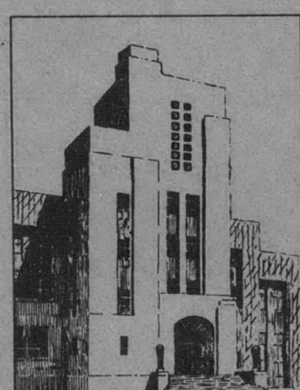
THE DAVID W. TAYLOR MODEL BASIN

UNITED STATES NAVY

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



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REPORT 518

NAVY DEPARTMENT

DAVID TAYLOR MODEL BASIN

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PERSONNEL

The accomplishments described in this report represent the combined efforts of members of the staffs of the Bureau of Ships, the New York and the Philadelphia Navy Yards, the General Electric Company, the Naval Research Laboratory, the David Taylor Model Basin, and a number of other activities.

The tests and investigations were coordinated by Captain A.G. Mumma, USN. The report is the work of R.T. McGoldrick and W.F. Curtis of the Taylor Model Basin staff. The digest was written by Captain H.E. Saunders, USN.

DIGEST*

On the first builders' trial of the new post-war battleship NORTH CAROLINA (BB55), conducted on 19 and 20 May 1941, severe fore-and-aft vibration of the propeller shafting, the main reduction gear units, the main turbines, the main condensers, and other parts of the propelling machinery was encountered. In fact, the vibration of the high-pressure and high-temperature main steam lines was so great that it was considered unsafe to run the vessel at a speed greater than 25 1/2 knots on the first trial.

This vessel and her sister ship, the battleship WASHINGTON (BB56), differed in one important respect from all previous large ships of this type in that, while the two outboard propeller shafts were carried by double-arm struts of orthodox design, the two inboard shafts were carried in deep skegs, as shown in Figures 1, 2, and 3. It was hoped in this design to obtain the high propulsive efficiency of single-screw ships for both inboard shafts. Although the early model tests indicated that this would not be achieved, the deep skegs were nevertheless retained to serve as docking keels for the after part of the vessel, to add to the longitudinal stiffness of the ship girder in a vertical plane, and to act as torpedo protection for the propellers on the unengaged side.

A study of the very considerable fore-and-aft movement of the main shafts while the vessel was running on the first trial showed that a great part of the difficulty lay in the existence of large variations in thrust, and in the presence of axial resonance of the propeller-shaft systems which accentuated those variations. The resulting periodic forces of large magnitude could not be taken care of by the main thrust bearings without causing fore-and-aft movement of those bearings and an accompanying fore-and-aft vibration of all parts of the propelling machinery. The vibratory forces were transmitted into the ship structure and manifested themselves elsewhere as rather serious vibrations of the foundations of the fire-control instruments.

A further study made it evident that the structure supporting the main thrust bearings and carrying the propeller thrust down into the hull of the ship was too limber. The use of underslung condensers and the building of the main thrust bearings into the forward ends of the large reduction-gear

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* This digest is a condensation of the text of the report, containing a description of all essential features and giving the principal results. It is prepared and included for the benefit of those who cannot spare the time to read the whole report.

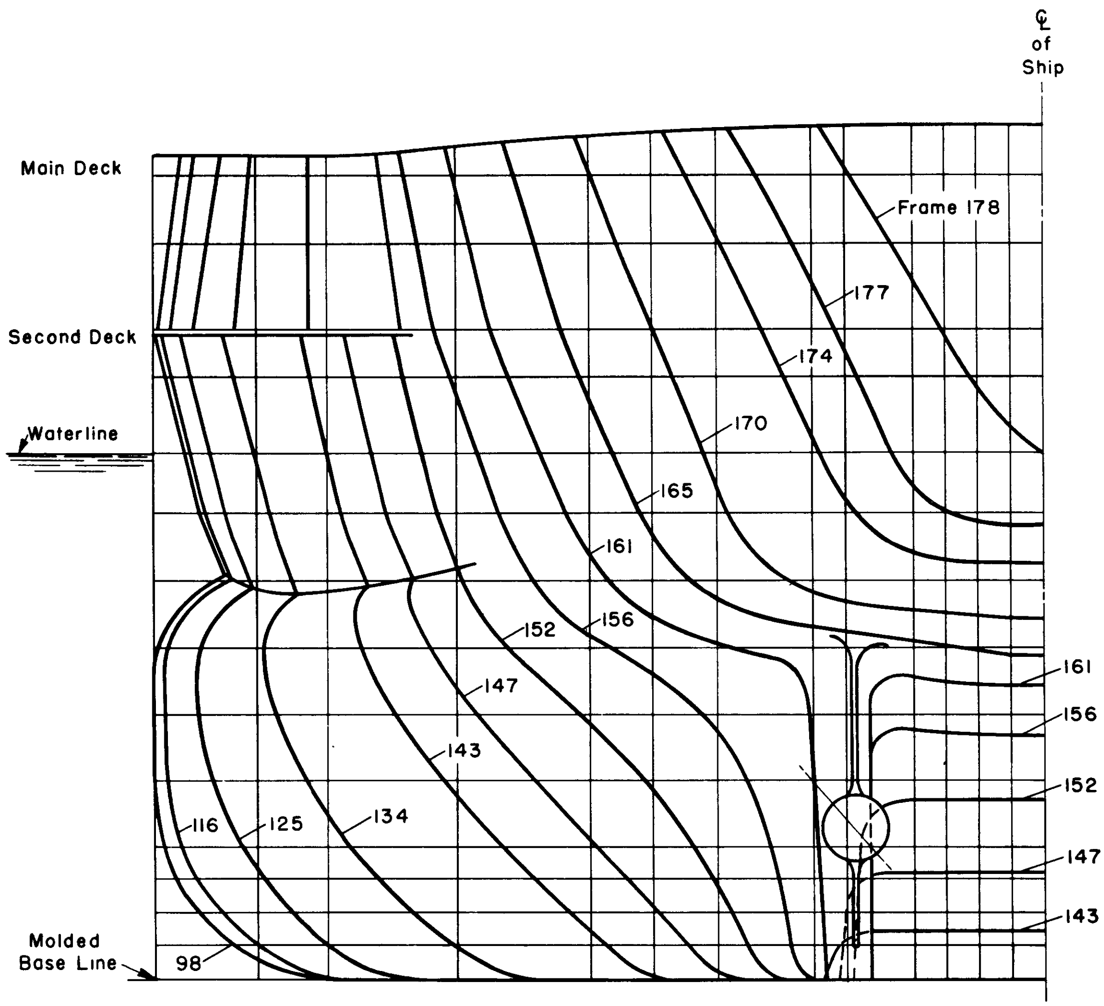


Figure 1 - Plan of Afterbody, Showing Shape of Skeg and Tunnel Between Skegs

The molded lines of the vessel are shown here. The numbers indicate frame stations, spaced 4 feet apart.

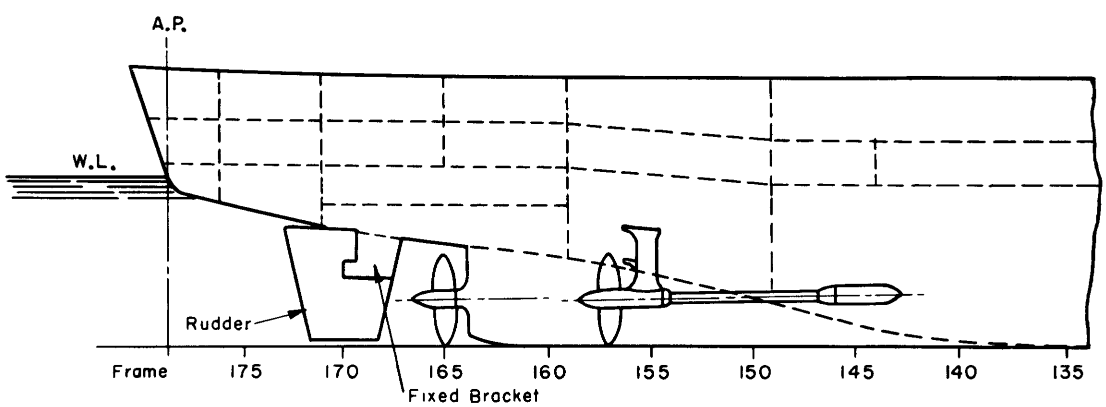


Figure 2 - Outboard Profile of Afterbody

The after propellers are inboard, the forward propellers outboard. The top of the tunnel between the skegs is shown by the broken line. There are two rudders, one behind each inboard propeller.

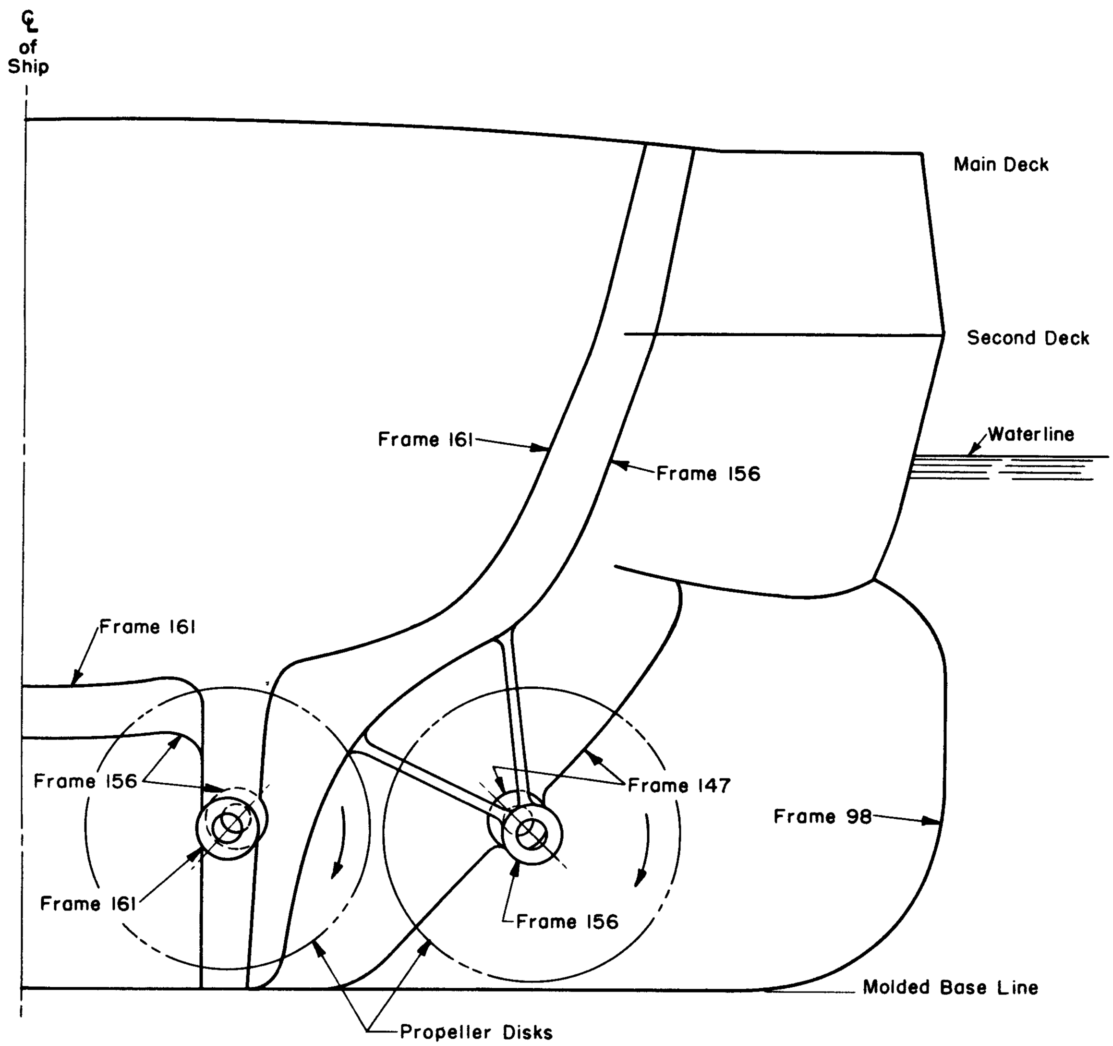


Figure 3 - Partial Body Plan Showing Position of Propellers with Reference to Stern of Ship

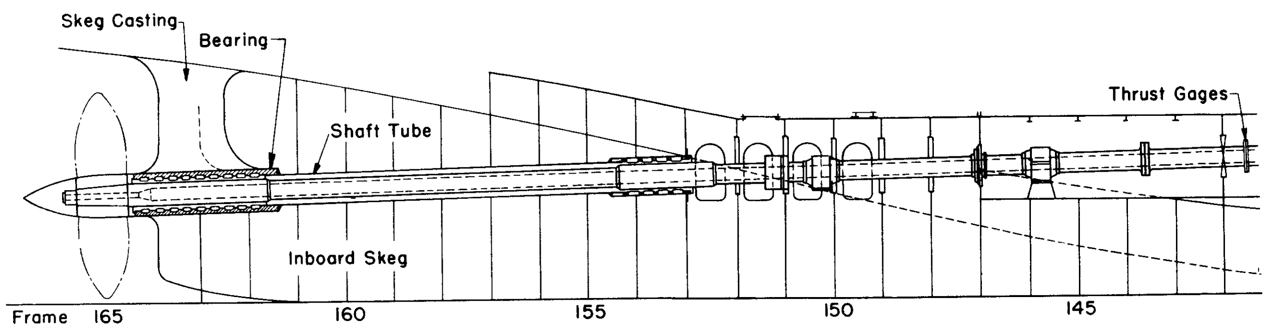
The positions of the frames, indicated by numbers on this diagram, are shown in Figure 2.

cases, as shown in Figure 4, combined to make it impossible to carry the thrust forces from the bearings into the ship structure in anything resembling a line which would be straight and which would at the same time lie at a reasonably small angle to the centerline of the shaft. The diagram of one of the main propelling units in Figure 4 indicates that this angle was in fact approaching 60 degrees.

The twin-skeg construction for carrying the inboard shafts, as described in the Introduction of the report and as referred to in subsequent sections, was indeed a novel feature for so large a ship. However, its performance had been quite thoroughly investigated by a long series of tests on

Figure 4 - General Arrangement of Propelling Machinery for Port Inboard Shaft, USS NORTH CAROLINA

This shaft is approximately 160 feet long, from the propeller to the main thrust bearing. It is the shortest shaft of the four in the ship.



self-propelled models and the benefits to be expected from it, as described previously on page iii, were considered to outweigh any possible disadvantages.

It was thought at the time of the first builders' trial on the NORTH CAROLINA that the presence of the two large rudders behind the inboard propellers might have been a contributing factor. However, large changes in rudder produced an increase in vibration which was no greater than that experienced in vessels in which vibration conditions were satisfactory. The fact that the rudders themselves were relatively steady indicated that the vibratory conditions in the propelling machinery would probably have existed even if the rudders had been removed. This matter is treated in somewhat greater detail in the report.

Taking all the foregoing matters into consideration, the design features thought likely to contribute to the longitudinal vibration encountered in the NORTH CAROLINA and WASHINGTON were:

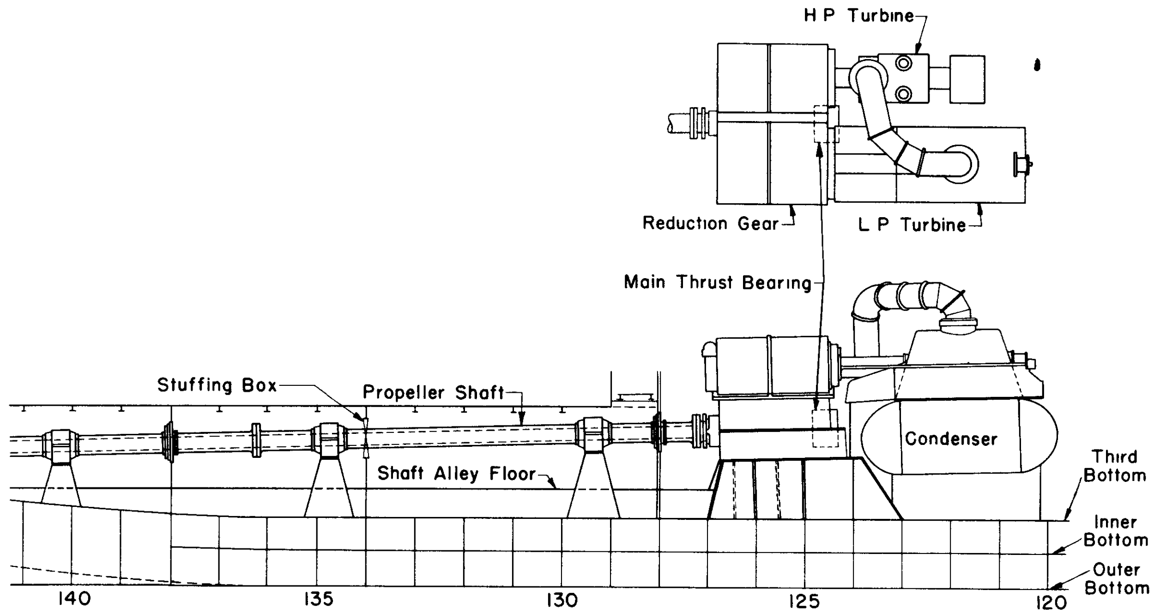
a. long propeller and line shafts, combined with the use of hollow shafts throughout,

b. high propeller speeds,

c. large engine and propeller masses,

d. high thrust-bearing foundations, inadequately braced because of the large gear case and the underslung condensers,

e. position of the inboard propellers behind the skegs and proximity of all propellers to the hull, and



f. shape of the skeg ends ahead of the inboard propellers.

Of these a, b, c, and d were thought to influence the frequency of the critical shaft speed and to account for its appearance in the running range while a, e, and f were considered responsible for the magnitude of the vibration at the critical speed.

With the analytical technique and procedure now available to the engineer (8),* it is possible to attack a problem of this kind by reducing it to fairly simple terms and then to estimate or determine by experiment the coefficients or factors needed for a solution.

For example, the propelling machinery shown in Figure 4 may be symbolized and represented by four bodies having certain equivalent masses, attached to each other by elastic members having certain stiffness characteristics. The schematic representation of this system is indicated in Figure 7.

The various masses and spring constants can be estimated, calculated, or determined by experiment, depending upon the length of time and the facilities available during the analysis. For example, it was possible, as explained in the body of this report, to isolate the units in one of the machinery spaces of the NORTH CAROLINA by uncoupling the line shaft directly abaft the main reduction gear and to set the engine and gear units in vibration by a special vibration generator. The system then became one of three bodies, as shown in Figure 8 on page 17 of the report.

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* Numbers in parentheses indicate references on page 87 of this report.

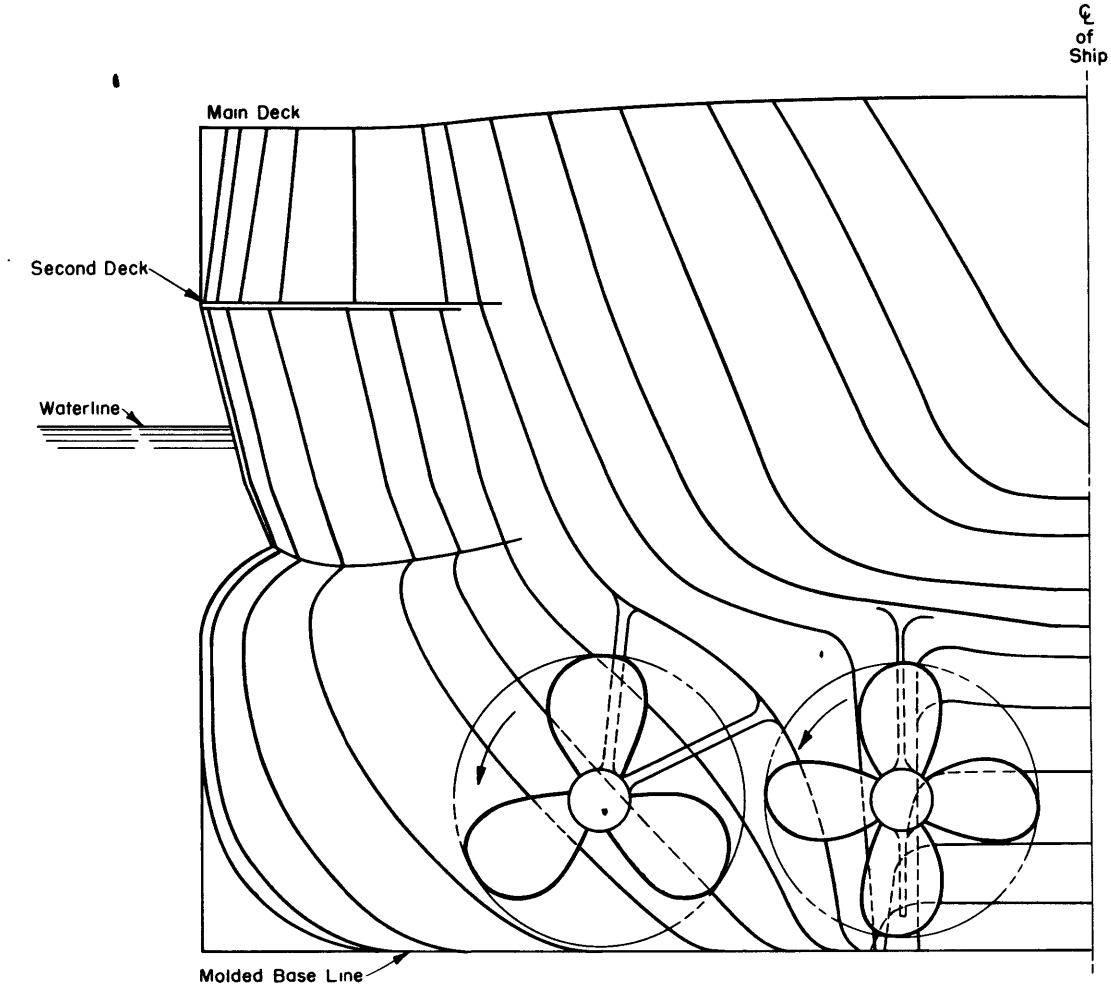

Figure 6 - Plan of Afterbody of NORTH CAROLINA Class, with Original Propellers and Struts Superposed

The propellers shown in this diagram are those fitted for the first builders' trial on each vessel; see Table 1 on page 31 of the report.

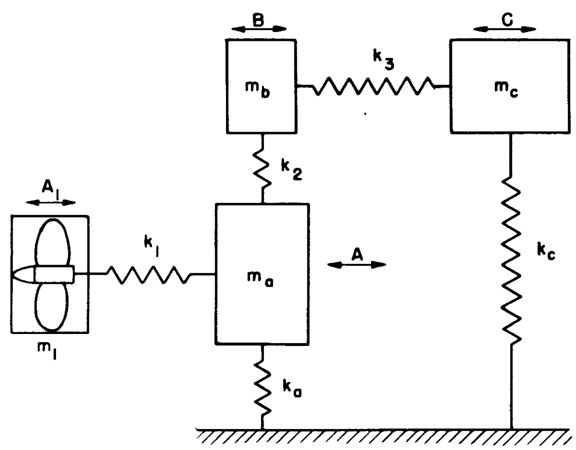

Figure 7 - Schematic Representation of a 4-Body Elastic System

This diagram represents the four principal parts of one unit of propelling machinery, corresponding to Figure 4 on pages vi and vii. The various masses and springs are described in the text of the report. The rectangle drawn around the propeller represents the entrained water vibrating with it. Although in this schematic representation some springs are vertical and some horizontal, their elastic constants in the *horizontal* direction are represented by the k's.

Special equipment was devised and installed on both the NORTH CAROLINA and the WASHINGTON, as illustrated diagrammatically in Figure 14, to measure the alternating components of thrust in the propeller shafts under all operating conditions; this gear is described in detail in Reference (3). With this aparatus it was possible to obtain records of the type shown in Figure 16, in which there is shown the variations in compressive strain in the propeller shaft, resulting from a combination of variable thrust from the propeller blades and resonant frequency in the axial mode of vibration of the shaft, propeller and engine.

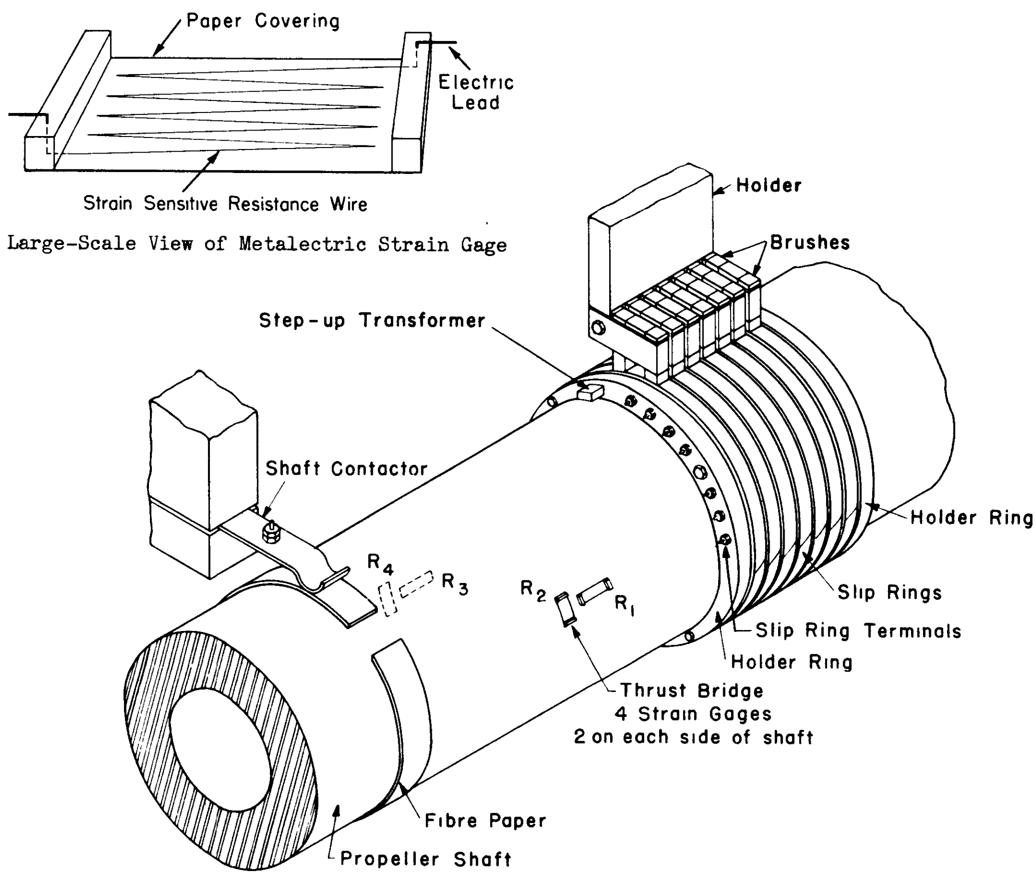

Figure 14 - Diagram of Thrust Gages and Accessories, Mounted on a Propeller Shaft

Four metalectric or wire-resistance strain gages, shown diagrammatically in the large-scale detail, are cemented to the shaft, two in the positions R_1 and R_3 and two in the positions R_2 and R_4. Signals from these gages are taken off through four slip rings, as explained in Reference (3), and are impressed on an oscillograph, which produces a record of the type shown in Figure 16.

The slip rings are insulated from the shaft and from each other. Three of the four extra rings are for torque-measuring gages, not shown here.

The contact maker at the left is for indicating on the record the position of each propeller blade with reference to the vertical.

See Figures 15 and 17 for details of the slip rings.

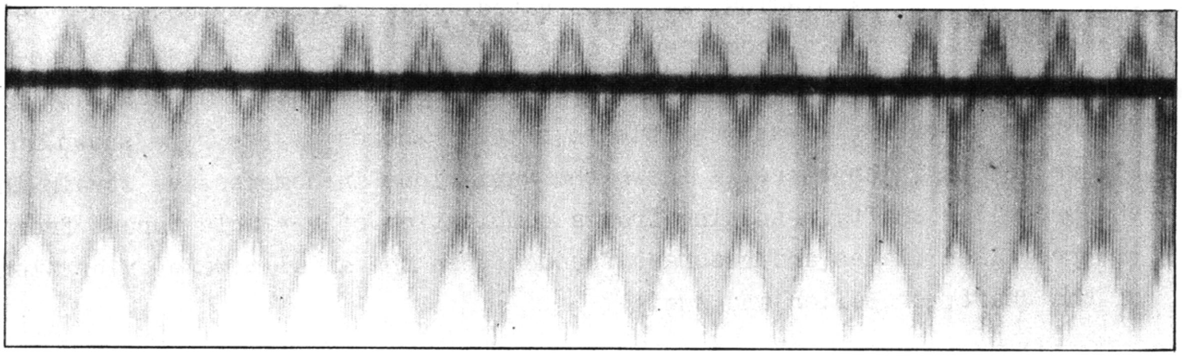

Figure 16 - Typical Oscillogram from Thrust Strain Gages, Showing Variations in Shaft Thrust

The rapid fluctuations are caused by the carrier wave. The variations in thrust for each blade are quite large.

Data were taken during the early trials of the NORTH CAROLINA with these thrust gages and other instruments and were quickly analyzed. From these data it was found that the resonant frequencies of the lowest mode of fore-and-aft vibration of all four of the propelling machinery systems, with the propellers then in use, were at or near full power, that the single amplitudes of vibration at the after ends of the reduction-gear cases were of the order of 0.030 to 0.038 inch, and that the thrust variations at blade frequency, especially on the inboard shafts, could be expected to reach plus and minus 60 per cent of the steady thrust at full power.* This magnitude of thrust variation was due to resonance; the actual variation in thrust at the propellers was less than 12 per cent.

Numerous remedies were proposed by all the activities working on this problem. Special trials were run on the ships and on the model, and tests were made to investigate various features of the problem. The results were then carefully analyzed, following which steps were taken simultaneously to work up several solutions, so that one would be available if the others failed.

The various remedial measures proposed were substantially as follows:

1. Stiffening the main thrust-bearing foundations. Some alterations of this kind were made on the WASHINGTON (BB56) between the first and second trials of that vessel; others were made on that vessel after those trials.

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* This meant that the thrust varied, at blade frequency, from 40 to 160 per cent of the mean thrust.

2. Reducing the fore-and-aft clearance in the main thrust bearings. However, this was already about as small as the design of the pivoted segmental bearing would permit.

3. Moving the thrust bearings from their positions in the forward ends of the main reduction-gear cases to positions in the shaft alleys as close as possible to the propellers.

4. Designing, constructing, and installing shaft-restraining blocks which would act as stationary dashpots around the rotating shafts, as indicated in Figure 9, and which would, by main force, restrain the shafts from vibrating or moving more than a negligible amount in a fore-and-aft direction.

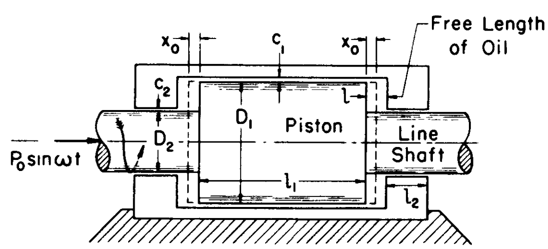

Figure 9 - Diagram of Shaft-Restraining Block

The piston, secured to the shaft, rotates with it in a fixed chamber with as small a diametral clearance as practicable. The interior of the chamber around the shaft and piston is filled with oil. Longitudinal movement of the shaft is only possible by squeezing the oil from one end of the chamber to the other.

5. Fitting propellers with numbers of blades such that the correspondence between the blade-frequency exciting force and the natural frequency of the propulsion system would occur either beyond the running range or at a low RPM in that range, where the powers and forces would be relatively small and the resultant vibration would be acceptable.

6. Installing pendulum dampers on the rotating shafts to counterbalance the vibratory forces.

7. Introducing one or more lengths of line shafting of greater axial resilience than the existing shafting, or one or more couplings with axial flexibility (9), to lower the resonant frequency of the shaft-propeller system in the longitudinal direction.

8. Modifying the after ends of the inboard skegs, especially the portion below the shafts, to provide greater clearance between the fixed ship structure and the moving propellers, and to smooth out the inequalities in the wake behind the skegs.

9. Increasing the propeller clearances by moving the propellers farther from the hull or raking the blades.

10. Skewing the propeller blades so that the inequalities of flow in the wake behind the skegs would, to a certain extent, be averaged out.

The second builders' trials of the USS WASHINGTON, built by the Philadelphia Navy Yard, were held on 21 June 1941. For these trials the original 3-bladed outboard propellers were cut down from a diameter of 17 feet 3 inches to a diameter of 16 feet 4 5/8 inches and they were installed on the inboard shafts, with the expectation that resonance in the longitudinal mode would then be sufficiently above the running range of the machinery

Figure 10 - BB55 and 56 Class, 3-Bladed Propeller Cut Down in Radius and Shifted from Outboard to Inboard Shaft

The inboard skeg is clearly visible behind the propeller.

to avoid objectionable vibration at full power. The original 4-bladed inboard propellers were installed on the outboard shafts without change; it was hoped by this change that resonance in these shaft systems would occur at such a low power as not to be objectionable. The propellers in these positions are shown in Figures 10 and 11.

Figure 11 - BB55 and 56 Class, 4-Bladed Propeller Shifted from Inboard to Outboard Shaft

This photograph shows the relation between the outboard shaft and the skeg. The rudder is visible in the background just behind the inboard propeller.

With this propeller combination a speed of over 27 knots was attained on the WASHINGTON. The vibration at that speed was not so severe as to endanger any part of the power plant but it was still sufficient to interfere seriously with operation of some of the gun directors. The complete test data are set forth on pages 34 to 52 of the report.

As the distribution of power among the four shafts was uneven, the inboard shafts taking more power than their share, the 3-bladed inboard propellers were cut down still further by taking 3 inches more off the tips, and additional trials were held on the WASHINGTON on 12 July 1941. The powers were found to be equalized fairly well, and the vibration was reduced sufficiently so that the vessels could work up to full speed and could carry on with their training exercises.

In the meantime new propellers of 5-bladed design were built for the inboard shafts, together with re-designed 4-bladed propellers for the outboard shafts. The characteristics of these propellers are listed in Table 1 on page 31 of the report.

Further trials of the WASHINGTON with these new propellers were held on 2 December 1941; a summary of the thrust forces observed during these trials is given in Table 7.

From the last group of entries in Table 7 it will be noted that although the variation of thrust on the inboard shafts with the new 5-bladed propellers was still of the order of plus and minus 60 per cent at resonance, this occurred well down in the speed range, where the powers were low. The actual thrust variations were of the order of only 48,000 to 50,000 pounds, whereas at resonance with the original 4-bladed propellers on the inboard shafts they had been of the order of 90,000 to 120,000 pounds.

Meanwhile a number of special tests were being carried out. During June and July of 1941 the shaft restraining block was studied intensively. This work included building and testing two scale models of restraining blocks at the David Taylor Model Basin (10). One engine of the NORTH CAROLINA was tested with a vibration generator on 15 July 1941, in order to evaluate some of the vibration constants of the system.

A 30-foot self-propelled model representing both ships was built, and special equipment was designed and constructed, as shown in Figure 20, to measure the longitudinal vibration-exciting forces in the four shafts with various model propellers. The first test of this nature was held at the Taylor Model Basin on 27 and 28 October 1941; this was chiefly useful in developing the method of measurement. Further model tests were made on 12 November 1941 and on 26 August 1942. Model tests to determine the effect of a possible skeg modification on this class was conducted on 20 October 1942.

TABLE 7

Summary of Thrust Forces in Shafts

Shaft Location	Port		Starboard	
	Outboard	Inboard	Inboard	Outboard
NORTH CAROLINA 20 May 1941				
Number of Propeller Blades	3	4	4	3
Maximum Variation from Steady Thrust, pounds single amplitude			71,300	
Speed at which Maximum Occurred, RPM			148††	
Indicated First-Mode Resonance, cycles per minute			*	
Corresponding Steady Thrust,† pounds			111,000	
Plus and Minus Variation of Thrust as Percentage of Steady Thrust			64	
NORTH CAROLINA 27 May 1941				
Number of Propeller Blades	3	4	4	3
Maximum Variation from Steady Thrust, pounds single amplitude	73,600	121,500	87,300	63,000
Speed at which Maximum Occurred, RPM	180	170	150	170
Indicated First-Mode Resonance, cycles per minute	*	680	600	510
Corresponding Steady Thrust,† pounds	176,000	157,000	114,000	157,000
Plus and Minus Variation of Thrust as Percentage of Steady Thrust	41	77	76	40
WASHINGTON 21 June 1941				
Number of Propeller Blades	4	3	3	4
Maximum Variation from Steady Thrust, pounds single amplitude	36,900**	61,200	92,350	16,240**
Speed at which Maximum Occurred, RPM	200	200	200	200
Indicated First-Mode Resonance, cycles per minute	600	*	600	520
Corresponding Steady Thrust,† pounds	236,000	236,000	236,000	236,000
Plus and Minus Variation of Thrust as Percentage of Steady Thrust	15	25	38	6
WASHINGTON 12 July 1941				
Number of Propeller Blades	4	3	3	4
Maximum Variation from Steady Thrust, pounds single amplitude	58,600**	No data	101,500	21,760**
Speed at which Maximum Occurred, RPM	190		190††	190
Indicated First-Mode Resonance, cycles per minute				520
Corresponding Steady Thrust,† pounds	205,000		205,000	205,000
Plus and Minus Variation of Thrust as Percentage of Steady Thrust	28		50	16
WASHINGTON 2 December 1941				
Number of Propeller Blades	4	5	5	4
Maximum Variation from Steady Thrust, pounds single amplitude	21,800	47,700	50,500	18,800**
Speed at which Maximum Occurred, RPM	150	136	120	195
Indicated First-Mode Resonance, cycles per minute	600	680	600	560
Corresponding Steady Thrust,† pounds	114,000	92,000	73,000	220,000
Plus and Minus Variation of Thrust as Percentage of Steady Thrust	19	51	69	8

* The test did not extend to a sufficiently high speed to determine resonance.

** Maximum occurred in the neighborhood of second-mode resonance, which is apparently near full speed.

† Taken from model tests.

†† The highest speed for which a reliable measurement is available.

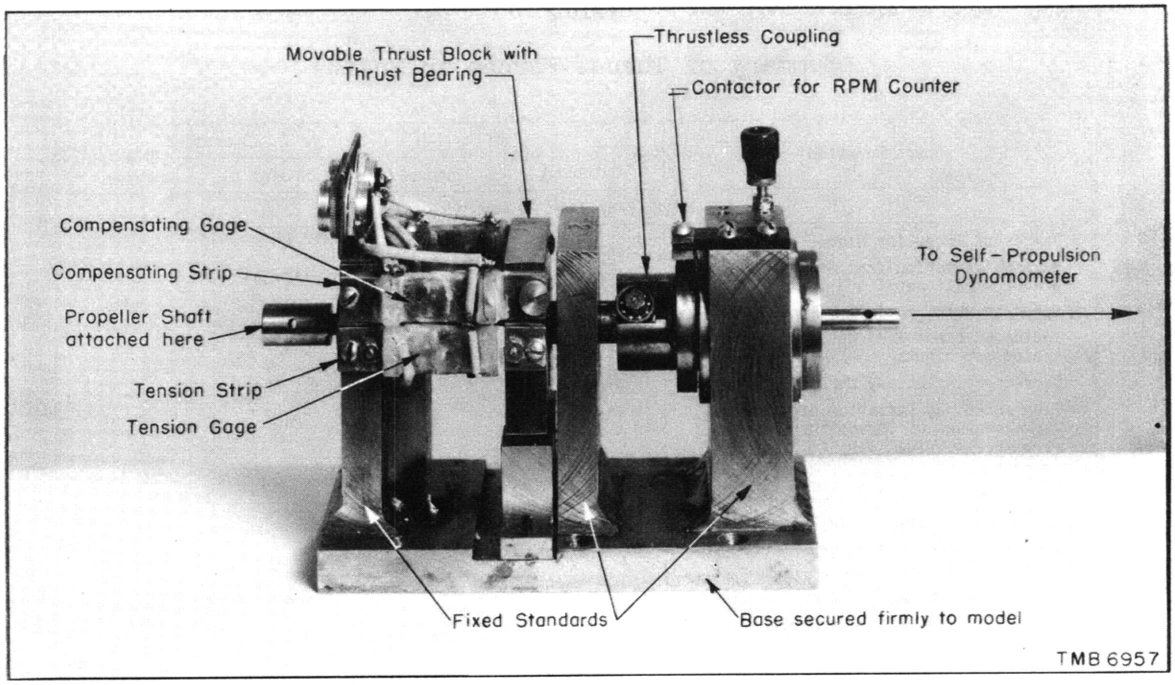

Figure 20 - Thrust Pickup Used on 30-Foot Self-Propelled Model

A short shaft section at the left is coupled to the propeller shaft of the model, and is mounted in journal bearings in the two fixed standards at the left. A thrustless fork coupling at the right-hand end of this short shaft enables the self-propulsion dynamometer to drive it at the required speed.

The left-hand shaft section is pushed against the movable thrust block through a thrust collar; all propeller thrust is thus delivered to this block. Ball thrust bearings are not used as these bearings generate strong vibrations as they rotate.

The thrust block is connected to the left-hand fixed standard by two bronze tension strips, one on each side abreast the shaft, upon which metalectric gages are mounted.

Compensating gages mounted on similar strips alongside, free from the effect of thrust, take care of temperature and other variations and form the third and fourth arms of the bridge.

Several features stand out rather prominently from a preliminary examination of the test data.

In spite of the more or less symmetrical disposition of the two parts of the inboard skegs, above and below the shaft centerline, and regardless of the number of blades on the inboard propellers, the exciting force generated in that propeller position is predominatly of blade frequency. This and other evidence bears out the original thought that the blade closest to the ship's main hull is the blade most responsible for the vibration.

Likewise, regardless of the angle between the two arms of the outboard struts and any correspondence between that angle and one of similar magnitude on an outboard propeller, the blade passing close to the hull is the one which generates the significant exciting force.

Because of the well-known phenomenon of magnification at resonance in an elastic vibrating system, a relatively small periodic exciting force at the propeller can produce dangerously high loads and amplitudes if the periods of the two coincide.

Whenever the blade frequency becomes equal to the resonant frequency of any elastically-mounted unit, such as a fire control instrument, that unit can be expected to vibrate in unison. In addition, if the blade frequency forces are magnified by resonance in the propelling machinery and its foundations, the vibration can be expected to be serious.

Blade frequency forces and the vibration produced by them, other things being equal, may be expected to increase with the ship speed and the propeller thrust, at a rate corresponding to the propeller thrust.

It is to be expected that in general structural modifications will change the natural frequencies of the structure and that propeller changes will modify the frequencies and amplitudes of the exciting forces.

Analysis of the effect of propeller changes shows that the use of either 3-bladed or 5-bladed propellers behind the inboard skegs respectively raised the critical speeds up to or beyond the running range or lowered them to points inside the range where the powers and forces were moderate. In the former case, the resonance RPM was above the running range for the port inboard shaft but not for the starboard inboard shaft. In the latter case, with 5-bladed propellers, the exciting forces were much smaller than at full power, with a corresponding reduction in the thrust variation forces at the thrust bearing, even though in this case the shaft systems were running at resonance.

Blade frequencies in the speed range will of course be higher with 5-bladed propellers, so that stiffer parts of the ship which did not vibrate in resonance when 3-bladed or 4-bladed propellers were fitted, may be found to vibrate within the running range at the higher frequency.

While there were not sufficient data available to evaluate accurately the various factors for the simplified 4-body, 3-body, and 2-body elastic systems representing the ship installation, an excellent beginning was made along lines which will in the future enable predictions to be made of the vibration characteristics of engine-shaft-propeller systems of vessels in the design stage.

In the same manner, more thorough studies of the wake and the behavior of propellers in disturbed water, given great impetus in this investigation, will continue until it will be possible to predict the magnitude of the exciting forces in the propeller under any combination of circumstances.

Of all the remedies which were proposed to eliminate the vibration on the NORTH CAROLINA and the WASHINGTON, only the one involving changing the number of blades and the design of the propellers was carried through as a reasonably systematic full-scale test program. Both the first and the second series of 5-bladed propellers showed efficiencies comparable with those of good 4-bladed and 3-bladed propellers of modern design, so that the use of propellers with 5 blades need no longer be limited by considerations of efficiency if there are other important advantages to be gained.

Although it was considered a practical impossiblity to relocate the thrust bearings aft near the propellers on the NORTH CAROLINA and the WASHINGTON, the problem of thrust-bearing location was studied theoretically in a quite general manner by a somewhat idealized representation of a ship's propulsion system in which the fore-and-aft location of the thrust bearing is regarded as a design parameter to be selected to suit the case. In practical propulsion systems the length of shaft from the propeller to the reduction gear, the cross section of the shaft, the mass of the reduction gear, thrust block and propeller, and the practicable stiffness of the thrust-block foundation, are all more or less fixed by considerations which have little to do with the vibration problem. If these quantities are assumed to be given, it is found that there is for each design a location of the thrust block that makes the frequency of the fundamental mode of axial vibration a maximum.

In some of the designs studied in this way, it was found that by using a 3-bladed propeller and making the resonant frequency of the lowest mode as high as possible, the critical shaft RPM would be well above the designed full-power RPM. This would usually be a much better solution than having the resonant frequency or critical shaft RPM within the running range.

Usually the main thrust bearing can be located at or near the position required for maximum frequency, as determined from a study of the elastic characteristics of the system. Where this can be done, and where the resonant frequency of the lowest mode can be placed above the highest exciting blade frequency, it may be a satisfactory solution of the problem of axial shaft vibration. As noted previously, the presence of thrust variation in itself seems not to be a problem except when this variation is greatly amplified by resonance. Proper placing of the main thrust bearing will often avoid resonance completely.

In this connection it may be well to point out here a distinct improvement which can be made in the design and installation of main thrust bearings in high-powered vessels, especially where the shafts are long and it is mandatory that the thrust bearing restrain the shaft from longitudinal movement at that point. Figure 45 shows a proposed type of foundation with a long base length, to prevent tilting and deformation, and a barrel-type thrust-bearing casing, designed to transmit the thrust load to the foundation in line with the center of the shaft, instead of a considerable distance below it, as is the case with existing installations.

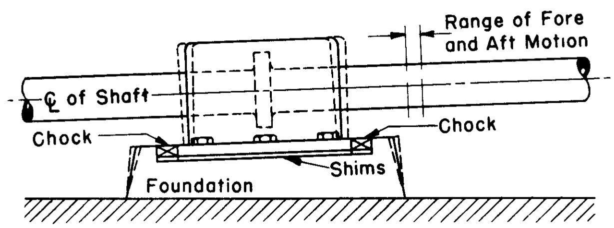

Figure 45a - Orthodox Thrust-Bearing Foundation with Short Base Length and Pedestal-Type Casing

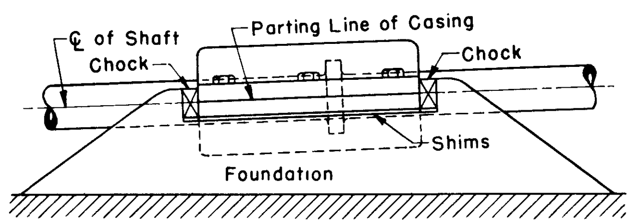

Figure 45b - Recommended Thrust-Bearing Foundation with Long Base Length and Barrel-Type Casing

Figure 45 - Types of Propeller-Shaft Thrust Bearings

The broken lines in Figure 45a are intended to represent, in exaggerated fashion, the manner in which an installation of this kind can deform under the influence of variable thrust forces.

The variations in thrust due to individual blades passing the after end of a skeg can be lessened appreciably by modifications to the skeg ending and by changing the clearances ahead of and abreast the propeller. They can also be modified by changing blade profiles and other characteristics of the propeller. The latter was done on the final design of 5-bladed propellers, but model tests showed that no appreciable improvement could be made to the NORTH CAROLINA class skegs without changes so drastic that they will probably never be undertaken.

In conclusion the following recommendations are made for future construction:

1. The main thrust bearings should preferably be located aft of rather than in the gear case; the particular location will depend upon a complete analysis of the elastic system.

2. A barrel type of thrust-bearing housing should be developed and adopted, with chocks at the level of the centerline of the shaft, and with a length of foundation at least five times its height.

3. The clearances between the propeller and the hull and the propeller and the skeg should be made as large as practicable. This has been the aim of ship designers for many years past.

4. Coupling between the propeller and the skeg should be reduced by offsetting the shaft center from the vertical axis of the skeg section, and by skewing the blade profiles.

5. Tests on large models should be made to predict the amount and nature of thrust variations in advance. This should be done for all major vessels during the design period.

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ANALYSIS OF VIBRATION IN THE PROPELLING MACHINERY OF THE BATTLESHIPS *NORTH CAROLINA* AND *WASHINGTON* (BB55 and BB56)

ABSTRACT

On the first sea trials of the new post-war battleship NORTH CAROLINA (BB55), severe fore-and-aft vibration of the main reduction-gear units, the main turbines, and the main condensers was encountered. Longitudinal vibrations of other engine-room structures were also in evidence, as well as vibration of certain fire-control directors and structures in other parts of the ship. Trials of the sister ship, the battleship WASHINGTON (BB56), produced exactly the same results.

The vibrations were found to be of blade frequency, excited by periodic fore-and-aft impulses from the propellers. The resonant peak of the vibration was encountered at a critical speed in the running range of the machinery, owing to the relatively long shafts and to excessive elasticity in the thrust-bearing foundations, which were mounted as part of the large reduction-gear housings.

The frequency and amplitude of the vibrations were measured at selected locations on the ship; maximum single amplitudes of the order of 0.038 inch were found. The machinery vibration was analyzed on the basis of 4-body, 3-body, and 2-body elastic systems, and the essential constants of the principal vibrating system were computed.

A large number of expedients for overcoming the vibration were considered. Of these several were discarded as being impractical to apply to ships already finished; still others were rejected for various practical reasons.

Of the remaining expedients, the most useful was changing the number of propeller blades. The original 3-bladed propellers on the outboard shafts were replaced by 4-bladed wheels, and the 4-bladed inboard propellers by 3-bladed wheels. This change, which did not produce entirely satisfactory results, nevertheless permitted the ships to fulfill their missions during the war. A final installation of 5-bladed propellers inboard, with 4-bladed propellers outboard, was made in April 1944. The vibrations in the ships while they are operating with these propellers are reduced to magnitudes which are quite acceptable.

INTRODUCTION

On the first builders' trial of the new post-war battleship NORTH CAROLINA (BB55), conducted on 19 and 20 May 1941, severe fore-and-aft vibration of the propeller shafting, the main reduction-gear units, the main turbines, the main condensers, and other parts of the propelling machinery was encountered. In fact, the vibration of the high-pressure and high-temperature main steam lines was so great that it was considered unsafe to run the vessel at a speed greater than 25 1/2 knots on the first trial. The designed trial speed was 27 1/2 knots.

This vessel and her sister ship, the battleship WASHINGTON (BB56), differed in one important respect from all previous large ships of this type in that, while the outboard propeller shafts were carried by double-arm struts of orthodox design, the two inboard propeller shafts were carried in deep skegs, as shown in Figures 1, 2, and 3. It was hoped in this design to obtain the high propulsive efficiency of single-screw ships for both inboard shafts.* Although the early model tests indicated that the results hoped for would not be achieved, the deep skegs were nevertheless retained to serve as docking keels for the after part of the vessel, to add to the longitudinal stiffness of the ship girder in a vertical plane, and to act as torpedo protection for the propellers on the unengaged side.

It had been anticipated before the first sea trials of the NORTH CAROLINA that some *transverse* vibration would be encountered in the skegs carrying the inboard shafts, especially as the resonant frequency of these skegs in that mode of vibration was known to be below the propeller-blade frequency at full power (1) (2).** Special instruments were installed in the skeg of the NORTH CAROLINA on the first trial for detecting and measuring the transverse vibrations, but the magnitude of these vibrations was found to be so small that they did not affect the operation of the ship.

The longitudinal vibration of the propelling machinery, on the other hand, was unexpected, although longitudinal thrust-measuring equipment had been installed on the shafts for this trial. All available activities were quickly concentrated on a study of the situation to determine the causes and to work out a remedy. When it is remembered that all this was taking place at the time that the German battleship BISMARCK was making its sortie into the North Atlantic, the seriousness of the situation as it appeared at the time can well be imagined.

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* A David Taylor Model Basin report describing the development of the twin-skeg stern for large vessels is now in preparation.

** Numbers in parentheses indicate references on page 87 of this report.

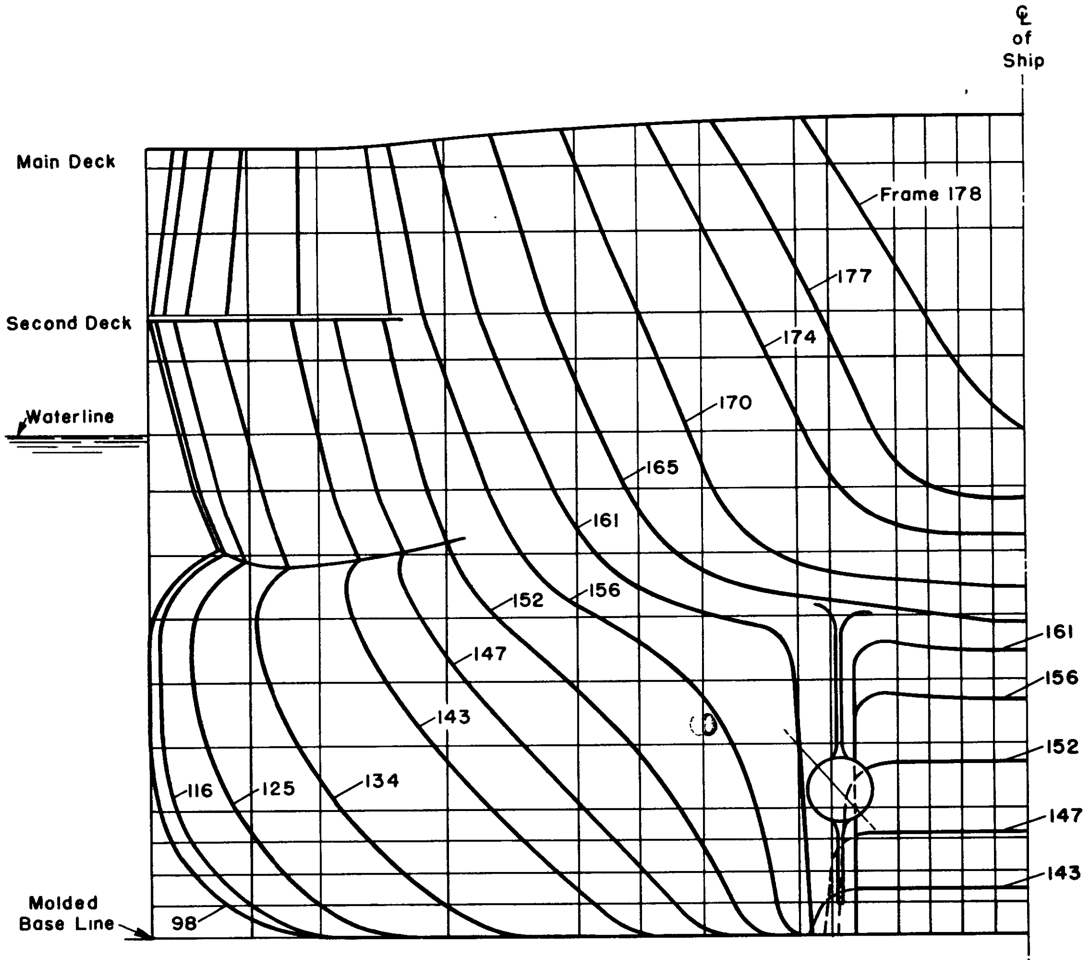


Figure 1 - Plan of Afterbody, Showing Shape of Skeg and Tunnel between Skegs

The molded lines of the vessel are shown here. The numbers indicate frame stations, spaced 4 feet apart.

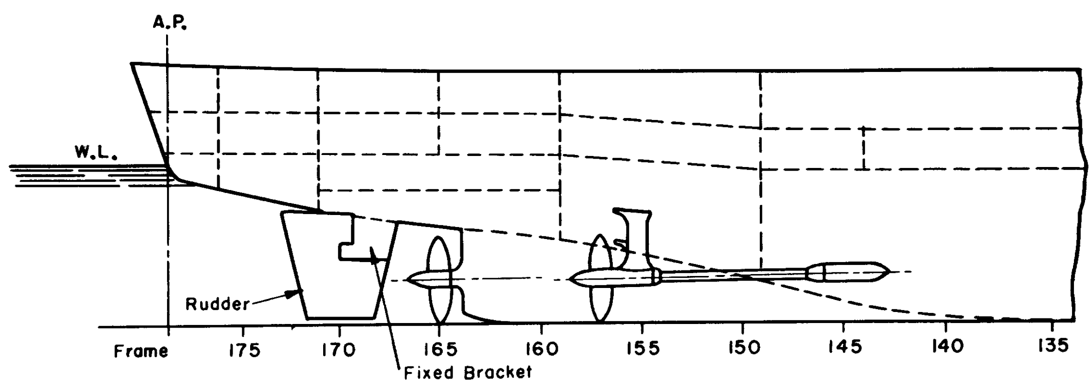


Figure 2 - Outboard Profile of Afterbody

The after propellers are inboard, the forward propellers outboard. The top of the tunnel between the skegs is shown by the broken line. There are two rudders, one behind each inboard propeller.

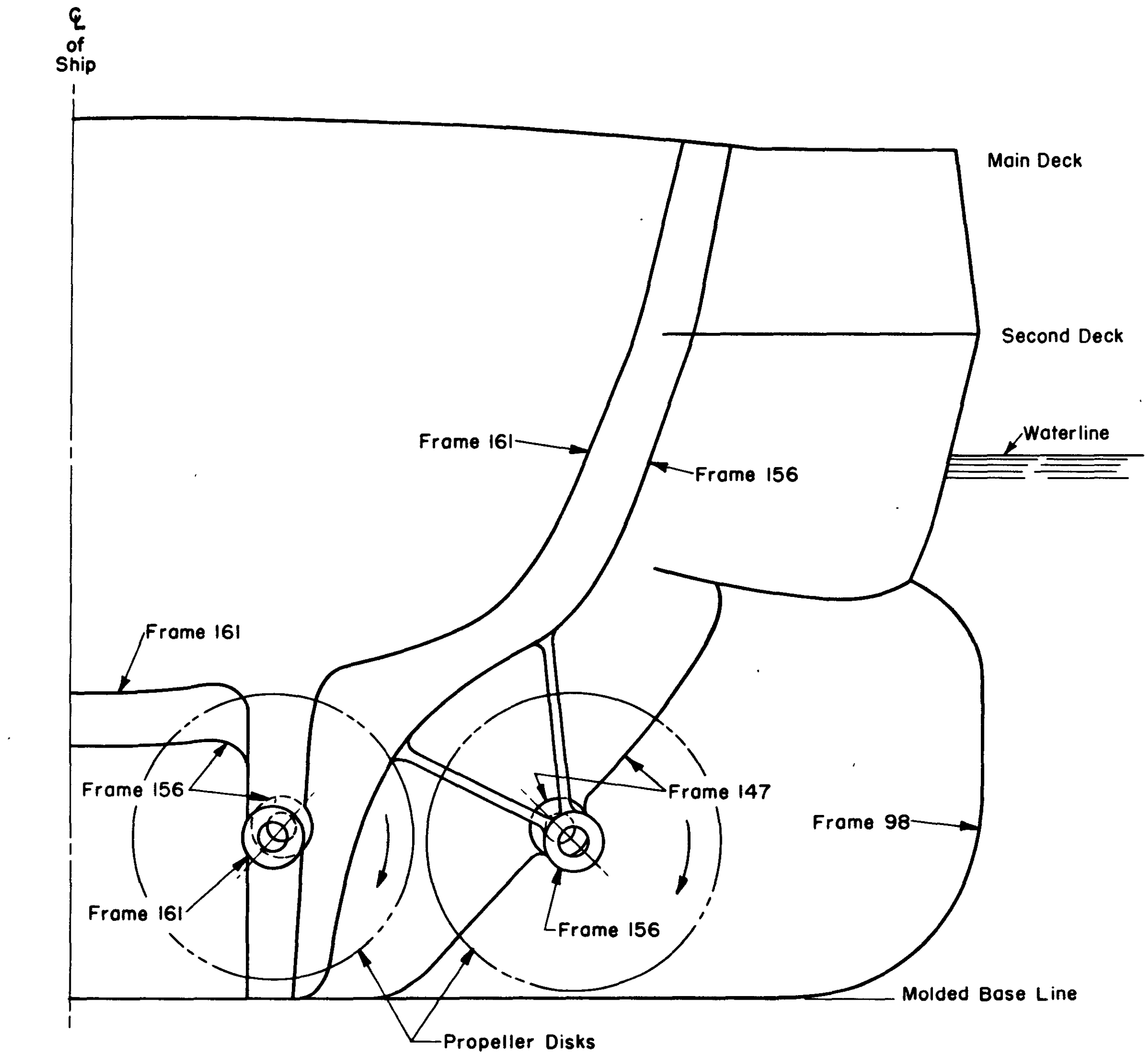


Figure 3 - Partial Body Plan Showing Position of Propellers with Reference to Stern of Ship

The positions of the frames, indicated by numbers on this diagram, are shown in Figure 2.

After observation of the very considerable fore-and-aft movement of the main shafts while the vessel was running on the first trial, it was quite evident that a great part of the difficulty lay in the existence of large variations in thrust, and of conditions of resonance which accentuated those variations. The resulting periodic forces could not be taken care of by the main thrust bearings without causing fore-and-aft movement of their foundations and an accompanying fore-and-aft vibration of all parts of the propelling machinery. The vibratory forces were transmitted into the ship structure and manifested themselves elsewhere as rather serious vibrations of the fire-control instruments and other equipment.

It so happened that at about this time the David Taylor Model Basin, following some preliminary work by Professor A.C. Ruge of the Massachusetts Institute of Technology, had developed a technique whereby it was possible to mount sensitive wire-resistance strain gages on a rotating shaft and to record, by oscillographic apparatus, the variations in torque and thrust which took place throughout one revolution of the shaft (3). Gages of this kind were mounted on the starboard inboard shaft of the NORTH CAROLINA during the first builders' trial on 19 and 20 May 1941. Before the second builders' trial on 27 May 1941, the experimental setup of this equipment on one shaft of the NORTH CAROLINA was expanded to include all four shafts, in an effort to determine the magnitude of the thrust variations by measurements in the propeller shafts themselves.

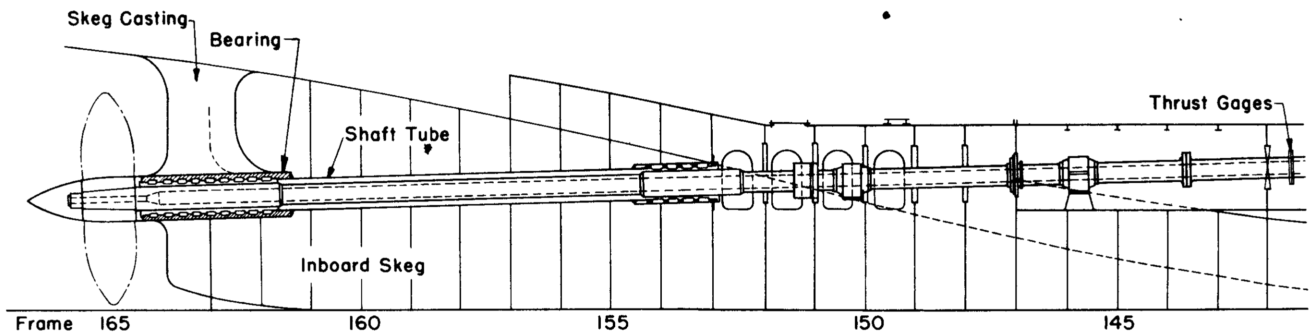
These measurements showed that the vibration was approaching resonance at the top speed attained, and that conditions would grow progressively worse as the power was increased. In an attempt to locate the cause of this resonance quickly, the natural frequency of the propeller-shaft system was estimated on the assumption that the long, hollow propeller shaft had sufficient longitudinal flexibility to permit the propeller to vibrate fore and aft, assuming the shaft to be fixed at the forward end. This computation showed that resonance in the vicinity of maximum speed most likely was the cause of the trouble.

It also became apparent at this time that the structure supporting the main thrust bearings and carrying the propeller thrust down into the hull of the ship was too limber. The use of underslung condensers and the building of the main thrust bearings into the forward ends of the large reduction-gear cases, as shown in Figure 4, combined to make it impossible to carry the thrust forces from the bearings into the ship structure in anything resembling a line which would be straight and which would at the same time lie at a reasonably small angle to the centerline of the shaft. The diagram of one of the main propelling units in Figure 4 indicates that this angle was in fact approaching 60 degrees.

In this respect, the NORTH CAROLINA installation was not much better than some previous thrust-bearing installations which were known to have given trouble. Among these may be mentioned the USS CALDWELL (DD69), a World War I destroyer, in which the thrust-bearing foundation deflected forward under the influence of the thrust and caused the bearing to bind on the shaft; the MS TRIUMPH, a converted and re-engined World War I merchant ship in which the main propeller shaft, connecting the electric driving motor of about 4000 HP to the single propeller, was observed to have a total longitudinal vibration or movement approaching 1/8 inch; and the HMS WARSPITE (4) (5), on

Figure 4 - General Arrangement of Propelling Machinery
for Port Inboard Shaft, USS NORTH CAROLINA

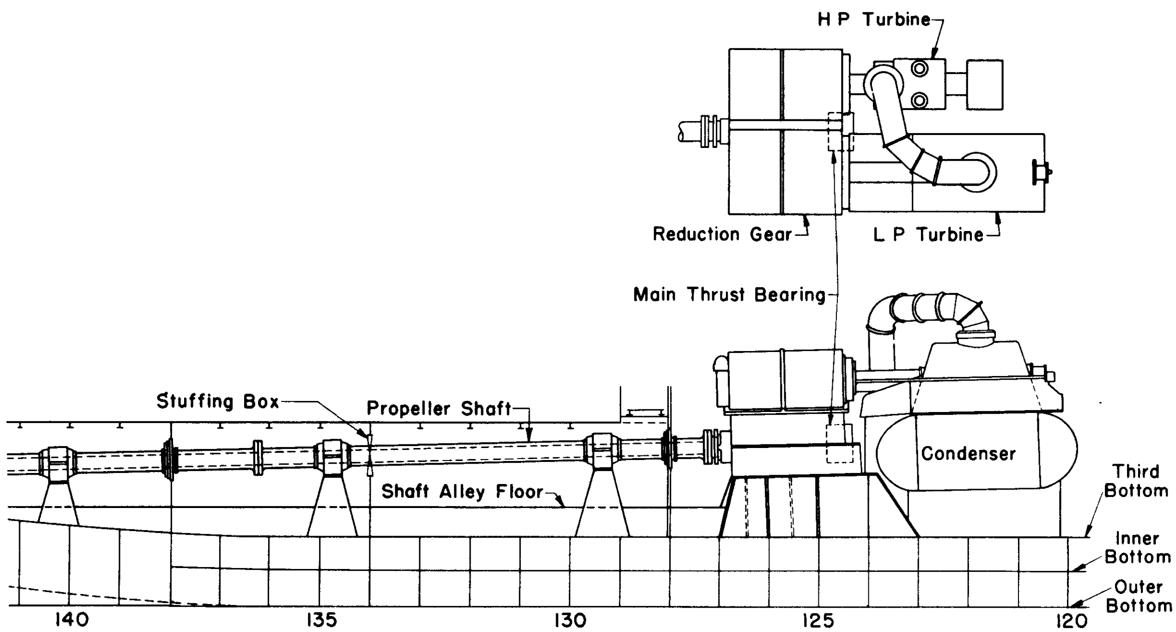
This shaft is approximately 160 feet long, from the propeller to the main thrust bearing.
It is the shortest shaft of the four in the ship.



which thrust-bearing foundations of insufficient rigidity are believed to have been indirectly responsible for much of the trouble encountered with the propelling machinery on that vessel.

A case of true fore-and-aft shaft resonance was noted on the SS SEA FOX, one of the C-3 vessels of the United States Maritime Commission, during sea trials held on 12 March 1940. In that case the fore-and-aft flexibility of the reduction-gear foundation, combined with the flexibility of the thrust plate of the Kingsbury thrust bearing, was approximately equal to the axial flexibility of the shaft. Both the foundation and the thrust plate were stiffened by simple means which shifted the resonant speed above the running range and which eliminated the trouble.

Although none of the printed literature discloses any reference to it, there is reason to believe that the French liner NORMANDIE, now the USS LAFAYETTE, experienced vibration of this type in spite of the fact that the principal difficulties during the early runs of this vessel were diagnosed as hull vibrations caused by vertical impulses from the propellers. This latter effect seems to have overshadowed any longitudinal shaft-and-engine vibration that might have been present, but the fact that the failures occurred at several points in the main thrust-bearing foundations on the ship's maiden voyage is considered significant. Of the many remedies applied to correct unsatisfactory conditions on the vessel, one was a considerable stiffening of the main thrust-bearing foundations in the fore-and-aft direction, and another was a change from 3-bladed to 4-bladed propellers.



Looking back from the NORTH CAROLINA builders' trials to the builders' trials of a number of older United States heavy cruisers and aircraft carriers, those who had witnessed the trials of both the new and the old vessels recalled that longitudinal vibration of the kind encountered on the NORTH CAROLINA had been observed on several previous occasions, but it had never been sufficiently severe to interfere with the operation of the vessel or to require any corrective measures, nor had the theoretical methods been developed for calculating or predicting the presence of the trouble in the new designs.

As a matter of historical interest, it may be noted that in John Bourne's famous treatise on the screw propeller, published in London in 1852 (6), he gave reproductions of the records from a recording thrustmeter fitted on the screw steamer RATTLER in 1845. It is significant that these early experiments showed clearly the variation in thrust as the blades of the 2-bladed propeller passed behind the stern post at each half-revolution.

GENERAL CONSIDERATIONS

UNUSUAL DESIGN FEATURES OF SHIP AND MACHINERY

These two vessels, the NORTH CAROLINA and the WASHINGTON, embodied a number of unusual features in their design and construction and there was a natural concern in some quarters, as the ships were being built, as to whether these features would prove themselves useful and advantageous in service.

The four machinery spaces, each with its complete boiler and engine unit and each extending entirely across the ship from holding bulkhead to holding bulkhead, were arranged in tandem. Farthest forward was the machinery for the starboard outboard shaft, then that for the port outboard shaft. Behind these was the machinery for the starboard inboard shaft and farthest aft that for the port inboard shaft. This arrangement necessitated the use of rather long lengths of line shafting for some of the engines. This shafting was hollow, in accordance with the practice for many years in the United States Navy, but it was in fact relatively only a little longer than the shafting employed in previous naval vessels with satisfactory results.

The twin-skeg construction for carrying the inboard shafts, as described in the Introduction and as referred to in subsequent sections, was indeed a novel feature for so large a ship. However, its performance had been quite thoroughly investigated by a long series of tests on self-propelled models, and the benefits to be expected from it, as described previously on page 2, were considered to outweigh any possible disadvantages.

Although the use of twin rudders on these two battleships was by no means novel, it was somewhat unusual to make them so large and to place them directly behind the two inboard propellers, instead of placing them between the propeller races, as had been done on previous capital ships of British and German design. Because the rudders were so large, it was expected that the periodic changes in pressure in the water around them, caused by rotation of the inboard propellers, might exert large vibratory forces on the stern structure of the vessel.

Turbines, Condensers, and Gears

Each engine consisted of one high-pressure and one low-pressure turbine in separate casings. Each turbine drove one pinion through a flexible coupling consisting of two flanges carrying internal gears, joined by a cylindrical member having external teeth at each end, matching the internal teeth on the flanges.

Each gear was of the double-reduction type, with the turbine pinions mounted at the upper forward end of the reduction-gear case. The turbine foundations were designed to afford fore-and-aft flexibility, especially the supports at the forward ends, but each of the turbine casings was connected to the upper end of the reduction-gear case by a substantial spacer piece bolted to both members. The condenser was slung underneath the low-pressure turbine, with flexible connecting pieces between the condenser and the main injection and discharge pipes respectively.

Main Thrust Bearings

These bearings were of the pivoted segmental type in extensive use for many years throughout the world on all classes of vessels; they were manufactured under the Kingsbury patents. Following a practice which had been adopted on destroyers in World War I, the thrust bearing for each shaft was mounted in a recess at the forward end of the reduction-gear case.

The original design embodying these features was found advantageous on destroyers because it saved the weight of a separate thrust-bearing foundation, it removed the thrust bearing from the rather crowded shaft alley, and it placed the thrust bearing at a point where lubrication and frequent inspection was an easy matter.

This general design, having functioned successfully on many destroyers, was later copied for cruisers and aircraft carriers built in the 1920's and the 1930's. It was found to function equally well on those vessels and so was copied in the designs for the machinery of the NORTH CAROLINA and the WASHINGTON.

Upon careful consideration, it becomes obvious that the forward end of a reduction-gear case, built to house a large gear which may be 10 or more feet in diameter, is not a particularly rigid structure, due to lack of through longitudinal members to which to attach a main thrust bearing, unless the bearing housing can be well braced forward to some rigid portion of the ship. As explained previously in the Introduction and as shown in Figure 4 on pages 6 and 7, this was not possible in the NORTH CAROLINA and WASHINGTON because of the presence of the underslung condensers.

Although not properly a part of this report, it is of interest to note that the main thrust bearings on the SOUTH DAKOTA class of battleships, BB57 to 60, similar in design to those on the NORTH CAROLINA class but stiffened and held somewhat more rigidly, have so far performed without particular difficulty. It may be said, however, that this is the result of low percentage variations in the propeller thrust rather than of any inherent superiority in the arrangement or the design of the thrust bearing and its foundation.

In the IOWA class of battleships, BB61 to 66, which have considerably greater power per shaft and longer shafts than either of the two preceding classes, the main thrust bearings were moved aft as far as possible in the shaft alleys, as a result of experience on the NORTH CAROLINA class.

Twin Skegs and Inboard Propellers

As explained previously, the use of twin skegs for the inboard shafts was a novel feature in the design of the NORTH CAROLINA and WASHINGTON, and there was some anxiety as to whether these skeg structures would vibrate transversely as cantilevers to an objectionable degree. Accordingly, estimates were made of natural frequencies of the skegs, and as soon as the hulls were completed, their natural frequencies both in and out of water were experimentally determined (1) (2) by vibration generators.

The twin-skeg design had originally been developed to provide a region of high wake in way of the inboard propellers, in an endeavor to obtain for these two propellers the high propulsive efficiencies characteristic of single-screw vessels of modern design. Because of the proximity of the two skegs to each other, this result was not achieved; the space between the skegs acted as a tunnel in which the water velocity was high and the wake fraction correspondingly low. In fact, outboard of the skegs the water velocity was also relatively high; see Figure 5a.

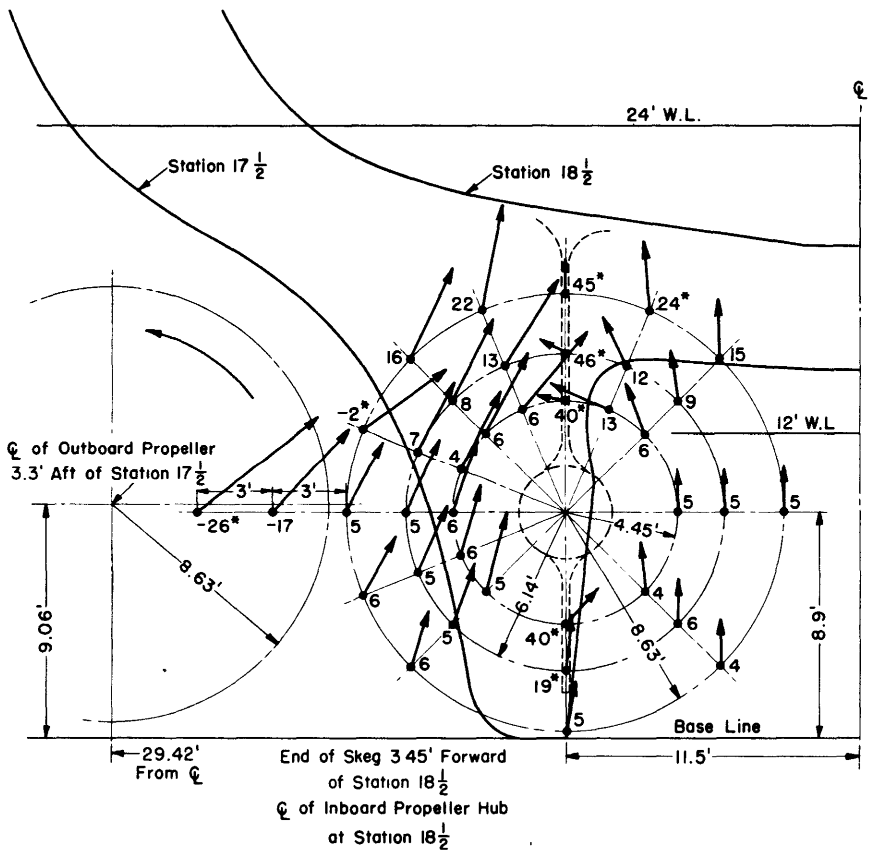


Figure 5a - Wake in Way of Inboard Propeller, with Outboard Propeller Working

This diagram, as well as Figure 5b, was made from the results of tests on a model composed of the bow of Model 3460 and the alternate stern of Model 3556. The struts, shafts, and fairwaters were in place, and a dummy hub was fitted on the inboard shaft. This test was made under conditions representing a displacement of 42,000 tons, a fore-and-aft trim of zero, and a speed of 27 knots. Figure 5c explains the vector system and notation.

Values marked with an asterisk are approximate.

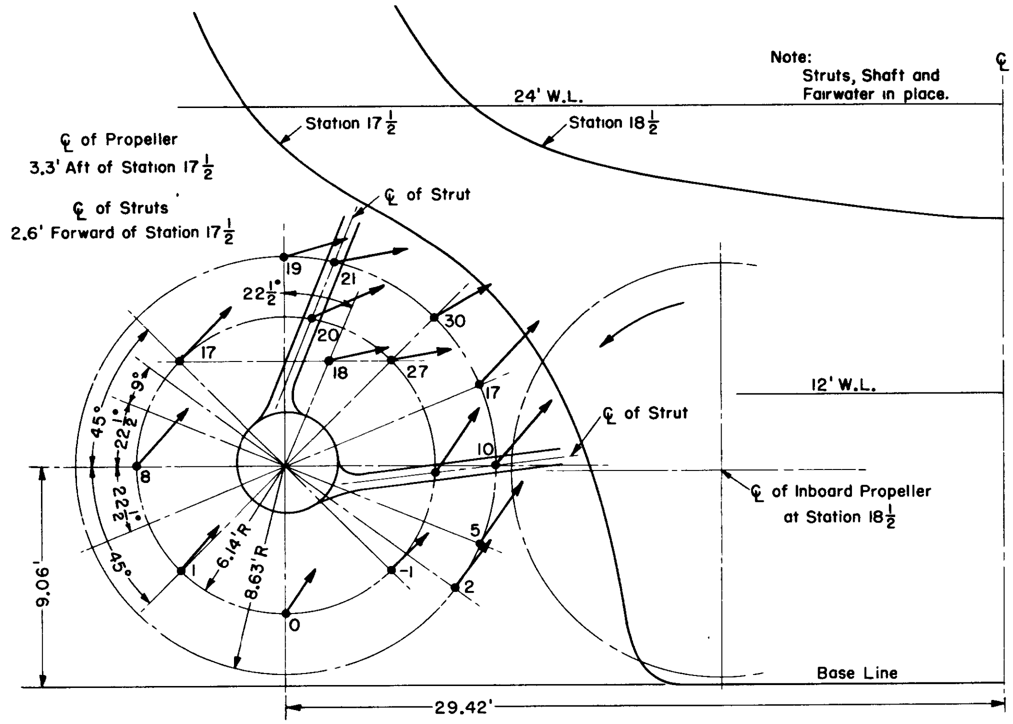


Figure 5b - Wake in Way of Outboard Propeller, with Inboard Propeller Working

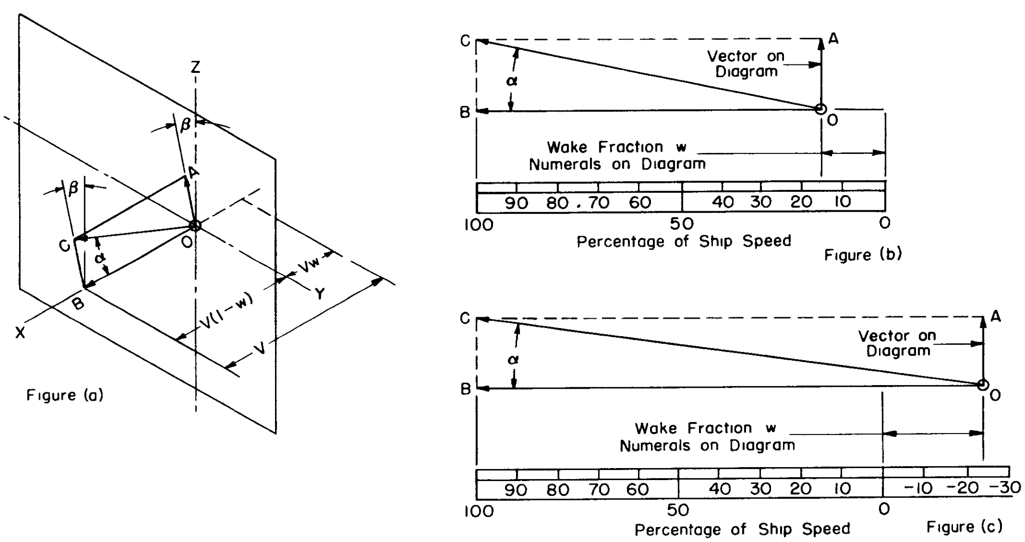


Figure 5c - Diagram Illustrating the Significance of the Wake Diagram Vectors of Figures 5a and 5b

The Y-Z plane in Figure (a) represents a transverse section through the ship at the point of measurement.

Vector $\overline{OA}$ lies in the Y-Z plane and is shown in the flow diagram. This vector indicates the transverse flow components in magnitude and direction.

Vector $\overline{OB}$ is parallel to the centerline of the ship. Its length is equal to the ship speed multiplied by $(1 - w)$, where w is the wake fraction.

Vector $\overline{OC}$ is the resultant velocity of flow relative to the ship. Figures (b) and (c) are drawn in the plane passing through $\overline{OA}$, $\overline{OB}$, and $\overline{OC}$.

The relationships between the vectors $\overline{OA}$, $\overline{OB}$, and $\overline{OC}$ are shown in these figures for positive and negative wake respectively. The scales permit expressing the flow magnitudes in terms of ship speed.

Directly behind the skegs, as was to be expected, the wake fraction was large, and the usual region of high wake was observed in the boundary layer near the ship. The wake in way of the outboard propellers, whose shafts were supported by a single pair of strut arms of conventional design, was quite normal, as shown in Figure 5b.

In the preparation of the contract plans for these two new battleships, the first which had been designed and built for the United States Navy in the period following World War I, considerable thought was given to the question of vibration (7). Estimates of the natural frequencies of the hull structure and of the torsional natural frequencies of the propeller-shaft-engine systems led to the belief that serious vibration would not be encountered in the modes considered. However, there was little to suggest that trouble would be encountered from longitudinal vibration of the propeller-shaft-engine system. In fact, except for a few instances such as those mentioned in the Introduction, longitudinal vibrations had rarely given rise to serious difficulty in propeller-driven ships. Literally thousands of single-screw merchant ships with 4-bladed propellers were known to be in satisfactory operation. There must have been thrust variations in all these vessels, but apparently they were so small as not to be troublesome unless magnified by resonance in the propulsion system.

In accordance with the general practice then current for high-powered naval vessels, the outboard propellers were made 3-bladed. Because of their lesser area, these appeared to have slightly higher efficiencies than propellers with more blades, and they had proved uniformly satisfactory provided serious cavitation could be avoided.

The design of the inboard propellers working behind the skegs was somewhat more of a problem. On the basis that a propeller with narrower blades would produce less pressure variation and less exciting force per blade than one with wide blades, a 4-bladed design was chosen for the original installation. It was recognized that with a skeg more or less symmetrical about a propeller diameter through the shaft center, two blades would be passing by the end of the skeg simultaneously, but it was felt that the upper blade only, passing close to the hull, would be the one to produce most of the undesirable forces. As explained previously in this section, it was considered that the successful use of 4-bladed propellers on hundreds of single-screw merchant vessels of moderate and high powers was ample justification for the use of 4-bladed propellers in what amounted to a double single-screw combination.

A half section of the afterbody, reproduced in Figure 6, shows the general arrangement and relative position of the hull and propellers at the time of the first sea trials on the NORTH CAROLINA.

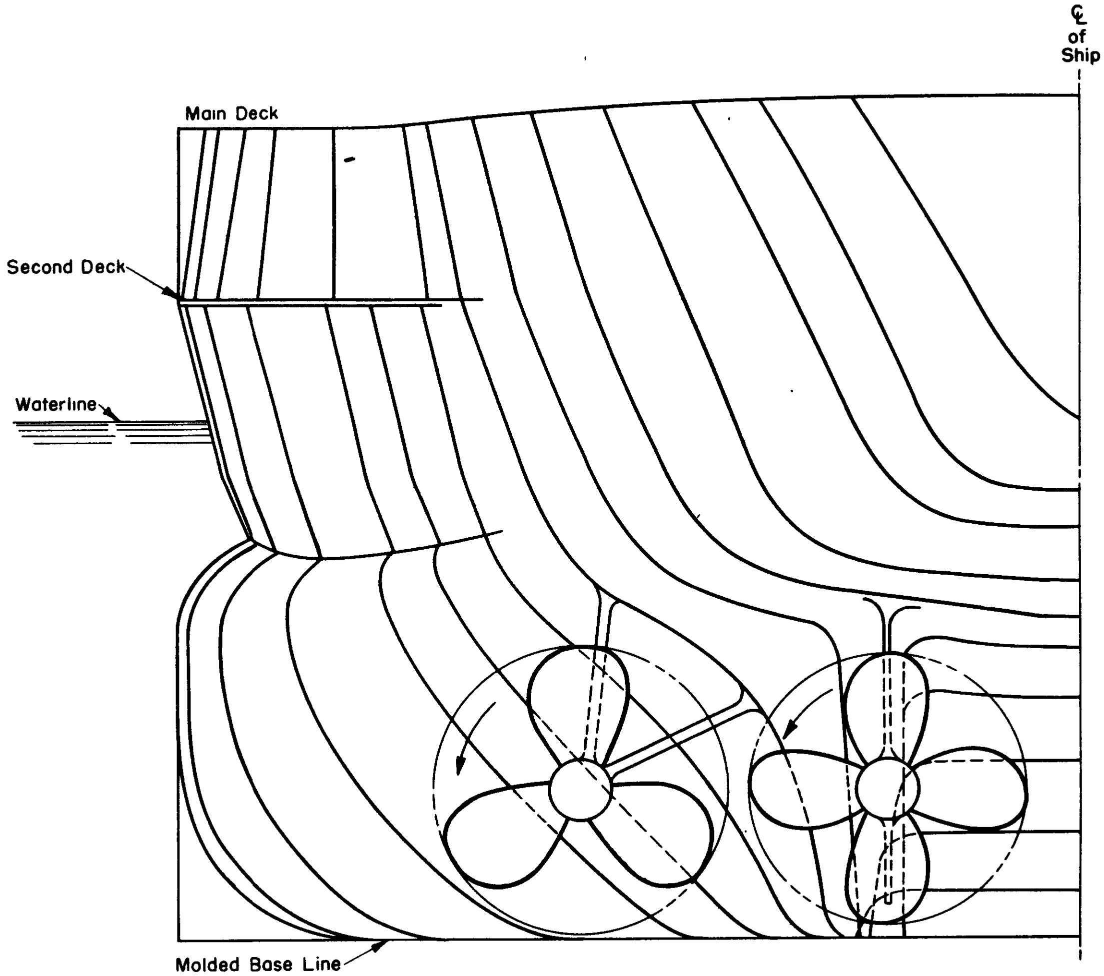


Figure 6 - Plan of Afterbody of NORTH CAROLINA Class, with Original Propellers and Struts Superposed

The propellers shown in this diagram are those fitted for the first builders' trial on each vessel; see Table 1 on page 31.

Twin Rudders Behind Inboard Propellers

It was considered at the time of the first builders' trials on the NORTH CAROLINA that the presence of the two large rudders behind the inboard propellers might have been a factor contributing to the vibration, as was suspected in the WARSPITE case. However, the fact that large changes in rudder angle produced only a moderate* increase in vibration and the fact that the rudders themselves were relatively steady indicated that the vibratory conditions in the propelling machinery would probably have existed even if the rudders had been removed.

* The vibration with full rudder was about twice the vibration with rudder amidships. Such an increase of vibration with rudder angle is often found on ships which are entirely satisfactory as regards vibration.

PROBABLE CAUSES OF VIBRATION

Taking all the foregoing matters into consideration, the design features thought likely to contribute to the longitudinal vibration encountered in the NORTH CAROLINA and WASHINGTON were:

a. long propeller and line shafts, combined with the use of hollow shafts throughout,

b. high propeller speeds,

c. large engine and propeller masses,

d. high thrust-bearing foundations, inadequately braced because of the large gear case and the underslung condensers,

e. position of the inboard propellers behind the skegs and proximity of all propellers to the hull,

f. shape of the skeg ends ahead of the inboard propellers.

Of these a, b, c, and d were thought to influence the frequency of the critical shaft speed and to account for its appearance in the running range while a, e, and f were considered responsible for the magnitude of the vibration at the critical speed.

VIBRATION CHARACTERISTICS OF PROPELLING MACHINERY

Regarding the large variations in thrust, set up initially as the individual blades of the propellers pass through regions of high wake behind the outboard struts and behind the inboard skegs, calculations made between the first and second builders' trials of the NORTH CAROLINA indicated that there was sufficient axial resilience in the hollow propeller shaft so that, if the shaft were assumed fixed to an infinite mass at the main thrust bearing, the shaft-and-propeller system would have vibrated in resonance longitudinally at about the worst frequency encountered. Owing to the magnification factor involved when an elastic system is vibrating in resonance, a small variation in thrust at the propeller would then have manifested itself as a large variation in force at the main thrust bearing.

A rough comparison of the longitudinal type of vibration with the more familiar torsional type, such as that encountered in recent years with diesel engines, is included here to give an idea of the relative frequency factors involved. Assuming a node at the reduction gear for both longitudinal and torsional vibration, the ratio between the two types of critical speeds for this simple one-mass-and-spring system, comprising the propeller and the shaft, is given by the following expression, based on common values of the elastic moduli for steel:

$$\frac{\text{critical longitudinal RPM}}{\text{critical torsional RPM}} = \frac{R}{D}\sqrt{\frac{20}{1 + \left(\frac{d}{D}\right)^2}} \qquad [1]$$

where D and d are the outer and the inner shaft diameters, respectively, and R is the radius of gyration of the propeller. The ratio R/D is approximately 4. The factor $1 + (d/D)^2$ is of the order of 1.2, so that the critical shaft speed for longitudinal vibration in an average design is some 15 times the torsional critical speed. In most cases of ordinary steam-turbine design the lowest torsional critical speed is less than one-fourth of the running speed, so that the longitudinal critical speed is frequently above the running range. On the NORTH CAROLINA and WASHINGTON, however, the combination of long shafts, high RPM, and heavy masses dropped the longitudinal critical speed into the running range.

Once the critical shaft speed for longitudinal vibration lies within the operating range, the severity of the vibration is determined by the severity of excitation, and this in turn depends upon the nonuniformities in the flow within the propeller disk.

FUNDAMENTAL ANALYSIS OF THE VIBRATORY SYSTEM

With the analytical technique and procedure now available to the engineer (8), it is possible to attack a problem of this kind by reducing it to fairly simple terms and then to estimate or determine by experiment the coefficients or factors needed for a solution.

For example, the propelling machinery shown in Figure 4 on pages 6 and 7 may be represented by four bodies having certain equivalent masses attached to each other by elastic members which have certain stiffness characteristics. The schematic representation of this system is indicated in Figure 7.

Here the propeller shaft is considered as a massless spring k_1, and its mass is lumped at the two ends; that is, half is added to the propeller and half to the mass of the lower part of the reduction-gear unit. As the propeller moves back and forth longitudinally, a certain amount of water is entrained with it, and this mass must be added to that of the propeller. The spring k_a represents the stiffness of the gear-case foundation, k_2 the stiffness of the part of the gear case above the shaft, k_3 the stiffness of the connection between the gears and the turbines, and k_c the stiffness of the turbine foundation. The mass m_1 is then equal to the mass of the propeller

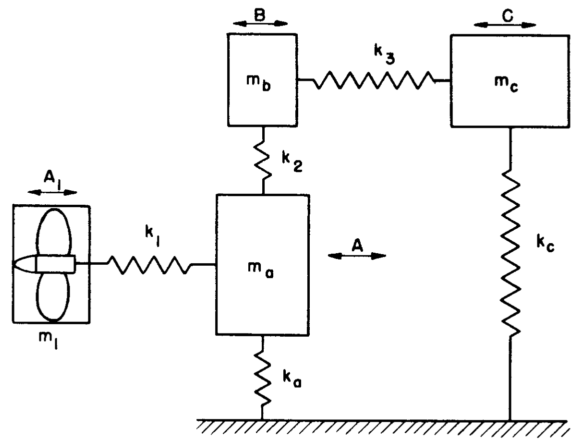


Figure 7 - Schematic Representation of a 4-Body Elastic System

This diagram represents the four principal parts of one unit of propelling machinery, corresponding to Figure 4 on pages 6 and 7. The various masses and springs are described in the text of the report. The rectangle drawn around the propeller represents the entrained water vibrating with it. Although in this schematic representation some springs are vertical and some horizontal, their elastic constants in the ***horizontal*** direction are represented by the k's.

plus half the mass of the shaft plus the mass of the entrained water; m_a represents the mass of that part of the gears and gear case which vibrates as a unit with the end of the shaft plus half the mass of the shaft; m_b the mass of the upper part of the gear unit attached to the turbines; and m_c the mass of the high-pressure turbine, low-pressure turbine, and condenser combined. The symbol A_1 represents the amplitude of vibration of the propeller, A the amplitude of the lower part of the reduction-gear unit, B the amplitude of the upper part of the gear case, and C the amplitude of the turbine and condenser.

If this system is considered to be vibrating freely, the acceleration of each body times the mass of the body must equal the resultant of the forces due to the deflection of the attached "springs." Frictional damping forces are neglected here as having a negligible effect on the natural frequencies.

A system such as that shown in Figure 7 is capable of simple harmonic motion at its natural frequencies. The relative accelerations and displacements during such vibration at the lowest natural frequency can be readily calculated. The conditions for satisfying Newton's laws can be expressed in the following equations:

$$-m_1\omega^2 A_1 = -k_1(A_1 - A) \quad [2]$$

$$-m_a\omega^2 A = k_1(A_1 - A) - k_2(A - B) - k_a A \quad [3]$$

$$-m_b\omega^2 B = k_2(A - B) - k_3(B - C) \quad [4]$$

$$-m_c\omega^2 C = k_3(B - C) - k_c C \quad [5]$$

in which ω is 2π times the natural frequency.

These four simultaneous equations can be solved to give the lowest natural frequency and the ratios of the vibration amplitudes in terms of the various masses and stiffnesses. If desired, the natural frequencies of the higher modes can be derived from the same basic equations. There is a total of four natural frequencies for the 4-body system.

The various masses and spring constants can be estimated, calculated, or determined by experiment, depending upon the length of time and the facilities available during the analysis. For example, it was possible, as explained subsequently in this report, to isolate the units in one of the machinery spaces of the NORTH CAROLINA by uncoupling the line shaft directly abaft the main reduction gear and to set the engine and gear units in vibration by a special vibration generator. The system then becomes one of three bodies, as shown in Figure 8.

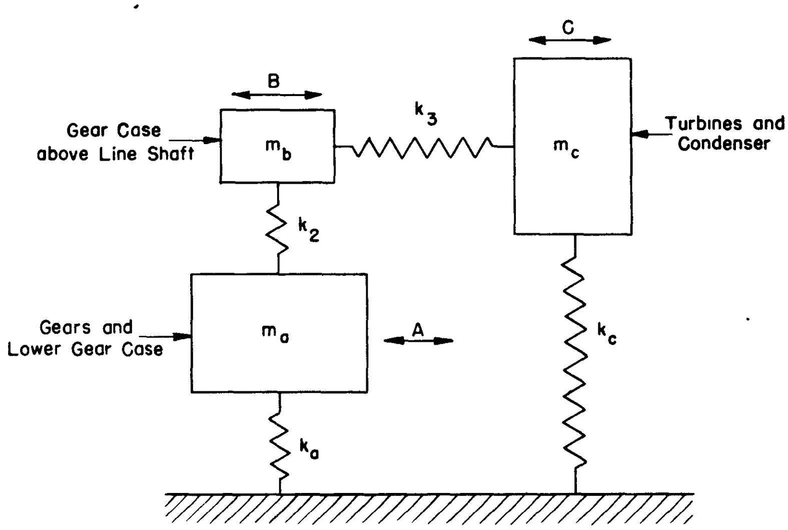


Figure 8 - Schematic Representation of a 3-Body Elastic System

This diagram represents the machinery and foundations after the connection between the reduction gear and the propeller shaft has been broken.

In this particular treatment of the problem the fundamental natural frequency and the relative amplitudes were determined from experimental data obtained in the test with the vibration generator. The mass and spring constants were then evaluated by a cut-and-try process so as to fit equations similar to those given previously for the 4-body system.

ANALYSIS OF THRUST VARIATION

A theoretical approach to the analysis of thrust variation, although made after many of the ship tests had been completed, may conveniently be described here for a better understanding of some of the procedure described in the sections following.

It was assumed that the usual formulas for obtaining propeller thrust from RPM and speed of advance hold for the instantaneous thrust in non-uniform flow as well as for steady thrust in uniform flow. Applying these laws to a propeller rotating at constant speed in a non-uniform wake, it was found that the ratio of the alternating to the steady component of thrust should be the same for model and full-scale ship, and the same for all speeds, provided the following conditions are satisfied:

(a) the propeller slip is the same for all speeds within the range investigated,

(b) the flow around the propeller is geometrically similar at all speeds, and

(c) the rotational speed does not change during the course of revolution.

Condition (a) is only approximately met, but as the ratio in question does not change much with small changes in slip ratio the approximation is quite acceptable. Condition (b) is a basic condition for all model testing. Condition (c) is fulfilled quite exactly for all speeds well in excess of the critical speed for torsion, which in the case of these two vessels occurred in the neighborhood of or below 50 RPM.

The equality between the ratio of alternating thrust to steady thrust for model and ship, theoretically derived in this manner, made it possible to predict the magnitude of the excitation forces due to various ship propellers by using the results of model experiments, and to check the forces thus found by analysis of the measurements made on the actual vessel. On the NORTH CAROLINA and WASHINGTON the full-scale testing was completed before this special technique for model testing had been completely worked out. For this reason, it was not possible to realize the full advantage of the model tests on these ships, but on subsequent classes of battleships predictions based on model tests have been instrumental in eliminating propeller-excited vibrations.

PROPOSED SOLUTIONS OF THE PROBLEM

At the end of the second builders' trial on the NORTH CAROLINA, on 28 May 1941, a conference was held on the ship to propose possible remedies and to prepare recommendations for future action.

Enough data had been taken during the trials and quickly analyzed to establish the facts that the resonant frequencies of all the propelling-machinery systems, with the propellers then in use, were at or near full power; that the single amplitudes of vibration at the after ends of the reduction-gear cases were of the order of 0.030 to 0.038 inch; and that due to the variation at the propellers, combined with the effect of resonance, the thrust variations at the thrust block, especially on the inboard shafts, could be expected to reach plus and minus 60 per cent of the steady thrust at full power.

After listing the numerous remedies proposed at the conference and subsequently, this report will describe the special tests made in connection with this project, give the analysis of the results of these tests, and then discuss in detail the relative value, estimated or determined, of the numerous remedial measures proposed.

These were substantially as follows:

1. Stiffening the main thrust-bearing foundations. Some alterations of this kind were made on the WASHINGTON (BB56) before that vessel was sent on her second trials; others were made on that vessel after those trials.

2. Reducing the fore-and-aft clearance in the main thrust bearings. However, this was already about as small as the design of the pivoted segmental bearing would permit.

3. Moving the thrust bearings from their positions in the forward ends of the main reduction-gear cases to positions in the shaft alleys as close as possible to the propellers.

4. Designing, constructing, and installing shaft-restraining blocks which would act as stationary dashpots around the rotating shafts, as indicated in Figure 9, and which would, by main force, restrain the shafts from vibrating or moving more than a negligible amount in a fore-and-aft direction.

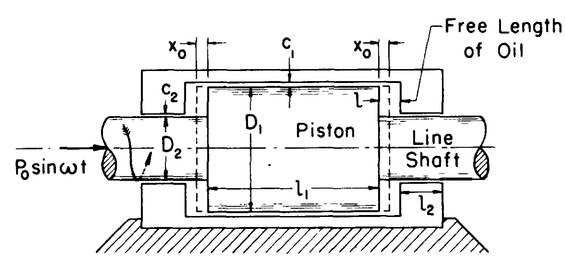


Figure 9 - Diagram of Shaft-Restraining Block

The piston, secured to the shaft, rotates with it in a fixed chamber with as small a diametral clearance as practicable. The interior of the chamber around the shaft and piston is filled with oil. Longitudinal movement of the shaft is possible only by squeezing the oil from one end of the chamber to the other.

5. Fitting propellers with numbers of blades such that the correspondence between the blade-frequency exciting force and the natural frequency of the propulsion system would occur either beyond the running range or at a low RPM in that range, where the powers and forces would be relatively small and the resultant vibration would be acceptable.

6. Installing pendulum dampers on the rotating shafts to counterbalance the vibratory forces.

7. Introducing one or more lengths of line shafting of greater axial resilience than the existing shafting, or one or more couplings with axial flexibility (9), to lower the resonant frequency of the shaft-propeller system in the longitudinal direction.

8. Modifying the after ends of the inboard skegs, especially the portion below the shafts, to provide greater clearance between the fixed ship structure and the moving propellers, and to smooth out the inequalities in the wake behind the skegs.

9. Increasing the propeller clearances by moving the propellers farther from the hull or raking the blades.

10. Skewing the propeller blades so that the inequalities of flow in the wake behind the skegs would, to a certain extent, be averaged out.

SCHEDULE OF SPECIAL TESTS

As already noted, the first trials of the NORTH CAROLINA, which was built by the New York Navy Yard, were held on 19 and 20 May 1941. They disclosed longitudinal vibrations of such magnitude that it was considered unsafe to run the ship at speeds above 25 1/2 knots. As the seriousness of the longitudinal vibration had not been anticipated, it was possible to make measurements in only a few locations, and further information was urgently needed. Accordingly, preparations were made to measure longitudinal vibrations at a large number of selected stations; at the same time, structures which had exhibited the severest vibrations during the first trials, as for example the main steam lines to the turbines, were strengthened locally. The second trials were held on 27 May 1941 and a speed of more than 26 1/2 knots was attained. The propellers used during both these trials were the propellers originally designed for this class of ship; the inboard propellers were 4-bladed and the outboard propellers were 3-bladed.* All propellers turned outward when going ahead.

* The characteristics of these propellers are given in Table 1 on page 31.

The second builders' trials of the USS WASHINGTON, built by the Philadelphia Navy Yard, were held on 21 June 1941. For these trials the original 3-bladed outboard propellers were cut down from a diameter of 17 feet 3 inches to a diameter of 16 feet 4 5/8 inches and they were installed on the inboard shafts, with the expectation that resonance in the longitudinal mode would then be above the running range of the machinery. The

Figure 10 - BB55 and 56 Class, 3-Bladed Propeller Cut Down in Radius and Shifted from Outboard to Inboard Shaft

The inboard skeg is clearly visible behind the propeller.

original 4-bladed inboard propellers were installed on the outboard shafts without change; here it was hoped that resonance would occur at such a low power as not to be objectionable. Photographs of this installation are shown in Figures 10 and 11.

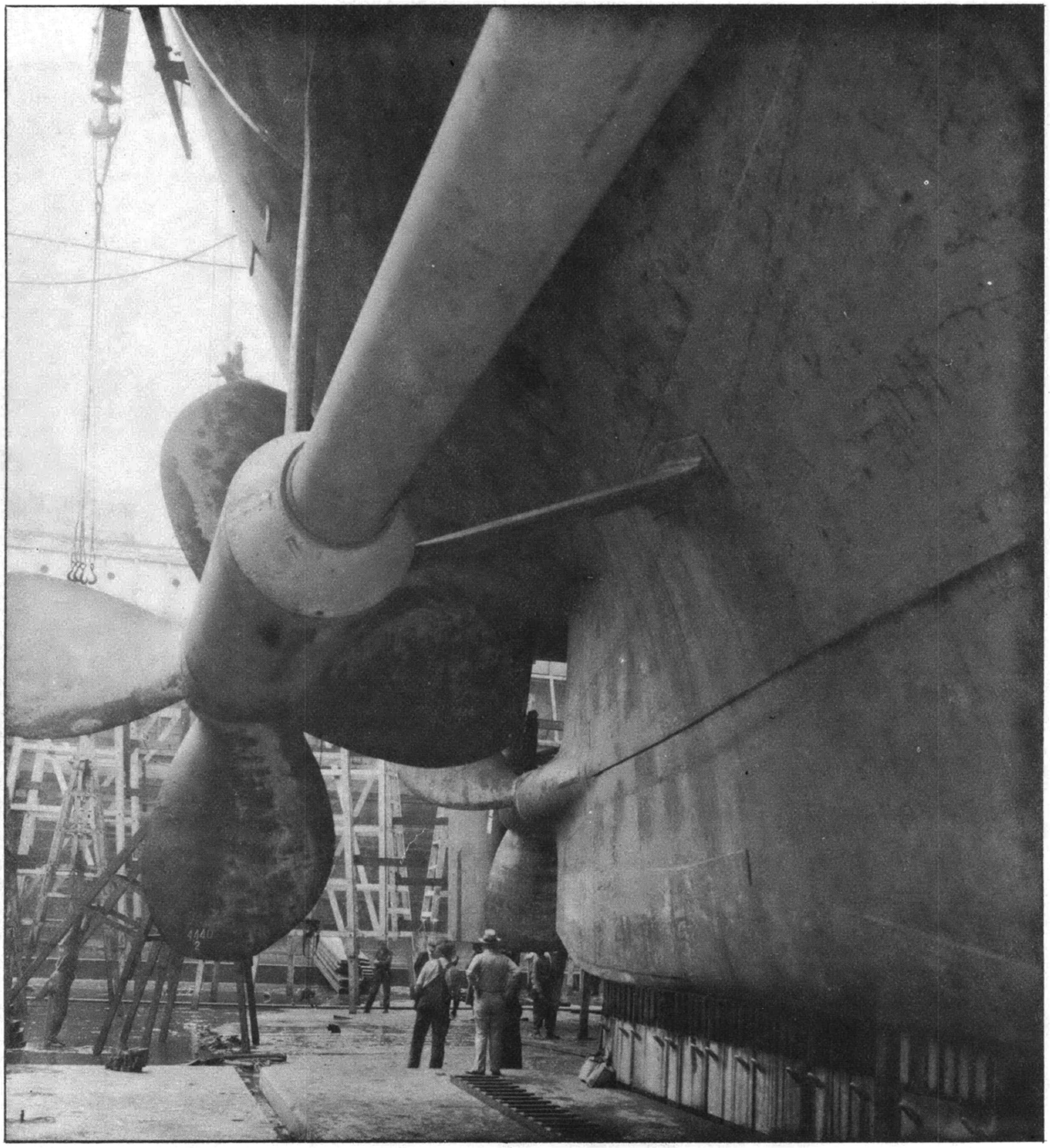


Figure 11 - BB55 and 56 Class, 4-Bladed Propeller Shifted from Inboard to Outboard Shaft

This photograph shows the relation between the outboard shaft and the skeg. The rudder is visible in the background, just behind the inboard propeller.

With this propeller combination a speed of over 27 knots was attained on the WASHINGTON. The vibration at that speed was not so severe as to endanger any part of the power plant but it was still sufficient to interfere seriously with operation of some of the gun directors.

As the distribution of power among the four shafts was uneven, the inboard shafts taking more power than their share, the 3-bladed inboard propellers were cut down still further by taking 3 inches more off the tips, and additional trials were held on 12 July 1941. During a later trial of the WASHINGTON, held on 2 December 1941, there were 5-bladed propellers on the inboard shafts and 4-bladed propellers on the outboard shafts. Both sets of propellers were of new design, as listed in Table 1 on page 31. Still further improvement was found on trials held on 23 April 1944, with later designs of 5-bladed propellers inboard and 4-bladed propellers outboard.

Meanwhile a number of special tests were being carried out. During June and July of 1941 the restraining block was studied intensively. This work included building and testing two scale models of restraining blocks at the Taylor Model Basin (10). One engine of the NORTH CAROLINA was tested with a vibration generator on 15 July 1941, in order to evaluate some of the vibration constants of the system.

A 30-foot self-propelled model representing both ships was built, and special equipment was designed and constructed to measure the longitudinal vibration-exciting forces in the four shafts with various model propellers. The first test of this kind was held on 27 and 28 October 1941; this was useful chiefly in developing the method of measurement. Further model tests were made on 12 November 1941 and on 26 August 1942. Model tests to determine the effect of a possible skeg modification on this class were conducted on 20 October 1942.

TEST APPARATUS

VIBRATION-INDICATING AND -RECORDING GEAR

Sperry-MIT Portable Type

This equipment, a complete description of which is given in Reference (11), consists of a relatively small pickup unit attached to the structure, as shown in Figure 12, and a special recording oscillograph. The response is made proportional to the amplitude by an integrating circuit incorporated in an amplifier between the pickup and the oscillograph. The space inside the pickup is filled with oil for damping, and this damping is adjusted to about 0.6 critical so that, theoretically, amplitudes are recorded accurately at frequencies as low as 1 1/2 times the natural frequency of the element.

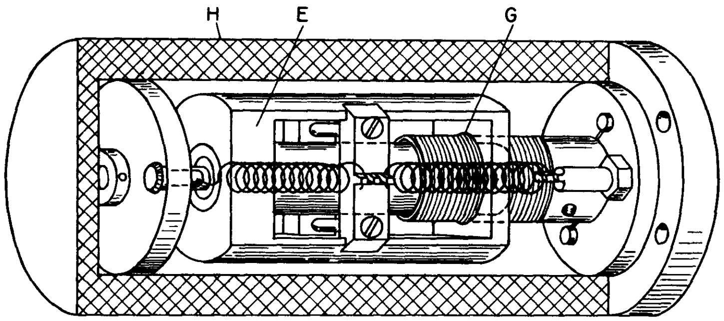


Figure 12 - Essential Parts of Sperry-MIT Vibration Pickup

The pickup unit comprises a case H containing a spring-supported seismic element E which is itself a permanent magnet whose flux cuts the turns of a coil G fixed in the case. A voltage is generated in the coil proportional to the relative velocity between the coil and the seismic element.

The integrating circuit consists simply of a high resistance in series with a condenser of low impedance. The current through the resistance is proportional to the generated voltage, and the voltage across the condenser is determined by its charge, which is the integral of the current with respect to time. The condenser voltage is fed into the amplifier whose output goes to the oscillograph which is of the magnetic type with photographic recording. A calibrating gear, consisting of an adjustable eccentric driven by a motor, is supplied with the equipment.

Geiger Torsiographs

The Geiger torsiograph consists of a heavy flywheel connected by a spiral spring to a shaft on which a light pulley is mounted. The pulley is driven by a belt from the rotating shaft. If there is no torsional vibration, the pulley and the flywheel will rotate together at uniform speed with no relative movement between them. If, however, torsional oscillations of sufficiently high frequency are present, the flywheel will not follow them, owing to its inertia. The relative motion between the flywheel and the pulley actuates a pen which makes a record on a moving strip of paper; a time record is also made on this paper. The natural torsional frequency of the flywheel is about 90 cycles per minute.

This instrument can be modified for use as a linear vibrograph by substititing for the flywheel a single weight which acts as a pendulum of low natural frequency. If the instrument is attached to a structure vibrating at a frequency several times the natural frequency of the pendulum, the weight remains practically fixed in space and the relative movement actuates the recording mechanism.

TMB Pallographs

The pallographs used on these tests, of which the vertical type is shown in Figure 13, are seismic instruments similar in principle to the Geiger instrument when used as a linear vibrograph. The pallographs were designed for recording very low natural frequencies and large amplitudes, and they have small magnification. A waxed paper and a stylus are used for recording, and a magnetically operated stylus is used for timing.

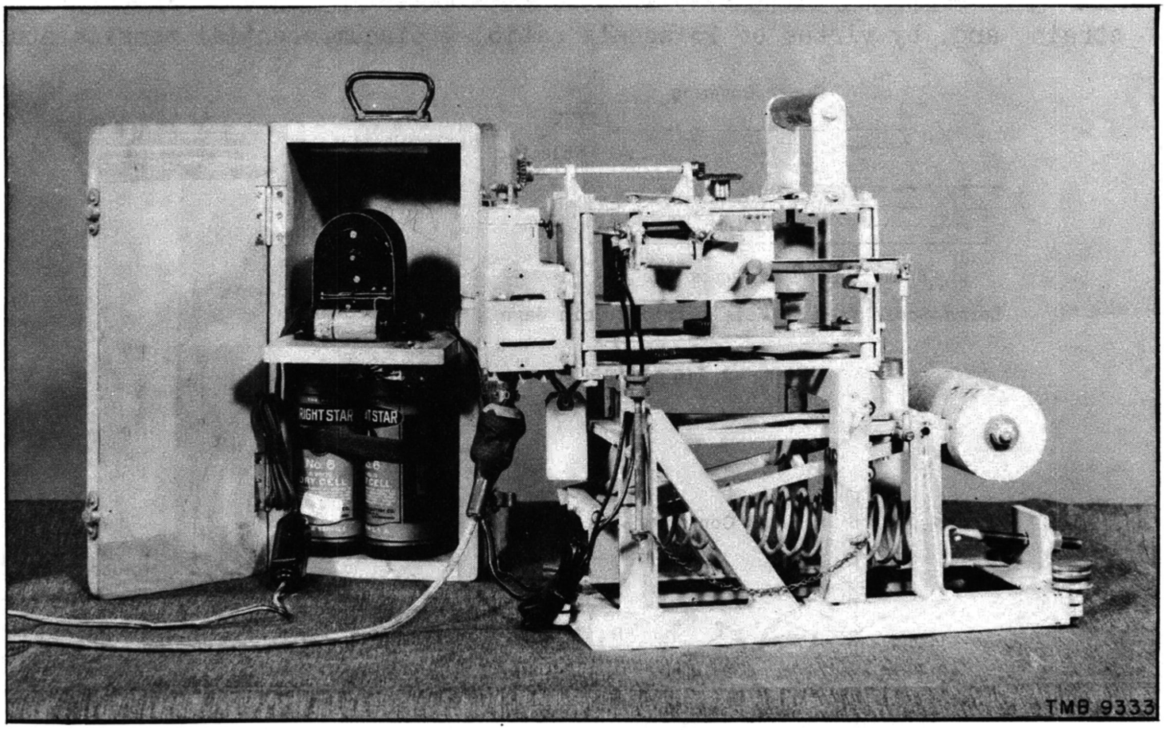


Figure 13 - TMB Pallograph, Vertical Type

A cylindrical inertia weight or seismic element at the right is carried on a horizontal arm pivoted at the left end of the instrument. A spring attached to a bell crank holds the weight in equilibrium. The recording stylus and paper are at the top of the instrument. This instrument can be used for recording horizontal vibration by mounting it on end in a frame, removing the balancing spring and allowing the weight to hang as a pendulum.

Metalectric Strain Gages for Full-Scale Thrust Variations

The type of strain gage used for measuring the alternating component of thrust, the so-called metalectric gage, consists of a length of resistance wire of very small diameter, about 0.001 inch, wound back and forth in multiple-W fashion between two lucite terminal blocks carrying the external connecting wires; see Figure 14. The strain-sensitive wire is covered with thin paper for mechanical protection and electrical insulation, and the entire assembly is in convenient form for cementing to the member under test. The gage extends or compresses to conform to any strains which appear in the surface to which the wire is cemented. A compression of the wire produces a reduction in its electrical resistance, and a tension produces an increase in its resistance. The relationship between the strain and the resistance is established by the manufacturer by sampling experiments made on each batch of gages so that stresses in members to which the gages are attached can be evaluated directly from the electrical measurements.

In a propeller shaft, the thrust produces an axial compressive strain, and, by virtue of Poisson's ratio, a circumferential tensile strain.

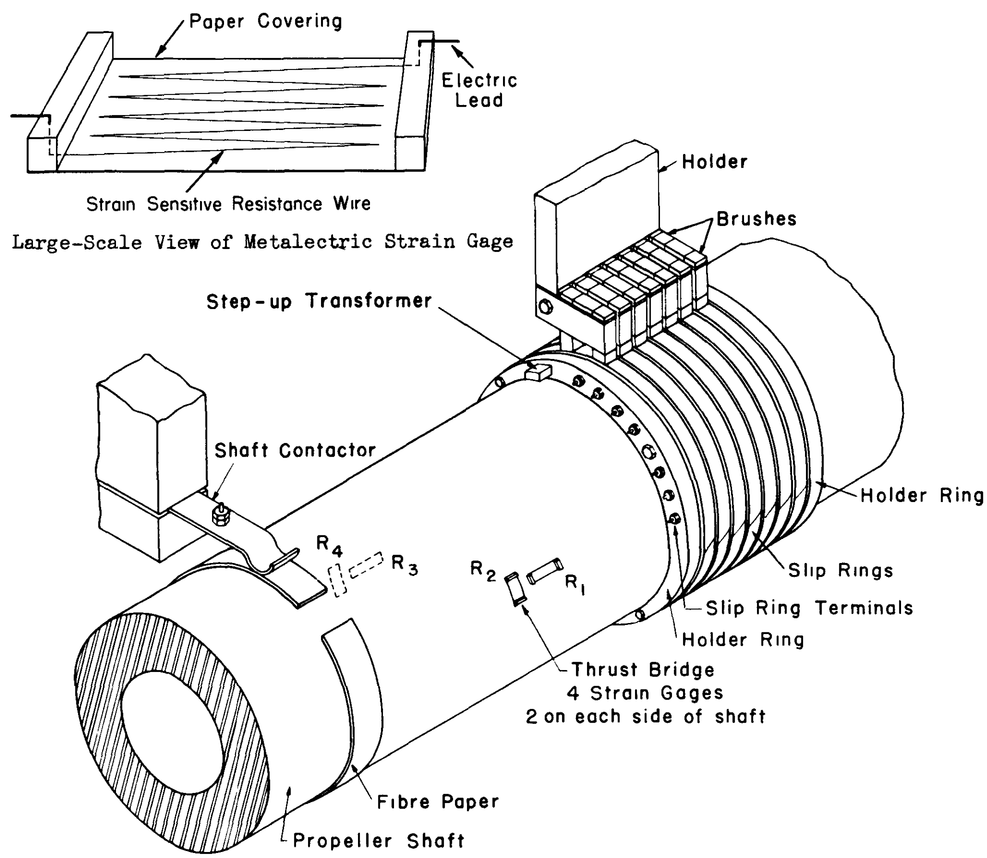


Figure 14 - Diagram of Thrust Gages and Accessories, Mounted on a Propeller Shaft

Four metalectric or wire-resistance strain gages, shown diagrammatically in the large-scale detail, are cemented to the shaft, two in the positions R_1 and R_3 and two in the positions R_2 and R_4. Signals from these gages are taken off through four slip rings, as explained in Reference (3), and are impressed on an oscillograph, which produces a record of the type shown in Figure 16.

The slip rings are insulated from the shaft and from each other. Three of the four extra rings are for torque-measuring gages, not shown here.

The contact maker at the left is for indicating on the record the position of each propeller blade with reference to the vertical.

See Figures 15 and 17 for details of the slip rings.

The torsion in the shaft does not produce any strain in these directions. To measure thrust, four gages are cemented to the shaft, two along the length of the shaft and two around its circumference, as shown in Figure 14. The four gages are connected into a Wheatstone bridge circuit, the four terminals of which are brought off the shaft through slip rings and brushes, as indicated in Figures 14 and 15. The use of four gages in this manner minimizes the effect of varying temperature, bending stress in the shaft, and the varying contact resistances of the brushes.

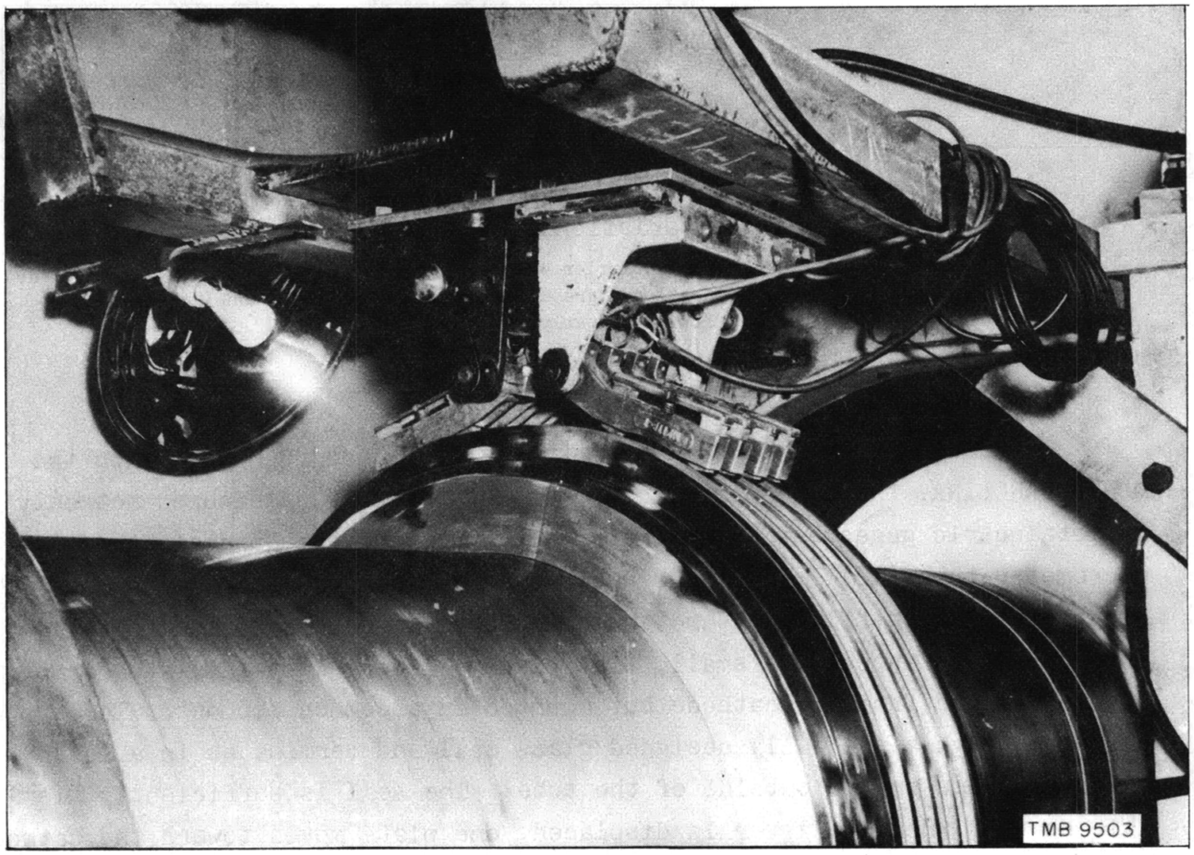


Figure 15 - Slip Rings and Brushes used in Measuring Thrust Variations, as Installed on USS NORTH CAROLINA

These rings were taken from a Ford torsionmeter. Two sets of brushes were used to cut down resistance variations at this point. The photograph was taken while the shaft was in motion.

For the measurements on the NORTH CAROLINA and the WASHINGTON alternating current was used to supply the strain-gage bridge, in order to simplify the amplifier design, to eliminate the effect of variable thermal voltages at the brushes, to facilitate calibration, and to make it possible to measure steady stresses. The output of the strain-gage bridge after amplification was recorded on an oscillograph of the galvanometer type. The details of the entire process are explained in full in Reference (3). A sample oscillogram is shown in Figure 16. More recently these measurements have been made successfully using d-c supply.

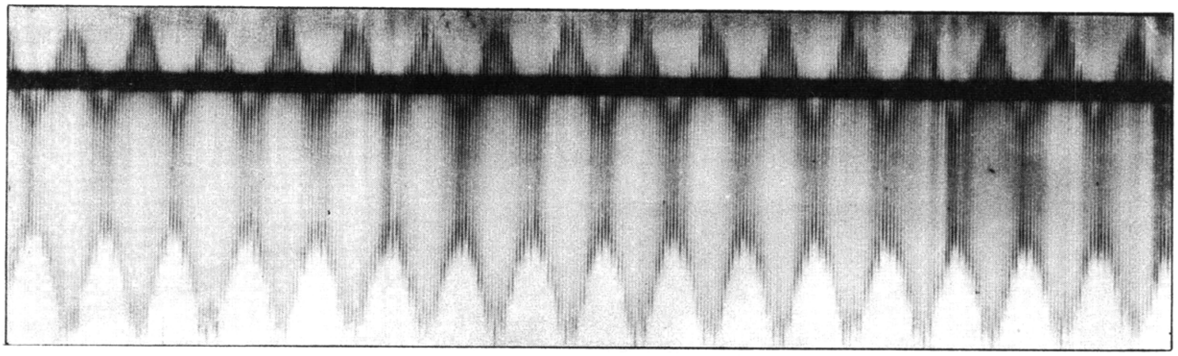


Figure 16 - Typical Oscillogram from Thrust Strain Gages, Showing Variations in Shaft Thrust

The rapid fluctuations are caused by the carrier wave. The variations in thrust for each blade are quite large. If the various blades are different, and are not producing the same action, this feature shows up clearly in an oscillogram of this kind.

NRL TORSIONMETER

During the first builders' trials of 19 and 20 May 1941 on the NORTH CAROLINA, the alternating component of torque was measured not only by metalectric gages but by an instrument employing the Gunn electrical-micrometer tube (12). This tube employs a cathode consisting of a hot, electron-emitting filament, and two anodes, all inside an evacuated glass envelope. The anodes are small, flat plates electrically insulated from each other and from the cathode but mounted on a common support. The support passes through a specially designed glass seal and terminates in a lever which projects on the outside of the tube. The seal is sufficiently flexible so that when the lever is displaced, one plate moves toward the cathode and the other plate away from the cathode a corresponding distance. The electrical resistance from a plate to a cathode is a function of the distance between the plate and the cathode. By the use of a Wheatstone bridge circuit, the output current can be used as a measure of the displacement of the lever. The device is sensitive to displacements of the order of 0.0001 inch and will accurately follow alternating strains having a frequency of several thousand cycles per minute.

For the measurement of torque, this tube is mounted on a yoke which is clamped around the shaft. A second yoke 8 inches from the first one carries a cantilever arm which engages the actuating lever on the tube.

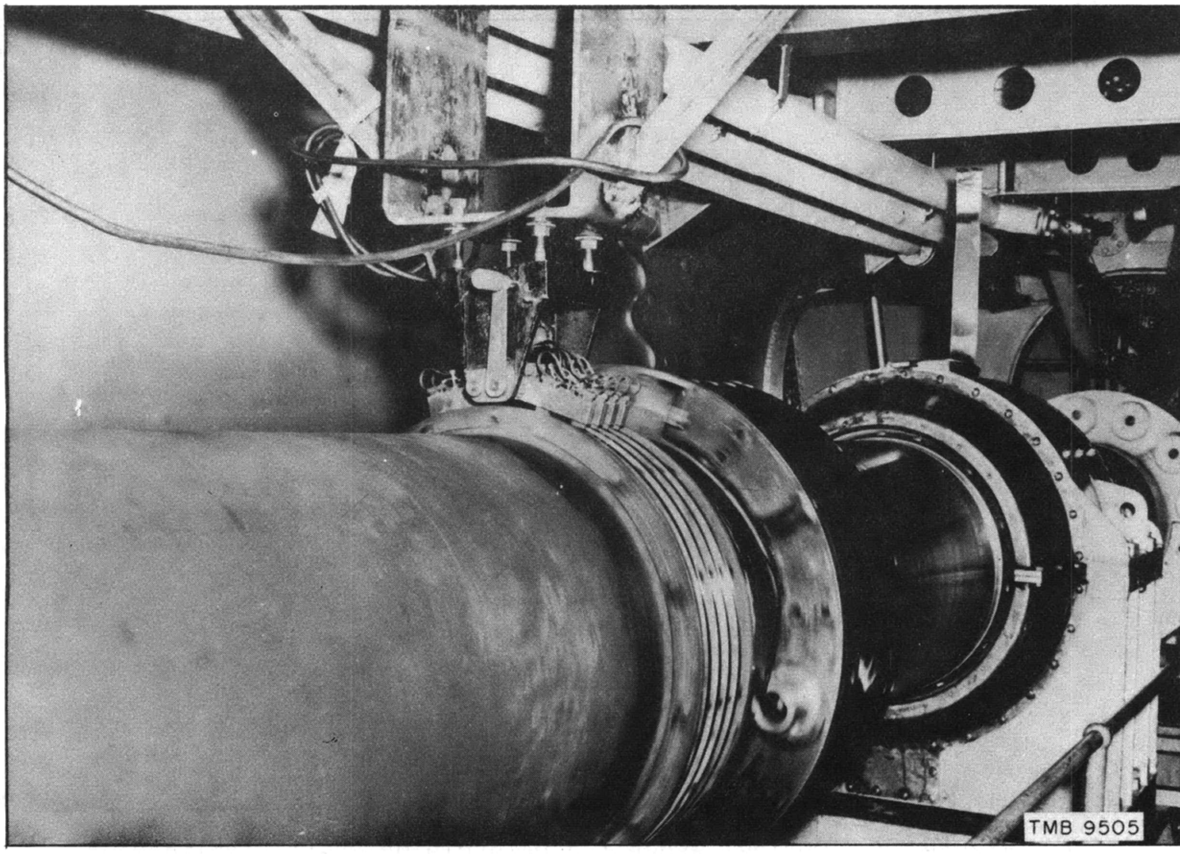


Figure 17 - NRL Torsionmeter, as Installed on USS NORTH CAROLINA

The four slip rings are in the foreground, and behind them is the case over the Gunn tube and the mechanical gear. In the background is a shaft steady bearing and a shaft coupling.

Power-supply and output circuits are carried through slip rings and brushes, as illustrated in Figure 17. The tube thus measures the instantaneous torsional deflection in an 8-inch length of shaft. A detailed description of the whole device will be found in References (12) and (13).

APPARATUS FOR MEASURING SHAFT DISPLACEMENT

For the trials of 21 June 1941 on the WASHINGTON an optical wedge was mounted near one of the shaft couplings. A motion-picture camera was provided to record the fore-and-aft displacement of the face of the coupling relative to the optical wedge as the shaft rotated. Figure 18 shows the arrangement of the apparatus. Two such devices were employed on the starboard inboard shaft, one in the engine room just abaft the thrust bearing and the other in the shaft alley about 100 feet farther aft, with the object of measuring comparative amplitudes.

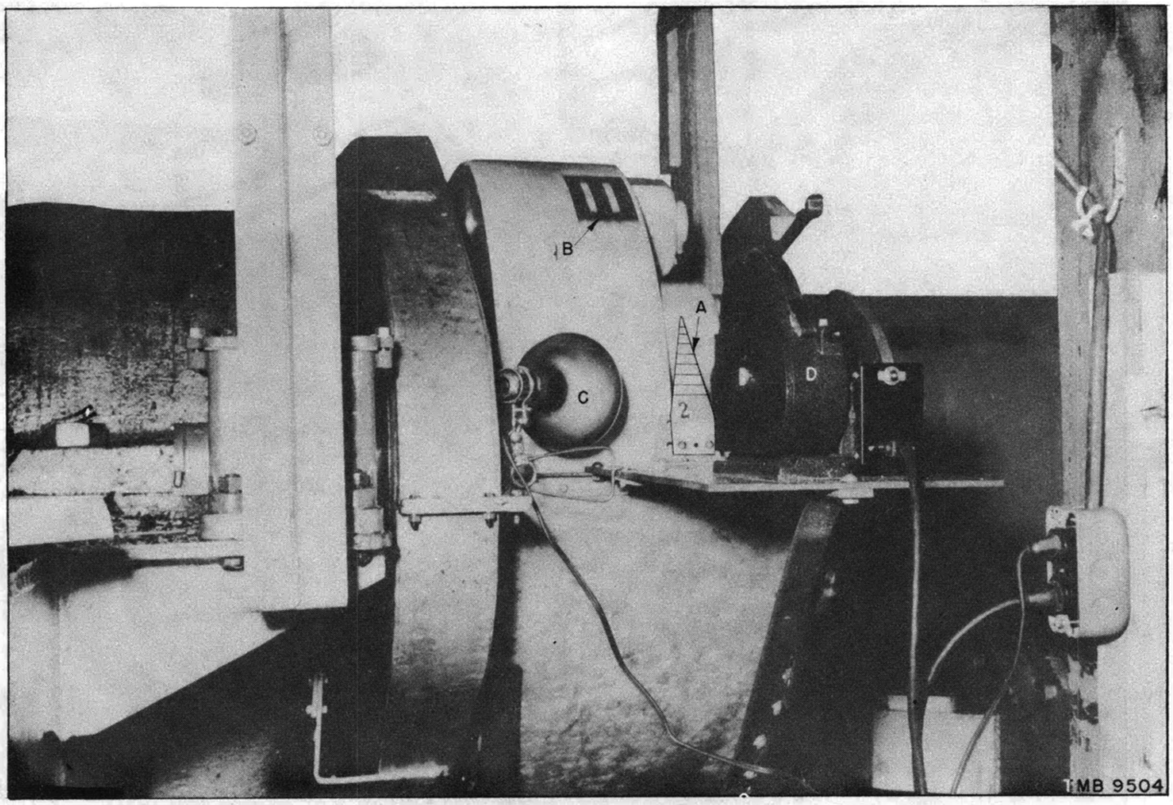


Figure 18 - Optical Wedge for Determining Shaft Displacement

A target B is fixed to one of the coupling flanges on the propeller shaft and moves axially with the shaft if there is axial displacement. A tapered wedge A, fixed to the ship structure, is illuminated by the lamp C and viewed by the camera D. Axial displacement is measured by using the graduations on the tapered wedge as a vernier when the wedge is viewed against the straight sides of target B.

FULL-SCALE PROPELLERS

The principal dimensions of all propellers used during the full-scale trials of both ships are given in Table 1.

APPARATUS FOR VIBRATION OF UNCOUPLED MACHINERY UNIT

The vibration generator used in this test on the NORTH CAROLINA was supplied by the General Electric Company. It consisted simply of a single adjustable eccentric driven by a d-c motor. It produced sinusoidal forces in two directions; however, measurements were made only of horizontal amplitudes in the fore-and-aft direction. As this machine produced peak exciting forces of the order of only 1000 pounds and as the amplitudes were too small to measure with anything but sensitive electrical pickups, the Sperry-MIT vibrations-measuring equipment previously described was used for amplitude measurements in these tests.

TABLE 1

Principal Dimensions* of Propellers Used in Full-Scale Trials

	Port		Starboard	
	Outboard	Inboard	Inboard	Outboard
Original propellers installed on USS NORTH CAROLINA for trials of 19 May 1941 and 27 May 1941 and on USS WASHINGTON for trials of 29 May to 2 June 1941				
Serial Number (USS WASHINGTON)	4442	4441	4440	4439
Number of Blades	3	4	4	3
Diameter, feet and inches	17-3	16-7 1/2	16-7 1/2	17-3
Pitch at 0.7 R, feet and inches	17-2 1/4	16-4 1/2	16-4 3/4	17-2 5/8
Mean Width Ratio	0.327	0.283	0.283	0.327
Blade Thickness Fraction	0.050	0.054	0.054	0.050
Projected Area, square feet	98.41	105.53	105.20	98.61
Helicoidal Area, square feet	112.99	120.71	120.34	113.21
Propeller Weight, pounds	33,660	34,965	34,780	34,080
Weight of Cap, Gland, Studs, etc.	1,261	1,219	1,229	1,211
Propellers with first modification as installed on USS WASHINGTON for trials of 21 June 1941				
Serial Number	4441	4546	4545	4440
Number of Blades	4	3	3	4
Diameter, feet and inches	16-7 1/2	16-4 5/8	16-4 5/8	16-7 1/2
Pitch at 0.7 R, feet and inches	16-4 1/2			16-4 3/4
Pitch at 72.45 R, feet and inches		17-3	17-1 5/8	
Mean Width Ratio	0.283	0.349	0.349	0.283
Blade Thickness Fraction	0.054	0.0528	0.0528	0.054
Projected Area, square feet	105.53	94.64	94.24	105.20
Helicoidal Area, square feet	120.73	108.99	108.53	120.34
Propeller Weight, pounds	34,965	33,420	33,875	34,780
Weight of Cap, Gland, Studs, etc.	1,219	1,261	1,211	1,229
Propellers with second modification as installed on USS WASHINGTON for trials of 12 July 1941				
Serial Number	4441	4546	4545	4440
Number of Blades	4	3	3	4
Diameter, feet and inches	16-7 1/2	15-10 1/2	15-10 1/2	16-7 1/2
Pitch at 0.7 R, feet and inches	16-4 1/2			16-4 1/2
Pitch at 72.45 R, feet and inches		17-3	17-1 5/8	
Mean Width Ratio	0.283	0.359	0.359	0.283
Blade Thickness Fraction	0.054	0.0533	0.0533	0.054
Projected Area, square feet	105.53	91.03	91.29	105.20
Helicoidal Area, square feet	120.73	105.14	105.44	120.34
Propeller Weight, pounds	34,965	No record	No record	34,780
Weight of Cap, Gland, Studs, etc.	1,219	1,261	1,211	1,229
Propellers of new design as installed on USS WASHINGTON for trials of 2 December 1941				
Number of Blades	4	5	5	4
Diameter, feet and inches	16-11	16-3 1/2	16-3 1/2	16-11
Pitch at 0.7 R, feet and inches		16-1 1/2	16-1 1/2	
Pitch at 0.65 R, feet and inches	16-5			16-5
Mean Width Ratio	0.285	0.250	0.250	0.285
Blade Thickness Fraction	0.0473	0.0476	0.0476	0.0473
Projected Area, square feet	109.6	110.3	110.3	109.6
Model Propeller Numbers	2208	2210	2209	2207
Propellers of new design as installed on USS WASHINGTON for trials of 23 April 1944*				
Number of Blades	4	5	5	4
Diameter, feet and inches	16-11	15-4	15-4	16-11
Pitch at 0.65 R, feet and inches	16-5	16-1 1/2	16-1 1/2	16-5
Mean Width Ratio	0.285	0.274	0.274	0.285
Projected Area, square feet	109.62	101.2	101.2	109.62
Projected Area/Disk Area Ratio	0.488	0.548	0.548	0.488
Helicoidal Area, square feet	127.58	122.40	122.40	127.58
Model Propeller Numbers	2208	2509	2508	2207

* The blade thickness fractions varied for all these propellers.

APPARATUS FOR TRANSVERSE VIBRATION MEASUREMENTS IN SKEGS

Metalectric strain gages similar to those used for measuring thrust in the shaft were applied to the shell plating at the tops of the skegs just forward of the strut castings. As the complications due to slip rings were absent, the gages were supplied with direct current from a battery. The electrical output from each gage was impressed on an amplifier whose output was connected to a cathode-ray oscillograph. Records were made by photographing the screen of the oscillograph. A horizontal pallograph was also installed in the bottom of one skeg on the first trial of the NORTH CAROLINA. The relation between stress and amplitude was compared with that found in the previous test made with the vibration generator (2) in which the transverse natural frequency of the skeg in water was determined.

MODEL RESTRAINING BLOCK

Two model restraining blocks were designed and built at the Taylor Model Basin, one to check the general theory and one to demonstrate the feasibility or non-feasibility of the design proposed for the ship installation. The scale factor of the first design was 0.23; that of the second design 0.30. Complete details of the models and the tests are given in Reference (10); a longitudinal section through one of the models is shown in Figure 19.

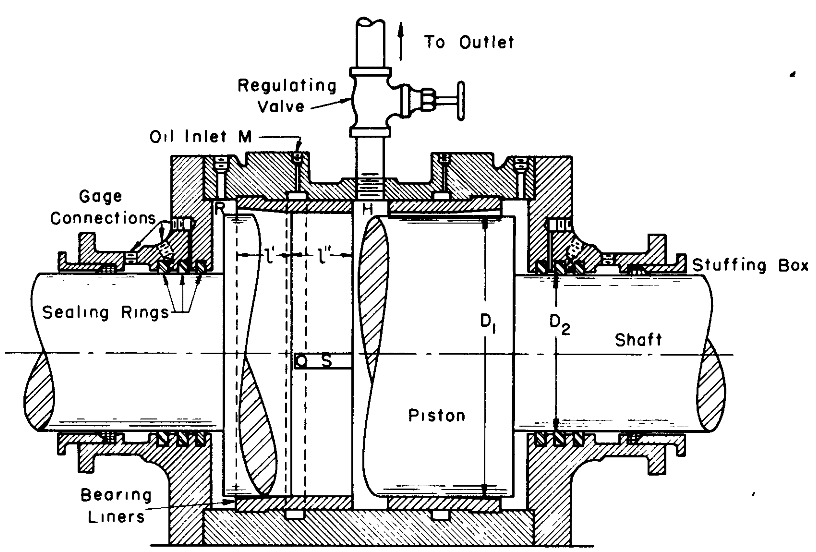


Figure 19 - Longitudinal Section Through 0.3-Scale Model Restraining Block

The oil clearances around the piston are exaggerated here to show the condition which obtains when the block is in operation. The block here serves as a shaft steady bearing as well as a restraining device.

APPARATUS FOR MEASURING THRUST VARIATION ON THE LARGE SELF-PROPELLED MODEL

Thrust was measured on a 30-foot self-propelled model of the BB55 and 56 class during tests in the model basin by an adaption of the means used to measure full-scale thrust on the ships. To avoid slip-ring difficulties and the difficulty of attaching strain gages to the 1/4-inch-diameter propeller shaft of the model, a special thrust-pickup unit was designed and constructed; see Figure 20. This replaced the thrust unit of the self-propulsion dynamometer ordinarily used in model tests; the thrust was taken on tension strips to which the metalectric gages were attached. A special coupling was provided for transmitting torque to the propeller shaft without

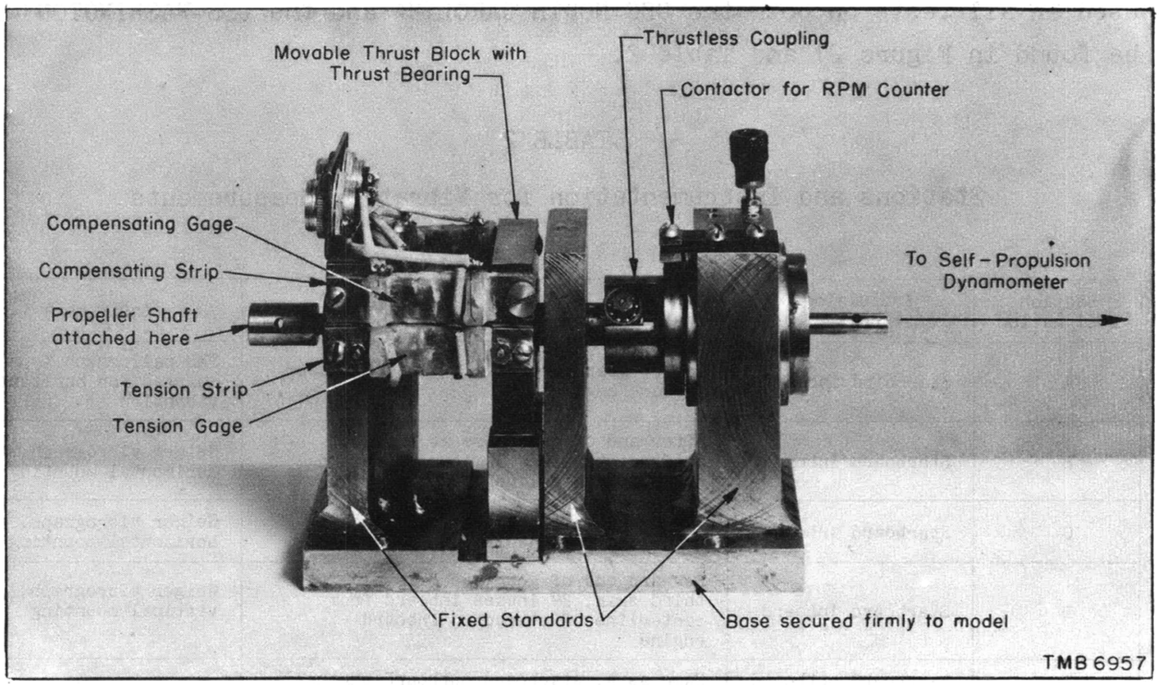


Figure 20 - Thrust Pickup Used on 30-Foot Self-Propelled Model

A short shaft section at the left is coupled to the propeller shaft of the model, and is mounted in journal bearings in the two fixed standards at the left. A thrustless fork coupling at the right-hand end of this short shaft enables the self-propulsion dynamometer to drive it at the required speed.

The left-hand shaft section is pushed against the movable thrust block through a thrust collar; all propeller thrust is thus delivered to this block. Ball thrust bearings are not used as these bearings generate strong vibrations as they rotate.

The thrust block is connected to the left-hand fixed standard by two bronze tension strips, one on each side abreast the shaft, upon which metalectric gages are mounted.

Compensating gages mounted on similar strips alongside, free from the effect of thrust, take care of temperature and other variations and form the third and fourth arms of the bridge.

affecting the action of the propeller thrust on the thrust pickup. The remainder of the equipment used in measuring thrust variation on the model was similar to that used in measuring thrust on the shafts of the full-scale ships.

TEST PROCEDURE

MEASUREMENTS MADE DURING SEA TRIALS

The quantities measured in the engine rooms were the amplitudes and frequencies of horizontal vibration in the fore-and-aft direction at a number of selected stations such as the top of the main reduction-gear case, the gear-case foundation, the high-pressure turbine, the low-pressure turbine, and the condenser. The exact locations and a key of the station designations used in all tests on both the USS NORTH CAROLINA and the USS WASHINGTON will be found in Figure 21 and Table 2.

TABLE 2

Stations and Instrumentation for Vibration Measurements

Station Designation	Propulsion Unit	Location	Instrument
A	Starboard inboard	Top of gear case at aft end, starboard inboard engine	TMB pallograph Type B adapted for horizontal vibration
B	Starboard inboard	After end of gear case at level of shaft centerline, 36 inches inboard of shaft centerline	Geiger vibrograph, horizontal mounting
C	Starboard inboard	Forward end of gear case, 24 inches below shaft centerline under shaft	Geiger vibrograph, horizontal mounting
D	Starboard inboard	Forward end of gear case, level of third skin, 24 inches inboard from centerline of starboard inboard engine	Geiger vibrograph, vertical mounting
E	Starboard outboard	Same as A, starboard outboard engine	
F	Port outboard	Same as A, port outboard engine	TMB pallograph Type C
H	Port inboard	Same as A, port inboard engine	
J	Starboard inboard	High-pressure-turbine foundation girder, 9 inches above shaft centerline starboard inboard engine	Sperry pickups for both vertical and horizontal vibration
K	Starboard inboard	After end of condenser near top, starboard inboard engine	
L		After end of low-pressure turbine near bottom	

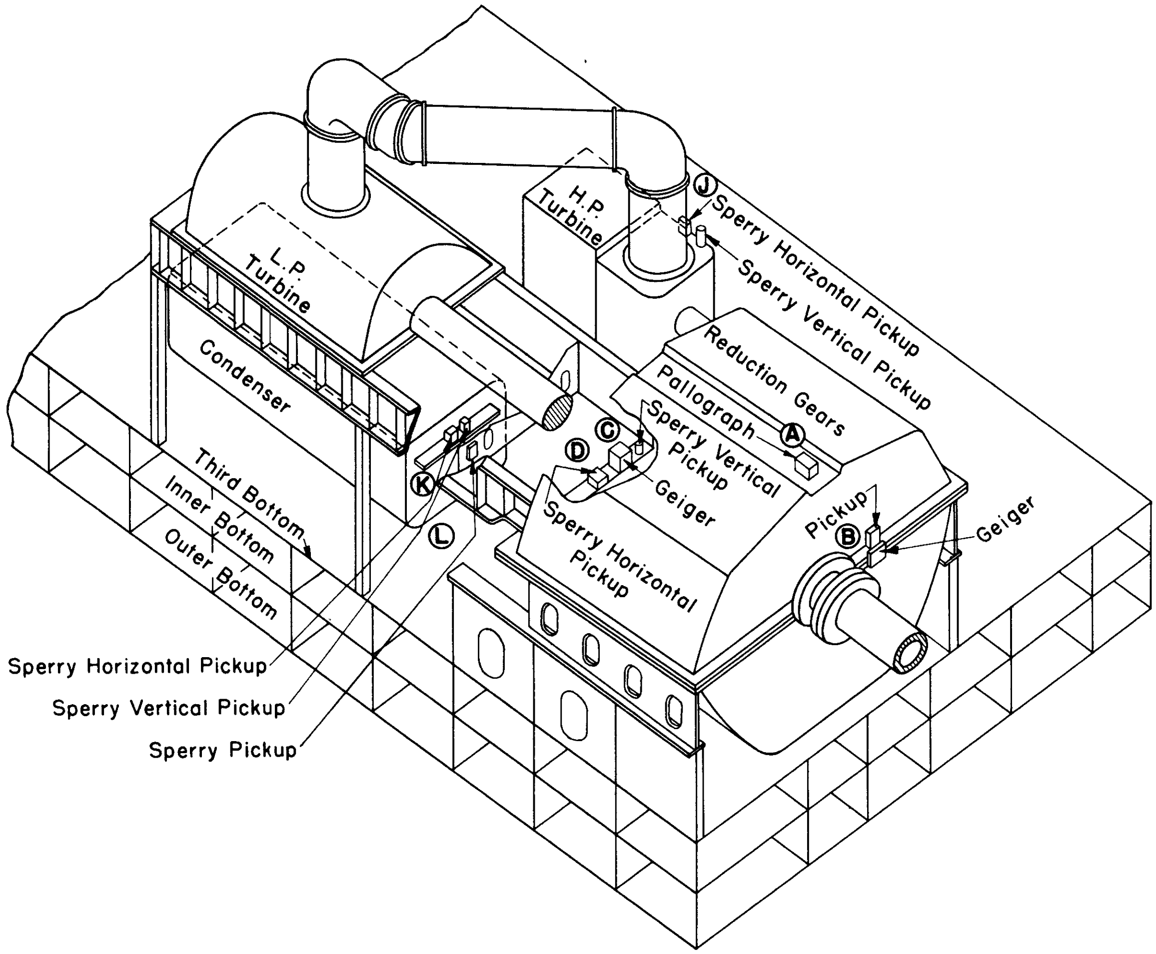


Figure 21 - Diagram Showing Location of Test Equipment for Starboard Inboard Engine, USS NORTH CAROLINA and WASHINGTON

The arrangement in other engine rooms was generally similar, but less equipment was used; see Table 2.

A special compartment on the third deck aft on the starboard side* was assigned to the Taylor Model Basin and the Naval Research Laboratory parties as an instrument room and test-control station on each ship. All the remote-recording instruments and their operating personnel were located in this compartment, which was centrally situated with respect to the stations where the measuring impulses originated.

The following apparatus was installed in this compartment: Ford torsionmeter receivers for all shafts; control equipment and oscillographs for measuring thrust; control equipment and oscillograph for measuring instantaneous torque by the Gunn method (12) (13).

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* On the NORTH CAROLINA the compartment used for this purpose was the large detention cell. On the WASHINGTON a nearby compartment, C-305-L, was used.

Observations at the remaining apparatus had to be made at the instruments themselves.

It was essential that all measurements be coordinated with the maneuvering of the ship. To make this possible a special sound-power telephone circuit was rigged which included the following stations: The central station for oscillographs, one station in each engine room, one station in each shaft alley, and one station in each skeg, when stress and displacement measurements were being taken in the skegs. The telephones at the central station were manned by an officer from the Taylor Model Basin who was in effective charge of coordinating all activities in connection with the measurements of shaft and engine vibration. The phones in the observation stations were manned by members of the crew who relayed the messages to the engineers.

The schedule of operations was decided upon, and all interested parties were informed before the ship reached the area of operations. During the actual tests the procedure was as follows: As soon as the ship had steadied at a selected speed, the Engineer Officer informed the Coordinating Officer in the central station for oscillographs, who then instructed the Ford-torsionmeter operators to take readings, meanwhile passing the word to all other stations to stand by. Electrical interference made it impossible to operate the Ford torsionmeters and the thrust-recording equipment simultaneously. As soon as the torsionmeter readings were completed, all other stations took readings, and after all stations had reported completion of measurements to the Coordinating Officer, the Engineer Officer proceeded to establish the next speed. In general, shaft speeds were varied in steps of 10 RPM, starting at some low value such as 50 RPM.

The various tests conducted at sea are listed in Table 3, which also gives the structural modifications tested, the propellers used, and the quantities measured in each test.

VIBRATION TEST OF ENGINE FOUNDATION

A special test with a vibration generator supplied by the General Electric Company, as described previously on page 30, was made on the NORTH CAROLINA to determine the masses and the spring constants of the individual members of the engine-foundation system as an aid in making an analysis of the normal modes of vibration.

The tests were carried out 15 July 1941 while the ship was moored to a dock. The port inboard engine was selected for this test as it was the only one in which the shaft could be uncoupled close to the main reduction-gear case. This change left the machinery free to vibrate independently of the shaft and the propeller.

TABLE 3

Summary of Sea Trials

Ship	Date	Conditions	Propellers[††] Inboard	Propellers[††] Outboard	Vibratory Quantities Measured	Speed Range RPM
BB55 NORTH CAROLINA	20 May 41	Original reduction-gear foundations, 38,000 tons displacement	4-bladed original	3-bladed original	Amplitude and frequency in Starboard Inboard Machinery Space; thrust on starboard inboard shaft; torque on all shafts	95-168
BB55 NORTH CAROLINA	27 May 41	Original reduction-gear foundations, 38,000 tons displacement	4-bladed original	3-bladed original	Amplitude and frequency at most stations; see Table 2; thrust on all shafts	70-180
BB55 NORTH CAROLINA	25 Aug 41	Condenser braced from bottom to forward end of thrust bearing foundation, 42,000 tons displacement	3-bladed cut down	4-bladed; original inboard wheels	Amplitude and frequency in Starboard Inboard Machinery Space	+
BB56 WASHINGTON	30 May 41	Original reduction-gear foundations	4-bladed original	3-bladed original	Amplitude and frequency in Starboard Inboard Machinery Space	+
BB56 WASHINGTON	21 Jun 41	*Gear foundations, Engine Rooms 3 and 4, stiffened by ties to bulkhead, inner bottom and third skin. Wire cables installed between inner bottom and main condensers.	3-bladed, modified spares; radius reduced by 5 1/4 inches	4-bladed; original inboard wheels	Amplitude and frequency at all stations; shaft amplitude and thrust on all shafts	20-198
BB56 WASHINGTON	12 Jul 41	Condition 1 Thrust-bearing clearance reduced. ** Condensers, Machinery Spaces 3 and 4, braced by 5-inch pipe strut to third skin. ** Extra brackets applied to gear foundations in Machinery Space 3 and 4. Displacement 38,600 tons. Condition 2 Displacement 42,000 tons. Otherwise same as Condition 1.	3-bladed; radius reduced by an additional 3 inches	4-bladed; original inboard wheels	Amplitude and frequency at all stations; shaft amplitude and thrust on all shafts	110-200
BB56 WASHINGTON	26 Jul 41	Condition 1 † Turbine foundations, Machinery Spaces 3 and 4, stiffened by ties to bulkhead and third skin. Two outer spacers between gear case and turbine foundation removed; center spacer left in position in Machinery Spaces 3 and 4. Additional stiffening between condenser and third skin. Condition 2 All three spacers between gear case and turbine foundation removed. Light longitudinal condenser ties in place. Otherwise same as Condition 1.	3-bladed; radius reduced by 8 1/4 inches	4-bladed; original inboard wheels	Amplitude and frequency in Starboard Inboard Machinery Space	+
BB56 WASHINGTON	3 Aug 41	Center spacer between gear case and turbine foundation replaced. 44,000 tons displacement.	3-bladed; radius reduced by 8 1/4 inches	4-bladed; original inboard wheels	Qualitative observation only; no recorded data	+
BB56 WASHINGTON	2 Dec 41	Center spacer between gear case and turbine foundation in position. Displacement 42,000 tons.	5-bladed new	4-bladed new	Amplitude and frequency at most stations; thrust on all shafts	110-202
BB56 WASHINGTON	23 Apr 44	Standard full load displacement. Smaller diameter and scimitar-shaped (skewed) 5-bladed propellers installed on inboard shafts.	5-bladed new; redesigned	4-bladed new	Amplitude and frequency at Stations A, E, and H	110-205

* Details of this installation may be found on BuShips plan 293223.

** Details are given on BuShips plans 292334 and 293225.

† See BuShips plan 293226.

†† See Table 1 on page 31 for further details of propellers.

\+ There was no TMB personnel aboard. Available data is incomplete.

The test procedure was as follows: The eccentricity of the vibration generator was adjusted to a safe value and the machine was run at successively increasing speeds. A Sperry-MIT vibration-pickup unit was placed at a selected point, such as the top of the gear case, and the amplitudes observed on the viewer of the oscillograph at each speed were recorded and plotted on a graph. The resonant speed was then determined from this graph and the vibration generator was run at this speed while amplitudes were measured at numerous points on the gear case and turbines by a "roving pickup," while the phase was checked against that of a fixed pickup. Thus the mode of motion and the relative amplitudes in different parts of the machinery were determined. This procedure was duplicated for the six conditions described in Table 4, in all of which the line shaft remained disconnected from the machinery. In all of the foregoing conditions the condenser was completely filled with water.

TABLE 4

Test Conditions for Vibration Test of Engine-Gear System

Condition	Main Thrust Bearing	Turbine Thrust Bearings	Spacers between Turbines and Gear Case	Condenser Support	Location of Vibration Generator
1	Working clearance taken up	Working clearance taken up	In place	As built	Forward end of main thrust bearing
2	Working clearance taken up	Clearance as designed	In place	As built	Forward end of main thrust bearing
3	Working clearance taken up	Clearance as designed	Removed	As built	Forward end of main thrust bearing
4	Working clearance taken up	Clearance as designed	Removed	As built	Outboard foundation rail of low-pressure turbine
5	Working clearance taken up	Clearance as designed	Removed	Braced to third bottom by 5-inch pipe struts	Outboard foundation rail of low-pressure turbine
6	Working clearance taken up	Clearance as designed	In place	Braced to third bottom by 5-inch pipe struts	Outboard foundation rail of low-pressure turbine

From the response curves obtained under these various conditions it was possible to deduce experimental values of the effective masses and spring constants of the members of the engine unit, as discussed more fully on pages 57, 58, 59, and 60.

MODEL TESTS

The thrust pickups described on page 26 were mounted in the 30-foot model of the BB55 and 56 class. The usual self-propulsion dynamometers for driving the propellers were used in these tests, except that in the case of shafts on which the thrust pickups were installed, the propeller thrust was absorbed by the pickup instead of by the dynamometer. The test procedure was similar to that for routine self-propelled model tests (14) with the following exceptions:

1. The thrust on the shafts which were provided with pickups was recorded by oscillograph instead of being read on the self-propulsion dynamometers.

2. It was found that at low speeds the thrust forces were so small that the indications from the pickups were unreliable, so that the tests were not extended to speeds below about 50 per cent of maximum speed. Torque and RPM on all shafts, as well as thrust on any shafts not provided with pickups, were read on the self-propulsion dynamometer in the usual manner. This provided a check on the values of the steady component of thrust obtained from the pickups.

A summary of the model tests is given in Table 5.

TEST RESULTS

VIBRATION TESTS MADE DURING SEA TRIALS

Transverse Skeg Vibrations

The vibrations found in the skegs were not excessive and the results of these measurements can be given quite briefly. Both the pallograph records and the strain oscillograms showed the movements of the skegs to be made up of several frequencies. However, the wave forms did not repeat themselves, in contrast with the more or less steady conditions observed in the engine rooms. No definite resonance was established from these data, and as the axial vibration of the shafts did not appear to be caused by the transverse vibration of the skegs, the analysis of the records was carried no further than the computation of the peak stress vibrations. These increased steadily with increasing speed.

TABLE 5

Schedule of Thrust-Variation Tests on Model 3689,
Representing the BB55 and 56 Class

Date	Thrust-Variation Data	Number of Propeller Blades		Remarks
		Inboard Shafts	Outboard Shafts	
27-28 Oct 1941	Starboard inboard shaft only	3	4	Preliminary experiment for developing the method.
		4	3	Use of cathode-ray oscilloscope gave records of length insufficient for analysis. Only one pickup available.
12 Nov 1941	Starboard inboard shaft only	4 2180* 2181	3 2178-2* 2179-2	Only one pickup available. Simulated conditions on NORTH CAROLINA during trials of 20 May and 27 May 1941.
		5 2209 2210	4 2207 2208	This test simulated conditions then contemplated and later realized on WASHINGTON during trials of 2 December 1941.
26 Aug 1942	Port and starboard inboard shafts	5 2209 2210	4 2196 2197	Simulated conditions approximately on WASHINGTON during the trials of 12 July 1941.
		3 2178-M 2179-M 4 2180 2181	4 2196 2197 4 2196 2197	Thrust pickups of improved design as shown in Figure 20. The primary purpose of these experiments was to find the effect of a change in the shape of the skegs. This was a control experiment with the origial skegs.
20 Oct 1942	All four shafts	3 2178-M 2179-M	4 2196 2197	Same pickups as used on test of 26 August. Skeg modified as shown in Figure 52. Note: The starboard outboard pickup developed mechanical trouble and was removed soon after the start of the first series of tests of this group.
	Port and starboard inboard shafts only	4 2180 2181	4 2196 2197	
	Port and starboard inboard shafts only	5 2209 2210	4 2196 2197	

* The 4-digit numbers are the serial numbers of the model propellers used.

The peak value of stress variation at the top of the skeg obtained at top speed was only about 200 pounds per square inch double amplitude. The maximum observed double amplitude of transverse motion at the bottom of the skeg was about 0.026 inch. These data check fairly closely the correlation between deflection and stress obtained in the special tests previously made (1) (2) to determine the natural frequency of the skegs. In the latter tests a double amplitude of 0.033 inch in the transverse direction at the shaft level produced a stress variation of 450 pounds per square inch. This amplitude represented about 0.066 inch at the bottom of the skeg.

As the blade frequencies at top speed, 200 RPM with 4 blades, were in excess of the natural frequency of the skegs, about 600 cycles per minute, with no indication of a serious resonant vibration, it may be concluded that the forces acting on the skegs partly counterbalance one another. In this connection, however, it should be pointed out that while the frequency test showed the damping to be only about 4 per cent of critical, this represents an appreciable absolute-damping constant when such large masses and spring constants are involved, and it is the absolute damping constant that determines the amplitude produced by a given driving force. For instance, it was shown in the frequency test that an excitation of 7400 pounds was required to produce a single amplitude of the skeg at the shaft level of 0.016 inch, measured horizontally in a transverse direction.

Torsional Vibration of Shafts

Torsional vibrations of the line shafts were of extremely small amplitude. In general, the torque variation was less than 4 per cent of the corresponding steady torque values; this is about the smallest value observable with either of the types of torsionmeters used. The accuracy of the results is therefore low, but it can be stated definitely that the amplitude of torsional oscillation is inappreciable. From visual observation made while the ship was getting underway, the torsional critical speeds of all shafts appeared to be less than 50 RPM.

Effect of Twin Rudders Behind Propellers

As there was little evidence during the early trials of the NORTH CAROLINA that the presence of the twin rudders directly behind the inboard propellers was responsible for the longitudinal shaft vibrations, and as the little evidence that appeared was somewhat contradictory, no extensive measurements were made while the rudders were in use. Recent trials made in April 1944 indicate that the maximum amplitude with rudders in use is about twice that found on a straight course. Obviously no sea trials could be made with the rudders removed.

Measurements of Longitudinal Shaft Movement

In Figures 22 and 23 are plotted the axial shaft movements observed on the two starboard shafts of the WASHINGTON on one trial. There were two stations along each shaft, one just behind the reduction-gear case and one in the shaft alley. Readings are given for all stations, with the condenser stays both tight and slack.

Measurements of Amplitude and Frequency of Vibration

Complete data on observed amplitudes of horizontal fore-and-aft vibration are given in Table 6 and in Figures 24 to 32. The most convenient stations for use as a basis of reference were those located at the after end of the top of the gear cases in each engine room. For the first trial run

(Text continued on page 48)

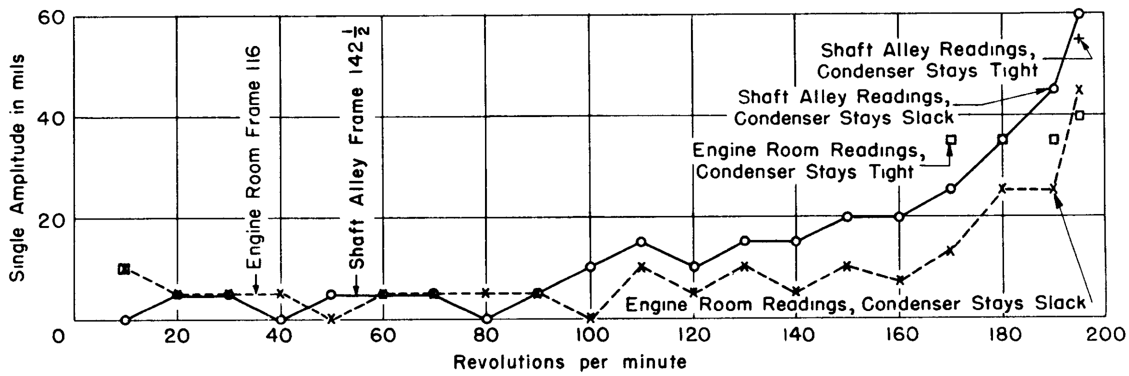

Figure 22 - USS WASHINGTON - Longitudinal Shaft Movement Plotted on RPM for Starboard Inboard Shaft - Trial Run of 21 June 1941

The engine-room readings were made close to the reduction gear.

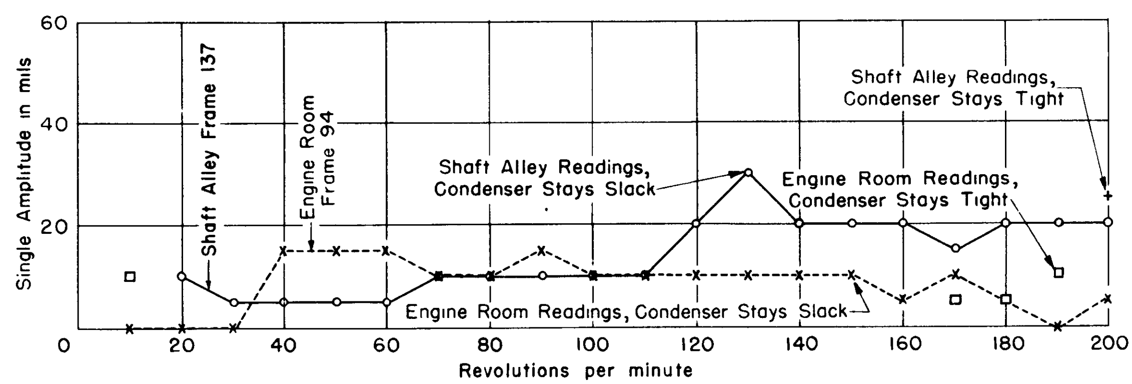

Figure 23 - USS WASHINGTON - Longitudinal Shaft Movement Plotted on RPM for Starboard Outboard Shaft - Trial Run of 21 June 1941

The engine-room readings were made close to the reduction gear.

TABLE 6

Summary of Observed Resonant Frequencies and Amplitudes

Ship	Date	Station†	Number of Propeller Blades: Inboard	Number of Propeller Blades: Outboard	Frequency of Observed Peak Amplitude (Probable Resonance) CPM	Corre-sponding Speed RPM**	Peak Single Amplitude at Resonance, Horizontal, Fore and Aft mils	Type of Instrument Used	Structural Modification See Table 3 on Page 37 for Details
BB55 NORTH CAROLINA	27 May 41	A B C E F H	4	3	600 600 600 510 540 680	200 200 200 170 180 223	30.0 11.3 9.7 30.0 30.0 35.0	Pallograph Vibrograph Vibrograph Vibrograph Vibrograph Pallograph	None
BB56 WASHINGTON	30 May 41	A B J	4	3	600 600 600	200 200 200	21.0 36.0 18.0	Oscillograph Oscillograph Oscillograph	None
BB56 WASHINGTON	21 Jun 41	A A A B B B C C E F H H J K L	3	4	580 580* 580 600 580* 580 590 590* 560 600 590 600* 580 570 570	145 145* 145 150 145* 145 147* 147* 187 200 147 150* 145 142 142	16.0 15.0* 16.5 9.5 8.0 58.0†† 12.0 12.0 5.0 9.0 27.0 22.5* 16.0 150.0†† 72.0††	Pallograph Pallograph* Oscillograph Vibrograph Vibrograph* Oscillograph Vibrograph Vibrograph* Pallograph Vibrograph Vibrograph Vibrograph* Oscillograph Oscillograph Oscillograph	Gear foundations stiffened. Wire-cable stays fitted from condensers to inner bottom to restrain vibratory motion of the condensers. The belief that these relatively flimsy stays could have any effect on a mass as large as the condenser shows that the fundamentals of the vibration phenomena were not well understood at the time. There is some evidence that the damping introduced by the viscosity of the water in the moving condenser was beneficial.
BB56 WASHINGTON	12 Jul 41	A E	3	4	585 520	146 173	20.0 3.5	Pallograph Pallograph	Additional stiffening of turbine and condensers foundation. Spacers between gear case and turbine foundation removed.
BB55 NORTH CAROLINA	25 Aug 41	A	3	4	580	145	19.0	Pallograph	Spacers replaced as on 12 July 41
BB56 WASHINGTON	2 Dec 41	A E H	5	4	600 525 680	120 131 136	13.0 6.0 13.0	Pallograph Pallograph Pallograph	
BB56 WASHINGTON	23 Apr 44	A E H	5	4	600 560 700	120 140 140	6.0 4.5 8.3	Vibrograph Vibrograph Vibrograph	None

* The condenser stays mentioned in the last column were tight when these observations were taken. All other observations were taken with the stays either slacked off or absent altogether.

** Maximum propeller RPM of these ships is 200.

† Station designations are given in Table 2 on page 34.

†† Questionable readings.

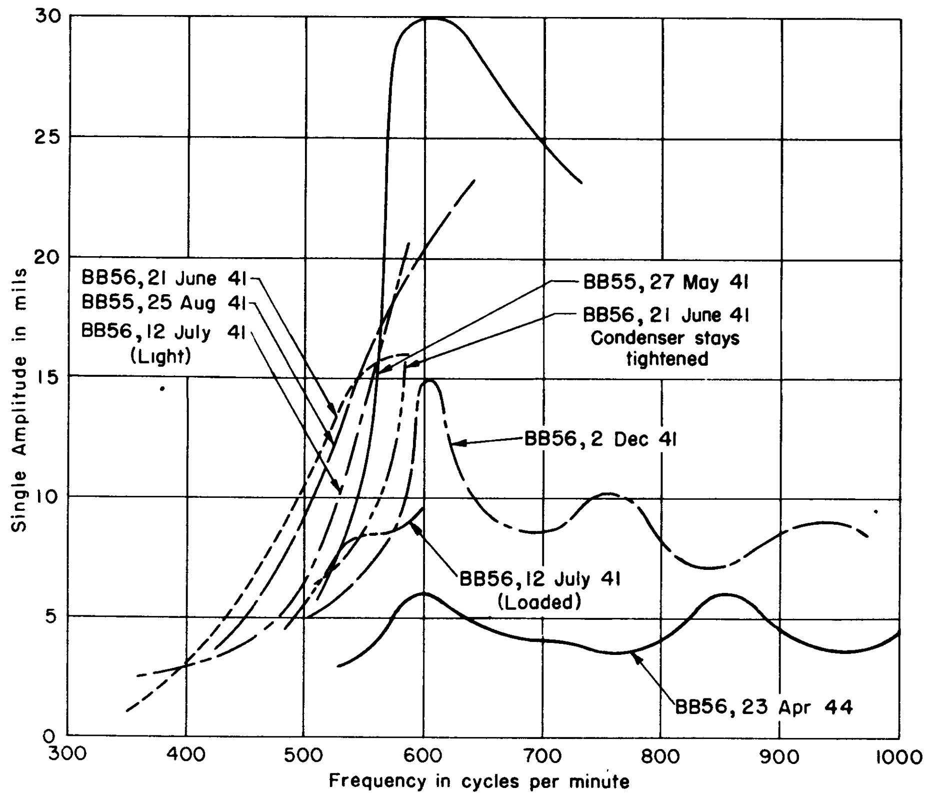

Figure 24 - USS WASHINGTON and USS NORTH CAROLINA - Vibration Measurements Made on Top of Starboard Inboard Gear Case (Station A) during Trial Runs

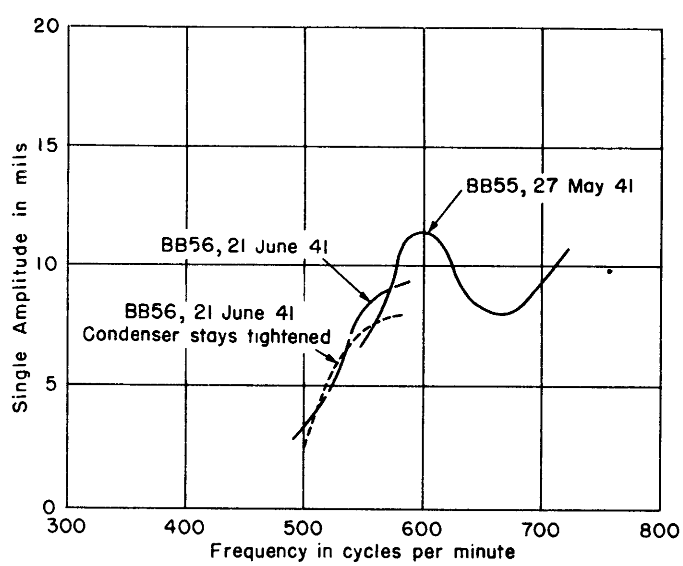

Figure 25 - USS WASHINGTON and USS NORTH CAROLINA - Measurements Made of Horizontal Fore-and-Aft Vibration at After End of Starboard Inboard Level of Shaft Centerline (Station B) during Trial Runs

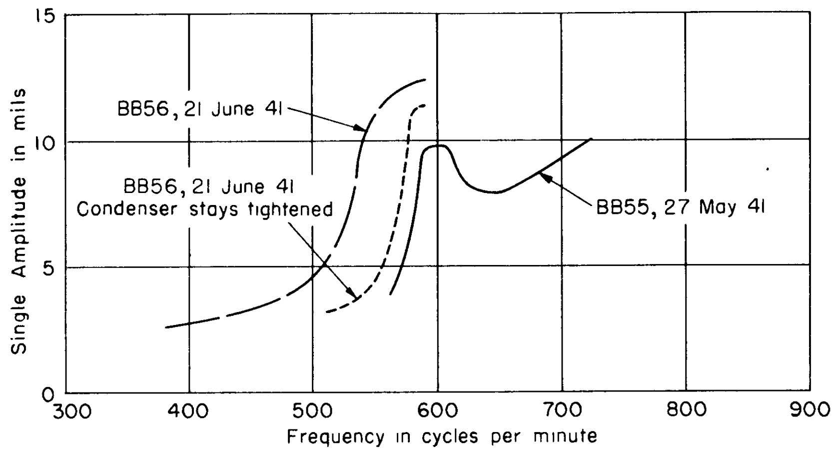

Figure 26 - USS WASHINGTON and USS NORTH CAROLINA - Vibration Measurements Made at Forward End of Starboard Inboard Gear Case (Station C) during Trial Runs

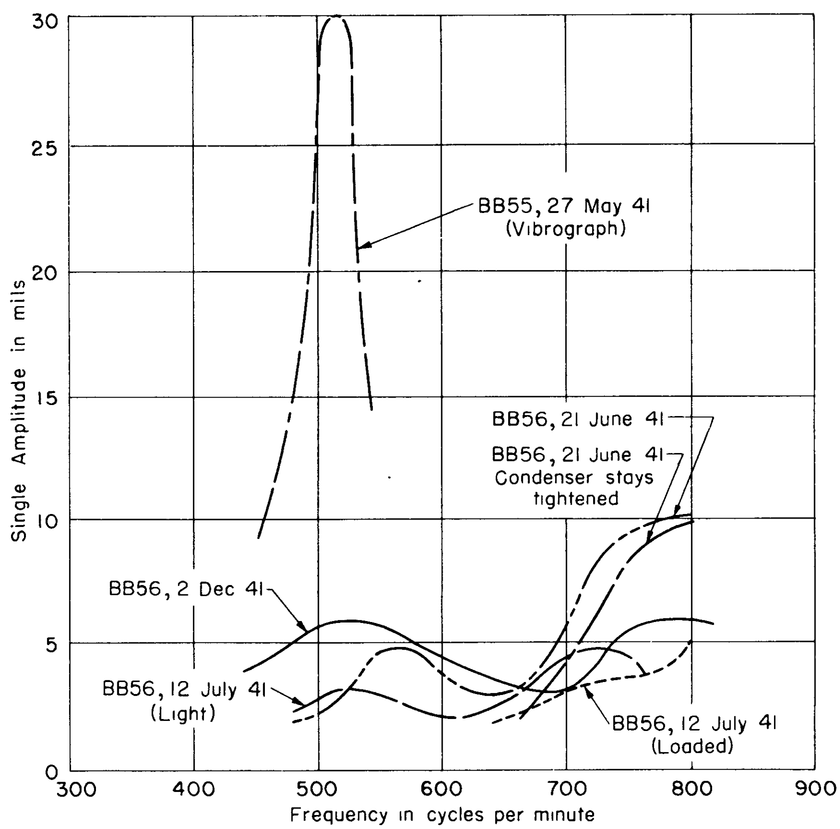

Figure 27 - USS WASHINGTON and USS NORTH CAROLINA - Vibration Measurements Made on Top of Starboard Outboard Gear Case (Station E) during Trial Runs

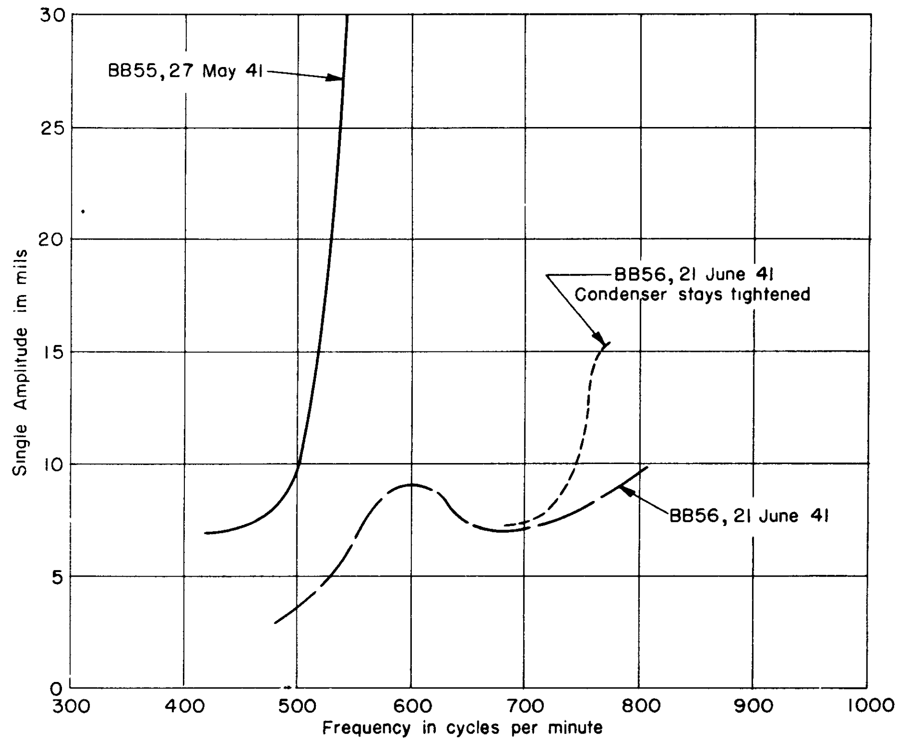

Figure 28 - USS WASHINGTON and USS NORTH CAROLINA - Measurements of Horizontal Fore-and-Aft Vibration Made on Top of Port Outboard Gear Case (Station F) during Trial Runs

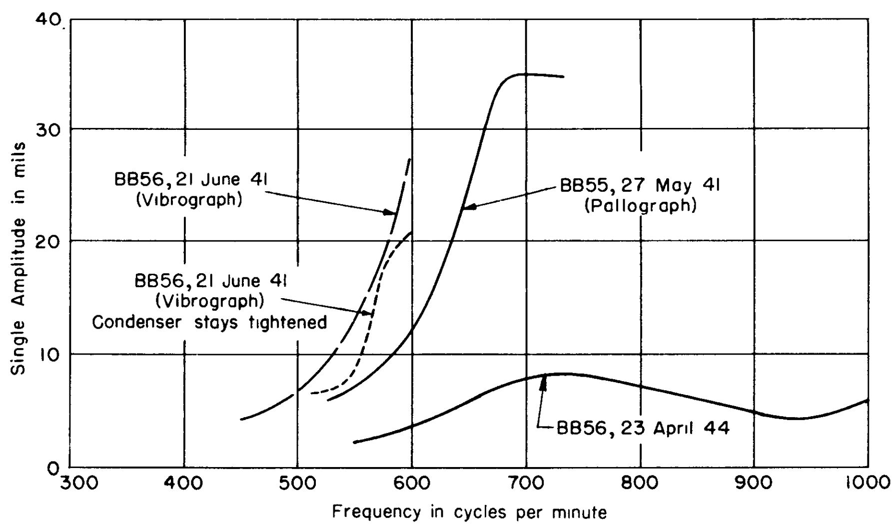

Figure 29 - USS WASHINGTON and USS NORTH CAROLINA - Measurements of Horizontal Fore-and-Aft Vibration Made on Top of Port Inboard Gear Case (Station H) during Trial Runs

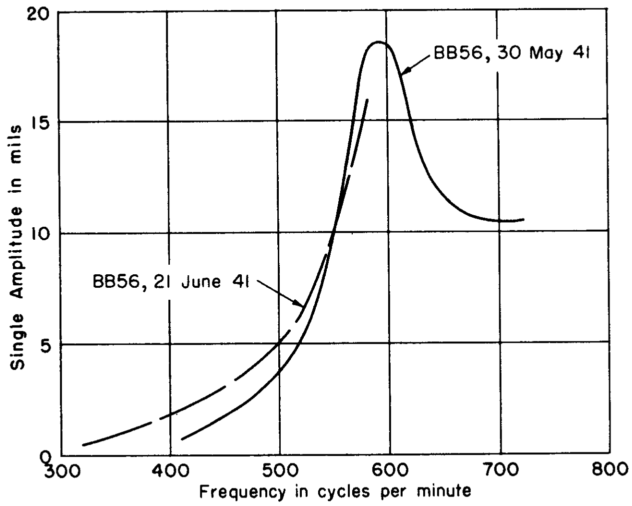

Figure 30 - USS WASHINGTON - Vibration Measurements Made on High-Pressure Turbine Foundation (Station J) during Trial Runs

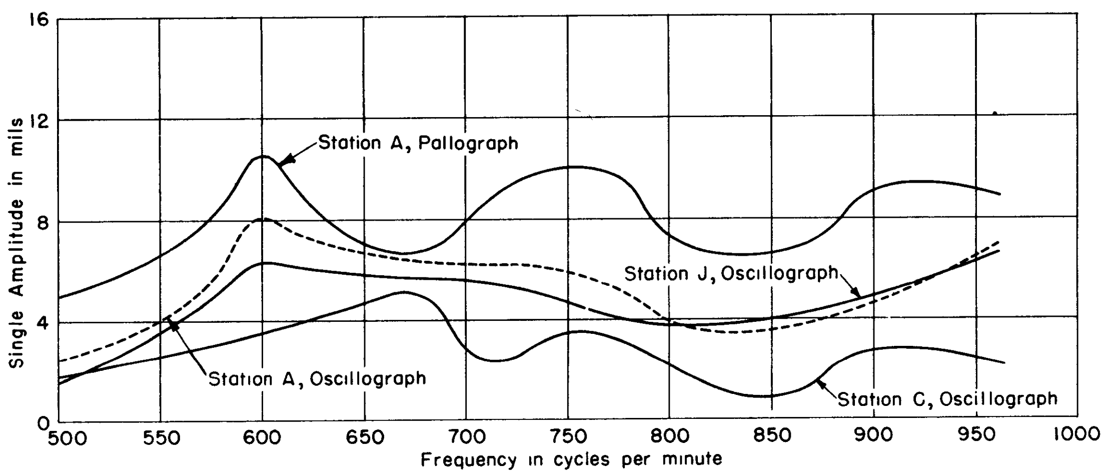

Figure 31 - USS WASHINGTON - Vibration Measurements Made in Starboard Inboard Engine Room during Trial Run of 2 December 1941 with 5-Bladed Propellers on Inboard Shafts

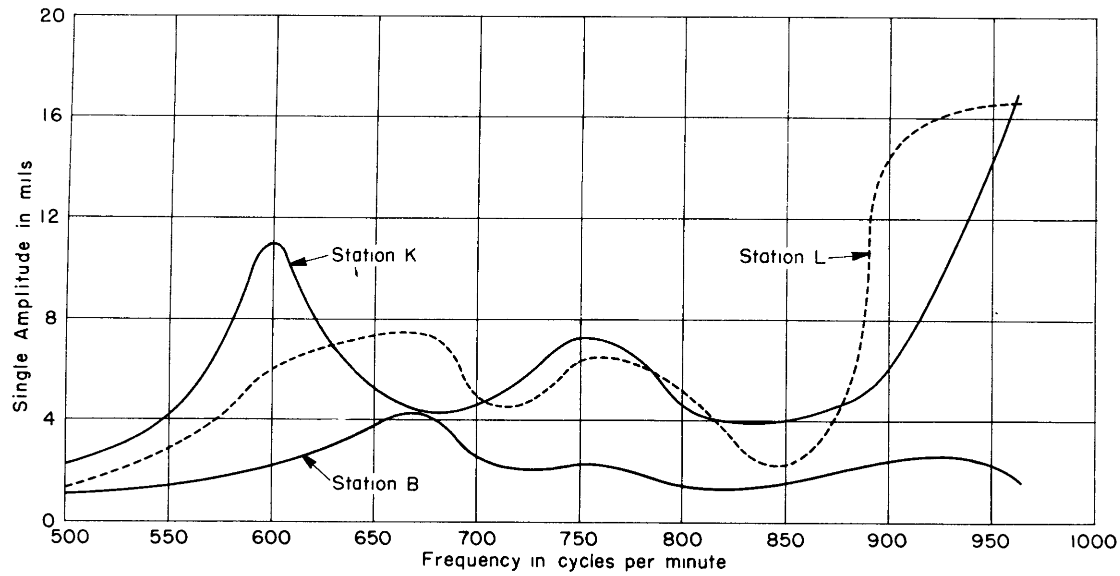

Figure 32 - USS WASHINGTON - Vibration Measurements Made in Starboard Inboard Engine Room during Trial Run of 2 December 1941 with 5-Bladed Propellers on Inboard Shafts

on the NORTH CAROLINA the highest single amplitude recorded was 35 mils for the port inboard engine; the other three engines all showed vibration amplitudes of 30 mils.

The observed resonant frequencies, determined from the average of a number of trials, were as follows: starboard outboard engine, 520 CPM; starboard inboard engine, 600 CPM; port inboard engine, 680 CPM; port outboard engine, 600 CPM. Blade frequencies at full power, 200 RPM, were 600 CPM, 800 CPM, and 1000 CPM, respectively, for 3-, 4-, and 5-bladed propellers.

These resonant values were checked by several different instruments in the engine rooms, including pallographs, vibrographs, and electrical vibration pickups, as well as wire-resistance thrust indicators on the individual shafts.

As was predicted before the stiffening was fitted, the structural changes on the WASHINGTON affected these frequencies very slightly, if at all.

Measurements of Thrust Variations in Shafts

Averaged results of the measurements of the total alternating components of thrust are given in Figures 33 to 37. This information is summarized in Table 7.

TABLE 7

Summary of Thrust Forces in Shafts

Shaft Location	Port		Starboard	
	Outboard	Inboard	Inboard	Outboard
NORTH CAROLINA 20 May 1941				
Number of Propeller Blades	3	4	4	3
Maximum Variation from Steady Thrust, pounds single amplitude			71,300	
Speed at which Maximum Occurred, RPM			148††	
Indicated First-Mode Resonance, cycles per minute			*	
Corresponding Steady Thrust,† pounds			111,000	
Plus and Minus Variation of Thrust as Percentage of Steady Thrust			64	
NORTH CAROLINA 27 May 1941				
Number of Propeller Blades	3	4	4	3
Maximum Variation from Steady Thrust, pounds single amplitude	73,600	121,500	87,300	63,000
Speed at which Maximum Occurred, RPM	180	170	150	170
Indicated First-Mode Resonance, cycles per minute	*	680	600	510
Corresponding Steady Thrust,† pounds	176,000	157,000	114,000	157,000
Plus and Minus Variation of Thrust as Percentage of Steady Thrust	41	77	76	40
WASHINGTON 21 June 1941				
Number of Propeller Blades	4	3	3	4
Maximum Variation from Steady Thrust, pounds single amplitude	36,900**	61,200	92,350	16,240**
Speed at which Maximum Occurred, RPM	200	200	200	200
Indicated First-Mode Resonance, cycles per minute	600	*	600	520
Corresponding Steady Thrust,† pounds	236,000	236,000	236,000	236,000
Plus and Minus Variation of Thrust as Percentage of Steady Thrust	15	25	38	6
WASHINGTON 12 July 1941				
Number of Propeller Blades	4	3	3	4
Maximum Variation from Steady Thrust, pounds single amplitude	58,600**	No data	101,500	21,760**
Speed at which Maximum Occurred, RPM	190		190††	190
Indicated First-Mode Resonance, cycles per minute				520
Corresponding Steady Thrust,† pounds	205,000		205,000	205,000
Plus and Minus Variation of Thrust as Percentage of Steady Thrust	28		50	16
WASHINGTON 2 December 1941				
Number of Propeller Blades	4	5	5	4
Maximum Variation from Steady Thrust, pounds single amplitude	21,800	47,700	50,500	18,800**
Speed at which Maximum Occurred, RPM	150	136	120	195
Indicated First-Mode Resonance, cycles per minute	600	680	600	560
Corresponding Steady Thrust,† pounds	114,000	92,000	73,000	220,000
Plus and Minus Variation of Thrust as Percentage of Steady Thrust	19	51	69	8

* The test did not extend to a sufficiently high speed to determine resonance.

** Maximum occurred in the neighborhood of second-mode resonance, which is apparently near full speed.

† Taken from model tests.

†† The highest speed for which a reliable measurement is available.

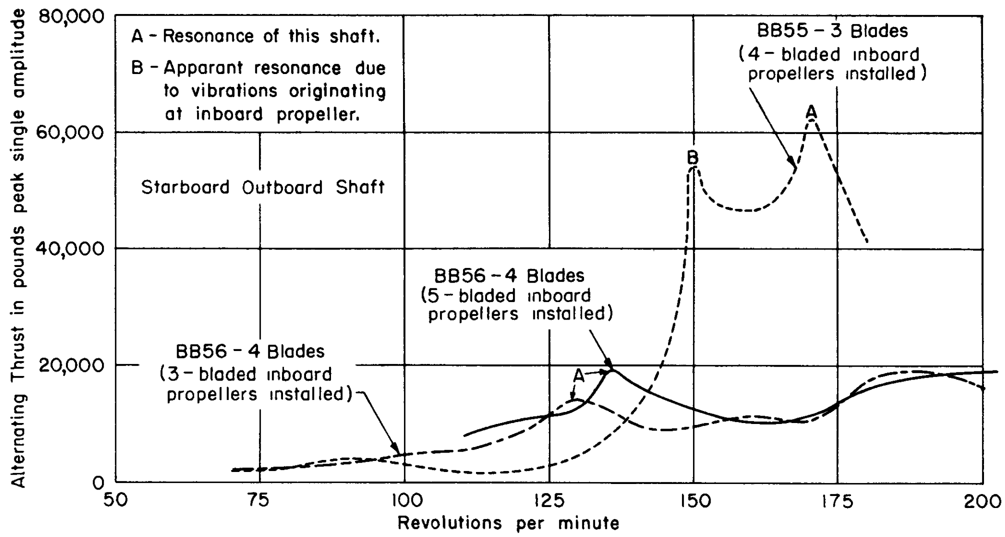

Figure 33 - USS WASHINGTON and USS NORTH CAROLINA - Alternating Thrust in Shaft, Plotted on Propeller RPM

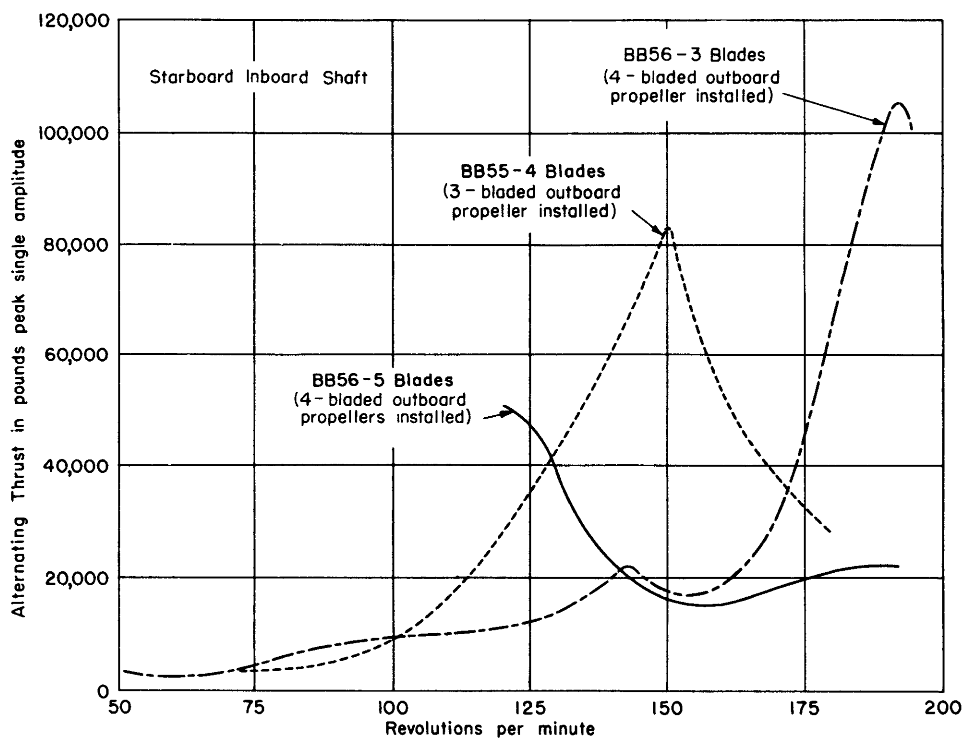

Figure 34 - USS WASHINGTON and USS NORTH CAROLINA - Alternating Thrust in Shaft, Plotted on Propeller RPM

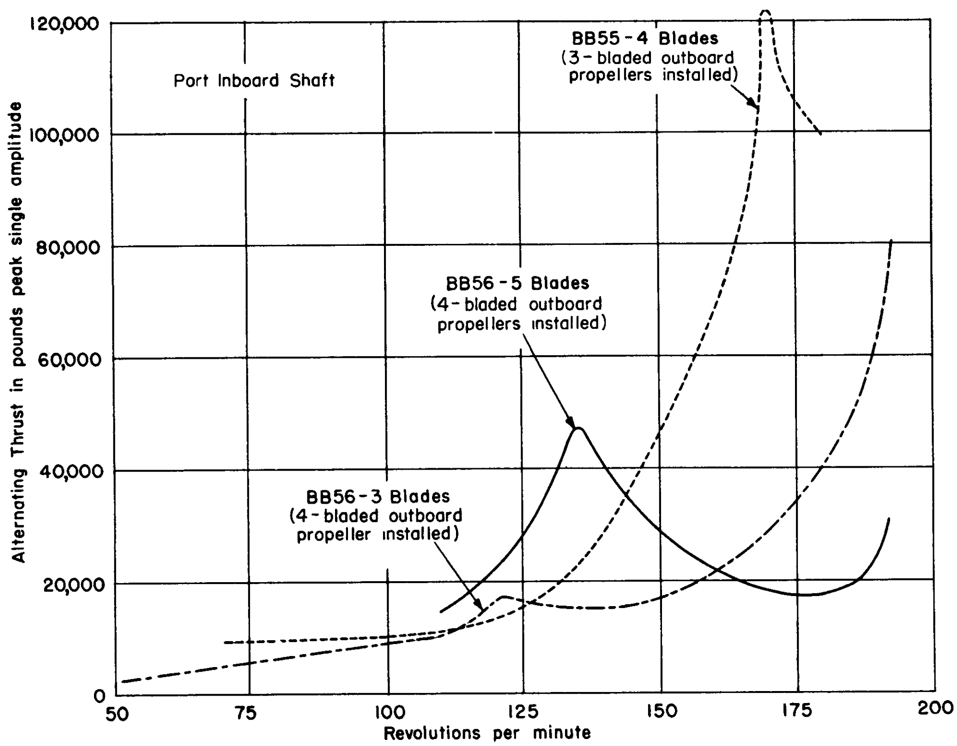

Figure 35 - USS WASHINGTON and USS NORTH CAROLINA - Alternating Thrust in Shaft, Plotted on Propeller RPM

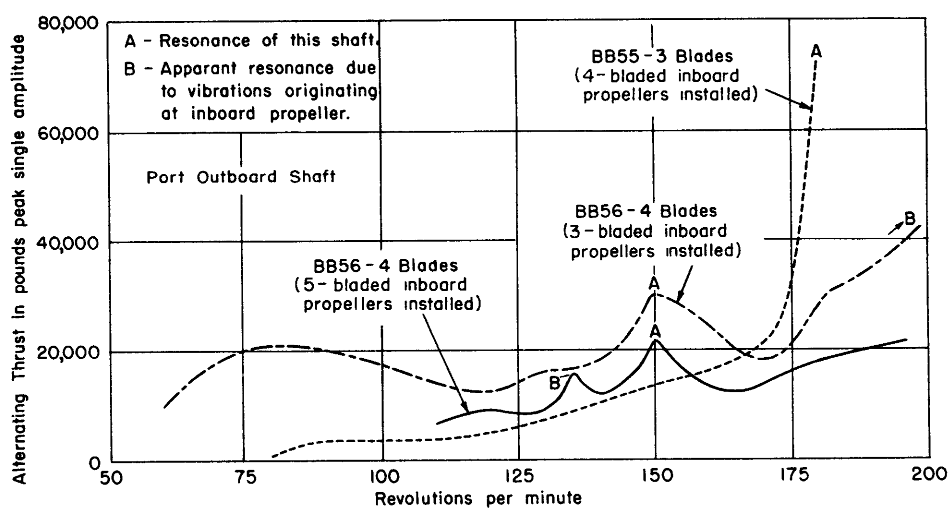

Figure 36 - USS WASHINGTON and USS NORTH CAROLINA - Alternating Thrust in Shaft, Plotted on Propeller RPM

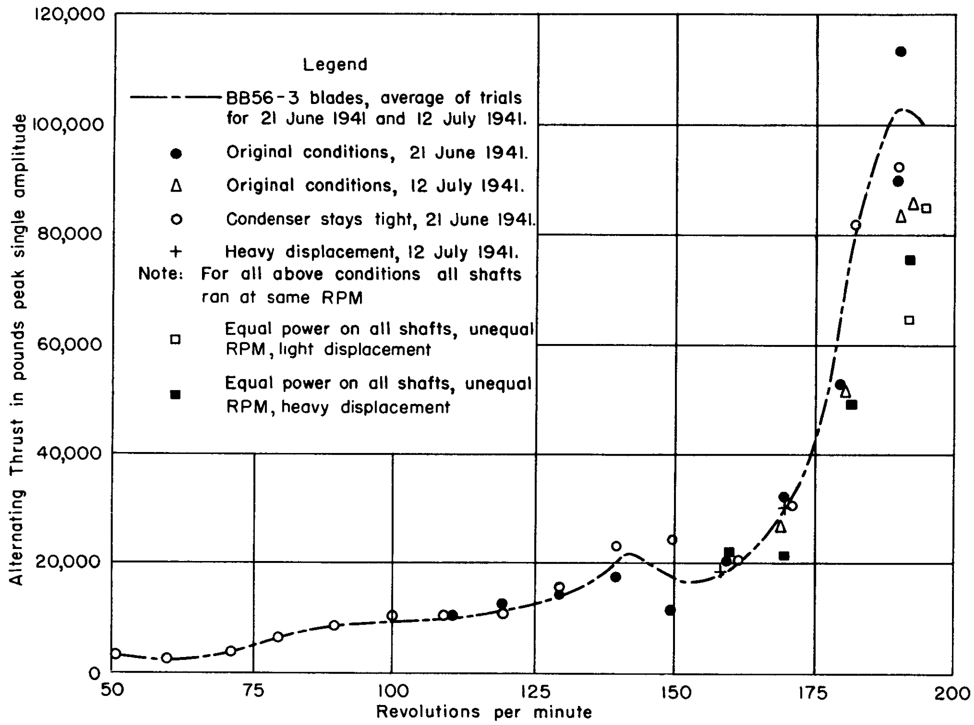

Figure 37 - USS WASHINGTON - Effect of Structural Modifications and Power Balance between Inboard and Outboard Shafts

FORCED VIBRATION OF UNCOUPLED ENGINE

The natural frequencies of the engine units as determined in the test with the vibration generator on the NORTH CAROLINA are given in Table 8.

Representing the three masses by m_a, m_b, and m_c, as shown in the diagram of Figure 8 on page 17, the ratio of the amplitudes when vibrating in resonance in Condition 1, at 900 CPM, was 1 : 1.5 : 3.0. All three masses vibrated in phase with each other.

MEASUREMENTS OF THRUST VARIATIONS ON THE MODEL

The lowest longitudinal natural frequency of the propulsion system used in the model would have been in resonance at about twice the maximum speed of the model. For this reason the model results are of no significance prior to analysis. The results of the model experiments are therefore included in the analysis of results which follows.

TABLE 8

Test of Port Inboard Engine, USS NORTH CAROLINA with Vibration Generator

For this test the line shaft was uncoupled just aft of the gear case.

Condition	Main Thrust Bearing	Turbine Thrust Bearing	Spacers between Turbines and Gear Case	Condenser Support	Location of Vibration Generator	Natural Frequency CPM
1	Working clearance taken up	Working clearance taken up	In place	As built	Forward end of main thrust bearing	900
2	Working clearance taken up	Clearance as designed	In place	As built	Forward end of main thrust bearing	900
3	Working clearance taken up	Clearance as designed	Removed	As built	Forward end of main thrust bearing	1500
4	Working clearance taken up	Clearance as designed	Removed	As built	Outboard foundation rail of low-pressure turbine	650
5	Working clearance taken up	Clearance as designed	Removed	Braced to third bottom by 5-inch pipe struts	Outboard foundation rail of low-pressure turbine	760
6	Working clearance taken up	Clearance as designed	In place	Braced to third bottom by 5-inch pipe struts	Outboard foundation rail of low-pressure turbine	940

ANALYSIS OF RESULTS

GENERAL

Several features stand out rather prominently from a preliminary examination of the test data.

In spite of the more or less symmetrical disposition of the two parts of the inboard skegs above and below the shaft centerline, and regardless of the number of blades on the inboard propellers, the exciting force generated by an inboard propeller is predominantly of blade frequency. This and other evidence bears out the original thought that the blade closest to the main body of the ship's hull is the blade most responsible for the vibration.

Likewise, regardless of the angle between the two arms of the outboard struts and any correspondence between that angle and one of similar magnitude between the blades of an outboard propeller, the blade passing close to the hull is the one which generates the significant exciting force.

Because of the well-known phenomenon of magnification at resonance in an elastic vibrating system, a relatively small periodic exciting force at the propeller can produce dangerously high loads and amplitudes if the periods of the two coincide.

Whenever the blade frequency becomes equal to the resonant frequency of any elastically mounted unit, such as a fire-control instrument, that unit can be expected to vibrate in unison. In addition, if the blade-frequency forces are magnified by resonance in the propelling machinery and its foundations, the vibration can be expected to be serious.

Blade-frequency forces and the vibration produced by them, other things being equal, may be expected to increase with the ship speed and the propeller thrust at a rate corresponding to the propeller thrust.*

It is to be expected that in general structural modifications will change the natural frequencies of the structure and that propeller changes will modify the frequencies and amplitudes of the exciting forces.

EFFECTS OF PROPELLER CHANGES AND STRUCTURAL ALTERATIONS

The simultaneous structural additions to the thrust-bearing foundations and the changing of the propellers on the WASHINGTON make it difficult to assess definitely the effect of each individual change on that vessel.

The effect of the first propeller change on these vessels, namely of shifting the original 4-bladed propellers from the inboard to the outboard shafts and installing modified 3-bladed propellers on the inboard shafts, was to reduce the amount of thrust variation, as shown in Figures 33 to 36 on pages 50 and 51.

Inboard Shafts

The original 4-bladed propellers used on the inboard shafts produced resonance near full speed in the case of the port inboard unit; the harmful effects of this have previously been described. Installing 3-bladed propellers on the inboard shafts placed this resonance peak at a shaft speed 4/3 as great as the critical speed with the 4-bladed propellers, i.e., slightly beyond the running range for the port inboard shaft at the top of the range for the starboard inboard shaft.

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* The blade frequency forces vary directly as the thrust, approximately as the torque, and directly as the square of the ship speed.

The 3-bladed propellers, originally designed for the outboard positions, absorbed more than their share of power when placed behind the inboard skegs. The effect of the trimming of the blade tips on the 3-bladed propellers when mounted on the inboard shafts was not noticeable except in decreasing the power and increasing the revolutions for those shafts.

The exciting force with 3-bladed propellers was found to be considerably less than that with 4-bladed propellers, and nearly the same as that with 5-bladed propellers, but the damping constant of the 3-bladed propeller was less than the damping constant of the 5-bladed propeller. When using these 5-bladed propellers, resonance occurred well below full speed and the absolute magnitude of the resulting force in the shaft was moderate. The result was that the maximum alternating force measured in the starboard inboard shaft when coupled to the 3-bladed propeller was 91 per cent of that measured when this shaft was coupled to the 4-bladed propeller, and 173 per cent of that measured when the shaft was coupled to the 5-bladed propeller.* The alternating force measured in the port inboard shaft when coupled to the 3-bladed propeller was 50 per cent of that measured when coupled to the 4-bladed propeller and 128 per cent of that measured when coupled to the 5-bladed propeller. In other words, the effect of installing the original 5-bladed propellers on the inboard shafts was to reduce the maximum thrust variation in the starboard inboard shaft to 58 per cent of its original value, and that in the port inboard shaft to 39 per cent of its original value.

Vibrations in the superstructure were slightly less with 3-bladed propellers than with the first 5-bladed propellers used, when running at cruising speed, but the 3-bladed propellers caused considerably greater vibration in the superstructure at maximum speed. The latest design of 5-bladed propellers has reduced the superstructure vibration to negligible amplitudes.

Outboard Shafts

The original 3-bladed propellers used on the outboard shafts caused resonant vibration very near full speed. Replacing these with 4-bladed propellers caused a reduction in the critical speed to 75 per cent of its original value. This change in critical speed would have reduced the vibratory force to 56 per cent of the original value if there had been no other factors to consider. In addition, the exciting force with 4-bladed propellers in the wing positions was slightly less than the corresponding force with 3-bladed propellers. The result was that the maximum alternating force measured in

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* These figures represent the average of all measurements made; see Table 7 on page 49.

the outboard shaft when coupled to the 4-bladed propellers was 52 per cent of that measured when these shafts were coupled to 3-bladed propellers.*

The figures given include the effect of the second-mode resonance, which was approached when driving with 4-bladed propellers. The use of 4-bladed instead of 3-bladed propellers on the outboard shafts therefore produced a clear reduction of shaft displacement and vibration of nearly 50 per cent. No further complications attended the use of 4-bladed propellers on the outboard shafts, except an unbalance in power, as described elsewhere. It should be noted that the outboard shafts were not housed in skegs but were carried by conventional struts.

In general it may be stated that the effect of propeller changes on the resonant frequencies of the propelling machinery systems was negligible, as the masses of all the propellers were nearly the same. Most of the changes in observed amplitudes of vibration can be explained principally by the change in frequency of the exciting force due to the propeller changes.

The installation of the new 5-bladed inboard propellers and the new 4-bladed outboard propellers on the WASHINGTON prior to the trial run of 23 April 1944 reduced all vibrations to acceptable values.

Effects of Foundation Reinforcement and Various Other Changes

The effect of foundation reinforcements on the resonant frequencies was not perceptible. This result agrees with theoretical results based on the vibration constants of the system as derived on page 59. As foundation reinforcement can affect the results principally by changes in natural frequency and only secondarily by damping, the vibration amplitudes are believed to have been only slightly affected by the foundation reinforcements carried out. In fact, in view of the theoretical results given on page 59, no foundation changes practicable on the finished vessel could be expected to produce a significant improvement.

The effect of structural modifications on the thrust variation was negligible. This was to be expected in view of the fact that the structural changes had no appreciable effect on the resonant frequencies and no effect whatever on the wake variation.

The effect of changing the power distribution among the shafts, after the propellers were first interchanged, by running the outboard shafts 10 RPM faster than the inboard shafts, was a slight reduction of the vibration in the inboard engine rooms, principally because the outboard propellers

* These figures represent the average of all measurements made; see Table 7 on page 49.

then removed some of the overload from the inboard propellers. The thrust variation in the inboard shafts was, however, reduced by about 20 per cent for both heavy and light displacements; the increased thrust variation on the outboard shafts which was expected turned out to be inappreciable. There was no effect on the resonant frequencies.

On one of the trial runs on the WASHINGTON it was observed that increasing the displacement from 38,600 tons to 42,000 tons reduced the amplitude of vibration in the starboard engine rooms by an additional 25 per cent. However, as this occurred simultaneously with a redistribution of power among the shafts, the result was not entirely conclusive.

EVALUATION OF VIBRATION CONSTANTS

As explained on pages 15 and 16 the longitudinal vibration properties of the propulsion system of these vessels can be reproduced approximately by a mechanical system like that in Figure 7.

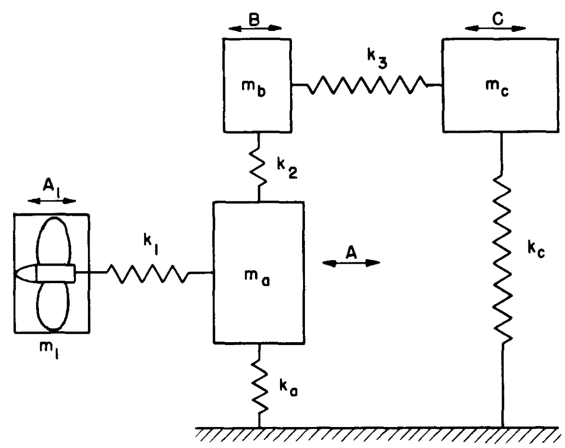

Figure 7 - Schematic Representation of a 4-Body Elastic System

The advantage in setting up this equivalent system lies in its simplicity; the actual propulsive system is too complicated to analyze mathematically. The entire response of the equivalent system can be described by equations of a tolerable degree of complexity; from these equations the results of any suggested changes, such as stiffening the engine foundations, changing the number of propeller blades, or applying damping, can be readily predicted once the constants of the system have been evaluated.

In this scheme, the effective propeller mass m_1 and the effective shaft stiffness k_1 can be obtained from simple measurements, to be explained presently. The other quantities pertain to structures of such complexity that other means must be employed. In the case under consideration, the constants were determined by a test with the vibration generator as described on page 32.

A certain amount of complexity attaches even to the constants of the propeller and the shaft, which are comparatively simple structures. The shaft has mass as well as elasticity, and both the mass and the elasticity are distributed throughout the length of the shaft. It can be shown (15) that the principal longitudinal vibration properties of such a shaft are the same as those of masses, each equal to 1/2 the mass of the shaft, connected by a spring having a constant equal to the actual spring constant of the shaft plus $1/6\ m\omega^2$, where m is the mass of shaft and ω is 2π times the frequency f in cycles per second. The longitudinal stiffness of the shaft is readily calculable from its dimensions and Young's modulus for the material; k_1 then follows readily.

As the shaft stiffness is inversely proportional to the shaft length, a different value of k_1 is obtained for each shaft on the NORTH CAROLINA class. As all the other quantities are in general the same for all shafts and engines, this variation in k_1, in addition to minor variations in k_a due to the particular position in the ship, is responsible for the fact that the four fundamental frequencies of the four propulsion systems of each ship are not all the same.

As mentioned on page 15, the equivalent propeller mass is equal to the actual weighed mass of the propeller plus 1/2 of the mass of the shaft plus the entrained mass of the water which accompanies the propeller during its fore-and-aft vibration. Simple experiments on vibrating model propellers under water* showed that the mass of the water entrained was 30 per cent of the propeller mass, if the propeller was free to rotate in synchronism with the fore-and-aft vibrations, and 60 per cent if the vibrating propeller was not free to twist or "screw" through the water. As the torsional critical speeds of the shafts of these vessels were well below the longitudinal critical speeds, the propellers rotated independently of the fore-and-aft vibration. This gave an estimate of 315 pound-seconds squared per inch for m_1, a value which was consistent with all observed data.

The tests made with the vibration generator in the port inboard engine room, during which the propeller shaft was disconnected from the gears, showed conclusively that although the gear case, turbines, and condenser vibrated in phase, they did not vibrate as a unit. Owing to flexibility in the connecting members there was considerable variation in the

* These experiments were not recorded in detail so that no unpublished or published data are available.

individual amplitudes. Analysis of the motion must therefore, from the beginning, take account of more than one degree of freedom.

The solution for the natural frequencies of this 3-body system, neglecting damping, can be derived theoretically by setting up equations satisfying Newton's laws during free vibration.* This results in the following three equations in which ω is 2π times the natural frequency f and the other quantities refer to Figure 8.

$$-m_a\omega^2 A = -k_2(A-B) - k_a A \quad [6]$$

$$-m_b\omega^2 B = k_2(A-B) - k_3(B-C) \quad [7]$$

$$-m_c\omega^2 C = k_3(B-C) - k_c C \quad [8]$$

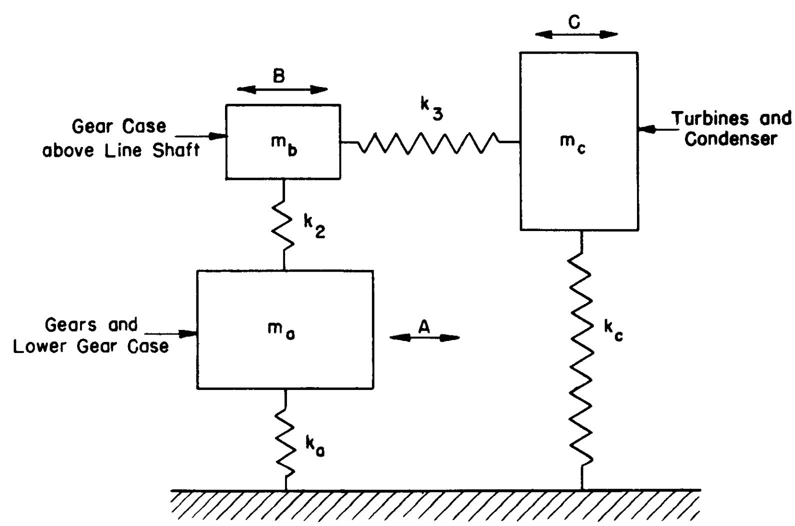

Figure 8 - Schematic Representation of a 3-Body Elastic System

This diagram represents one engine unit and its foundation after the connection between the large reduction gear and the line shaft has been broken.

In this scheme k_a represents the stiffness of the gear-case foundation, m_a the mass of that part of the gears and gear case considered to vibrate as a unit with the shaft, k_2 the rigidity of the part of the gear case above the shaft, k_3 the rigidity of the connection between the gears and the turbines, k_c the rigidity of the turbine foundation, m_b the mass of the upper part of the gear unit attached to the turbines, m_c the mass of the high-pressure turbine, low-pressure turbine, and condenser combined. It will be noted that the turbines and condenser are considered to vibrate as one unit, while the gear case is divided into two other units which have different amplitudes.

The capital letters indicate the amplitudes in arbitrary units. The test with the vibration generator indicated that the lowest natural frequency of this 3-body system with all masses moving in phase was 900 cycles per minute and that the relative amplitudes were $A = 1$; $B = 1.5$; $C = 3.0$.

* This is fundamentally the same method of analysis as that presented on page 15, except that here the calculations are applied to a 3-body instead of a 4-body system, on account of the fact that the propeller and shaft were disconnected from the rest of the system during these tests.

In this particular treatment of the problem the natural frequency and the relative amplitudes were assumed to be known from experimental data obtained in the special vibration test, and the mass and spring constants were evaluated by a cut-and-try process to fit the preceding equations.

The following values of the constants were obtained for the port inboard engine:

$k_a = 10.0 \times 10^6$ pounds per inch

$k_2 = 21 \times 10^6$ pounds per inch

$k_3 = 6.8 \times 10^6$ pounds per inch

$k_c = 0.20 \times 10^6$ pounds per inch

$m_a = 300$ pound-seconds squared per inch

$m_b = 90$ pound-seconds squared per inch

$m_c = 414$ pound-seconds squared per inch

With these constants introduced in the equations the ratios of the amplitudes A, B, and C becomes 1 : 1.33 : 2.18, whereas the experimental values obtained in the test with the vibration generator were 1 : 1.5 : 3.0. The disparity is an index of the validity of the simulation of the actual system shown in Figure 8 by the schematic model.

The constants thus obtained for the 3-body system could not be checked by the frequencies obtained when the spacers between the turbine foundations and the gear foundations were removed. This might be due to the fact that the foundations were not entirely independent and it was impossible to eliminate all coupling between m_c and m_a. The engine system could not be further broken down into its individual elements, but further verification of the constants is obtained by combining the engine-room system with the shaft and propeller to check the observed resonance of the whole propeller-shaft-engine system.

Since the engine foundations in all the machinery spaces in this class of battleships are substantially the same, with the exception of the port inboard unit, these same constants may be applied to the others with a reduced value of k_a, owing to the fact that the other gear foundations are longer than that in the port inboard machinery space.

When the propeller and shaft are combined with the engine, the 4-body system shown in Figure 7 is obtained. The following equations, repeated from page 16, must be satisfied for free harmonic vibration of the system:

$$-m_1\omega^2 A_1 = -k_1(A_1 - A) \qquad [2]$$

$$-m_a\omega^2 A = k_1(A_1 - A) - k_2(A - B) - k_a A \qquad [3]$$

$$-m_b\omega^2 B = k_2(A - B) - k_3(B - C) \qquad [4]$$

$$-m_c\omega^2 C = k_3(B - C) - k_c C \qquad [5]$$

The following values were found to satisfy these equations for the starboard inboard propeller-shaft-engine system:

m_1 = 315 pound-seconds squared per inch

m_a = 300 pound-seconds squared per inch

m_b = 90 pound-seconds squared per inch

m_c = 414 pound-seconds squared per inch

k_1 = 1.68×10^6 pounds per inch

k_a = 8.4×10^6 pounds per inch

k_c = 0.20×10^6 pounds per inch

k_2 = 21×10^6 pounds per inch

k_3 = 6.8×10^6 pounds per inch

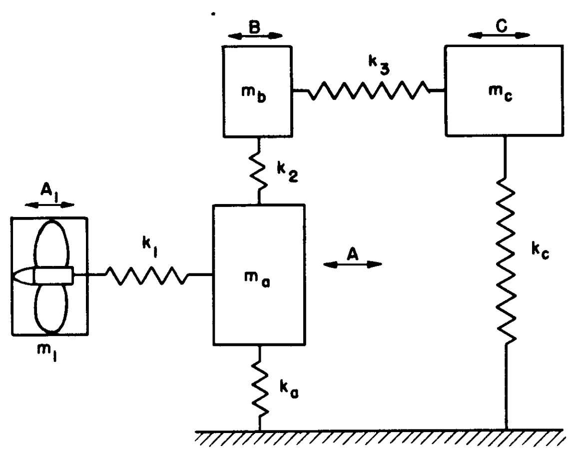

Figure 7 - Schematic Representation of a 4-Body Elastic System

These values when used in the equations listed give theoretical relative amplitudes as follows:

$$A_1 : A : B : C = 1 : 0.26 : 0.29 : 0.36$$

The value of the natural frequency of the lowest mode consistent with these quantities is 600 cycles per minute ($\omega = 62.8$). Strictly speaking, 1/2 the mass of the shaft should be added to the gear case as was done in the case of the propeller, in accordance with Reference (15), but in the calculation of the lowest mode where all the masses in the engine room move in phase this correction was found to make very little difference.

Two-Body Analysis

For the purpose of simplifying the analysis the various masses and spring constants in the engine system can be combined into a single effective mass and effective spring, but these effective values apply at only one frequency. These values can be obtained from energy considerations. The effective mass of the engine unit is equal to that of a mass which, if vibrating at the same amplitude as the main thrust bearing, would have the same kinetic energy as the whole engine system has when passing through midposition in its vibratory motion. The effective spring constant of the engine unit is the constant of a spring such that if the equivalent single spring had a deflection equal to the single amplitude at the thrust block the energy stored would be equal to the elastic energy actually stored by the whole engine system.

For the starboard inboard engine system the effective mass m was found from computations based on these relations to be 1190 pound-seconds squared per inch and the effective spring constant k to be 8.8×10^6 pounds per inch at 600 CPM. At other frequencies within the running range, the replacement of the actual system by the 2-body system shown in Figure 38 represents a reasonable approximation.

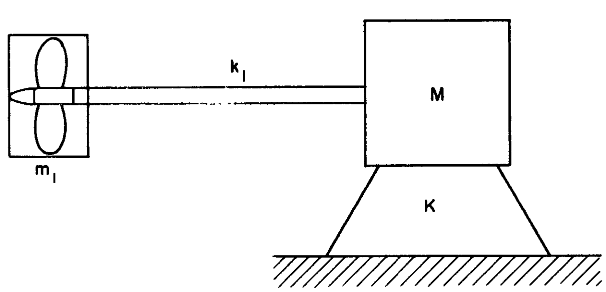

Figure 38 - Two-Body System Representing Propeller-Shaft-Engine System

In this system m_1 is the mass of the propeller, plus the virtual mass of the water, plus 1/2 the mass of the shaft; k_1 is the spring constant of the shaft; M is the effective mass of the entire engine, including 1/2 the mass of the shaft; K is the effective spring constant of all gear and turbine foundations.

ANALYSIS OF THRUST VARIATIONS AND WAKE

The elimination of propeller-excited vibrations on these vessels as well as on any other ships experiencing this difficulty would be expedited if the vibratory forces associated with propellers of various numbers and shapes of blades could be predicted in advance. The first attempt to do this for the NORTH CAROLINA and the WASHINGTON was based on wake measurements. If a propeller is rotating at constant speed in a stream whose velocity is not uniform, the resultant thrust and torque must vary throughout a revolution.

Relative velocities at various points within the propeller-disk area are known from pitot-tube measurements made on the model when it is being towed without the propellers; see Figures 5a and 5b on pages 10 and 11. The propeller thrust and torque for various speeds of advance are also known from model propeller experiments in open water. Combining these separate measurements should give an estimate of the thrust at each instant of the propeller cycle and hence of the alternating force on the propeller.

The results so derived did not agree well with the measurements made on the ships, probably due to inaccuracies in the assumptions made and to the following difficulties: The velocity near the midlength of a propeller blade affects the thrust more than the velocity near the root or very near the tip; also velocities near the edge of the blade have less effect than velocities near the blade axis. The proper method to be used in accounting for these effects is one requiring complete mathematical and practical investigation, for which neither time nor personnel have been available.

A second attempt to analyze the thrust variations developed by a multibladed propeller followed the methods used in the analogous and well-known case of the vibration of a multicylinder internal combustion engine. The sole specific result of this analysis was the information that the resultant vibration-exciting force contains only components of blade frequency and integral multiples of blade frequency. This investigation did help, however, to focus attention on the harmonic content of the vibration; the oscillograms of thrust in the shafts were eventually analyzed with a harmonic analyzer and the results thus obtained were most illuminating.

The vibrations experienced on the two battleships in question were predominantly of blade frequency but certain other periodic variations of thrust were observed, notably the following:

a. Changes occurring over a period much longer than the period of rotation of a propeller. These changes affected both the average thrust and the blade-frequency component. The period was rather indefinite but of the same order as the pitching and rolling periods of the ship, i.e., between 3 and 15 seconds; the changes are believed to be traceable to the action of the waves of the sea.

b. Vibrations of a frequency equal to the shaft frequency, caused by inequalities among the blades of the propeller, shaft misalignment, or other mechanical imperfections.

c. Thrust variations on an inboard shaft produced by a propeller on an outboard shaft. This effect was noticed on several occasions when favored by resonance; see Figure 33 on page 50. It was traceable because of the fact that the inboard propellers had a different number of blades than the outboard propellers on every test on the NORTH CAROLINA and the WASHINGTON. It was, of course, due to coupling between the shafts, either mechanically through the foundations and ship's hull or hydraulically through the water.

d. Apparent thrust and torque variations due to chatter in the water-lubricated shaft-tube bearing adjacent to the thrust-gage installation. These occurred only at shaft speeds below 50 RPM.

None of these features was particularly objectionable as far as causing vibrations of the ship's structure was concerned, but taken together they did produce enough complications to make the raw observed data appear nearly unintelligible. Accordingly, attention was concentrated on the blade-frequency component. The effect of the extremely low-frequency variations mentioned was minimized as much as possible by averaging the results of harmonic analyses for intervals of 8 to 12 revolutions, whenever this could be done.

It is an unfortunate situation that in this case the underlying cause of the difficulty, the variable exciting force developed by the propeller, cannot be directly measured at the propeller by any technique yet developed. It is possible only to measure the response made by a complicated mechanical system, the propulsion system of the vessel, when the force is applied. At frequencies sufficiently below the lowest natural frequency of the system, the force transmitted by the propeller to the thrust block is very nearly equal to the exciting force as the resonant magnification approaches unity. However, on the NORTH CAROLINA and WASHINGTON the alternating forces at very low speeds were so small that the accuracy with which they could be measured was very poor.

The only way that the exciting forces can now be inferred from the full-scale tests is by a curve-fitting process. For simplicity, the system is assumed to be a 2-body system. A formula* for the force in the spring equivalent to the propeller shaft is found to involve

a. the frequency and magnitude of the exciting forces,

b. the masses and spring constants of the system, and

c. two damping constants, one representing hydrodynamic friction on the propeller plus ordinary friction in the stern tube and steady bearings, and the other representing friction opposing the vibration of the system representing the engine.

Confining attention to the blade-frequency vibration, the frequency of the exciting force is the RPM times the number of propeller blades. In line with the ideas just explained, the magnitude of the exciting force is the ratio of the exciting force to the steady thrust, a value to be determined, multipled by the steady thrust. The steady thrust is known from experiments with self-propelled models; see Figure 39. The masses and spring constants are known approximately or can be determined. The two damping constants are not previously known but the formula* shows that they are important only in the immediate neighborhood of the two natural frequencies of the 2-body system.

The unknown constants must have a value such that shaft forces calculated from the formula* are found to agree with the measured forces within

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* The general equation, derived from Newton's law, for the 2-body system of Figure 38, when solved for the ratio of the force in the shaft to the original exciting force, can be expressed in the form,

$$\frac{F_s}{F_e} = \frac{\omega_1^2\left[\left(\omega_2^2-\omega^2\right)^2+\frac{C^2}{M^2}\,\omega^2\right]}{\sqrt{\begin{aligned}&\left[\left(\omega_1^2-\omega^2\right)\left(\omega_2^2-\omega^2\right)^2-\omega_3^2\left(\omega_2^2-\omega^2\right)\omega^2+\frac{c_1}{M_1}\,\frac{C}{M}\,\omega_3^2\,\omega^2+\frac{C^2}{M^2}\left(\omega_1^2-\omega^2\right)\omega^2\right]^2\\&+\left[\frac{c_1}{M_1}\left(\omega_2^2-\omega^2\right)\left(\omega_2^2+\omega_3^2-\omega^2\right)\omega+\frac{C^2}{M^2}\,\omega^3\left(\omega_3^2+\frac{c_1}{M_1}\,\frac{C}{M}\right)\right]^2\end{aligned}}}$$

where F_s is the peak value of alternating force in the shaft,

F_e is the peak value of the exciting force on the propeller,

ω is 2π times the frequency of the exciting force, i.e., the exciting force on the propeller, which is $F_e \sin\omega t$,

$F_s \sin(\omega t + \phi)$ is the force in the shaft, where ϕ is a phase angle which could be computed but is not important to the operation of the ship,

$\omega_1 = \sqrt{\frac{k_1}{M_1}}$,

$\omega_2 = \sqrt{\frac{K}{M}}$,

$\omega_3 = \sqrt{\frac{k_1}{M}}$,

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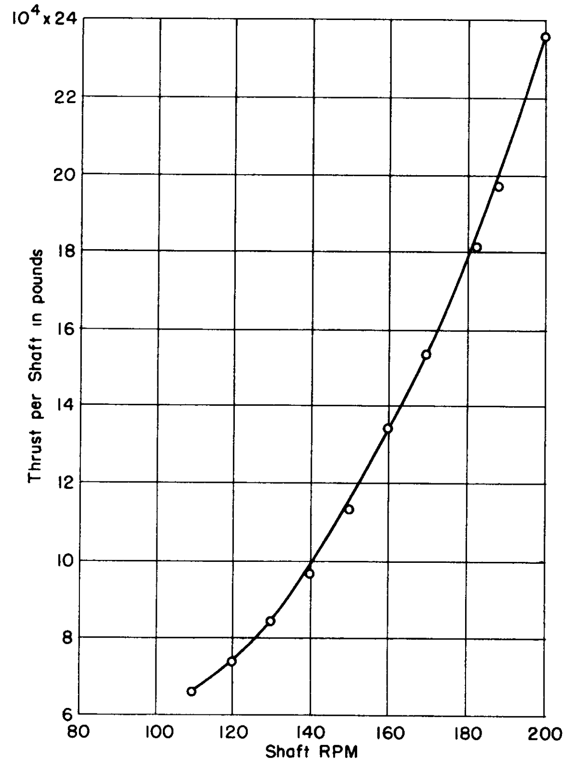

Figure 39 - USS NORTH CAROLINA, Average Thrust per Shaft Plotted on Shaft RPM

This curve, obtained from tests with 30-foot model, was used in determining alternating thrust as a percentage of steady thrust.

the limits of experimental error. The required values of these constants were found by a curve-fitting process involving repeated trials of assumed constants in a more or less systematic manner.

It was found early in the present investigation that large values must be assigned to both damping constants to explain the experimental results. The values of these damping constants represent additional information which could not be obtained otherwise. This information is important

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k_1 is the effective stiffness of propeller shaft in 2-body system of Figure 38,

M_1 is the effective mass of propeller in 2-body system of Figure 38,

K is the effective stiffness of engine foundation in 2-body system of Figure 38,

M is the effective mass of engine in 2-body system in Figure 38,

c_1 is the damping constant applicable to motion of m_1, and

C is the damping constant applicable to motion of M.

In the 2-body equivalent system adopted here, the force in the shaft is equal to the force on the thrust bearing. In the actual system, the alternating force transmitted varies slightly from point to point along the shaft and hence the force appearing in the shaft is not exactly equal to the force on the thrust bearing. This effect is rather slight and has been neglected.

because at resonance the restraining forces due to the stiffness of the foundations and to other elastic restraint are completely canceled by inertial reactions, so that the damping becomes the sole restraint on the system.

The damping associated with the part of the system representing the engine is peculiar to these vessels. In the construction employed, the underslung condensers participated in the vibration and it is probable that hydrodynamic damping in the condenser was responsible for a large fraction of this damping. Major changes in the design of the engine would be likely to change the constant by a large and unpredictable factor.

The other damping factor, however, is believed to be largely due to hydrodynamic damping of the propeller. If so, it should be fairly typical of a large number of propellers. Dimensional reasoning shows that the propeller damping should be proportional to the blade area, although the sharpness of the blade edges may affect its value to a considerable extent.

The agreement between the forces computed from the constants finally selected and the forces actually measured can be seen from Figures 40 to 43. In these figures the points were computed from oscillograms of thrust in the shaft and the curves from the formula. The values of exciting-force ratio and propeller damping used in these calculations are given in Tables 9 and 10.

TABLE 9

Vibration Constants of Propulsion Systems, USS NORTH CAROLINA and WASHINGTON

	Shaft			
	Starboard Outboard	Starboard Inboard	Port Inboard	Port Outboard
Resonance lowest mode, CPM (experimental)	520	600	680	600
Resonance second mode, CPM (experimental)	970	1020	1030	1090

TABLE 10

Vibration Constants of Propellers, USS NORTH CAROLINA and WASHINGTON

The system is assumed to be composed of two bodies, as in Figure 38 on page 62.

Location of Propeller	Number of Blades	Blade Frequency Exciting Force at Propeller as Per Cent of Steady Thrust from Full-Scale Tests	Propeller Damping pounds per inch per second
Inboard behind skeg	3 4 5	7.4 7.7 8.0	800 1600 1800
Outboard no skeg	3 4	2.4 2.1	600 1600

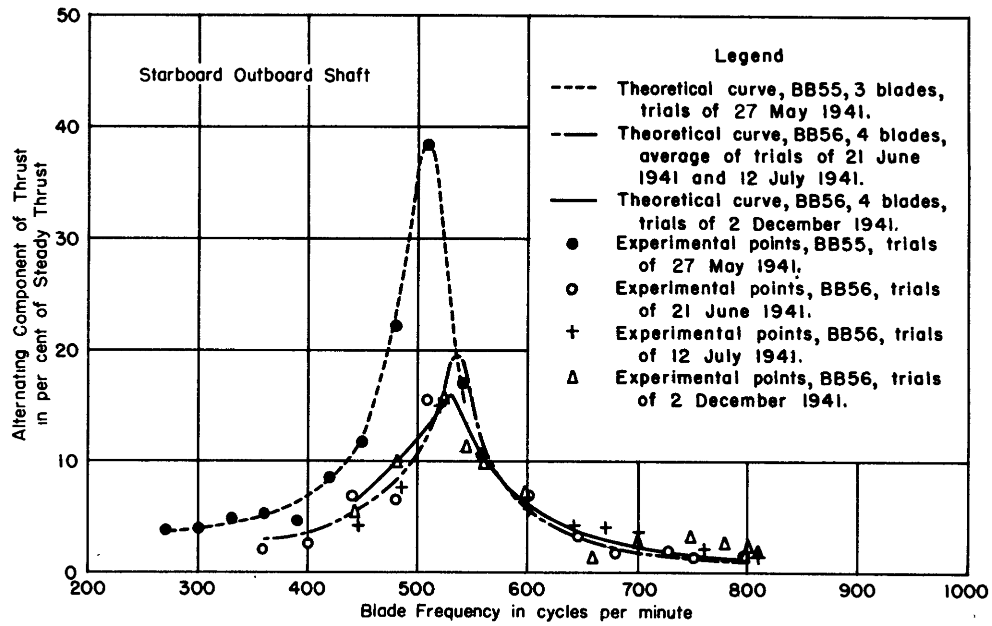

Figure 40 - USS NORTH CAROLINA and USS WASHINGTON, Curves of Thrust Variation at the Thrust Block, Showing High Resonance Peaks

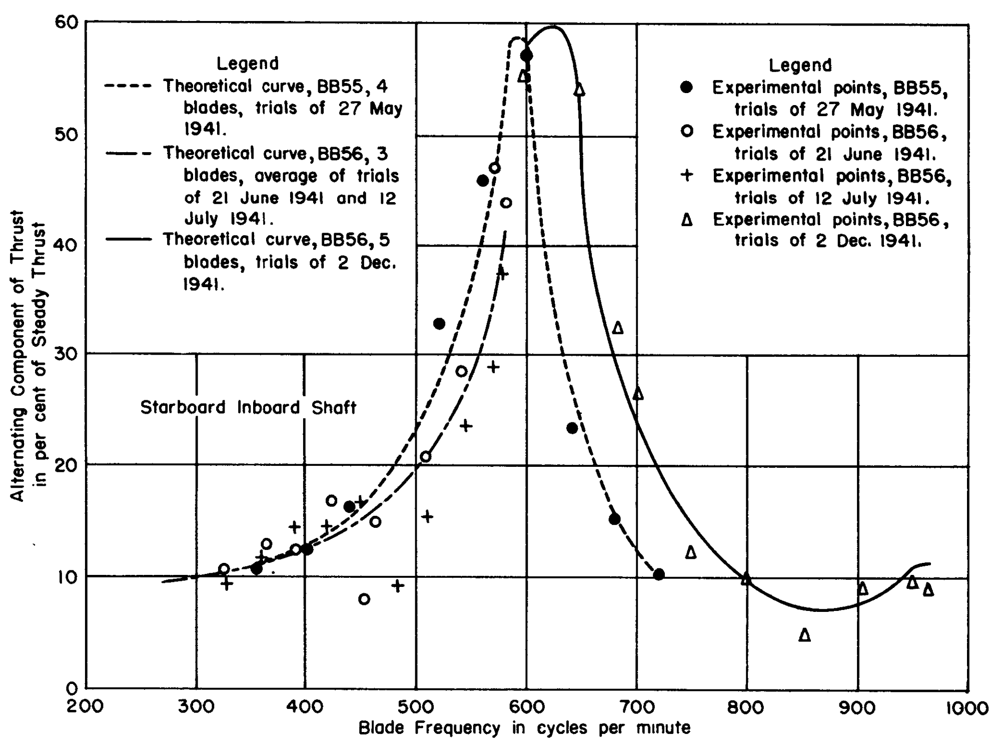

Figure 41 - USS NORTH CAROLINA and USS WASHINGTON, Curves of Thrust Variation at the Thrust Block, Showing High Resonance Peaks

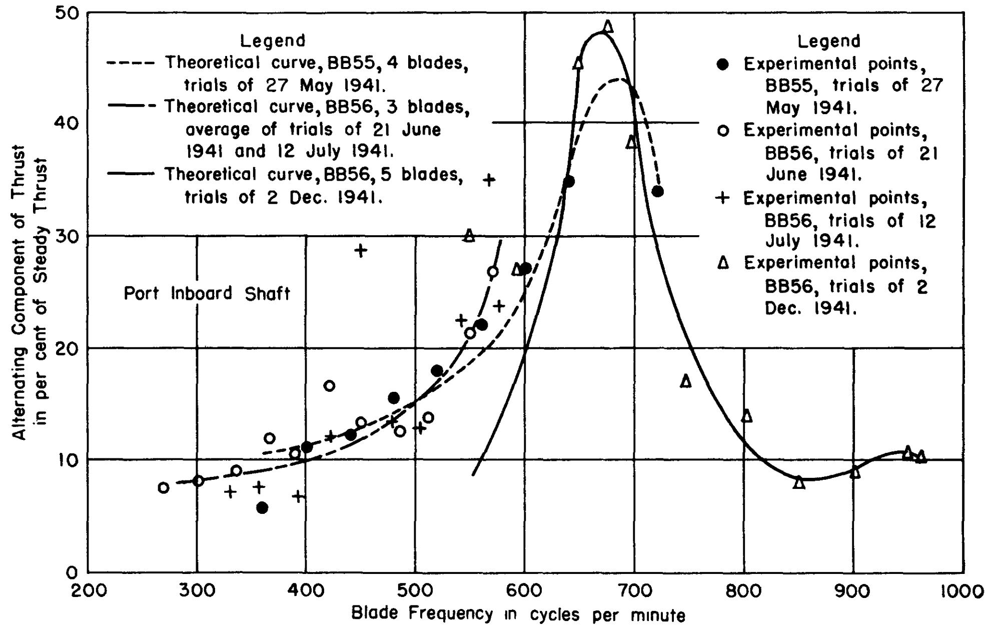

Figure 42 - USS NORTH CAROLINA and USS WASHINGTON, Curves of Thrust Variation at the Thrust Block, Showing High Resonance Peaks

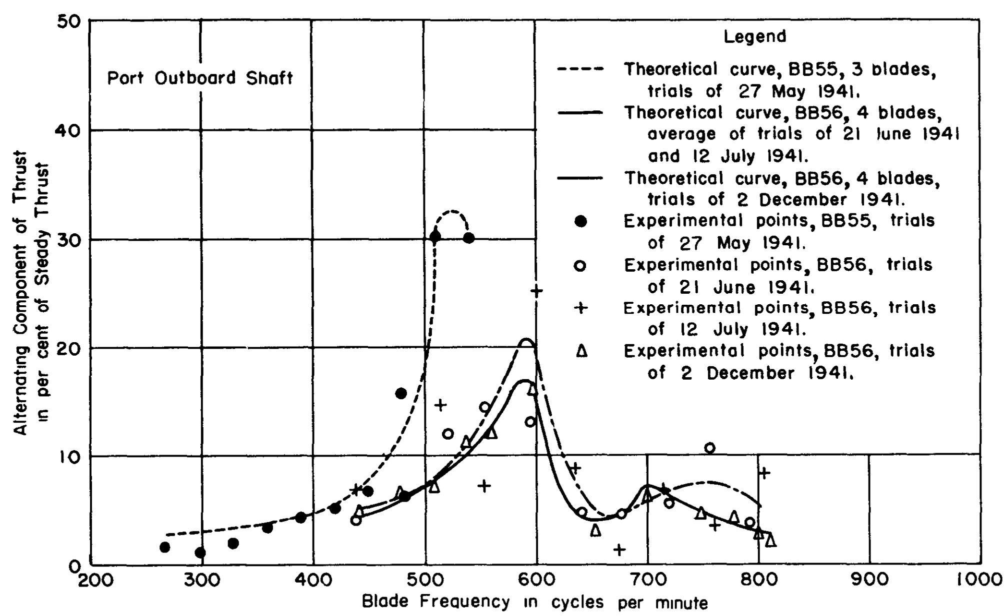

Figure 43 - USS NORTH CAROLINA and USS WASHINGTON, Curves of Thrust Variation at the Thrust Block, Showing High Resonance Peaks

The longitudinal natural frequency of the propulsion system used on the model did not follow the same law of comparison as the usual quantities involved in model testing. The natural frequency of the model propulsion system was partially determined by the design of the thrust pickup, but as an attempt to design for frequencies corresponding to the full-scale frequencies would have led to great difficulties, the thrust pickups were designed to give the highest practicable natural frequency so that the blade-frequency force measured would be equal to the blade-frequency exciting force to a sufficient degree of approximation. This object was attained, which simplified the computations and made it possible to take the data at or near top speed where the forces were largest and the accuracy highest. The natural frequency attained, however, was about twice the blade frequency corresponding to full-speed operation. This accentuated some of the harmonics of the blade-frequency force to such a degree that it was necessary to determine the magnitude of the blade-frequency force by harmonic analysis before using the results. Model and full-scale data are compared in Table 11.

TABLE 11

Comparison of Alternating Component of Propeller-Exciting Forces Measured during Model and Full-Scale Tests, USS NORTH CAROLINA and WASHINGTON

Number of Propeller Blades	Location of Propeller	Vibration-Exciting Force at Propeller as Per Cent of Steady Thrust		
		Model Test		Full-Scale Test from Table 10
		Preliminary Tests**	Later Tests*	
4	outboard		2.8	2.1
3	inboard		5.4	7.4
4	inboard	12.3	9.0	7.7
5	inboard	7.4	6.3	8.0

* Tests from which these values were determined were made on 26 August 1942.

** Tests from which these values were determined were made on 12 November 1941 and are considered inferior to those of 26 August 1942 in reliability.

The model results differ from the full-scale results by less than 2 per cent of the steady thrust except in case of the model test of 12 November 1941; this showed a large discrepancy which must be attributed to some error peculiar to the earliest experiments. The model results for 3- and 5-bladed propellers (odd number of blades) were consistently low, whereas the model results for 4-bladed propellers (even number of blades) were consistently high. This suggests that the wake and flow around the model was not accurately similar to that around the ship.

DISCUSSION OF REMEDIES PROPOSED AND TRIED

VARIATIONS IN STIFFNESS OF ENGINE FOUNDATIONS AND THRUST-BEARING FOUNDATIONS

From the 2-body analysis previously mentioned on page 62 it is obvious that a change in the resonant frequencies could have been effected by changing the stiffness of the engine and thrust-bearing foundations; in fact, as shown in Table 3 on page 37, numerous changes were actually made in these foundations on the WASHINGTON. In view of the massiveness of the structure, however, and the fact that the frequencies vary only as the square root of k, the amount of reinforcement required to make an appreciable increase in the frequency would have been enormous unless the condensers could have been relocated.

Weakening of the foundations in an attempt to lower the resonant frequencies would theoretically have been another possible solution, but the excessive deflections resulting would undoubtedly have caused trouble from bearing and shaft misalignment. Experiments were not made in this direction on the foundations proper, although the spacers were removed between the gear case and the turbine foundations without noticeable effect on the fundamental frequency.

REDUCTION OF THRUST-BEARING CLEARANCE

Reduction of the fore-and-aft clearance in the main thrust bearings tried on the WASHINGTON during the trials of 12 July 1941 did not produce any measurable effect. As the alternating component of thrust was always less than the steady thrust, and as the thrust never actually reversed, the forward surface of the thrust collar was always against the bearing segments and hence no effect was to be expected.

RELOCATION OF THRUST BEARINGS AFT

It was considered a practical impossibility, because of delays and shortages in labor and critical materials, to relocate the thrust bearings of these vessels, as each ship was substantially complete when the vibration was discovered. The following brief discussion of this expedient is included for the sake of completeness and because it has been applied in several subsequent designs.

The problem of thrust-bearing location was studied theoretically in a quite general manner by a somewhat idealized representation of a ship's propulsion system in which the fore-and-aft location of the thrust bearing is regarded as a design parameter to be selected to suit the case. In practical propulsion systems the length of shaft from the propeller to the reduction gear, the cross section of the shaft, the mass of the reduction gear, thrust block and propeller, and the practicable stiffness of the thrust-block foundation, are all more or less fixed by considerations which have little to do with the vibration problem. If these quantities are assumed to be given, it is found that there is for each design a location of the thrust block that makes the frequency of the fundamental mode of axial vibration a maximum. Figure 44 shows the various components which must be taken into consideration in a study of this kind.

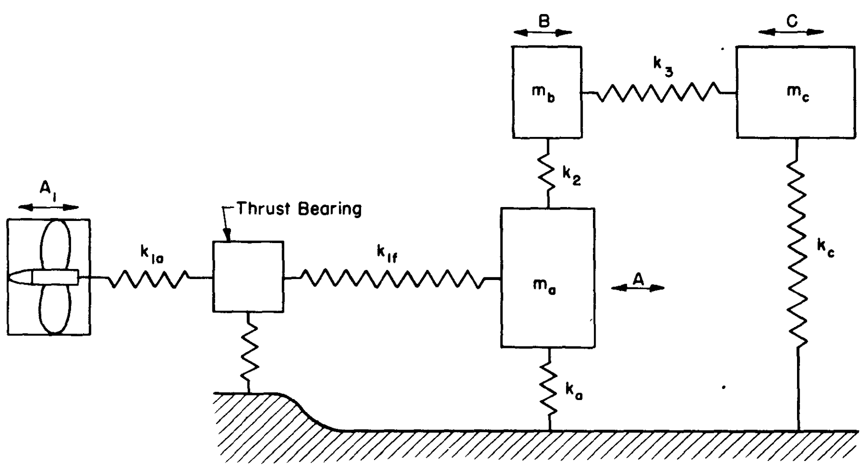

Figure 44 - Schematic Representation of the 4-Body Elastic System of Figure 7, with a Thrust Bearing in the Line Shaft

In some of the designs studied in this way it was found that by using a 3-bladed propeller and making the resonant frequency of the lowest mode as high as possible, the critical shaft RPM would be well above the designed full-power RPM. This would usually be a much better solution than having the resonant frequency or critical shaft RPM within the running range.

Usually the main thrust bearing can be located at or near the position required for maximum frequency, as determined from a study of the elastic characteristics of the system. Where this can be done, and where the thrust bearing and foundation are made stiff enough, it may be a satisfactory solution of the problem of axial shaft vibration. As noted previously, the presence of thrust variation in itself seems not to be a problem except when this variation is greatly amplified by resonance. Proper placing of the main thrust bearing will often avoid resonance completely.

In this connection it may be well to point out here a distinct improvement which can be made in the design and installation of main thrust bearings in high-powered vessels, especially where the shafts are long and it is mandatory that the thrust bearing restrain the shaft from longitudinal movement at that point. Figure 45 shows a proposed type of foundation with a long base length, to prevent tilting and deformation, and a barrel-type thrust-bearing casing, designed to transmit the thrust load to the foundation in line with the center of the shaft, instead of a considerable distance below it, as is the case with existing installations.

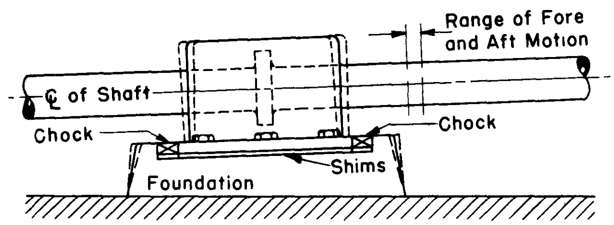

Figure 45a - Orthodox Thrust-Bearing Foundation with Short Base Length and Pedestal-Type Casing

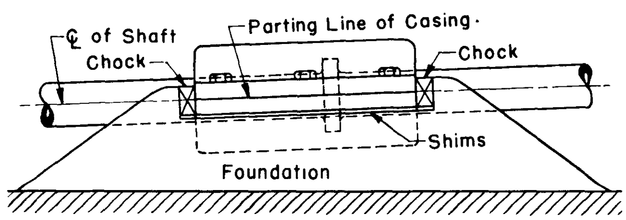

Figure 45b - Recommended Thrust-Bearing Foundation with Long Base Length and Barrel-Type Casing

Figure 45 - Types of Propeller-Shaft Thrust Bearings

The broken lines in Figure 45a are intended to represent, in exaggerated fashion, the manner in which an installation of this kind can deform under the influence of variable thrust forces.

RESTRAINING BLOCK

When serious shaft vibration was first observed on the NORTH CAROLINA, one of the remedies suggested was the installation of a shaft-restraining block as far aft as possible in the shaft alley. The function of such a device is to introduce a restraint into the system, reducing the axial vibration by increasing the damping, while at the same time the steady component of thrust is still being taken by the thrust block in the engine room. The results of extensive model experiments and the complete theory of this device are given in Reference (10). It was found that a large reduction in the amplitude of longitudinal vibration could be obtained with this apparatus. At best, however, it is only a makeshift for relocating the thrust bearing near the propeller.

Lack of full-scale experience with any similar equipment led to considerable uncertainty about the maintenance and vulnerability of the restraining block, but a full-scale installation was actually under manufacture for the NORTH CAROLINA when it was found that the difficulties could be eliminated by other means.

PROPELLER CHANGES

It has already been noted that resonance may be shifted away from an undesirable speed by changing the number of propeller blades. It has also been noted that propeller changes have produced more improvement than any other change actually tried on the NORTH CAROLINA and WASHINGTON. Such an expedient is readily acceptable if it involves the use of 3- and 4-bladed propellers only. The use of propellers having more than 4 blades is quite unusual, however, so that in suggesting the fitting of 5-bladed propellers as a means for curing a vibration difficulty, the effect of the proposed change on the propulsive efficiency had to be given special consideration.

Five-Bladed Propellers for Inboard Shafts

Multibladed propellers have always been attractive to a designer interested in reducing the magnitude of the individual blade impulses and hence the exciting forces. It is obvious that if the same thrust is apportioned among 5 blades instead of among 3 or 4, the individual blade impulses will be proportionately smaller if all other conditions remain the same, especially if the propeller absorbs a given or fixed amount of power.

Changing the number of propeller blades is an old trick for eliminating vibration troubles. It was proposed by Taylor in his second 1933 edition of "The Speed and Power of Ships" (16), in which the following statement appears on page 99:

> "Five- or six-bladed propellers are not unknown, but their use would seem justified only if both three-bladed and four-bladed (propellers) caused excessive vibration."

This was based upon a statement of Taylor just preceding the foregoing quotation, in which he stated that:

> "For actual ships, two-bladed propellers, even if slightly more efficient under the conditions, are barred by reason of their great tendency to cause vibration."

The use of propellers having more than 4 blades was undoubtedly discouraged in the past by the low efficiency ascribed to multibladed propellers. This was in turn laid to the greatly increased area whereas if there was any unavoidable loss in efficiency, it must have been due rather to blade interference.

In the past it was not uncommon* to make a comparison between propellers of 2, 3, and 4 blades on the basis of equal mean width ratios and identical blades, regardless of the number. On such a basis it is obvious that the multibladed propeller suffers considerably because the increased total area of the blades results in lowered efficiency. Actually, multibladed propellers designed for the same conditions would have narrower blades and usually there would be only a small increase in total blade area.

There are still left unanswered, however, major objections to the multibladed propeller, in the form of blade interference and increased tip losses. Blade interference is a function of blade-section shape and blade shape and spacing, as well as of pitch ratio.

Much of the interference on the older propellers which employed ogival sections and constant pitch, was caused by the sharply varying pressure distribution present over the blade sections. The present use in propeller design of hydrofoil blade sections with "lifted" or "washed back" leading edges near the hub has greatly ameliorated these conditions, so that a much more uniform pressure distribution is now present in the average propeller. This decreases the blade interference to such an extent that the 5-bladed propeller performs comparably with a 3- or 4-bladed one in respect to efficiency. Figure 46 shows the blade sections and other features of one of the most recent 5-bladed propellers designed in 1943 and 1944 by the Bureau of Ships for the NORTH CAROLINA class, and Figure 47 is a photograph of one of the completed propellers of the first (1941) design.

While the use of fewer blades undoubtedly involves less waste of power from tip losses, this effect is probably offset in the case of more blades by the more uniform circulation through the propeller disk.

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* See page 99, Section 127, of Reference (16).

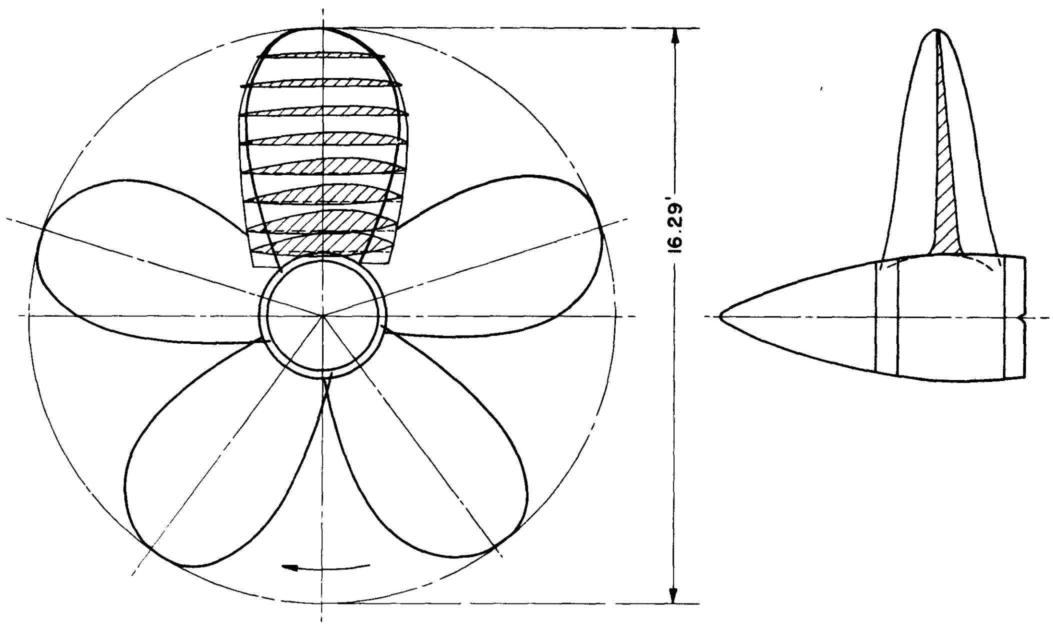

Figure 46 - General Characteristics of a 5-Bladed Propeller Designed for USS NORTH CAROLINA Class

This illustrates the final design of inboard propellers for this class.

The conclusion to be drawn from this theoretical discussion is that there is little if any loss in efficiency to be expected in going from 4 to 5 blades, so that the 5-bladed propeller can be adopted if it lessens the vibration amplitudes. Model tests with 5-bladed propellers designed for BB55 and BB56 showed an efficiency slightly better than that of the 3- and 4-bladed propellers previously designed, as indicated in Figure 48. No full-scale standardization trials have been made to verify these results but service at sea indicates approximately the same speed with 5-bladed as with 3-bladed propellers on the inboard shaft. There was no attempt made to attain full power with the original 4-bladed propellers on the inboard shafts because of the severity of the vibration.

The 5-bladed propeller was expected to reduce the vibration amplitude in two ways. The critical shaft speed with 5 blades is only 80 per cent of the critical shaft speed with 4 blades. It has been shown on page 18 that the exciting force at the propeller is proportional to the steady thrust. Hence the exciting force varies approximately as the square of the shaft RPM, so that the vibratory force at the critical shaft speed with a 5-bladed propeller would be only 64 per cent of the value with a 4-bladed wheel if the exciting force as a percentage of the steady thrust were independent of the

Figure 47 - View Showing Completed Inboard Propeller for USS NORTH CAROLINA Class, Manufactured by Navy Yard, Philadelphia

This photograph shows the forward side of a right-hand propeller, one of those made for the 1941 trials on this class.

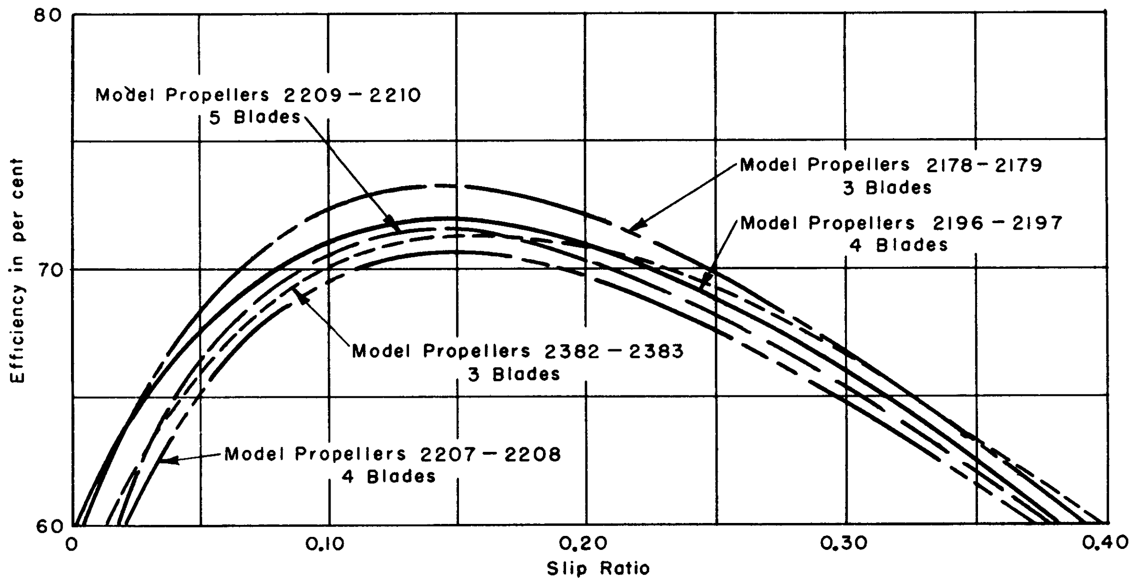

Figure 48 - Curves Showing Relative Maximum Efficiencies for Various Propeller Designs for the NORTH CAROLINA Class

These curves are plotted from the results of open-water propeller tests of the model propellers listed by serial numbers on the diagram.

number of blades. Any reduction in the vibratory-exciting force due to increasing the number of blades would produce a further reduction in vibration.

As has been noted, it would appear that distributing the thrust over more blades would result in smaller vibration-producing impulses per blade. There are, it is true, more blades to consider, but there is a greater opportunity for fluctuations to average out in the resultant. On the other hand, some of this advantage must be lost on account of the narrower blades required to maintain the efficiency. Considering the problem from the viewpoint of Fourier analysis, it is evident that the natural tendency of the Fourier coefficients to decrease with increasing order is favorable and that a skeg which approaches symmetry with respect to any plane through the propeller axis should give exciting forces which are smaller for an odd number than for an even number of blades. In short it seemed very probable that there would be a reduction in the percentage of exciting force if 4-bladed propellers were replaced by 5-bladed propellers on this class of battleships.

It was further expected that the 5-bladed propeller would have a higher damping constant than the corresponding 4-bladed propeller, which would in turn reduce the multiplication of force, between the propeller and the main thrust bearing, in the neighborhood of resonance. It has been found by observation of "singing" propellers made by the Newport News Engineering, Technical, and Hydraulic Laboratory staff over a period of years, during trial trips and on model turbine-runner tests, that the most probable cause of singing is blade vibration excited by the pulsations of the trail of vortices shed by a blunt trailing edge (17). If, as appears likely, sharpening the edge to reduce or eliminate singing produces a significant increase in damping, then since the total length of blade edge is greater in the 5-bladed propeller, the damping should become greater also.

Thrust analysis of the ship-trial data showed that the vibration-exciting forces and the damping constants were both slightly higher for the first 5-bladed than for the 4-bladed propeller. At resonance these effects operated to counteract each other and the variable thrust forces were only slightly less than those predicted as due solely to the change in the frequency of the exciting force, i.e., the maximum measured alternating force with 5-bladed propellers was 53 per cent of what it was with 4-bladed propellers (average of all determinations). If the difference had been entirely due to the change in frequency this ratio would have been 64 per cent as previously noted. The vibration amplitudes found when using the redesigned 5-bladed propellers were less than 50 per cent of the vibration amplitudes when using the first 5-bladed propellers. This was due to better power distribution when using the new design, to increased tip clearances produced by the reduced diameter of the new wheels, and to the skew effect produced by cutting back the leading edges of the blades.

When the first 5-bladed propellers were fitted on the inboard shafts, the fact that an exciting force having a frequency of 1000 CPM was present at the maximum speed of 200 RPM caused a number of engine-room auxiliaries to vibrate that would have remained relatively quiet under 600-CPM excitation when making full-power revolutions with 3-bladed propellers on the inboard shafts. Furthermore, units with a resonant frequency of, say, 660 CPM, which would be quiet throughout the speed range with 3-bladed propellers driving, would vibrate in the cruising range at a shaft speed of 132 RPM with 5-bladed propellers mounted.

For this reason, neither the original 5-bladed propellers nor any 3-bladed propellers appeared to be entirely satisfactory. The vibration of the engines and equipment produced by the first 5-bladed propellers was a serious disadvantage at cruising speed. The vibration previously experienced with the modified 3-bladed propellers inboard, while not satisfactory, was more satisfactory at cruising speed. Accordingly, the ship operated with the modified 3-bladed propellers inboard and the newly designed 4-bladed propellers outboard from January 1942 to April 1944.

The final run with the new design of 5-bladed propellers inboard and 4-bladed propellers outboard on the WASHINGTON proved so satisfactory that it has been adopted as the final solution for this class.

NEUTRALIZERS AND PENDULUM DAMPERS

The theory of neutralizers is thoroughly discussed by den Hartog in his book "Mechanical Vibrations" (8). Simply stated, the neutralizer consists of a mass attached to the vibrating structure by an elastic support, such as a spring. If the natural frequency that the mass and spring combination would have if attached to a rigid support is the same as the frequency of the exciting force acting on the structure, then the attached mass will take up the vibration and set up a reaction that balances the external force so that the structure does not vibrate. A diagram of such a device is shown in Figure 49.

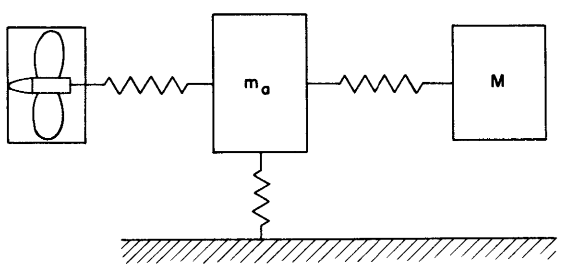

Figure 49 - Diagram of a Spring-and-Mass Type of Vibration Neutralizer Applied to a Propeller-Shaft Thrust-Block System

The practical application of a piece of apparatus of this kind, however, is not a simple problem. The neutralizer, while it eliminates vibration at the original resonant frequency, introduces other resonances at which the vibration may be just as objectionable. Hence use of the device is restricted to constant frequency conditions unless it is provided with variable tuning or with auxiliary equipment for arresting the element at the other critical speeds thus introduced. It is also obvious that the design of the springs must be safe against rather severe fatigue conditions.

A form of vibration neutralizer with automatic tuning is the so-called pendulum damper. If a pendulum and its support rotate as a unit about an axis in such a way that the plane of oscillation of the pendulum also rotates about the same axis, the pendulum may be considered as a gravity pendulum with the gravitational field replaced by a centrifugal field; or, in other words, with the gravitational acceleration g replaced by an acceleration $r\omega^2$, where r is the distance from the axis of rotation to the center of mass of the pendulum, and ω is the angular velocity of rotation.

Since the natural frequency of the gravity pendulum varies as $\sqrt{g}$ the natural frequency of the centrifugal pendulum varies as $\sqrt{r\omega^2}$ or directly as ω. From this it follows that if the angular velocity of the pendulum system, i.e., the angular velocity of the shaft, is proportional to the frequency of the disturbing vibration, the pendulum, if tuned to the exciting frequency at any one speed, will remain tuned to it at all speeds.

A possible application of this principle to the shaft-vibration problem would be the use of a pair of heavy weights suspended so as to be free to oscillate in a plane parallel to the shaft axis and rotating with the shaft; see Figure 50. Substituting $r\omega^2$ for g in the formula for the frequency of a pendulum we get

$$n = \frac{1}{2\pi}\sqrt{\frac{r\omega^2}{r'}} = \frac{\omega}{2\pi}\sqrt{\frac{r}{r'}}$$

where ω is the angular velocity of the shaft,

r is the distance of the center of mass of the pendulum from the axis, and

r' is its distance from the point of suspension.

It is clear then that if it is desired to tune the pendulum to the blade frequency, it is necessary only to make r/r' equal to the square of the number of blades. To maintain balance of the shaft it would be necessary to use elements of this type in pairs, as shown schematically in Figure 50.

As a practical consideration the angle of oscillation must be kept small. The larger the masses, the smaller will be the required angle. If a is the linear amplitude of the attached masses and m the mass of either, then for 100 per cent neutralizing action $2ma\omega_1^2$ must equal the exciting force at the propeller, where in this case ω_1 is the circular frequency of the exciting force, or 2π times the blade frequency. For example, to neutralize blade frequency thrust variations of 20,000 pounds single amplitude at 600 CPM (three-bladed propellers) with two 1-ton weights would require a single amplitude at the weights in the axial direction of the shaft of about 1/4 inch. Practically, at such low shaft speeds, the

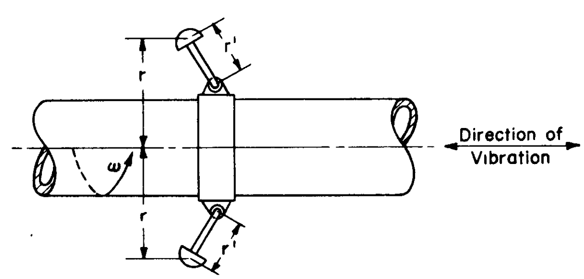

Figure 50 - Schematic Diagram of a Pendulum Damper on a Rotating Shaft

gravitational field would not be negligible in proportion to the centrifugal field and a spring support would be necessary to prevent the weights from pounding when starting up.

FLEXIBLE SECTION IN SHAFT

From the 2-body analysis, it was found that both natural frequencies of the system would be lowered by lowering k, the propeller-shaft constant. Obviously if the resonant frequency could be lowered from 600 CPM to, say, 400 CPM, resonance would occur at such a low shaft speed that there would be very little alternating thrust excitation from the propellers. Hence the vibration trouble would be remedied, provided that the change did not bring the second mode of vibration within the operating range of blade frequencies. Since the shaft must continue to transmit torque and thrust, weakening of a section means a large deflection due to the steady thrust, as well as a lowering of the torsional natural frequencies.

The shaft could be weakened axially without weakening it too much torsionally, by introducing a bellows-like section of larger diameter than

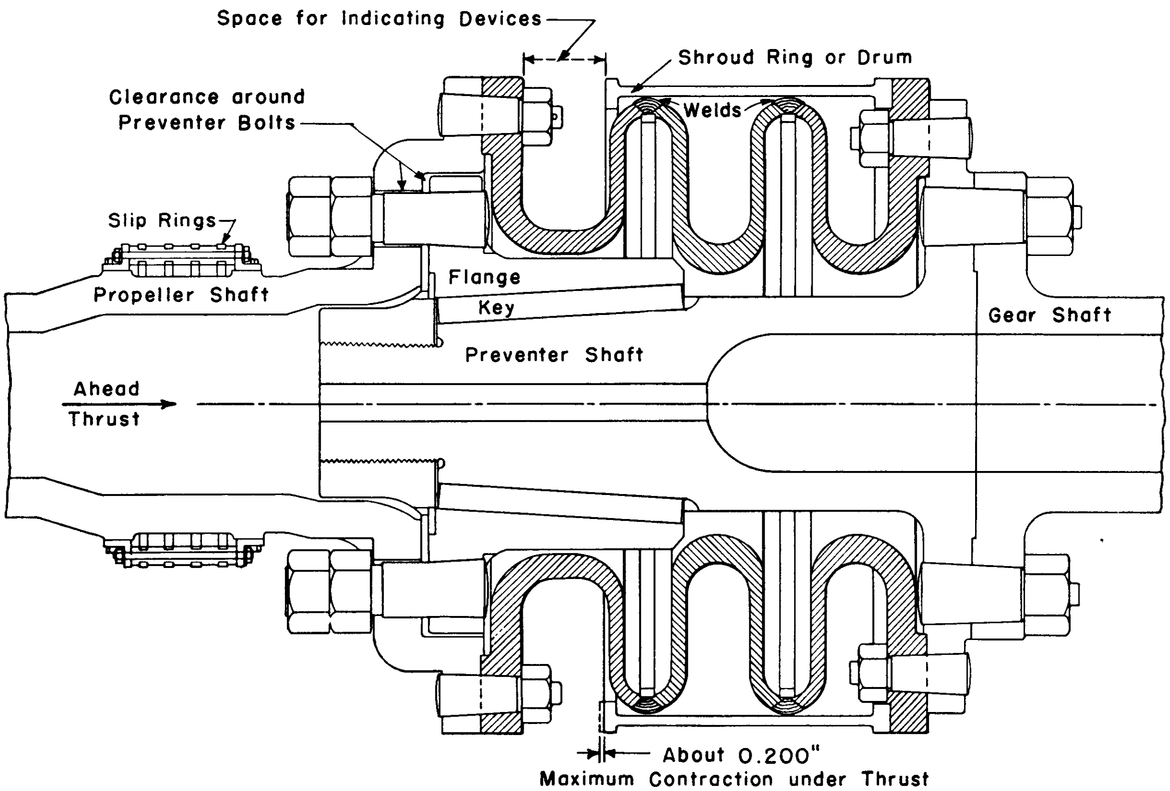

Figure 51 - One Type of Elastic Shaft Section to Increase Axial Flexibility

This design was prepared as a bellows-type thrustmeter but it serves to illustrate the details of one type of elastic shaft section.

the shaft. This would be somewhat similar to a type of thrustmeter at one time under development at the U.S. Experimental Model Basin (9) (18), illustrated in Figure 51. Before such a scheme is tried a careful analysis of the stresses in the flexible section would have to be made to insure against fatigue failures.

MODIFICATION OF SKEGS

Several expedients for modifying and improving the skegs are quite obvious. These may be classified as follows:

Modification 1. While preserving the general shape of the transverse sections of the skeg, narrow the after end, maintaining the same transverse rigidity. This might or might not be accompanied by a general shortening of the skeg, in an effort to reduce the variations in the wake and their effect on the propeller. To accomplish this modification on the NORTH CAROLINA and the WASHINGTON would necessitate new skeg castings and a rather expensive rebuilding of the stern. Its effects would be broadly similar to the effect of moving the propeller aft, as discussed on page 85.

Modification 2. While preserving the dimensions of the upper part of the skeg, reduce the dimensions of the portion below the shaft. It can be shown that this results in an asymmetrical distribution of velocity in the wake which, however, might cause the thrust variations to be less for some particular number of blades.

Modification 3. While maintaining the upper portion of the skeg intact, increase the dimensions of the portion below the shaft. It can be shown that if the wake in the upper and lower halves of the propeller disk are such as to produce symmetry about any plane through the shaft axis a propeller with an odd number of blades will produce a small blade-frequency component of exciting force.

Modification 4. Add two vanes similar to contra-propeller vanes, one on either side of the after edge of the skeg and directed downwards at a slight angle from the vertical. These vanes would be formed so as to deflect the water flowing aft in such a way that its sternward velocity would be partially transformed into a rotation contrary to the rotation of the propeller. The decrease in relative axial velocity and the increase in relative rotational velocity would both act to produce an increase in thrust during the interval when a blade would be near its lowest position, i.e., at the time when the production of a blade-frequency force by a 3- or 5-bladed propeller requires that the thrust be a minimum. As a contra-propeller generally produces a slight increase in propulsive coefficient, the arrangement cannot be objected to on the grounds of efficiency. The rudder action of the vanes on one skeg would be balanced by the oppositely directed rudder action of the vanes on the other skeg.

A combination of Modification 1 and 2 was tested on the 30-foot model of the NORTH CAROLINA class. Thrust variations due to 3-, 4-, and 5-bladed propellers were carefully measured by the methods described on pages 25, 26, and 33 of this report. The lower portions of the skegs of the model were then cut away to shorten and fine the skegs, as shown in Figure 52. As the modification chiefly affected the lower portion of the skeg, it produced a certain amount of asymmetry with respect to the horizontal plane in the distribution of flow. As more deadwood was cut from the outboard than from the inboard side of the skeg, the modification was equivalent to moving the center of the skeg away from the shaft center. This has particularly favorable effects, as shown in the discussion on page 85. The modification was carefully restricted to an amount of work that could reasonably be undertaken on the actual ships if they were ever docked for extensive hull repairs.

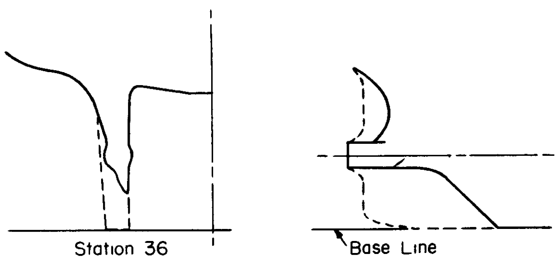

Figure 52 - Diagram of Proposed Skeg Modification for BB55 and BB56

Thrust variation was measured both before and after modification. Table 12 shows the results obtained. The modifications made conditions worse when a 3-bladed propeller was used and better when a 4-bladed propeller was used. This was in accordance with predictions based on the theory. The modifications also improved conditions with 5-bladed propellers.

The effect of these changes is disappointing. The observed changes are no larger than the average discrepancy between model and full-scale results. It is probable that the lack of improvement resulted from the fact that it was impossible to reduce the width of the upper portion of the skeg sufficiently. As mentioned previously in this report, there are strong indications that the dead water abaft the upper portion of the skeg is the major source of difficulty. Therefore, cutting away the lower portion of the skeg without changing the upper portion may have actually increased the wake variation enough to outweigh any beneficial effects from the effective displacement of the axis of the skeg.

While the modifications might produce greater improvement on the actual ship than the model results predicted, it is highly improbable that

TABLE 12

Effect of Skeg Modifications
Model Tests of 26 August and 20 October 1943

Four-bladed propellers were used on outboard shafts in all tests.

Exciting Force as Per Cent of Steady Thrust		
Number of Propeller Blades on Inboard Shafts	Original Skeg	Modified Skeg
3	5.4	6.6
4	9.0	8.0
5	6.3	5.6

this modification would produce sufficient improvement to warrant the change, particularly in view of the fact that the modification increased the shaft horsepower necessary to drive the model, and hence the ship, at any given speed, and it considerably reduced the bearing area of the skeg when used as a docking keel.

CHANGING POSITION OR BLADE OUTLINE OF PROPELLER

The most obvious way of changing the position of the propeller is simply to move it aft along the line of the propeller-shaft axis. This probably would have little effect in reducing the exciting force unless the propeller were moved an impracticably large distance, as with a well streamlined skeg the wake pattern changes rather slowly in proceeding aft from the end of the skeg.

There is, however, good reason to suppose that offsetting the center of the propeller from the center of the skeg will produce an appreciable decrease in thrust variation. The region of high wake which is produced by a skeg is chiefly concentrated around a vertical plane through the axis of the skeg. If the center of the propeller coincides with the skeg axis, this plane lies along a diameter of the propeller disk, and all parts of a propeller blade enter the high-wake region at once, which produces a rather sudden change in thrust. If, however, the center of the propeller is placed to one side of the skeg axis the tip of the blade enters the high-wake region while the rest of the blade still is in a region of more normal flow. If the offset is sufficient the tip will pass completely out of the high-wake region at the instant when the center of the blade is in the region of maximum wake, so that there will be a tendency to average out the thrust variation to which each blade is subjected. Further, the blade tips will not enter the region in which the wake is the highest of all, i.e., the region near the top of the skeg.

Obviously a change in the shaft lines of a large vessel already built is almost out of the question. However, it was possible to produce the same result, at least to a certain extent, by a change in propellers. It can be readily seen that a skewed propeller blade produces results very similar to those discussed for the offset propeller center. The amount of skew which is employed is of course limited by strength and efficiency considerations. The design of the 5-bladed propellers finally adopted embodied a moderate amount of skew.

CONCLUSIONS AND RECOMMENDATIONS

CONCLUSIONS SPECIFICALLY APPLICABLE TO THE BATTLESHIPS NORTH CAROLINA AND WASHINGTON

These vessels, as originally designed and constructed, were subject to longitudinal vibration of the propelling machinery so severe as to render it inadvisable to run them up to full power and speed.

Local stiffening of steam lines and other items overcame some of the most objectionable features of the vibration but the vessels remained unsatisfactory as regards vibration in general. Investigations showed that no practicable amount of stiffening of the machinery foundations could produce any appreciable improvement.

Replacement of the original 3-bladed outboard propellers with 4-bladed propellers resulted in operating conditions which were satisfactory as far as the outboard shafts and associated machinery were concerned.

Replacement of the original 4-bladed inboard propellers with 5-bladed propellers produced satisfactory conditions in the major machinery units. However, this caused troublesome vibrations in the same auxiliary machinery and in certain fire-control instruments at speeds between 17 and 20 knots.

Installation of 3-bladed propellers of reduced diameter on the inboard shafts resulted in tolerable conditions in both the engine rooms and the superstructure at all speeds. These propellers, in combination with 4-bladed propellers of new design on the outboard shafts, made it possible for the vessels to fulfill their missions in service, but the vibration conditions, as judged by modern standards, were not yet acceptable.

Installation of 5-bladed propellers of a later design on the inboard shafts reduced vibration throughout these vessels to acceptable amplitudes.

The relocation of the main thrust bearings on the inboard shafts would be a major alteration and is not now recommended for these vessels.

Modification of the skeg endings to increase the propeller clearance is not recommended unless modification or replacement of the large castings is necessary for other reasons.

RECOMMENDATIONS FOR FUTURE CONSTRUCTION

It is recommended that on future ships, especially those with skegs, the main thrust bearings be located as near the propellers as possible. It is further recommended that a barrel type of thrust-bearing housing be developed and adopted, with chocks at the level of the centerline of the shaft, and with a length of foundation at least five times its height.

The clearances between the propeller and the hull and the propeller and the skeg should be made as large as practicable. This has been the aim of ship designers for many years past.

Coupling between the propeller and the skeg should be reduced by offsetting the shaft center from the vertical axis of the skeg, by modifying the skeg section, and by skewing the blade profiles.

Tests on large models should be made to predict the amount and nature of thrust variations in advance. This should be done for all major vessels during the design period.

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