can be expressed ultimately, that is, after clearing fractions, as a sum of complex terms all of which are of the first degree in R, L, or $1/C$, it follows that if the impedance of each element is reduced to one-tenth its former value, the impedance of the network will be reduced to one-tenth its former value. The impedances of inductances and capacitances are functions of frequency. The impedance of an inductance is $j\omega L$, where j is equal to $\sqrt{-1}$ and ω is the circular frequency or 2π times the natural frequency, that of a capacitance is $-j/\omega C$, and that of a resistance is R. Hence a modified analogy will have one-tenth the impedance of the true analogy at 10 times the frequency if all its inductances have one-hundredth their value in the true analogy, if all capacitances are kept the same as in the true analogy, and if all resistances have one-tenth their value in the true analogy. Since it is the electrical charge that is analogous to mechanical displacement or amplitude, whereas the electrical impedance is universally defined as the ratio of impressed voltage to current, it follows that the modified analogy will have the same alternating charge for the same impressed voltage as the true analogy. This is true because the current I is the time derivative of the charge q, that is, $I = q\omega$. Since the modified analogy has one-tenth the impedance, it will have 10 times the current for the same impressed voltage as the true analogy, but since the frequency is 10 times as high the charge will be the same in both cases. Hence it follows that for 1 volt impressed on the modified analogy an alternating charge of 1 coulomb represents an amplitude of 1 inch for a 1-pound driving force in the mechanical system. It must be remembered that in the electrical system maximum and not root-mean-square values correspond to amplitudes as used mechanically, and hence ordinary electrical-meter readings must be converted from RMS (root-mean-square) to peak single values before they are converted to mechanical values. The mechanical and electrical quantities representing Model 5, Figure 22, for the starboard inboard propulsion unit of the IOWA are given in Table 5. In this table the following electrical symbols are used: E for voltage, R for resistance, L for inductance, C for capacitance, and q for charge.

By means of the modified analogy shown in Figure 22, and with the values listed in Table 5, the response of the starboard inboard propulsion

TABLE 5

Mechanical and Electrical Quantities Used in Representing the Starboard Inboard Propulsion Unit of Model 5 of Figure 22

Mechanical		True Electrical Analogy		Modified Electrical Analogy Used at Frequencies 10 Times As High	
Quantity	Numerical Value	Quantity	Numerical Value	Quantity	Numerical Value
Frequency range	5 to 50 CPS			Frequency range	50 to 500 CPS
P	30,000 pounds	E'	30,000 volts (peak single)	E	10 volts (RMS)
c_p	6000 lb-sec/in	R'_p	6000 ohms	R_p	600 ohms
m_p	370 lb-sec²/in	L'_p	370 henries	L_p	3.70 henries
k_{sp}	7.3×10^6 lb/ih	C'_{sp}	$\frac{1}{7.3 \times 10^6}$ farad	C_{sp}	0.137×10^{-6} farad
k_{sg}	5.9×10^6 lb/in	C'_{sg}	$\frac{1}{5.9 \times 10^6}$ farad	C_{sg}	0.170×10^{-6} farad
m_b	310 lb-sec²/in	l'_b	310 henries	L_b	3.10 henries
k_b	13.5×10^6	C'_b	$\frac{1}{13.5 \times 10^6}$ farad	C_b	0.0741×10^{-6} farad
m_{bg}	392	L'_{bg}	392 henries	L_{bg}	3.92 henries
c_g	2000 lb-sec/in	R'_g	2000 ohms	R_g	200 ohms
m_a	1108 lb-sec²/in	L'_a	1108 henries	L_a	11.08 henries
k_e	14.5×10^6 lb/in	C'_e	$\frac{1}{14.5 \times 10^6}$ farad	C_e	0.069×10^{-6} farad
x_{mp}	________inches	q_{mp}	________coulombs	q'_{mp}	________coulombs

system of the IOWA was explored by measuring the currents produced by known impressed voltages throughout the frequency range. The amplitudes thus obtained, which were based on a driving force of constant amplitude, were then corrected on the basis of a driving force at the propeller varying as the square of the frequency, the amplitude of the force being 10,000 pounds at 500 CPM. The frequency range was 300 to 2400 CPM, which covered both the single- and the double-blade frequency ranges with 4- or 5-bladed propellers.

Figure 23 shows the response at the propeller. It should be noted that the frequency of the first mode resonance shown in this figure agrees very closely with the blade frequency of 900 CPM (5 × 180) at the peak amplitude of alternating thrust measured on the NEW JERSEY as shown in Figure 19b. Although this graph shows a very large amplitude at the second-mode resonance it must be remembered that the primary excitation which occurs at the single-blade frequency extends only to 1000 CPM. The double-blade-frequency component of the alternating thrust at the propeller is usually

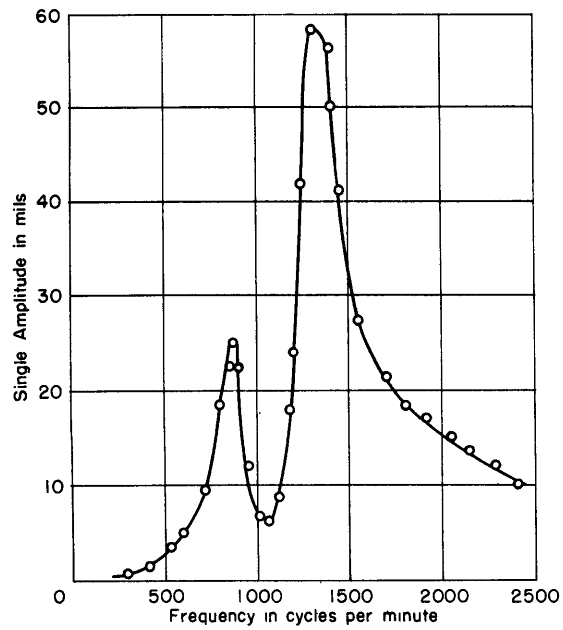


Figure 23 - Fore-and-Aft Amplitude of Propeller Determined from Electrical Analogy of Model 5, Starboard Inboard Propulsion Unit of BB61 through 66 Class

The amplitude of the driving force was $10{,}000\left(\frac{CPM}{500}\right)^2$ pounds.

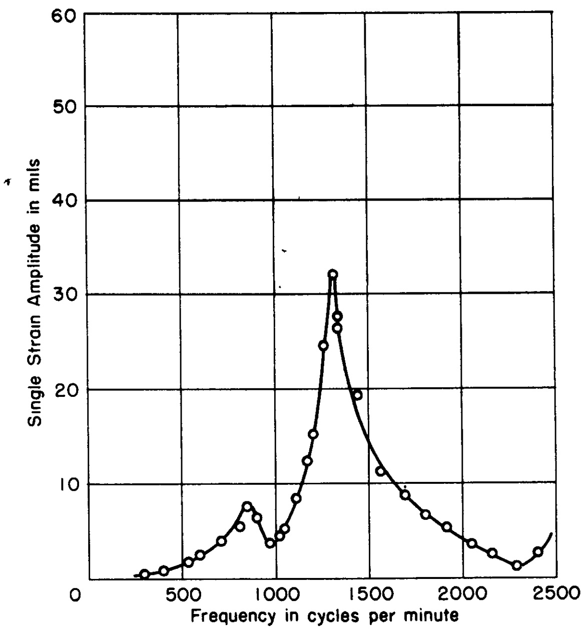


Figure 24 - Amplitude of Relative Axial Vibratory Motion between Propeller and Aft Thrust Block Determined from Electrical Analogy of Model 5, Starboard Inboard Propulsion Unit of BB61 through 66 Class

The amplitude of the driving force was $10{,}000\left(\frac{CPM}{500}\right)^2$ pounds.

considerably less than half the single-blade-frequency component, so that the large amplitudes shown on these curves for the second mode are only hypothetical.

Figure 24 shows the amplitude of the relative axial vibratory motion in the section of shaft between the propeller and the thrust bearing. This can be obtained by measuring the current through the condenser C_{sp}, Figure 22. The single amplitude of compression for the first-mode resonance is about 8 mils. This compression occurs in a shaft length of 1420 inches, the corresponding strain being 5.7×10^{-6} inch per inch. With a value of Young's modulus of 30×10^6 pounds per square inch, the corresponding stress is 170 pounds per square inch. As the area is 343 square inches, this corresponds to an alternating thrust of 58,000 pounds. The alternating-thrust measurements made on the ship by means of metalectric strain gages showed a thrust variation of about 45,000 pounds in the starboard inboard shaft of the NEW JERSEY at the first-mode critical of 180 RPM; see Figure 19c. Figure 25 shows the fore-and-aft amplitude of the after main thrust bearing and Figure

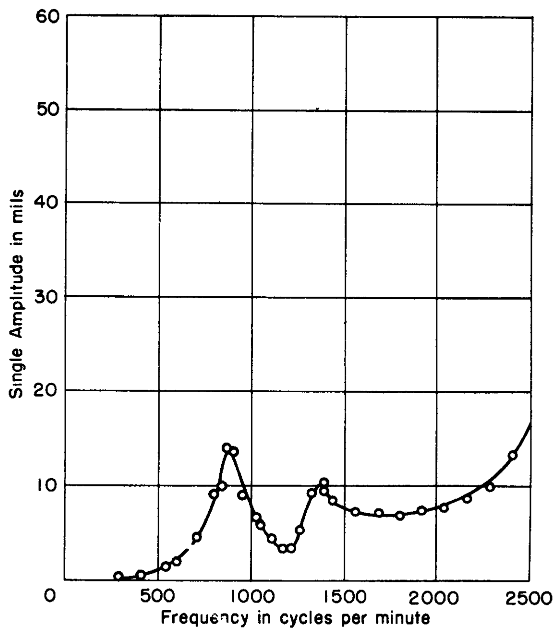


Figure 25 - Fore-and-Aft Amplitude of Thrust Block Determined from Electrical Analogy of Model 5, Starboard Inboard Propulsion Unit of BB61 through 66 Class

The amplitude of the driving force was $10,000 \left(\frac{CPM}{500}\right)^2$ pounds.

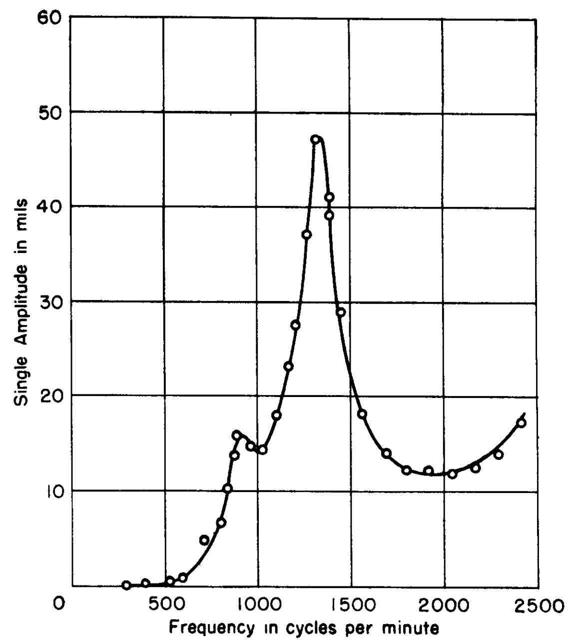


Figure 26 - Amplitude of Relative Axial Vibratory Motion between Thrust Block and Bull Gear Determined from Electrical Analogy of Model 5, Starboard Inboard Propulsion Unit of BB61 through 66 Class

The amplitude of the driving force was $10,000 \left(\frac{CPM}{500}\right)^2$ pounds.

27 shows the amplitude of the bull gear. By comparison with Figure 23 it is seen that the ratio of the amplitude of the propeller to that of the thrust bearing is 1.8 and the ratio of the amplitude of the bull gear to that of the thrust bearing is 2.1. The corresponding ratios computed in the vibration calculations of the normal modes of vibration for the undamped system, Table 2, are 1.7 and 2.1. Thus the ratios of the amplitudes of various members of the system can be estimated fairly well from the calculations for the undamped system, but to compute absolute values damping constants must be known or assumed.

Figure 26 gives the amplitude of the relative axial vibratory motion in the section of shaft forward of the after thrust bearing. The single amplitude of compression at the first-mode resonance as shown by the electrical analogy is 16 mils. As the shaft has a length of 1540 inches this corresponds to a strain of 10.4×10^{-6} inch per inch or to a stress of 300 pounds per square inch. As the area is 236 square inches this gives an alternating thrust of 71,000 pounds. The actual alternating thrust measured with metalectric strain gages was 50,000 pounds. Thus the electrical analogy

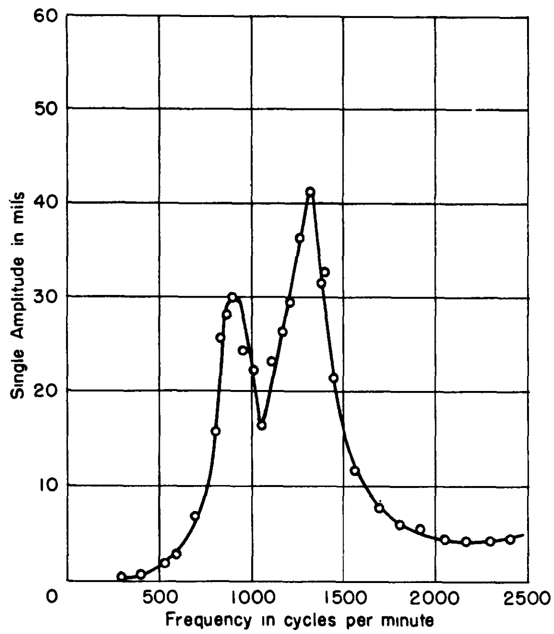


Figure 27 - Fore-and-Aft Amplitude of Bull Gear Determined from Electrical Analogy of Model 5, Starboard Inboard Propulsion Unit of BB61 through 66 Class

The amplitude of the driving force was $10{,}000 \left(\frac{CPM}{500}\right)^2$ pounds.

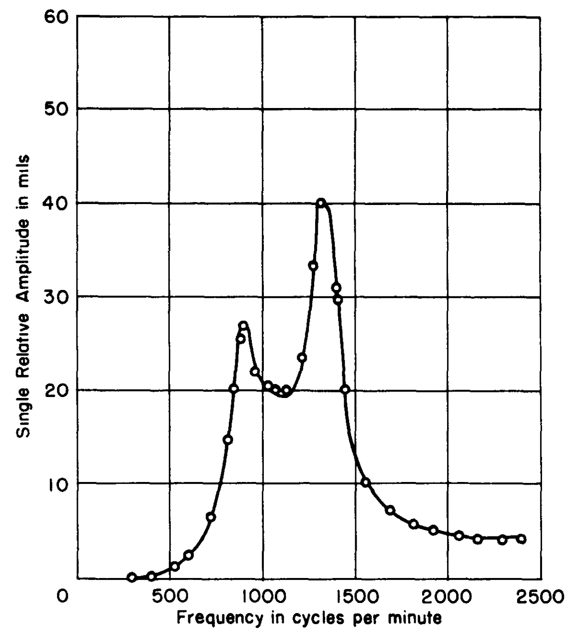


Figure 28 - Fore-and-Aft Amplitude of Bull Gear Relative to Its Bearings Determined from Electrical Analogy of Model 5, Starboard Inboard Propulsion Unit of BB61 through 66 Class

The amplitude of the driving force was $10{,}000 \left(\frac{CPM}{500}\right)^2$ pounds.

confirms the fact that the thrust variation can be greater forward of the thrust bearing than aft, a fact that was puzzling at the time that it was first observed. This phenomenon is due to the axial vibration of the relatively heavy bull gear.

The electrical analogy also permitted investigation of certain motions that could not be readily measured on the ship. For example, by assuming a reasonable value for frictional resistance to fore-and-aft motion of the bull gear as represented by c_g in Model 5, Figure 22, it was possible to investigate the relative motion between the bull gear and the gear case, and the probable motion that would have to be taken up in the turbine flexible couplings. This is shown in Figure 28, where a relative single amplitude of motion of 28 mils at the first-mode resonance is indicated.

Figure 29 shows that little vibration would be transmitted to the turbine under the assumed values of friction. Figure 15 shows that actually a single amplitude of 13 mils was reached on the port inboard low-pressure turbine on one of the trials of the IOWA but a local resonance was suspected in this case, as pointed out on page 18.

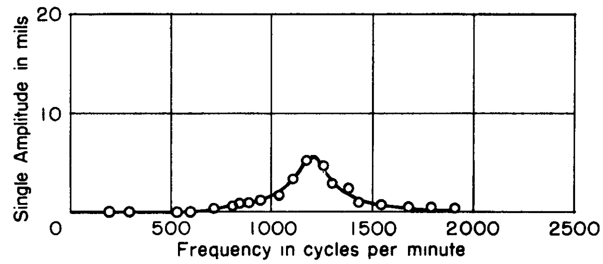


Figure 29 - Fore-and-Aft Amplitude of Turbines Determined from Electrical Analogy of Model 5, Starboard Inboard Propulsion Unit of BB61 through 66 Class

The amplitude of the driving force was 10,000 $\left(\frac{\text{CPM}}{500}\right)^2$ pounds.

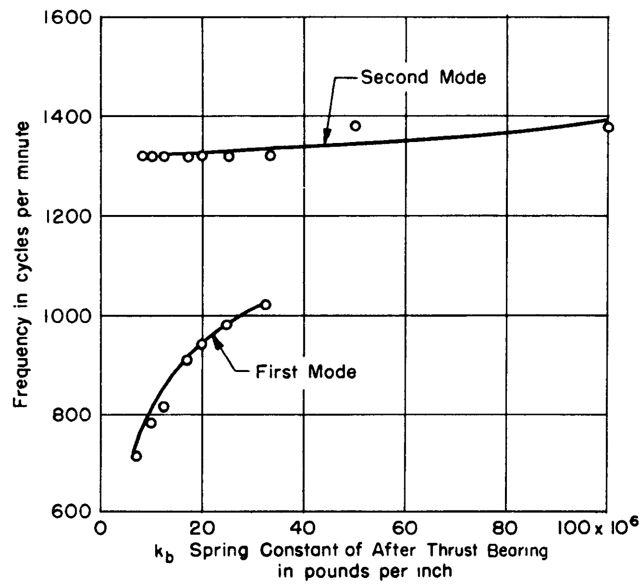


Figure 30 - Variations in the Critical Frequencies of Vibration of the Propulsion System as the Spring Constant of the Thrust Bearing Varies, Determined by Electrical Analogy of Model 5

The amplitude of the driving force was 10,000 $\left(\frac{\text{CPM}}{500}\right)^2$ pounds.

Figures 30 through 32 illustrate the effects of changing various values of the constants of the vibratory system. For example, Figure 30 shows the effect on the natural frequencies of stiffening the after thrust-bearing foundation, and it is seen that to bring the first-mode frequency above 1000 CPM would require trebling the spring constant. The effect on the second-mode frequency is negligible. Figure 31 shows the effect of various values of friction between the bull gear and its bearings on the amplitude

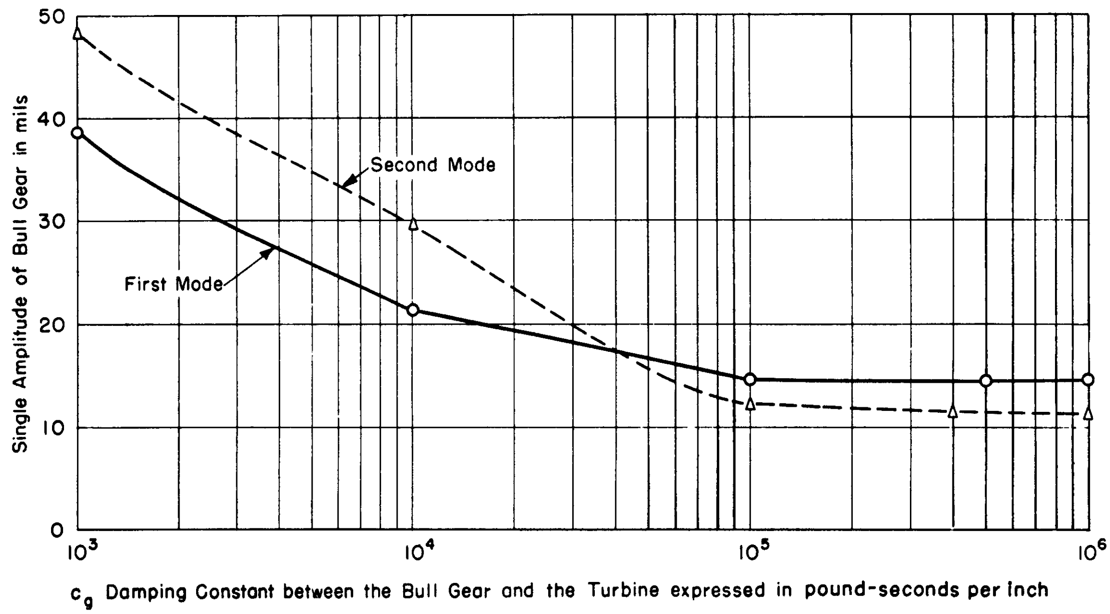


Figure 31 - Variation in the Amplitude of Vibration of the Bull Gear at the Critical Frequencies as a Function of Damping

The damping in the reduction-gear bearings is varied from 0 to 1,000,000 pound-seconds per inch. These data were determined for the starboard inboard propulsion unit of the BB61 through 66 Class from an electrical analogy of Model 5.

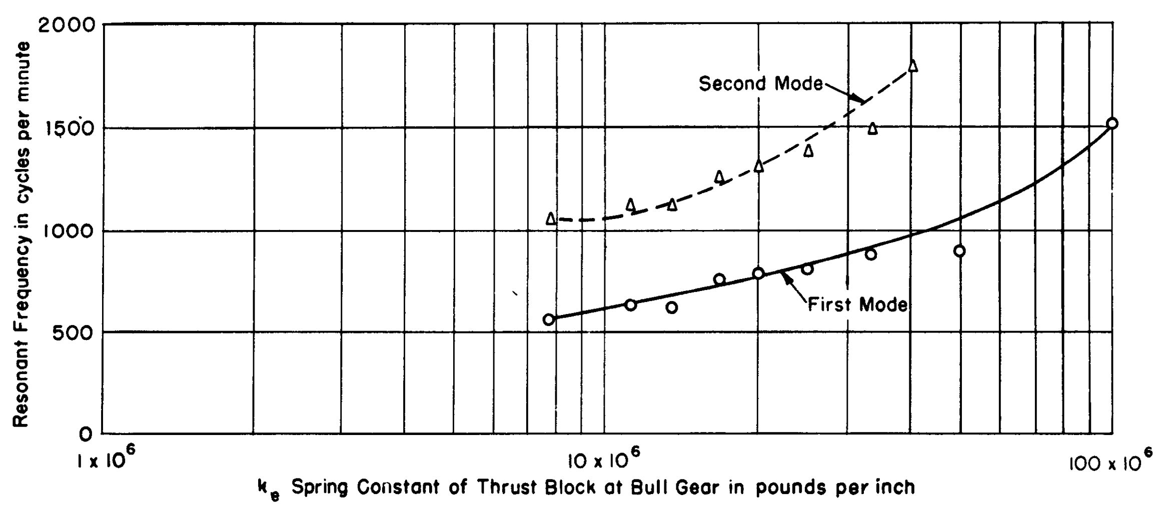


Figure 32 - Resonance Frequencies for Various Values of Stiffness of Machinery Foundations with Thrust Block at Bull Gear

of motion of the bull gear. Figure 32 shows the variation in the critical frequencies for various spring constants of the thrust-bearing foundation when the thrust bearing is located at the bull gear, as on the NORTH CAROLINA. It indicates that considerable stiffening would be required to bring the first-mode frequency above 1000 CPM.

CONCLUSIONS

From the behavior with respect to axial vibration of the propulsion systems of the three recent classes of battleships discussed in this report it is concluded that

1. The possibility of axial vibration of the propulsion systems must be considered in any future design of large vessels.

2. Model tests with pitot tubes should be made to determine the wake variation in the propeller races for any new design of a large vessel. If large wake variations occur, the possibility of corrective changes in the hull form aft should be investigated, in order to eliminate this source of excitation of axial vibration of the propulsion systems.

3. Estimates should be made of the first- and second-mode frequencies of axial vibration of all four propulsion units, the masses and shaft spring constants of thrust-bearing and machinery foundations being estimated from experimental results on similar installations by using data such as given in Table 4. These estimates should include an investigation of the effect of varying the fore-and-aft locations of the respective thrust bearings.

4. If calculations show that the first-mode resonance can be brought above the blade-frequency range by locating the thrust bearing at an optimum position in the shaft alley, serious consideration should be given to locating it in this position even though structural design difficulties might be involved.

5. Turbine flexible couplings for vessels of these sizes should be designed to resist wear due to axial vibration, as discussed in Reference (3).

6. The possible application of vibration neutralizers or pendulum dampers in reducing vibration of this type should be investigated, as discussed in Reference (1).

7. The electrical analogy provides a versatile method for investigating the effects of proposed changes in design parameters on the axial vibration of propulsion systems of naval vessels and should be used in conjunction with the accumulated data on system constants of recent ships when this type of vibration is under consideration in connection with new designs.

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