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AXIAL VIBRATION OF PROPULSION SYSTEMS OF BATTLESHIPS OF THE BB 57 THROUGH 60 CLASS

BY R. T. McGOLDRICK

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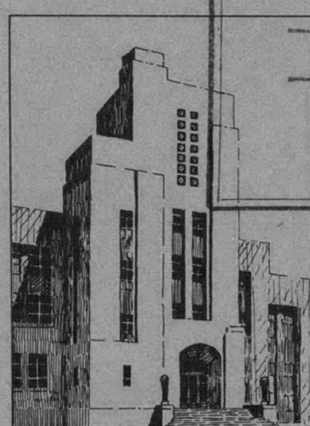

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AXIAL VIBRATION OF PROPULSION SYSTEMS OF BATTLESHIPS OF THE BB57 THROUGH 60 CLASS

ABSTRACT

An account is given of the measures taken to anticipate and evaluate the axial vibration of the propulsion systems on the battleships of the SOUTH DAKOTA Class (BB57 through 60) in the light of the excessive vibrations experienced with the propulsion machinery of the NORTH CAROLINA and WASHINGTON (BB55 and BB56).

Experimental data on the thrust variation include the amplitudes and resonance frequencies of the axial vibration of the shafts. It is shown that a fair estimate can be made of the frequencies of the first and second modes of axial vibration, provided a vibration test of the machinery unit alone can be made by a vibration generator, with the shaft uncoupled just aft of the reduction gear.

From observations of the machinery and the vibration data obtained at the time of the trials it was concluded that such axial vibration as did exist on these vessels was not serious. However, the recent appearance of excessive wear in the turbine couplings prompted further measurements on the ALABAMA (BB60) to determine the relative movements between the gears and the turbine rotors. These more recent measurements are discussed, as well as the 3-body system in which the flexibility in the main thrust bearing is taken into account.

It is concluded that, although the existing vibration is undesirable, it probably will not cause excessive wear in the newly designed flexible couplings, and that even if couplings must be replaced periodically the condition is not serious enough to warrant drastic modification of the present installation.

INTRODUCTION

The type of vibration discussed in this report consists of a fore-and-aft motion of the propeller shaft and the entire propulsion machinery, including both turbines, the condenser, and the reduction gears of each propulsion unit of the battleships of the BB57 through 60 Class. The system referred to is illustrated in Figure 1, which is an elevation of the port outboard shaft of the battleship SOUTH DAKOTA (BB57). The axial vibration is due primarily to the variation in thrust produced by each individual propeller blade in the course of a revolution. However, the effect of this thrust variation may be greatly magnified by resonance, which occurs when one or more

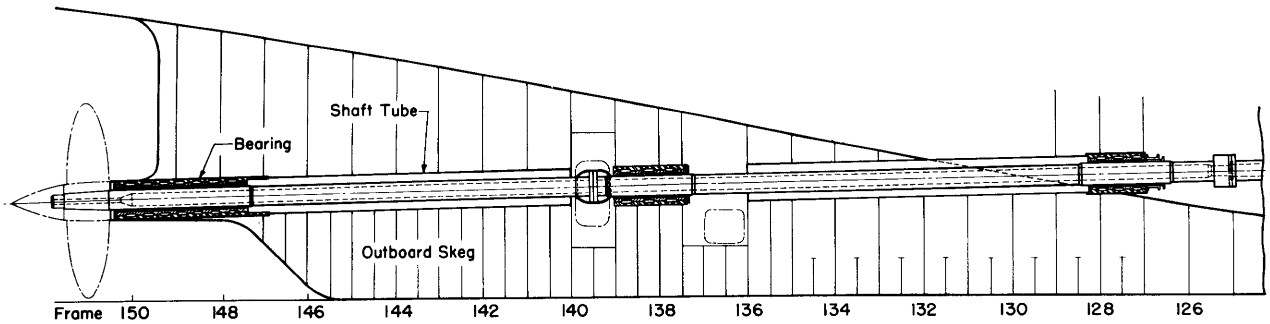

Figure 1 - General Arrangement of Propelling Machinery for Port Outboard Shaft on the USS SOUTH DAKOTA (BB57)

of the natural frequencies of the propulsion system fall within the operating range of blade frequencies.*

As a result of the difficulties encountered with axial vibration of the propulsion systems on the NORTH CAROLINA (BB55) and WASHINGTON (BB56) (1)** the possibility of a similar situation on the SOUTH DAKOTA Class (BB57 through 60) was anticipated. Although the propulsion systems of the latter class were essentially the same as those of the NORTH CAROLINA Class, the hull designs differed in an important respect as far as vibration is concerned, namely, in that the twin skegs or docking keels enclosed the outboard instead of the inboard shafts. The skeg arrangement for the BB57 through 60 Class is illustrated in Figures 2 and 3. There is a further difference in the machinery of two vessels of the BB57 through 60 Class which should be noted. The INDIANA (BB58) and the ALABAMA (BB60) have the nested system of gearing, whereas the SOUTH DAKOTA (BB57) and the MASSACHUSETTS (BB59) have the lock-train system as do BB55 and BB56. In the lock-train system of gearing there are axially flexible couplings between the bull gear and the first reduction gears, whereas in the nested system of gearing there are no flexible couplings within the gear system.

Model measurements of wake showed much less variation in wake fraction in the propeller races for the SOUTH DAKOTA than for the NORTH CAROLINA, and this applied both to the shafts supported by struts and to the shafts

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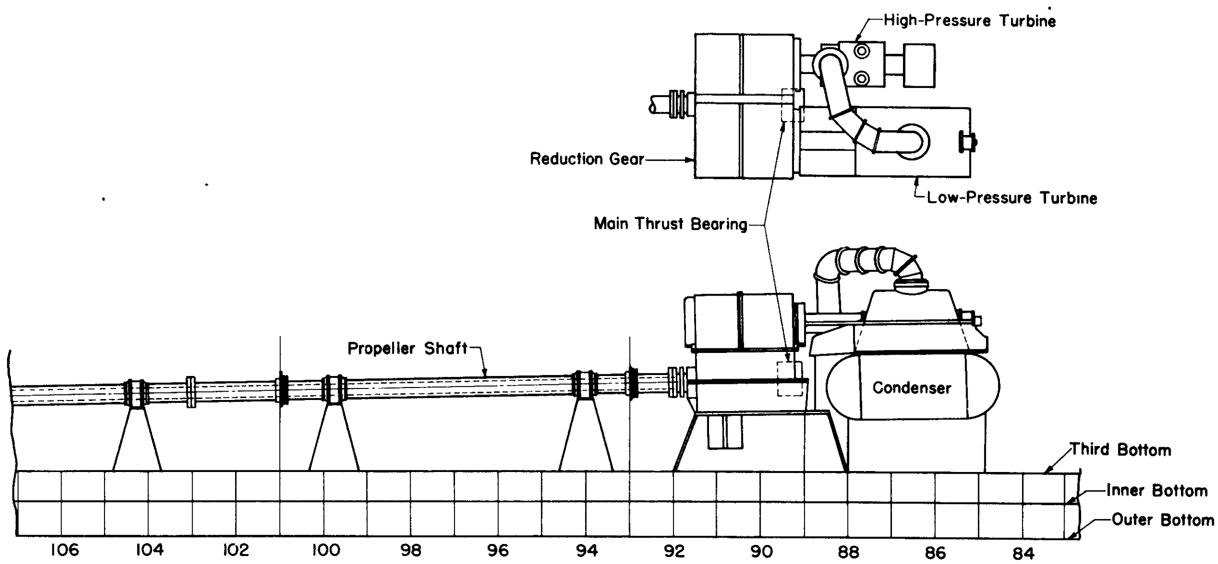
* The blade frequency is the product of the RPM by the number of blades on one propeller.

** Numbers in parentheses indicate references on page 30 of this report.



supported by skegs. Hence, although it was expected that resonance of the propulsion systems of the SOUTH DAKOTA Class would be encountered within the range of running speeds of the shafts, there was reason to believe that the vibration would be less serious than on the NORTH CAROLINA and WASHINGTON.

ESTIMATES OF RESONANCE FREQUENCIES BASED ON ANALYSIS OF A 2-BODY SYSTEM AND TEST WITH VIBRATION GENERATOR

The general theory of the type of vibration under discussion is given in an earlier report (1), where it is shown that for a first estimate the vibratory system may be treated as a 2-body system as shown schematically in Figure 4.

In Figure 4, m_p represents 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; k_s is the axial spring constant of the shaft; m_e is the effective mass of the entire engine unit, including high- and low-pressure turbines, wet condenser, reduction gears, and all machinery foundations, plus one-half the mass of the shaft; and k_e is the spring constant in the fore-and-aft direction of all the machinery foundations.

Of these constants m_p and k_s are obtained fairly easily from design data, but this is not the case with the effective mass m_e and the spring constant k_e. By the term "effective mass" is meant a mass which, if concentrated at the forward end of the shaft, would give the system the same natural frequencies as are actually found. This value obviously cannot be found by a simple summation of the weights of all the machinery items as they do not all

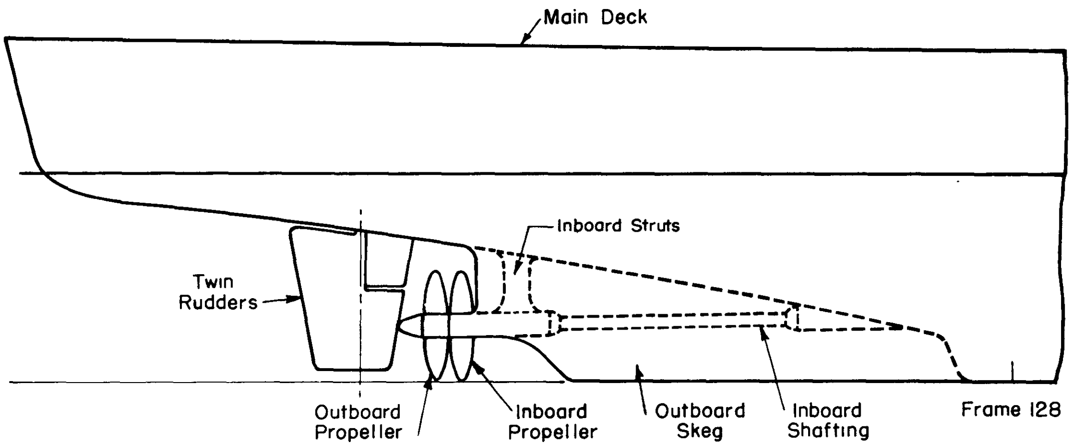


Figure 2 - Outboard Profile of Afterbody of BB57 through 60 Class

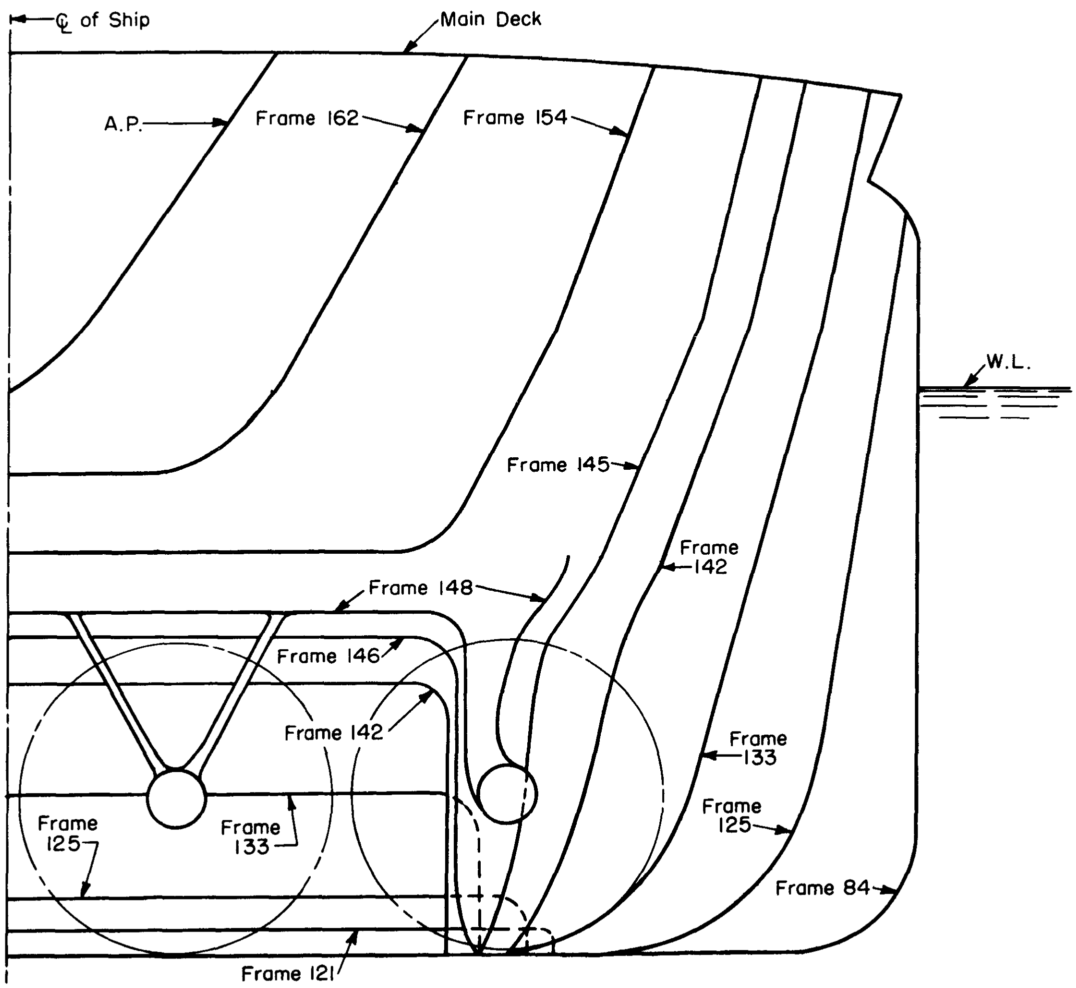


Figure 3 - Partial Body Plan of BB57 through 60 Class, Showing Position of Propellers with Reference to Stern of Ship

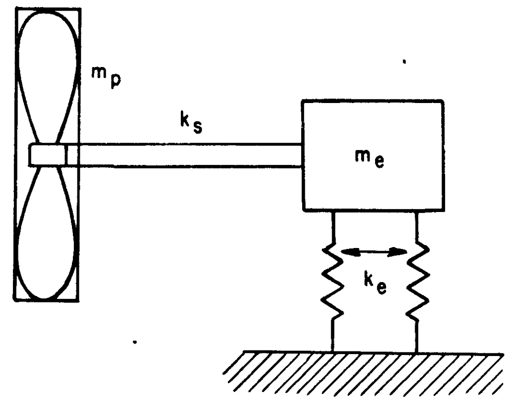


Figure 4 - Ideal 2-Body Vibratory System Used in Calculating Natural Frequencies of Axial Vibration of the Propulsion Systems on the BB57 through 60 Class

partake of the same motion. The effective spring constant k_e is an even more elusive quantity than the effective mass m_e. It is almost impossible to estimate the effective spring constant from foundation drawings because of the uncertainty of the fixity at the ends of the various structural members and because of the unexpected but definite flexibility of the surrounding hull structure.

By using the constants which could be readily calculated from design data, the remaining constants were obtained for the NORTH CAROLINA (1) from the experimental values of the natural frequencies. A set of values was found for each of the four propulsion systems which, when substituted in the frequency equation for a 2-body system, gave fair agreement between calculated and measured frequencies. Use was also made of the data obtained on a test of the port inboard engine with a vibration generator which gave an experimental value of the ratio k/m of the machinery unit alone.

To facilitate forecasting the resonance frequencies on the BB57 through 60 Class the David Taylor Model Basin conducted a test with a vibration generator on the INDIANA (BB58) similar to that conducted by the General Electric Company on the NORTH CAROLINA. The TMB medium vibration generator (2) was mounted at the forward end of the port inboard thrust bearing so as to produce vibration in the fore-and-aft direction. The shaft was uncoupled just aft of the reduction gear. Under this condition the fore-and-aft natural frequency of the machinery unit was found to be 875 cycles per minute.

The constants of the vibratory systems representing the four propulsion units and the frequencies calculated from them are given in Table 1, which also contains the average values of the resonance frequencies obtained on several trial runs. These values of resonance frequencies are based on the vibration and alternating-thrust data obtained on these trials. The frequencies of the second mode, most of which occur above the operating range of

TABLE 1

Constants of the Vibratory Systems Representing the Propulsion Systems of BB57 through 60 Class Considered as 2-Body Systems Such as Illustrated in Figure 4

Propulsion Unit	Resonance Frequencies, CPM				Estimated Constants of 2-Body System			
	First Mode		Second Mode		m_p $\frac{\text{lb-sec}^2}{\text{in}}$	k_s lb/in	m_e $\frac{\text{lb-sec}^2}{\text{in}}$	k_e lb/in
	Experimental	Theoretical	Experimental	Theoretical				
Starboard Outboard	520	530.	900	900	419	1.83×10^6	1200	8.0×10^6
Starboard Inboard	650	620	1100	1100	350	2.75×10^6	1130	8.0×10^6
Port Inboard	700	660	1240	1210	318	3.39×10^6	1100	8.0×10^6
Port Outboard	600	565	980	980	390	2.11×10^6	1180	8.0×10^6

blade frequencies, were found by plotting the double-blade-frequency component of alternating thrust obtained by harmonic analysis of the alternating-thrust oscillograms.

In Table 1 all units are given in the inch-pound-second system. The mass in this system is equal to the weight in pounds divided by the acceleration of gravity in inches per second squared, that is, it is equal to the weight in pounds divided by 386. The values tabulated were obtained in the following manner. The mass m of the machinery unit was first obtained by adding up the actual weights of the gears, gear case, high-pressure turbine, low-pressure turbine, and wet condenser, plus one-fourth the weight of the machinery foundations. The spring constant k of the machinery foundations was obtained by solving the equation $k = 4\pi^2\left(\frac{\text{CPM}}{60}\right)^2 m$. In this equation the frequency was taken as 875 CPM, the value actually found in the test with the vibration generator. The value m_e given in Table 1 was obtained by adding half the mass of the shaft to m. The value of k_e was taken equal to k. The spring constant k_s of the shaft was obtained directly from the drawings by computing the cross-sectional areas and using the formula $k = EA/l$, where E is Young's modulus in pounds per square inch, A is the cross-sectional area in square inches, and l is the length in inches. The effective mass m_p at the propeller was obtained by taking the actual propeller mass, plus 60 per cent of the propeller mass to allow for the virtual mass of the entrained water, plus one-half the mass of the shaft.

WAKE ANALYSIS

The skegs of the SOUTH DAKOTA Class differ from those of the NORTH CAROLINA Class not only in position but also in form. In profile they are cut back from the propellers so as to create less disturbance in the flow. The pitot-tube data show the wake fraction* to be practically zero at the 180-degree position on the SOUTH DAKOTA.

As in the case of the NORTH CAROLINA and WASHINGTON, wake variation within the area of the propeller race was determined for the BB57 through 60 Class from pitot-tube measurements made in the model basin. The variation in wake fraction at the propeller-tip radius is plotted against angular position, measured clockwise from the vertical, in Figure 5. Both outboard and inboard propeller races are included in these figures. For comparison the wake variations for the NORTH CAROLINA and WASHINGTON are also plotted on the same basis in Figure 6. Complete wake patterns for the SOUTH DAKOTA Class are shown in Figure 7.

It should be noted that, while the curves for the NORTH CAROLINA and SOUTH DAKOTA Classes are generally similar in form except for the difference due to the fact that the SOUTH DAKOTA has outboard skegs, the peak values of wake fraction are much lower for the SOUTH DAKOTA Class than for the NORTH CAROLINA. This is in conformity with the much smaller amplitudes of vibration actually observed, for it is reasonable to assume that the damping in the propulsion systems of the two classes is not greatly different and hence that the amplitudes will be proportional to the thrust variation.

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

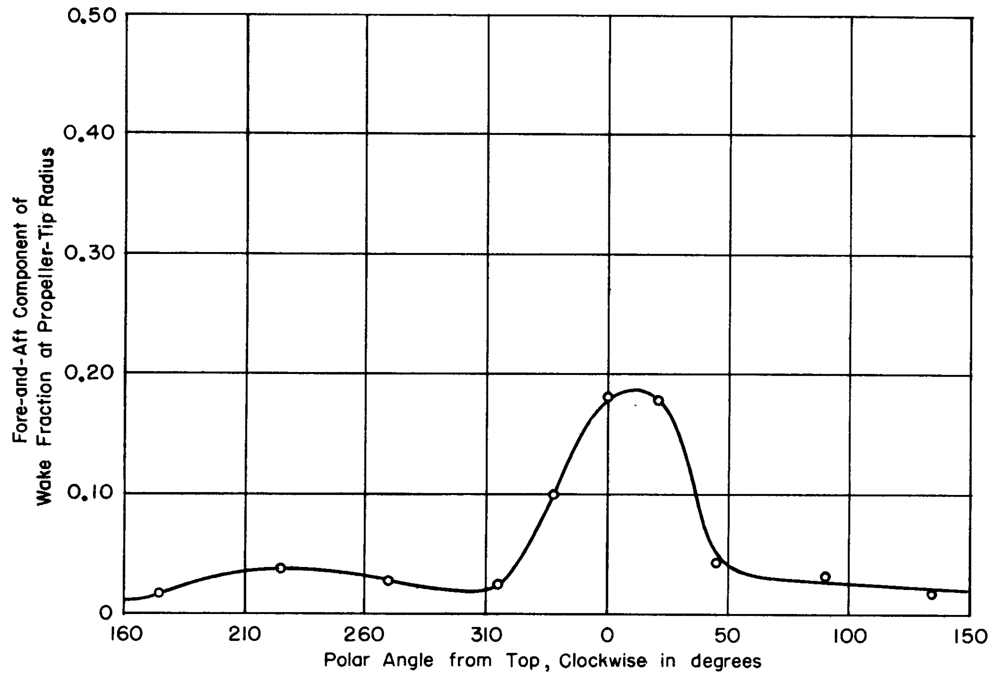

Figure 5a - Wake in Way of Inboard Propeller with Outboard Propeller Working

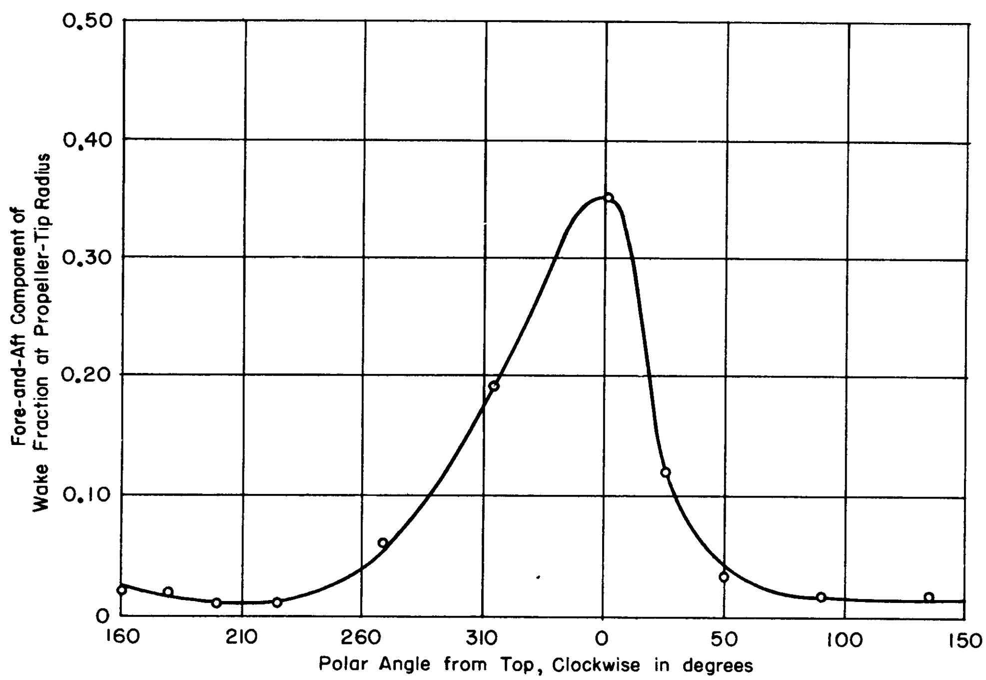

Figure 5b - Wake in Way of Outboard Propeller with Inboard Propeller Working

Figure 5 - Wake Variations from Model Tests of the SOUTH DAKOTA Class (BB57 through 60)

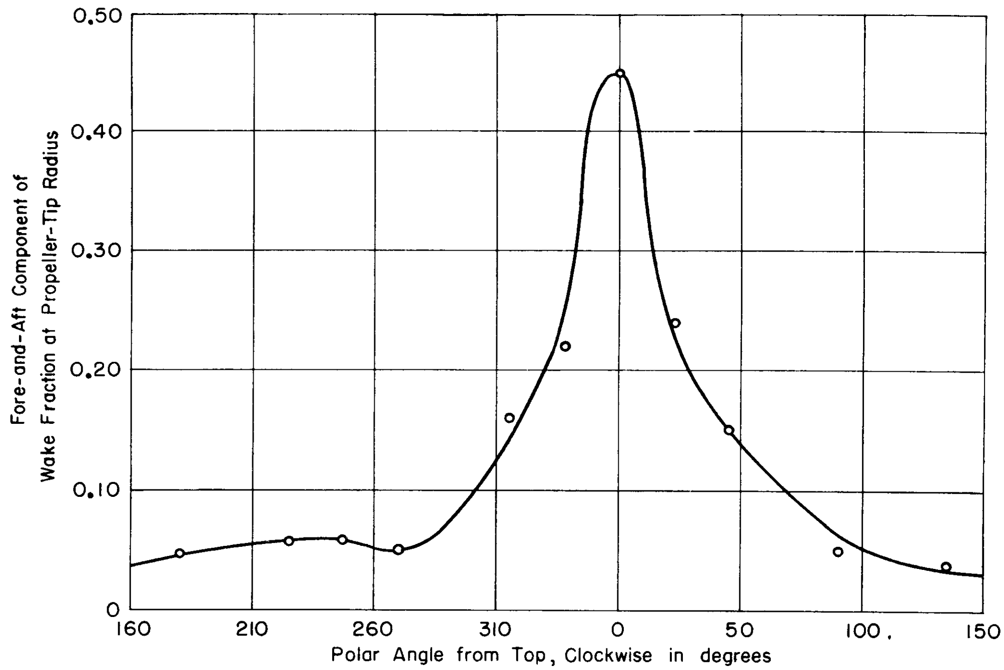

Figure 6a - Wake in Way of Inboard Propeller with Outboard Propeller Working

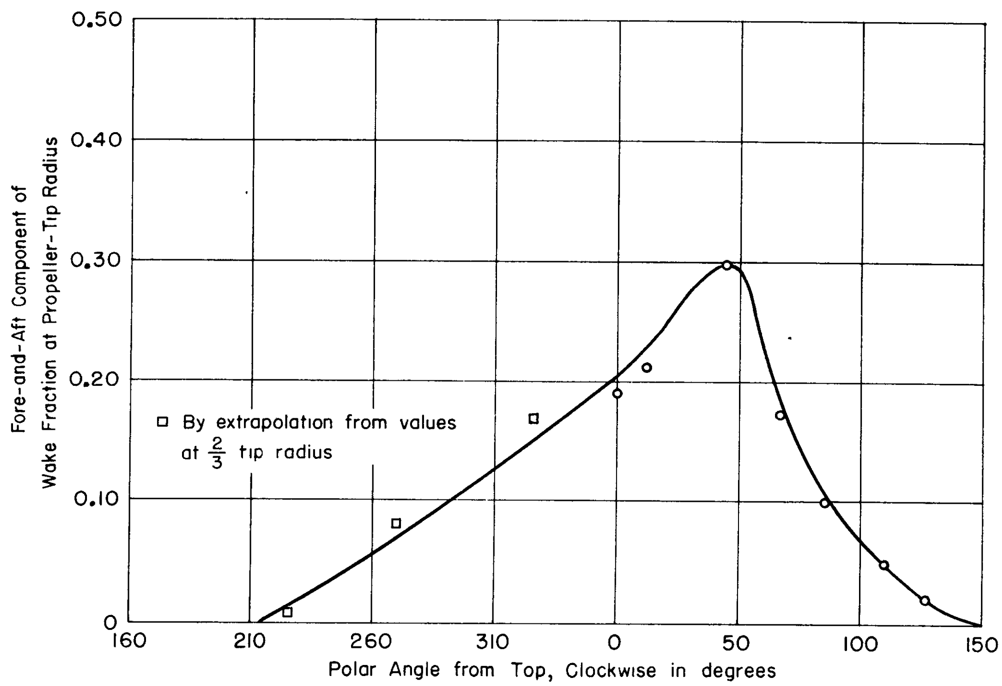

Figure 6b - Wake in Way of Outboard Propeller with Inboard Propeller Working

Figure 6 - Wake Variations from Model Tests of the NORTH CAROLINA and WASHINGTON (BB55 and 56)

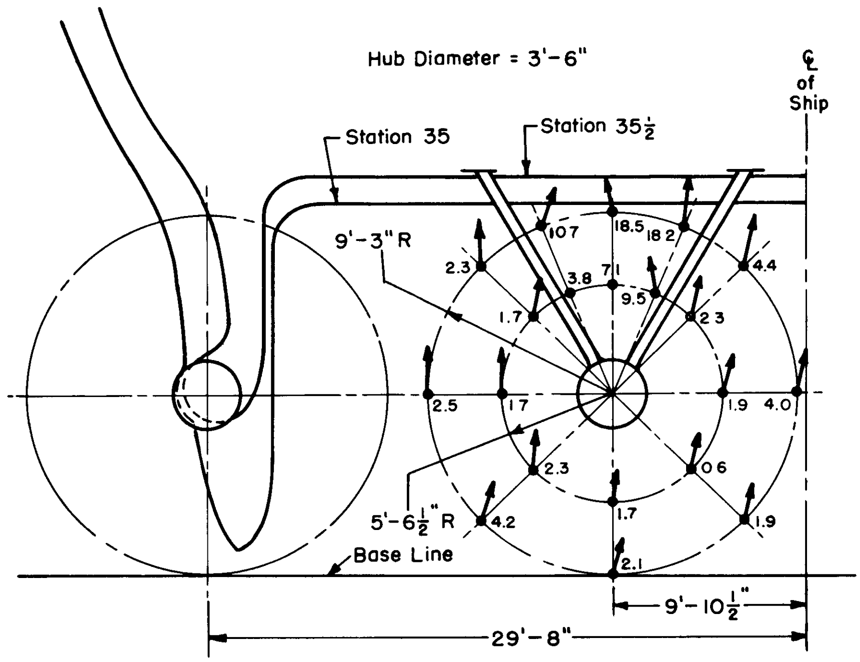

Figure 7a - Wake in Way of Inboard Propeller with Outboard Propeller Working

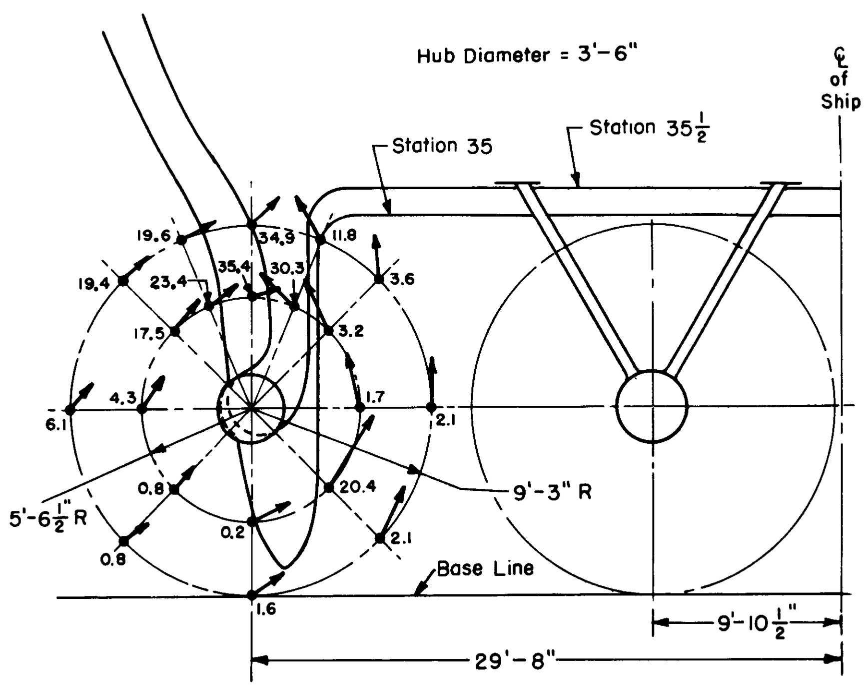

Figure 7b - Wake in Way of Outboard Propeller with Inboard Propeller Working

Figure 7 - Wake Patterns Obtained in Model Tests of the SOUTH DAKOTA Class (BB57 through 60)

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 during the sea trials of the BB57 through 60 Class was much less extensive than that carried out on BB55 and BB56. As the nature of the vibration was by this time better understood, the measurements were limited chiefly to the alternating components of thrust in the shafts (both blade frequency and double blade frequency) and to the fore-and-aft vibration at the thrust block. The thrust block on this class of vessels is located at the forward end of the reduction-gear unit as in the NORTH CAROLINA.

Thrust variations were measured on all four shafts on both the SOUTH DAKOTA (BB57) and the INDIANA (BB58). Vibration amplitudes were measured on the SOUTH DAKOTA, INDIANA, and ALABAMA (BB60). No measurements of axial vibration were made on the MASSACHUSETTS (BB59) by the Taylor Model Basin.

The strain-gage installation on the propeller shafts of BB57 and BB58 was essentially the same as that on BB55 and BB56 (3). However, the technique of making these measurements was still undergoing development at this time and a greatly improved form of brush rigging was used on BB58. The brushes themselves consisted of small pieces of braided copper wire attached to curved copper strips acting as springs to maintain the brush pressure. The use of carrier frequency in the electrical circuit was abandoned and the current through the metalectric strain gages was supplied by dry cells mounted directly on the rotating shafts. The voltage signals due to thrust variations were fed directly into amplifiers whose outputs were recorded on a 12-element string oscillograph.

The alternating-thrust oscillograms were analyzed for the blade-frequency and double-blade-frequency components of thrust variations by means of a harmonic analyzer.

In making the vibration measurements in the engine rooms of vessels of the BB57 through 60 Class, only mechanical instruments were used, in contrast with the large variety of instruments, both mechanical and electrical, used on BB55 and BB56. These mechanical instruments were TMB pallographs and Geiger vibrographs, the latter furnished by the New York Naval Shipyard. Both of these types of instruments are described in Reference (1). The location of the TMB pallograph on the starboard inboard engine is shown in Figure 8.

As on the trials of BB55 and BB56, speed changes during the trial runs were made in steps of 10 RPM. The tests were started at a low speed, usually about 120 RPM, and carried on up to full power, which was reached at about 185 RPM.

A central station was located on the third deck as in previous tests and the Taylor Model Basin officer in this station could communicate by

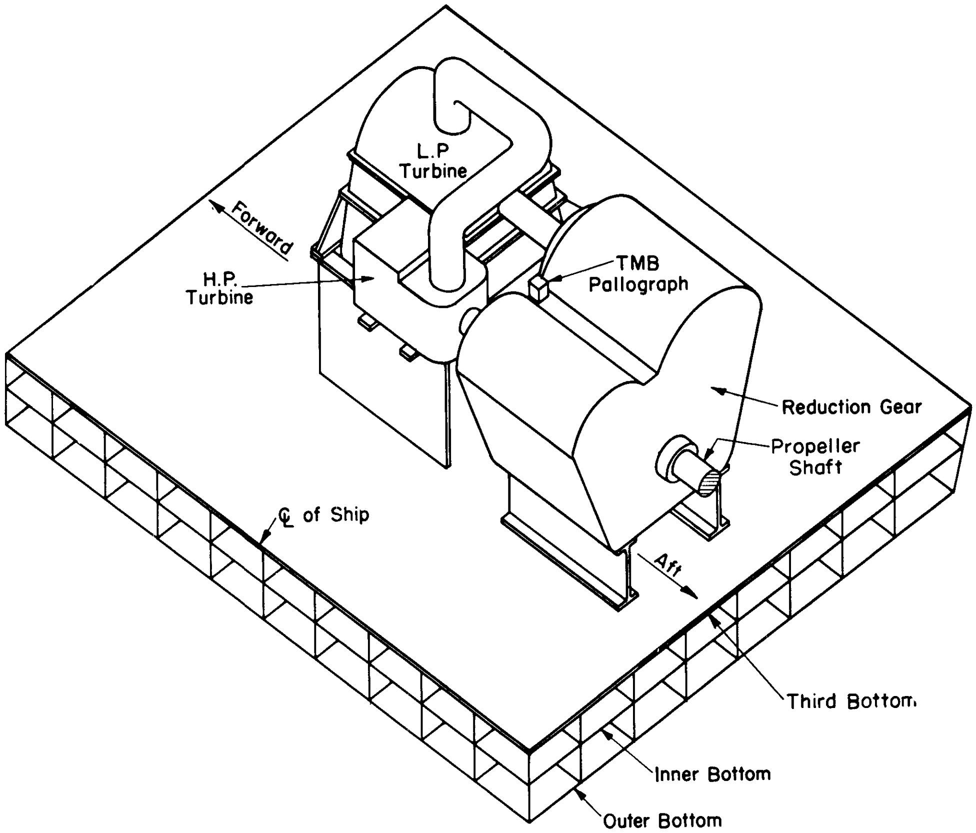

Figure 8 - Diagram Showing Location of TMB Pallograph for Starboard Inboard Engine of the USS SOUTH DAKOTA (BB57)

telephone with the test personnel in any of the shaft alleys or in any of the engine rooms. Each specified shaft speed was maintained until all vibration and alternating-thrust records had been obtained. The schedule of vibration tests on the SOUTH DAKOTA Class was as indicated in Table 2.

TABLE 2

Schedule of Trial Runs on the BB57 through 60 Class during Which Vibration Data Were Taken

Vessel	Date	Number of Propeller Blades		Measurements Made
		Inboard	Outboard	
SOUTH DAKOTA (BB57)	5 Jun 42	4	4	Thrust variation and fore-and-aft engine vibration.
INDIANA (BB58)	3 Aug 42	4	5	Thrust variation and fore-and-aft engine vibration.
INDIANA (BB58)	7 Sep 42	3	4	Thrust variation and fore-and-aft engine vibration.
ALABAMA (BB60)	19 Nov 42	4	5	Fore-and-aft engine vibration only.

RESULTS OF MEASUREMENTS DURING SEA TRIALS

The vibrations measured during the sea trials of the SOUTH DAKOTA Class were in general as anticipated from previous experience on the NORTH CAROLINA and WASHINGTON and from the wake variations measured in the model basin. Although resonance was encountered on all four propulsion systems within the operating range, the alternating component of thrust and the resulting vibration could safely be described as moderate. Sometimes the resonance effects were not sufficiently pronounced to clearly define the critical speed. The average values for the class as a whole, as previously listed in Table 1, page 6, are also given in Table 3, together with the accompanying critical speeds when the shafts were fitted with 4-bladed propellers inboard and 5-bladed propellers outboard. While there are some differences in the design of the machinery foundations and condensers on different vessels of this class, the overall variation in fore-and-aft rigidity does not appear to be sufficient to affect the resonance frequencies appreciably.

TABLE 3

Measured Resonance Frequencies and Critical Speeds of Axial Vibration of Propulsion Systems of BB57 through 60 Class, with 4-Bladed Propellers Inboard and 5-Bladed Propellers Outboard

Propulsion System	First Mode		Second Mode	
	Resonance Frequency CPM	Critical Speed RPM	Resonance Frequency CPM	Critical Speed RPM
Starboard Outboard	520	104	900	180
Starboard Inboard	650	163	1100	275*
Port Inboard	700	175	1240	310*
Port Outboard	600	120	980	196*

* These speeds are above the operating range and are the speeds which would be required for resonance at blade frequency. The second modes were actually excited by the double-blade-frequency components of thrust variation at speeds equal to half these values.

The thrust variation on the SOUTH DAKOTA Class was much less than on the NORTH CAROLINA and WASHINGTON, and it was therefore more difficult to obtain reliable data on the SOUTH DAKOTA Class. Metalectric strain gages were installed on all four shafts on both the SOUTH DAKOTA and the INDIANA for measuring the alternating component of thrust, but the data obtained on the SOUTH DAKOTA were somewhat questionable because of difficulties encountered

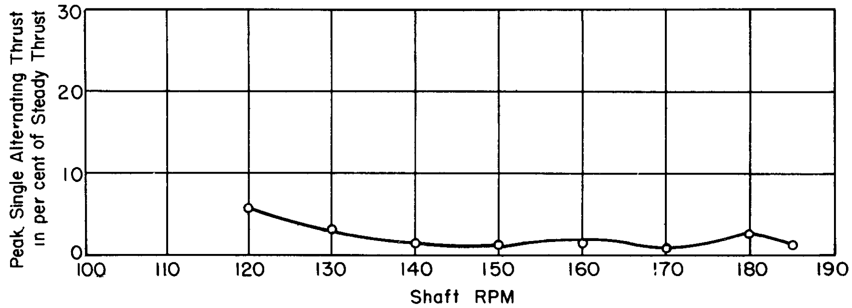

Figure 9a - Starboard Outboard Shaft, 5-Bladed Propeller

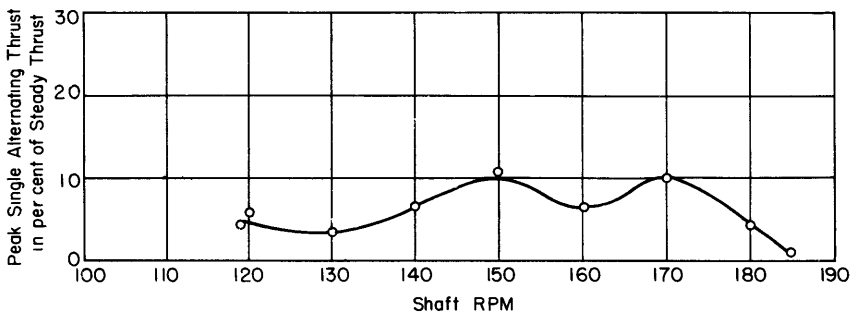

Figure 9b - Starboard Inboard Shaft, 4-Bladed Propeller

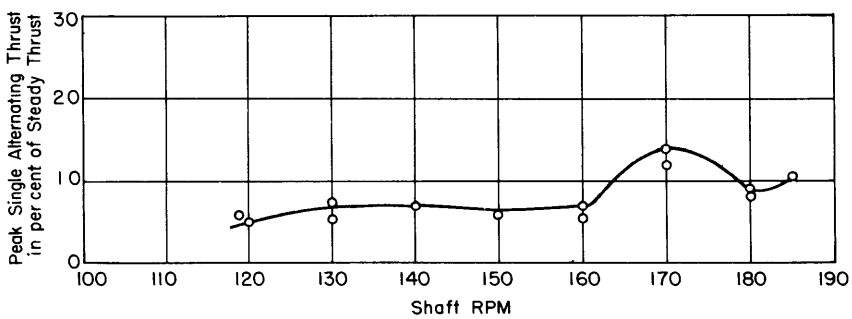

Figure 9c - Port Inboard Shaft, 4-Bladed Propeller

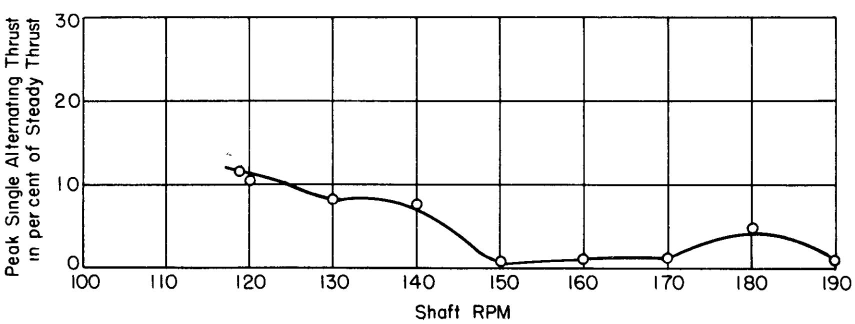

Figure 9d - Port Outboard Shaft, 5-Bladed Propeller

Figure 9 - USS INDIANA (BB58) - Blade-Frequency Component of Alternating Thrust, in Per Cent of Steady Thrust, Measured during Trial of 3 August 1942

with the brush rigging. However, with an improved design of brush and slip ring, fairly reliable data were obtained on two subsequent trial runs of the INDIANA.

The data obtained on the INDIANA are plotted against shaft RPM in Figures 9 and 10. These curves give the actual per cent variation in thrust in the shaft without correction for resonance effects. In general the critical speeds given in Table 3 are confirmed by the alternating-thrust curves, but as the values are quite low, with peak values of only about 10 per cent of the steady thrust, not all the curves give a clear indication of resonance. The steady-thrust values are shown in the curves of thrust plotted against shaft RPM in Figure 11. The second-mode resonance frequencies were determined by harmonic analysis of alternating-thrust oscillograms for the double-blade-frequency component.

Alternating thrust was also measured on the model of the BB57 through 60 Class by the same technique as that used for the BB55 and 56 Class. The average values from all tests indicated that the thrust variation at the

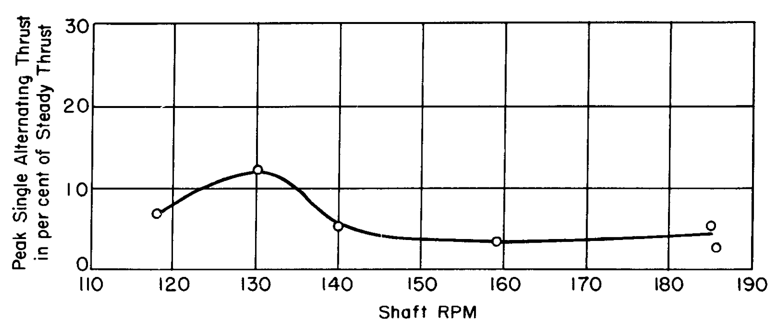

Figure 10a - Starboard Outboard Shaft, 4-Bladed Propeller

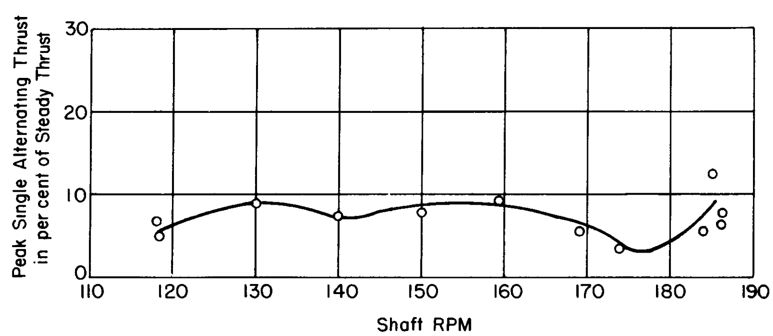

Figure 10b - Starboard Inboard Shaft, 3-Bladed Propeller

Figure 10 - USS INDIANA (BB58) - Blade-Frequency Component of Alternating Thrust, in Per Cent of Steady Thrust, Measured during Trial of 7 September 1944

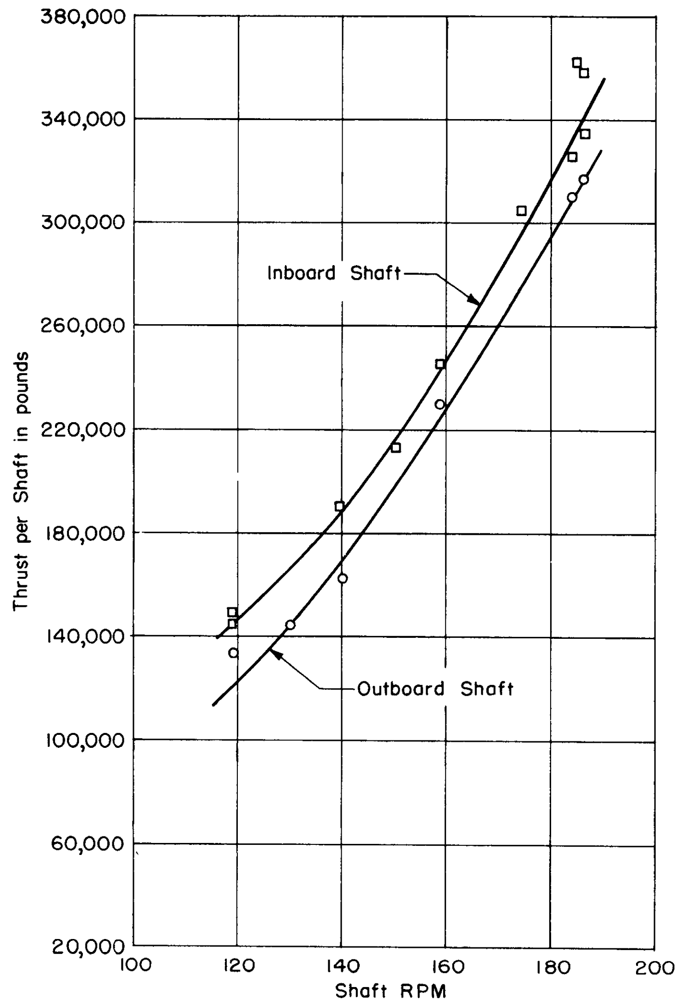

Figure 11 - USS INDIANA (BB58) - Curves of Steady Thrust per Shaft at Various Speeds, Based on Model Tests

propeller was 2.9 per cent of the steady thrust for the inboard shafts and 7.2 per cent for the outboard shafts with 3-bladed propellers. With 4-bladed propellers the percentages were 2.8 per cent inboard and 4.0 per cent outboard. With 5-bladed propellers only outboard data were obtained, the average being 6.0 per cent.

The amplitudes and frequencies of axial vibration were measured by means of Geiger vibrographs and TMB pallographs located on the gear cases just above the main thrust bearings. Figures 12 through 15 show the amplitudes of longitudinal vibration at the top of the reduction-gear cases plotted against shaft RPM for the SOUTH DAKOTA, INDIANA, and ALABAMA. As will be observed from the titles, combinations of 3-bladed and 4-bladed, all 4-bladed, and 4-bladed and 5-bladed propellers were tried. The arrangement finally adopted was 4-bladed propellers on all shafts for the SOUTH DAKOTA, and a combination of 4-bladed propellers inboard and 5-bladed propellers outboard on the remaining vessels of the class.

The amplitude at the main thrust bearing never exceeded half the value initially encountered on the NORTH CAROLINA, and it was usually much less than half. The vibration measurements gave a much clearer indication of resonance than the alternating-thrust data, and if account is taken of the number of propeller blades in each case, the resonance frequencies listed in Table 3 are generally confirmed. From the frequencies listed it is evident that most of the second-mode critical frequencies fall above the blade frequency developed at top operating speed.

In spite of the comparatively good performance of the BB57 through 60 Class with respect to axial vibration, these vessels were by no means free from vibration and at large rudder angles the condition was greatly aggravated, as is usual under such circumstances. Blade-frequency vibration was also amplified in some of the superstructure members, owing to local resonance. This resulted in some difficulty in operating the optical rangefinders.

(Text continued on page 24.)

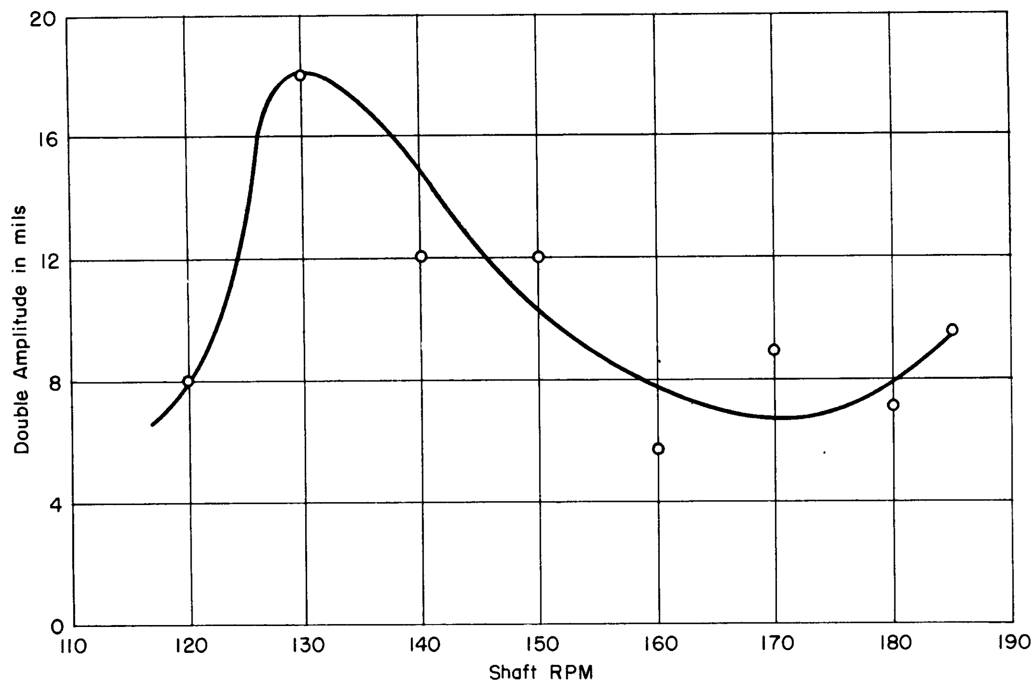

Figure 12a - TMB Pallograph Readings on the Starboard Outboard Gear-Case Shaft, 4-Bladed Propeller

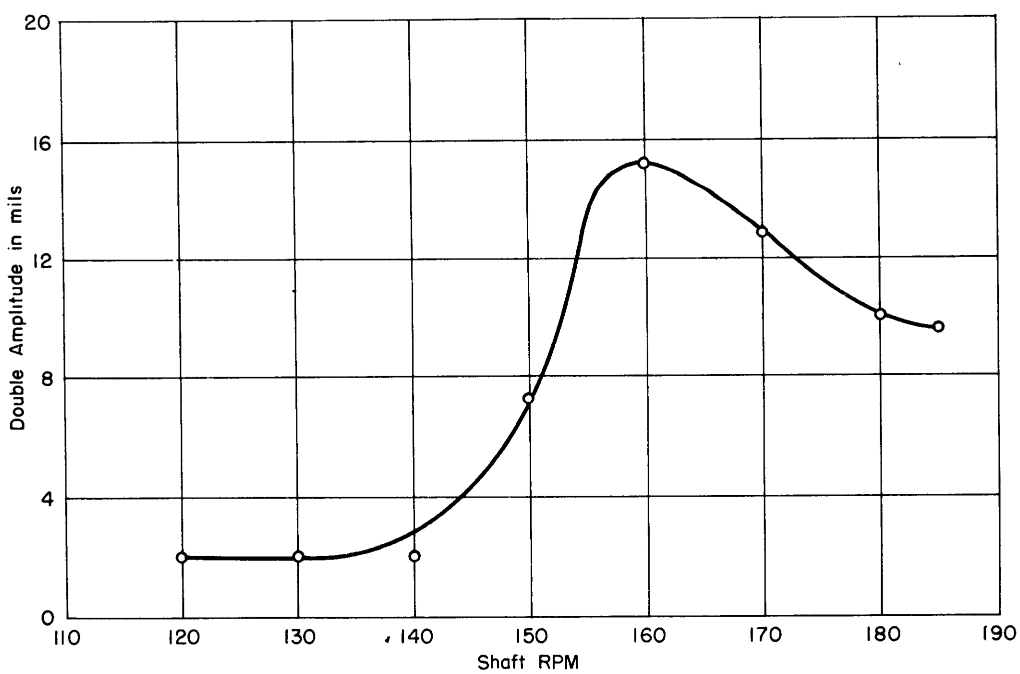

Figure 12b - TMB Pallograph Readings on the Starboard Inboard Gear-Case Shaft, 4-Bladed Propeller

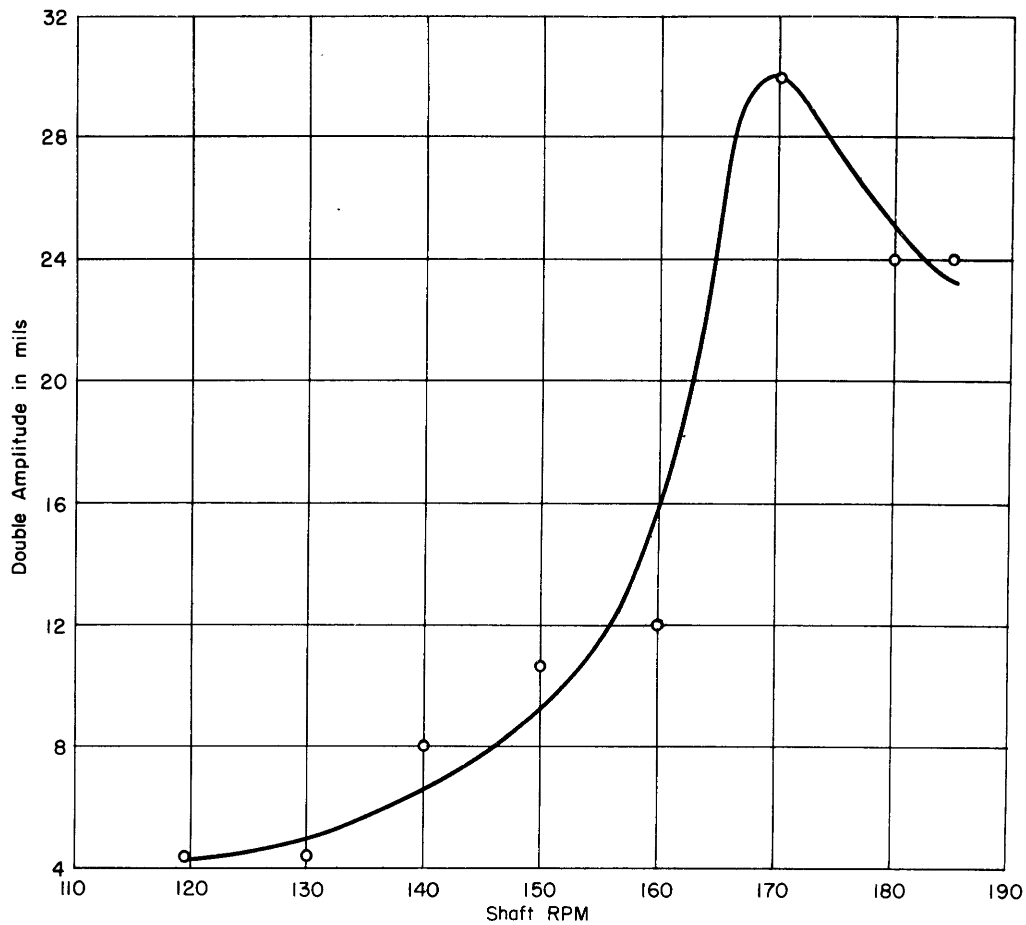

Figure 12c - Geiger Vibrograph Readings on the Port Inboard Gear-Case Shaft, 4-Bladed Propeller

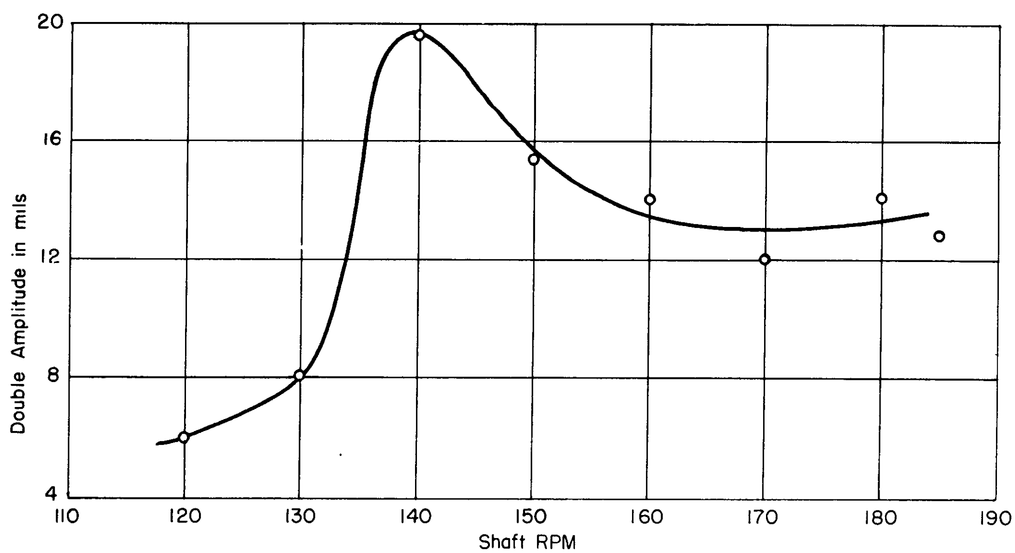

Figure 12d - Geiger Vibrograph Readings on the Port Outboard Gear-Case Shaft, 4-Bladed Propeller

Figure 12 - USS SOUTH DAKOTA (BB57) - Fore-and-Aft Vibration of Gear Case during Trial of 5 June 1942

Each instrument was mounted on the forward end of the gear case, just above the thrust bearing.

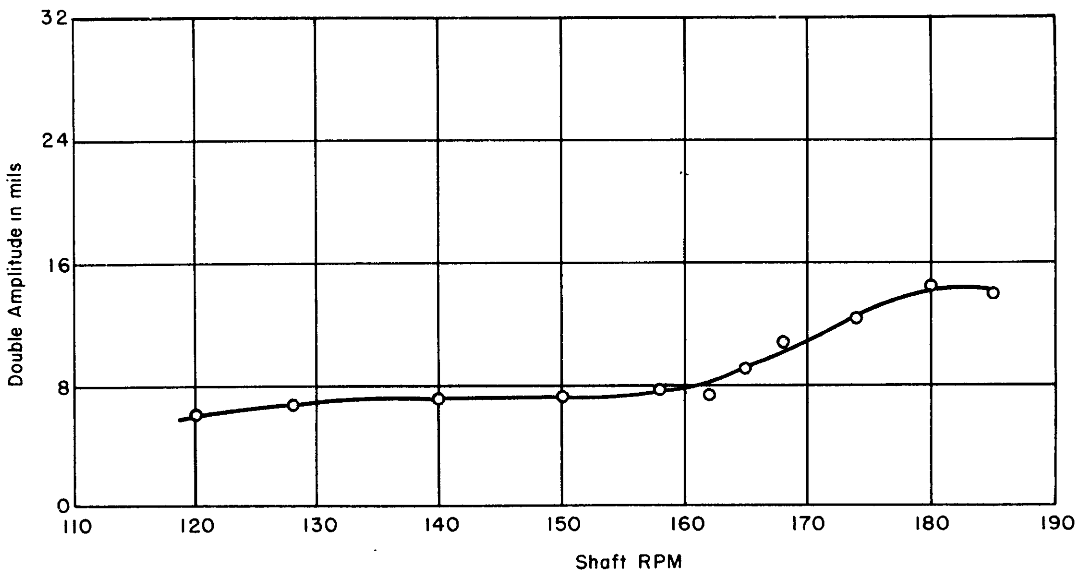

Figure 13a - Starboard Outboard Gear-Case Shaft, 5-Bladed Propeller

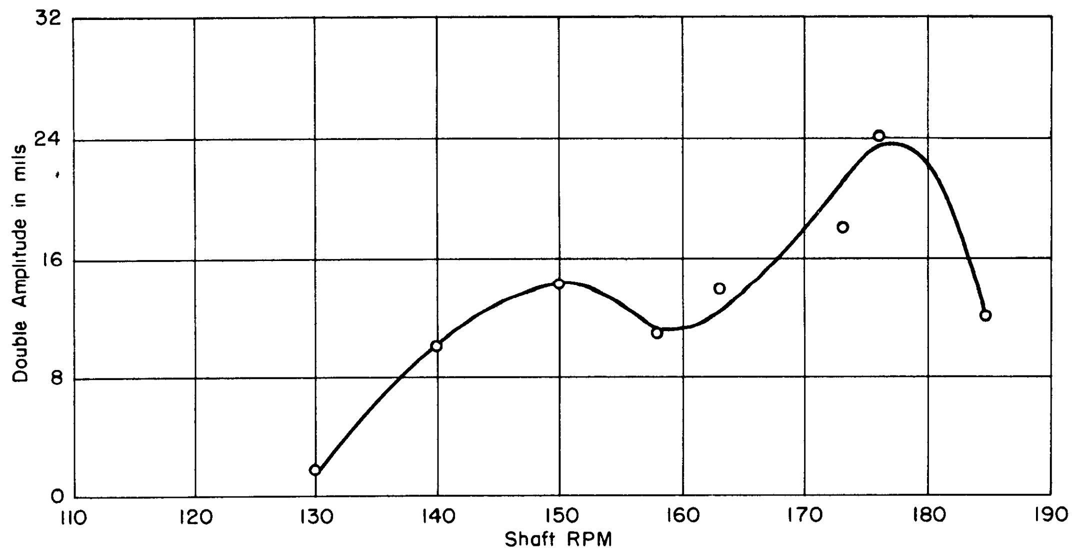

Figure 13b - Starboard Inboard Gear-Case Shaft, 4-Bladed Propeller

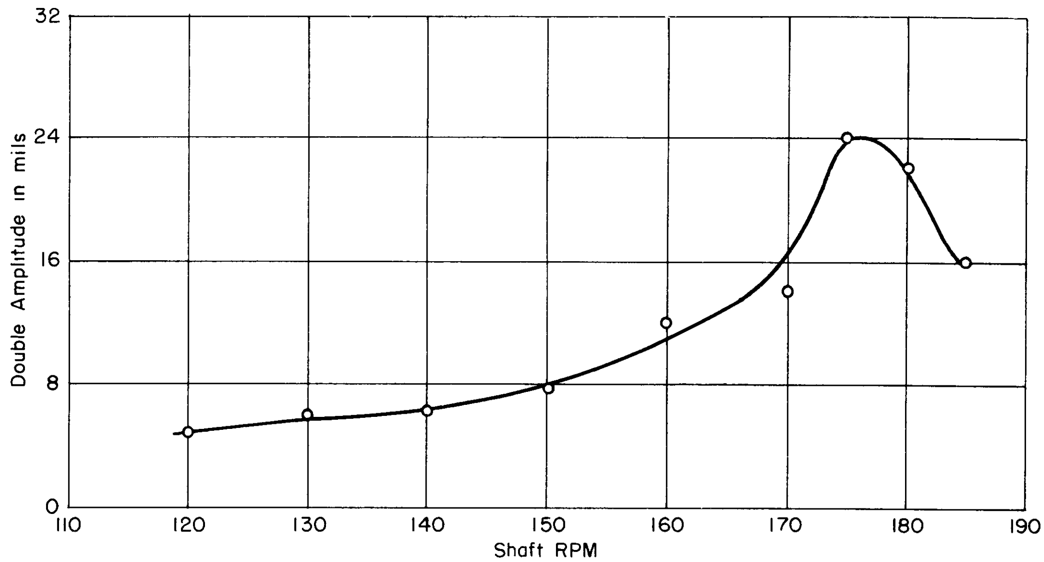

Figure 13c - Port Inboard Gear-Case Shaft, 4-Bladed Propeller

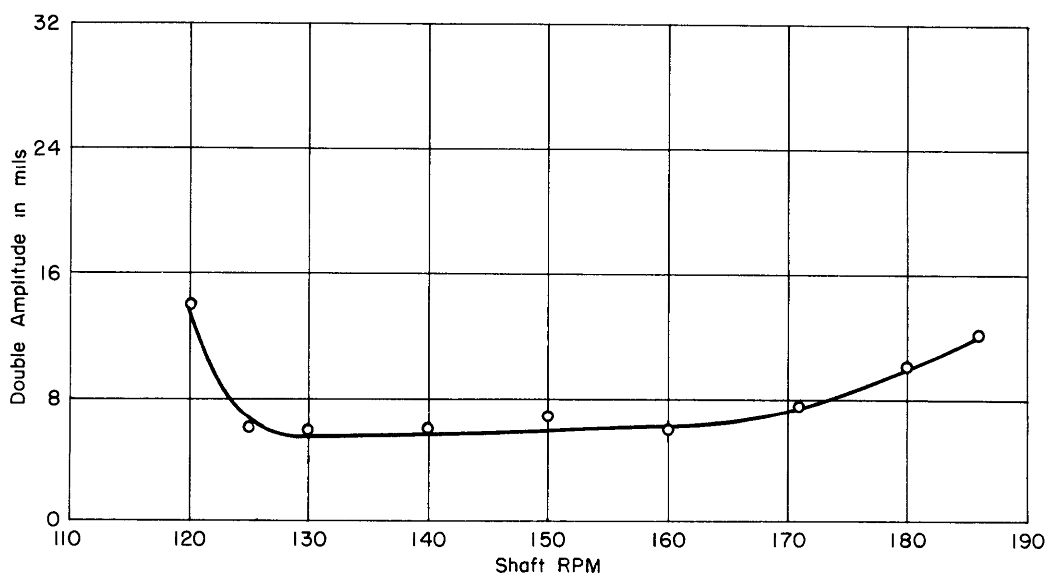

Figure 13d - Port Outboard Gear-Case Shaft, 5-Bladed Propeller

Figure 13 - USS INDIANA (BB58) - Fore-and-Aft Vibration of Gear Case during Trial of 3 August 1942

These measurements were made with Geiger vibrographs. The vibrographs were located on the forward end of the gear case, just above the thrust bearing.

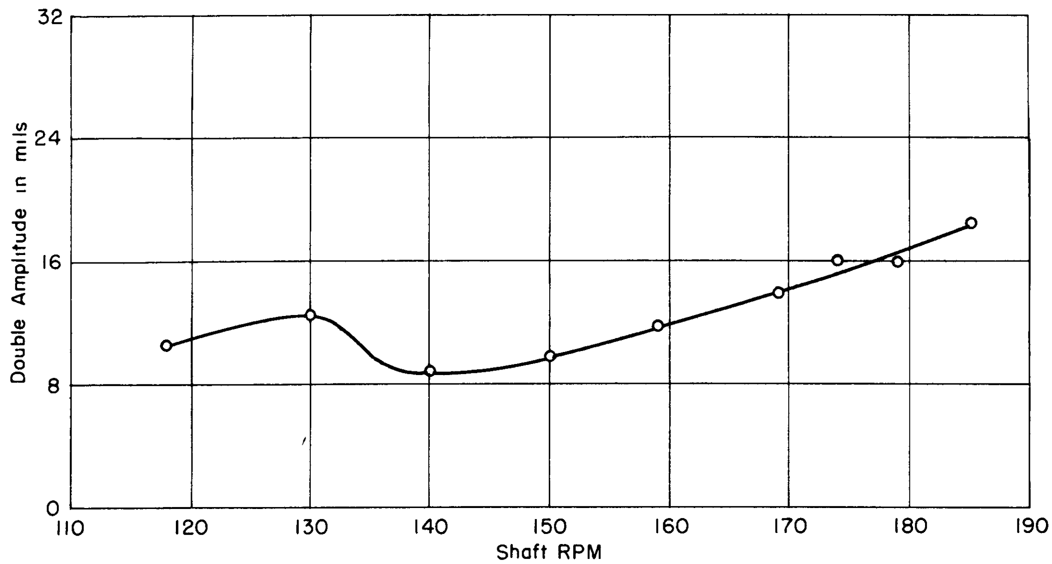

Figure 14a - Starboard Outboard Gear-Case Shaft, 4-Bladed Propeller

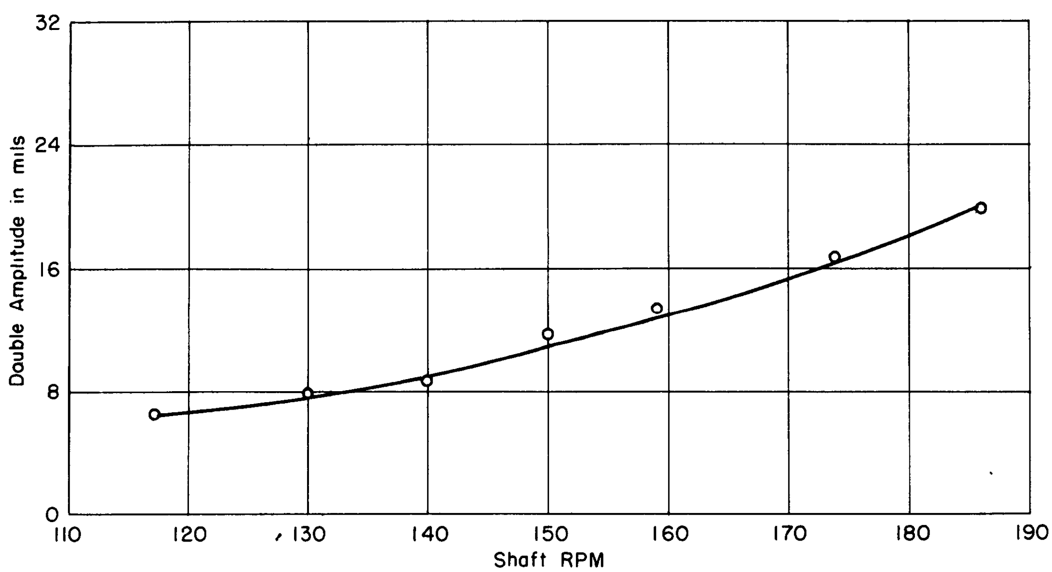

Figure 14b - Starboard Inboard Gear-Case Shaft, 3-Bladed Propeller

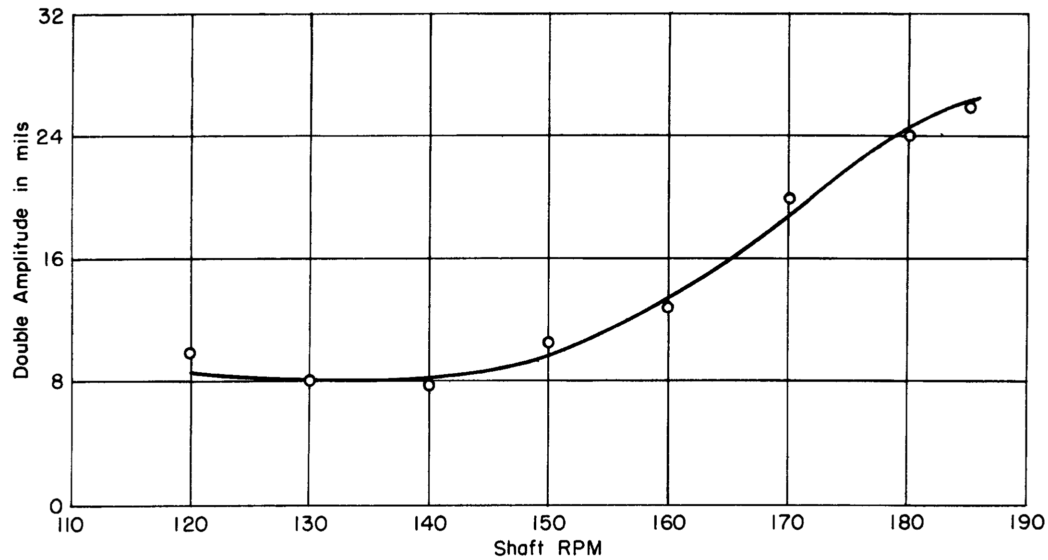

Figure 14c - Port Inboard Gear-Case Shaft, 3-Bladed Propeller

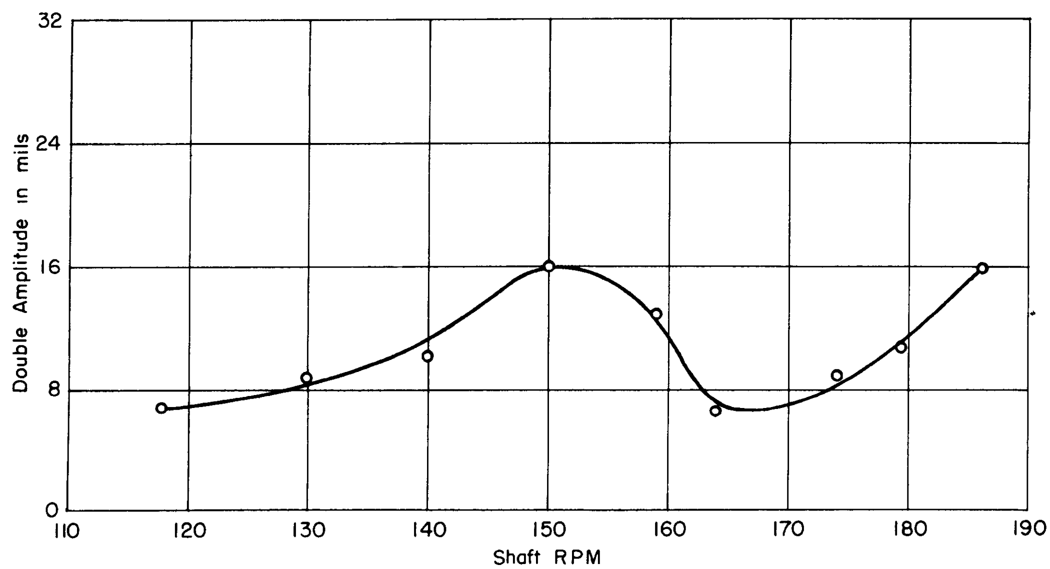

Figure 14d - Port Outboard Gear-Case Shaft, 4-Bladed Propeller

Figure 14 - USS INDIANA (BB58) - Fore-and-Aft Vibration of Gear Case during Trial of 7 September 1942

These measurements were made with Geiger vibrographs. The vibrographs were located on the forward end of the gear case, just above the thrust bearing.

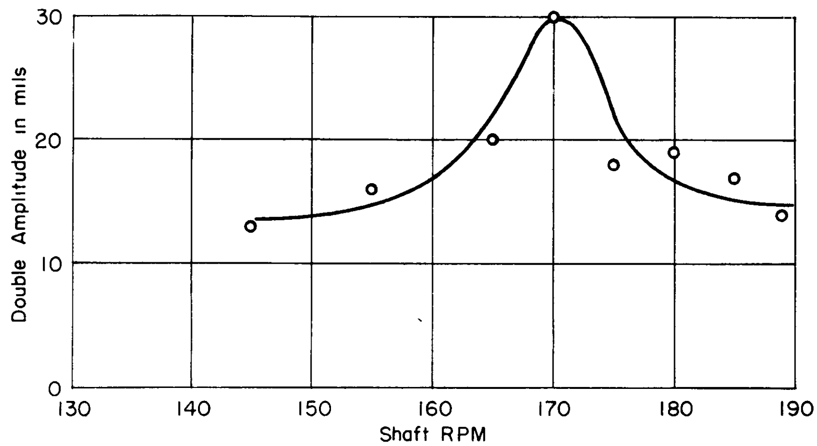

Figure 15 - USS ALABAMA (BB60) - Fore-and-Aft Vibration of Gear Case during Trial of 19 November 1942

These data are for the starboard inboard gear case when a 4-bladed propeller was mounted on the shaft. The measurements were made with a TMB pallograph which was located on the forward end of the gear case, just above the thrust bearing.

FURTHER INVESTIGATION DUE TO TURBINE COUPLING WEAR

Although, as a result of the measurements made during the engineering trials of the vessels, the axial vibration of the propulsion systems of the BB57 through 60 Class was considered moderate, it was recognized that vibration existed and that the amplitude increased considerably during turns. In the course of two years of operation, evidence that axial vibration was the cause of excessive wear in the flexible couplings accumulated. As a result of the suggestions of the Commanding Officer of the USS ALABAMA (BB60) (4), the Bureau of Ships decided that a further study of the fore-and-aft movement of the machinery components was necessary.

Engineers from the Westinghouse Electric and Manufacturing Company cooperated with personnel from the Vibration and Noise Reduction Section of the Mare Island Naval Shipyard and the Puget Sound Naval Shipyard in making a series of measurements of the absolute fore-and-aft movements of the shaft, gear case, turbine casings, high-pressure pinion, low-pressure intermediate gear, main thrust plate, and high- and low-pressure turbine rotors. They also recorded the movements of the gears and turbine rotors relative to their respective casings. In addition they measured lateral motions at both ends of the high-pressure flexible coupling and made a check of the torsional vibration in the propulsion system.

The chief aim in making these measurements was to determine whether the wear in the turbine couplings could be explained by the relative axial motions between the turbine rotors and the pinions to which they are coupled. The simple 2-body analysis given on page 3 is based on the assumption that the machinery and its housings move axially as a unit. If this were strictly true there would be no relative axial motion between the turbine rotor and the pinion. There was not only the wear in the couplings to support the view that the gears and turbine rotors move relative to their housings but also the fact that on inspection the turbine and main thrust bearings showed wear due to pounding.

RESULTS OF MEASUREMENTS ON GEARS AND TURBINES OF THE USS ALABAMA (BB60)

The vibration trials carried out on the USS ALABAMA at Puget Sound covered a 2-day period, 15 and 16 January 1945, during which measurements were made on both starboard engines. As the starboard inboard system has a shorter shaft and also fewer blades per propeller (4 instead of 5) its critical speed is much higher than that of the starboard outboard system. Hence considerably more attention was given to the analysis of the motions in the starboard inboard than to the starboard outboard engine.

Details of the investigation on the ALABAMA are given in References (5) and (6). On completion of these trials, inspection of the machinery revealed undue wear in the flexible couplings of both the high- and low-pressure turbines. It is believed that the explanation of this is found in the series of graphs, Figures 16 and 17, based on the data given in Reference (5). All these data refer to the starboard inboard engine, in which the first-mode critical was found at 150 RPM, the frequency being 600 CPM. This agrees with the value found on the INDIANA during the trials of 3 August 1942; see Figure 13b.

All records obtained on the starboard inboard engine showed that the elements, both stationary and rotating, vibrated in phase, but the gears were found to have a much higher amplitude than either the gear case or the turbine rotors. The relative motion of 0.012-inch double amplitude between the high-pressure pinion and the high-pressure rotor at the 150-RPM critical speed must have been taken up in the flexible coupling. From the negligible relative motion between the high-pressure pinion and the high-pressure rotor above 170 RPM, as shown in Figure 16, it was concluded that the coupling froze beyond this speed owing to the high torque.

Torsional vibration was found to be negligible in both the engines tested.

From the curves of Figures 16a and 17 it appears that at the first-mode resonance there is little relative motion between the shaft and the gear case, whereas there is a large relative motion between the high-pressure pinion and the gear case. This seems to indicate that the flexibility between

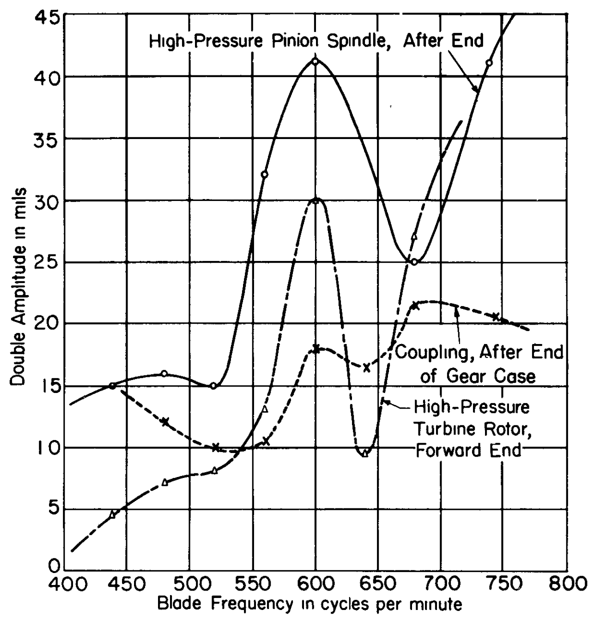

Figure 16a - High-Pressure Pinion, Coupling, and Rotor

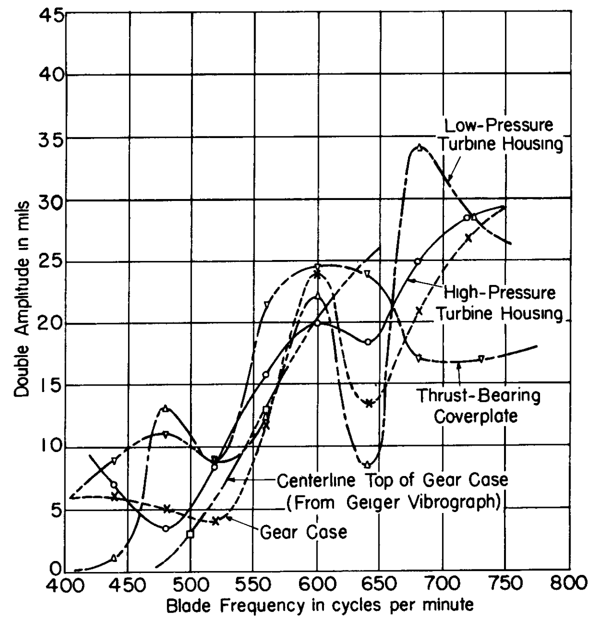

Figure 16b - Turbine and Gear Housings

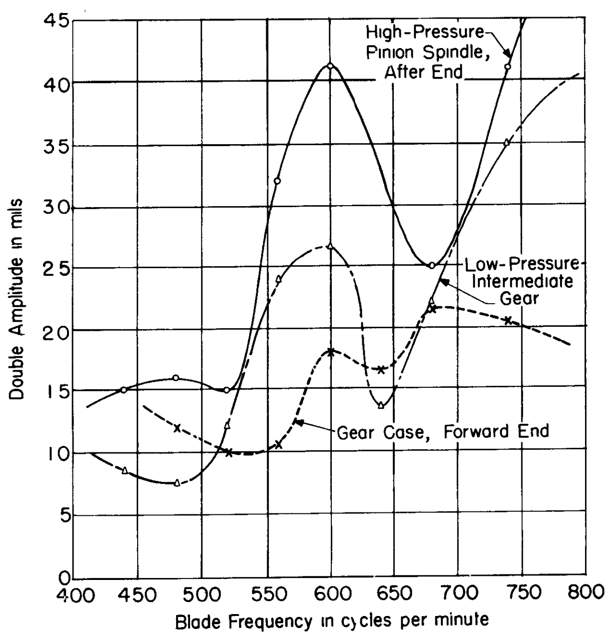

Figure 16c - Gear Case, Gears, and Pinions

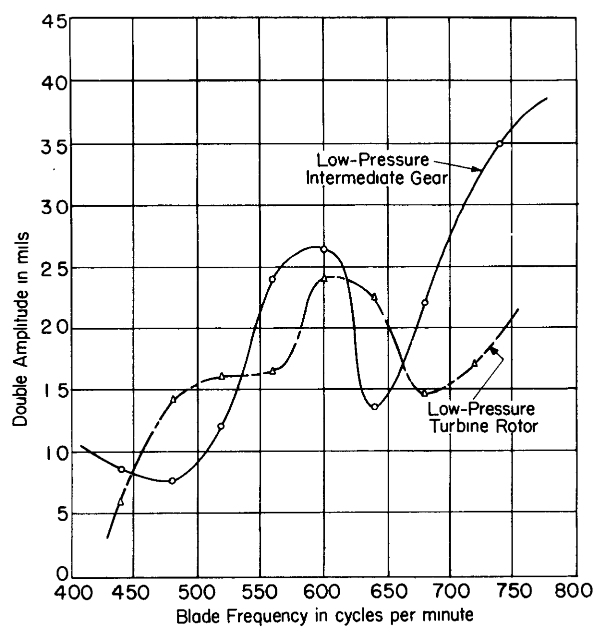

Figure 16d - Low-Pressure Rotor and Intermediate Gear

Figure 16 - USS ALABAMA (BB60) - Absolute Axial Motion of Various Machinery Components of the Starboard Inboard Engine, Plotted against Blade Frequency from Data Based on Westinghouse LE Vibrograph Records Made during Trial of 15 January 1945

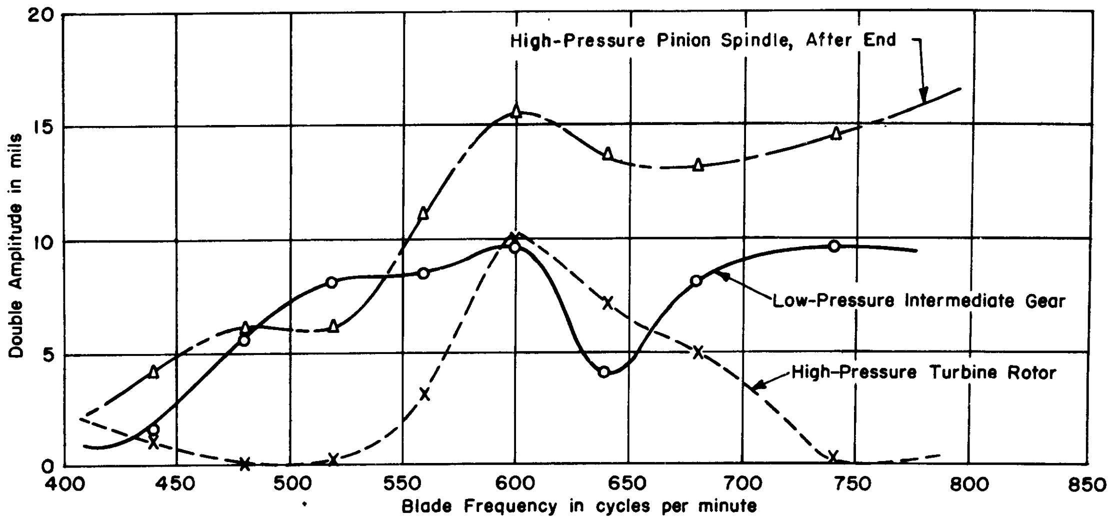

Figure 17 - USS ALABAMA (BB60) - Axial Motion of Various Machinery Components Relative to the Gear Case of the Starboard Inboard Engine, Plotted against Blade Frequency from Geiger Vibrograph Records during Trial of 15 January 1945

the pinions and the gear case lies chiefly in the gears themselves rather than in the support of the thrust bearing. This point is further discussed in the following section.

DECISIONS MADE AS A RESULT OF THE INVESTIGATION ON THE USS ALABAMA (BB60)

As a result of the measurements made on the USS ALABAMA in January 1945, the Bureau of Ships decided that extensive alterations in the propulsion systems of the BB57 through 60 Class, such as installing thrust bearings in the shaft alleys, were unnecessary. It was decided, however, to modify the design of the turbine flexible couplings to reduce the wear in the teeth. The modifications adopted were (a) to increase the coupling-tooth width from 1 3/4 inch to 2 3/4 inches; (b) to increase the hardness from 160 to between 300 and 350 Brinell for the external teeth and between 200 and 240 Brinell for the internal teeth; and (c) to increase lubrication of the coupling teeth. Further details of the modifications of the turbine couplings are given in Reference (7).

THREE-BODY ANALYSIS PROPOSED BY THE WESTINGHOUSE COMPANY

To account for the observed fore-and-aft flexibility between the rotating machinery and its casings and foundations, both the Westinghouse Company and the Newport News Shipbuilding and Dry Dock Company favored for the analysis the 3-body system shown in Figure 18. In this system m_p and k_s

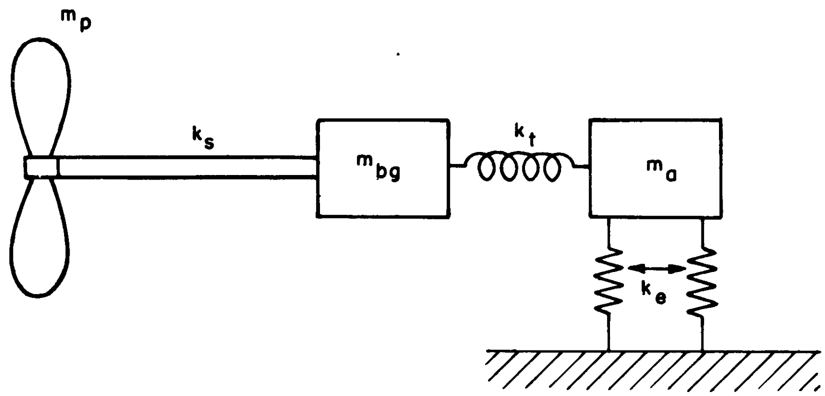

Figure 18 - A 3-Body System Representing the Propulsion System of the BB57 through 60 Class

are the same as in the 2-body system shown in Figure 3, but the machinery mass is broken up into two parts, m_{bg} representing the mass of the gear plus one-half the mass of the shaft and m_a representing the effective mass of all the rest of the machinery; k_t represents the flexibility which permits motion of the bull gear relative to the gear case and k_e represents the fore-and-aft flexibility of the machinery foundations.

In computing the natural frequencies of this lumped 3-body system the values indicated in Table 4 were used. The value of k_t given here is taken from Reference (5) and was originally computed by the Newport News Shipbuilding and Dry Dock Company for BB58. The remaining values are based on data at hand at the Taylor Model Basin and are consistent with the values previously given for the 2-body system in Table 1.

TABLE 4

Values of Constants of the Vibratory Systems Representing the Propulsion Systems of BB57 through 60 Class Based on 3-Body Analysis of Figure 18

Propulsion Unit	Constants of 3-Body System					
	m_p $\frac{\text{lb-sec}^2}{\text{in}}$	k_s $\frac{\text{lb}}{\text{in}}$	m_{bg} $\frac{\text{lb-sec}^2}{\text{in}}$	k_t $\frac{\text{lb}}{\text{in}}$	m_a $\frac{\text{lb-sec}^2}{\text{in}}$	k_e $\frac{\text{lb}}{\text{in}}$
Starboard Outboard	419	1.83×10^6	454	10×10^6	750	8.0×10^6
Starboard Inboard	350	2.75×10^6	380	10×10^6	750	8.0×10^6
Port Inboard	318	3.39×10^6	353	10×10^6	750	8.0×10^6
Port Outboard	390	2.11×10^6	426	10×10^6	750	8.0×10^6

TABLE 5

Comparison of Natural Frequencies Calculated for 3-Body Systems with Values Measured on BB57 through 60

Shaft	Calculated Frequencies, CPM		Measured Frequencies, CPM	
	First Mode	Second Mode	First Mode	Second Mode
Starboard Outboard	483	928	520	900
Starboard Inboard	565	1090	650	1100
Port Inboard	586	1184	700	1240
Port Outboard	509	978	600	980

In the estimation of the resonance frequencies, the 3-body system does not give values as near to the measured values as the 2-body system but it is of some help in forecasting the relative movements between the gears and the turbine rotors. Its success depends chiefly on the value of k_t, Figure 18, which is difficult to determine.

CONCLUSIONS

As a result of the vibration studies made on vessels of the BB57 through 60 Class and described in this report, it has been concluded that the axial vibration of the propulsion systems is not serious enough to require major alterations in the existing installations. It has, however, been decided to modify the design of the turbine couplings so as to reduce wear due to axial vibration.

For estimating the resonance frequencies of axial vibration of a propulsion system similar to that of the BB57 through 60 Class, the 2-body analysis appears to be adequate and affords a much simpler formula for the frequencies than an analysis involving three or more bodies. The apparent advantage of the 3-body system in predicting the relative amplitudes of the gears with respect to the gear case and turbines is outweighed by the uncertainty of the mass and spring constants to be used in the 3-body system.

In designing future vessels of similar proportions it probably would be preferable to locate the thrust bearings in the shaft alley, well astern, but a general assumption that this expedient will eliminate axial vibration is dangerous. When the thrust bearing is located astern, the bull gear and pinions, by virtue of the flexible couplings between the gears and the turbines, are free to vibrate axially. Analysis of such a system shows that at resonance large amplitudes can exist at the bull gear for very small amplitudes at the after thrust bearing. It is therefore essential, before making any final decision as to the location of the thrust bearing, to make a complete analysis of the vibratory system, using the best values of the mass and spring constants available from previously accumulated data.

ACKNOWLEDGMENTS

This report is based on measurements made by the David Taylor Model Basin and the Vibration Section of the New York Naval Shipyard during the initial sea trials of battleships of the BB57 through 60 Class and on subsequent measurements on the USS ALABAMA (BB60) by the Westinghouse Electric and Manufacturing Company and the Vibration and Noise Reduction Section of the Mare Island Naval Shipyard. The procedures followed on each trial were worked out in conferences with the ships' officers, in which the Bureau of Ships was represented by Captain J.P. Den Hartog, USNR, and the Taylor Model Basin by Captain A.G. Mumma, USN, and Captain J. Ormondroyd, USNR. The members of the Model Basin staff chiefly concerned with this work were F. Mintz, V.E. Benjamin, L.E. Wedding, F.B. Bryant, W.F. Curtis, and R.T. McGoldrick. The investigation of alternating thrust both at the Taylor Model Basin and on board ship was chiefly the responsibility of F.B. Bryant and W.F. Curtis, under the direction of Captain Mumma.

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(1) "Analysis of Vibration in the Propelling Machinery of the Battleships NORTH CAROLINA and WASHINGTON (BB55 and BB56)," by R.T. McGoldrick and W.F. Curtis, TMB CONFIDENTIAL Report 518, March 1945.

(2) "Construction and Operation of the Taylor Model Basin 5000-Pound Vibration Generator," by E.O. Berdahl, TMB Report 524, April 1944.

(3) "Electronic Methods of Observation at the David W. Taylor Model Basin. Part 2 - Measurements of Steady and Alternating Stresses in Rotating Shafts," by W.F. Curtis and W.J. Sette, TMB Report R-54, January 1942.

(4) Commanding Officer, USS ALABAMA letter BB60/S41-1 of 5 November 1944 to Chief, BuShips, TMB file C-S87-19/A11-(2), Subject: Fore and Aft Vibration of Main Engines and Its Effect on Main Turbine Thrust Bearings.

(5) "Vibration Investigation on USS ALABAMA, BB60 - Tests Conducted on January 15 and 16, 1945 Out of Puget Sound Navy Yard," Westinghouse Electric and Manufacturing Company Report.

(6) United States Navy Yard, Mare Island, California, Industrial Laboratory Report 2738-45, 1 February 1945.

(7) Navy Department, Bureau of Ships, Memorandum Report of Conference on 10 February 1945, TMB file C-S87-19/A11-(2).

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Further information on the subject can be gained from the following sources:

Bureau of Ships Bulletin of Information No. 9, dated 1 January 1943.

Puget Sound Navy Yard, Bremerton, Washington, CONFIDENTIAL letter BB60/S41 of 17 January 1945, "Preliminary Report on Vibration Test on Main Propulsion Machinery Held in Puget Sound 15-16 January 1945."

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