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INVESTIGATION OF HULL VIBRATIONS OF USCGC PONTCHARTRAIN (WPG70)

by

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DAVID TAYLOR MODEL BASIN

PERSONNEL

The tests described in this report were performed by V.S. Hardy and R.B. Allnutt of the Taylor Model Basin staff, in cooperation with Lt. Comdr. S.W. Lank, USCGR, and Lt. E.F. Noonan, USCGR. The report was written by Mr. Allnutt.

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ABSTRACT

Because two rather severe critical series of vibrations were encountered during the trials of a recent class of U.S. Coast Guard vessels, vibrations produced in the USCGC PONTCHARTRAIN (WPG70) by a vibration generator were measured to determine the various critical frequencies of the hull.

The test results include resonance curves of hull vibrations for vertical and horizontal excitation through the blade-frequency range of the ship for 3-, 4-, or 5-bladed propellers, and profile curves of the amplitude of vibration at the present objectionable horizontal and vertical critical frequencies of the hull. Also included is a discussion of the probable effect of substituting a 4- or 5-bladed propeller for the original 3-bladed one.

It is concluded that a 4- or 5-bladed propeller would reduce the hull vibration to such an extent that it would no longer be objectionable.

INTRODUCTION

At the request of the Bureau of Ships (1),* representatives of the David Taylor Model Basin investigated vibrations on the USCGC PONTCHARTRAIN (WPG70) at Curtis Bay, Maryland, on 25 April 1945 and again on 1 May 1945. U.S. Coast Guard cutters of the WPG39 through 44 and the WPG64 through 70 Classes have a normal displacement of 2000 tons and are turboelectric powered. They develop 4000 horsepower at a maximum speed of 18 knots and are propelled by a single 3-bladed propeller turning at a maximum speed of 180 RPM.

Prior to this test and as reported in Reference (2), the USCGC OWASCO (WPG39) had experienced severe vibrations of blade frequency** which caused maximum vertical motions of the hull at 170 shaft RPM and maximum horizontal motions of the hull at 100 shaft RPM, with the ship at normal displacement. Observations of hull vibrations aboard the USCGC MENDOTA (WPG69) at a displacement of 1675 tons on 24 April 1945 confirmed this report. On the MENDOTA objectionable vibrations of blade frequency in the vertical direction were observed at 180 shaft RPM and in the horizontal direction at 100 shaft RPM. The vertical vibration at full power, at 180 RPM, was considered especially objectionable in the engine room.

The purpose of the test on the PONTCHARTRAIN was to obtain information concerning the characteristics of the vibration of ships of this type with a view to studying the effect of changing the number of propeller blades

* Numbers in parentheses indicate references on page 9 of this report.

** Blade frequency is the propeller RPM times the number of blades.

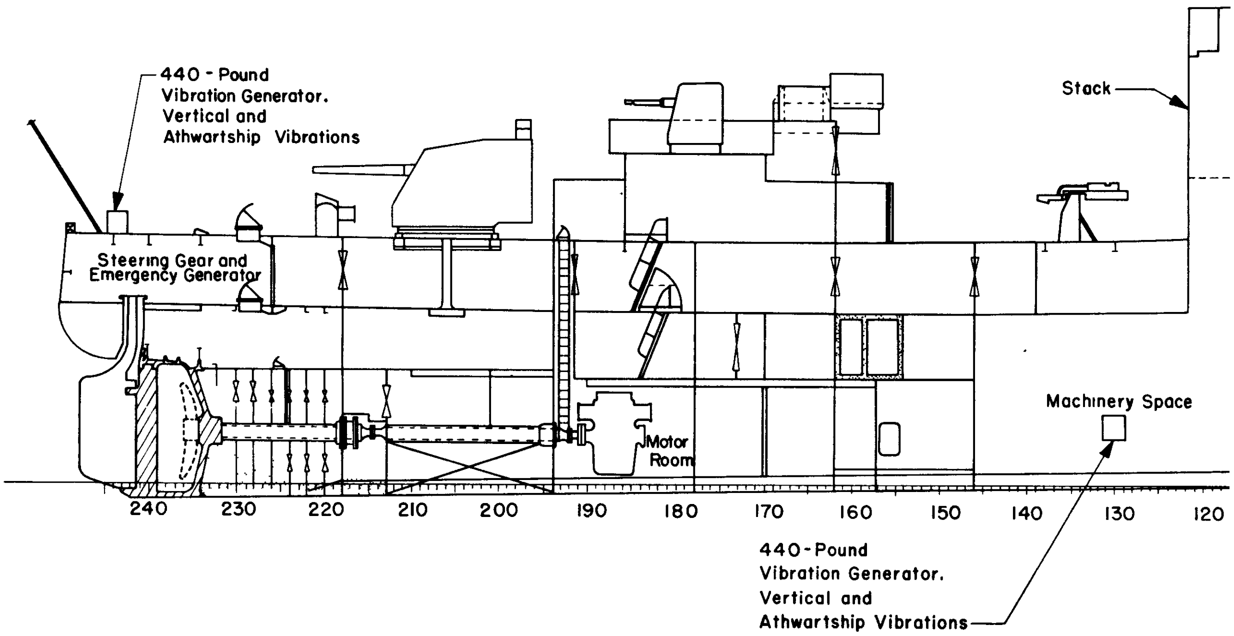


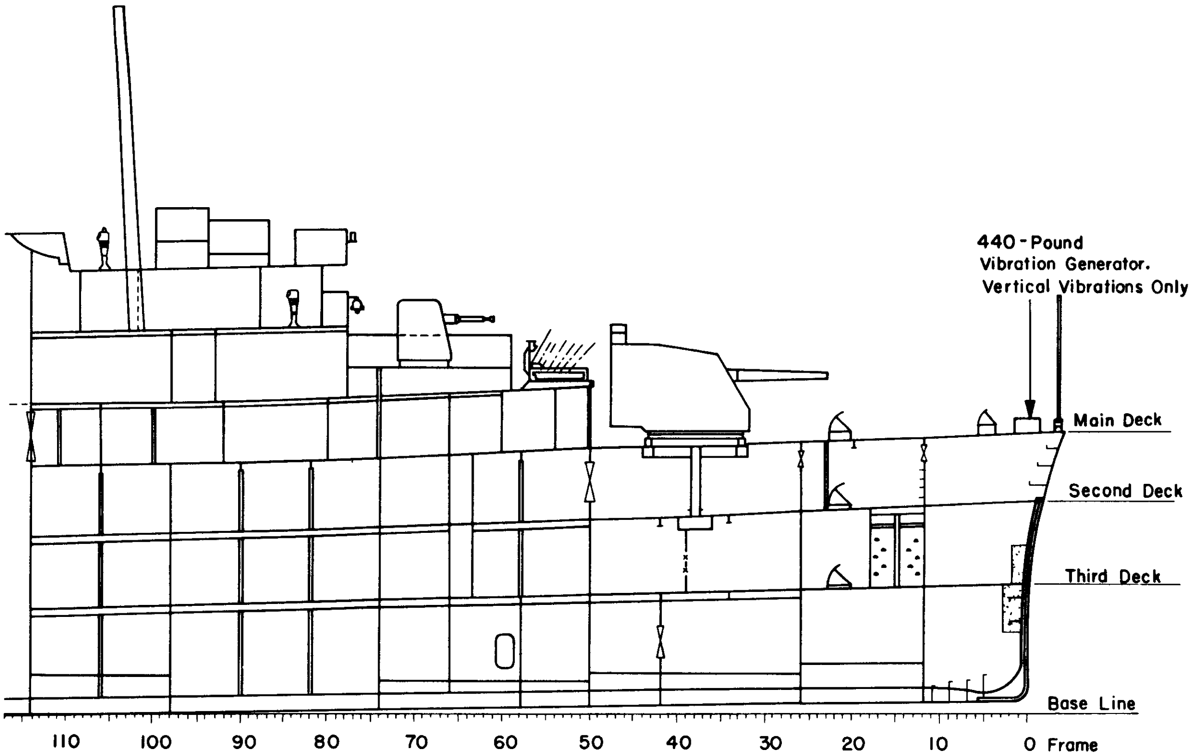
Figure 1 - Inboard Profile of USCGC PONTCHARTRAIN (WPG70)

from 3 to 4, or from 3 to 5. It was planned to find the vertical and horizontal resonant frequencies of the hull, to determine certain of the various modes of vibration, and to obtain information regarding the critical frequencies of the turbogenerator foundation. It should be noted, however, that the displacement of the PONTCHARTRAIN during this investigation was about 1470 tons, whereas its normal displacement is 2000 tons. Consequently the resonant frequencies were higher than they would have been if the ship had been loaded to normal displacement.

TEST APPARATUS AND PROCEDURE

The equipment used during this investigation consisted of a Losenhausen vibration generator (3), which weighs 150 pounds and generates a rated force of 440 pounds,* and a General Radio vibration meter with a piezoelectric crystal pickup attached. Prior to this test, the largest vessel vibrated with this machine for the purpose of determining the natural frequencies of the

* Vibration generators are generally rated in terms of the maximum peak force they are designed to produce under continuous operation.



entire hull had a normal displacement of only 250 tons. At that time the maximum force in pounds which could be produced by the generator was 2.20 x $(CPS)^2$, where the maximum speed permissible was 1500 CPM (25 CPS). Since that time additional eccentric weights had been added, enabling the vibration generator to produce a force in pounds given by the expression 4.41 x $(CPS)^2$ up to a frequency of 1200 CPM (20 CPS). At times during the tests on the PONTCHARTRAIN, the vibration generator was producing a force of only 50 pounds, and the amplitudes produced were therefore extremely small. The use of a General Radio vibration meter with a piezoelectric crystal pickup attached made it possible to record these slight vibrations and to determine six definite critical frequencies of the hull. To avoid erroneous readings of these small amplitudes, however, it was necessary to have the ship absolutely quiet in the water. The electric power necessary to operate the vibration generator was obtained from a shore generator installation. No other loads were imposed on the shore installation during the tests so that a constant frequency of the vibration generator could be maintained when desired. The speed of the vibration generator was measured with a Jagabi hand tachometer.

Vertical vibrations of the hull were produced with the vibration generator mounted on the main deck, first at the fantail and then at the bow. Athwartship vibrations were produced with the vibration generator mounted only at the fantail. Later it was mounted on the machinery platform at Frame 130, as shown in Figure 1, to produce vertical and athwartship vibrations of the turbogenerator foundation.

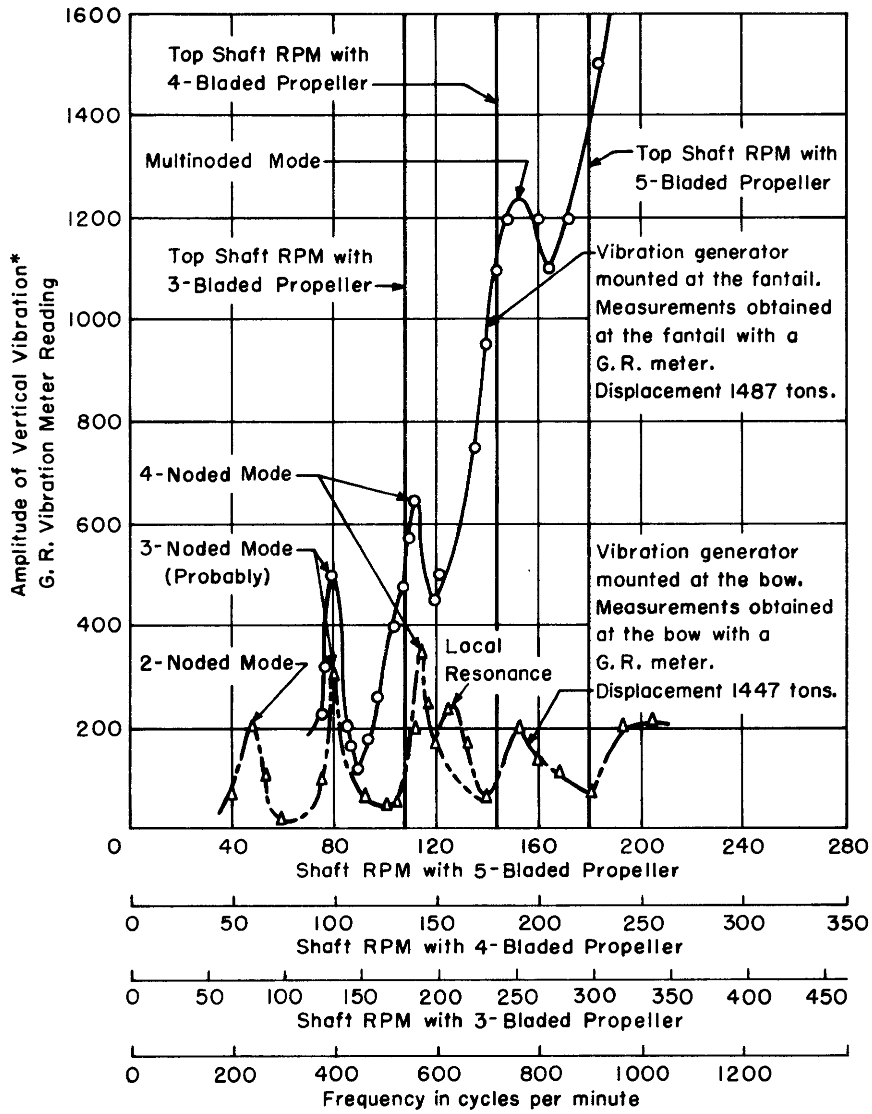


Figure 2 - Amplitude-Resonance Curves for Vertical Vibrations Measured on the Main Deck at the Fantail and at the Bow

The vibration generator was mounted first at the fantail, and then at the bow. The exciting force in pounds produced by the vibration generator was 4.41 $(CPS)^2$. The displacement of the vessel during these tests ranged from 1447 tons to 1487 tons.

* The General Radio vibration meter indicates root-mean-square amplitude. The true single amplitude in microinches is obtained by multiplying the meter reading by $\sqrt{2}$.

TEST RESULTS

Critical frequencies of the hull in vertical vibration were found at 240, 400, 565, and 760 CPM, as shown in Figure 2. It is apparent from the upper curve in Figure 2 that the amplitude of vibration of the fantail increased steadily above 820 CPM. To determine whether this behavior was due to a general characteristic of the entire hull, it was necessary to obtain an amplitude-resonance curve of the vertical vibration of the hull with the vibration generator mounted at the bow. The lower curve in Figure 2 shows that the vibration produced at the fantail above 820 CPM was local and was not characteristic of the entire hull.

The amplitudes of vertical vibration at the 565-CPM critical hull frequency were measured at various intervals along the centerline of the main deck from the bow to the fantail. At this critical hull frequency four definite nodal points were found, and an antinode was located opposite the machinery space, as shown in Figure 3. This condition explains the objectionable vibrations reported in the engine room at 170 CPM aboard the OWASCO at normal displacement. Since 565 CPM was found to be the frequency for the 4-noded vertical vibration of the hull, it could be concluded from Figure 2 that the resonance which occurs at 240 CPM is probably the 2-noded vertical flexural mode of the hull.

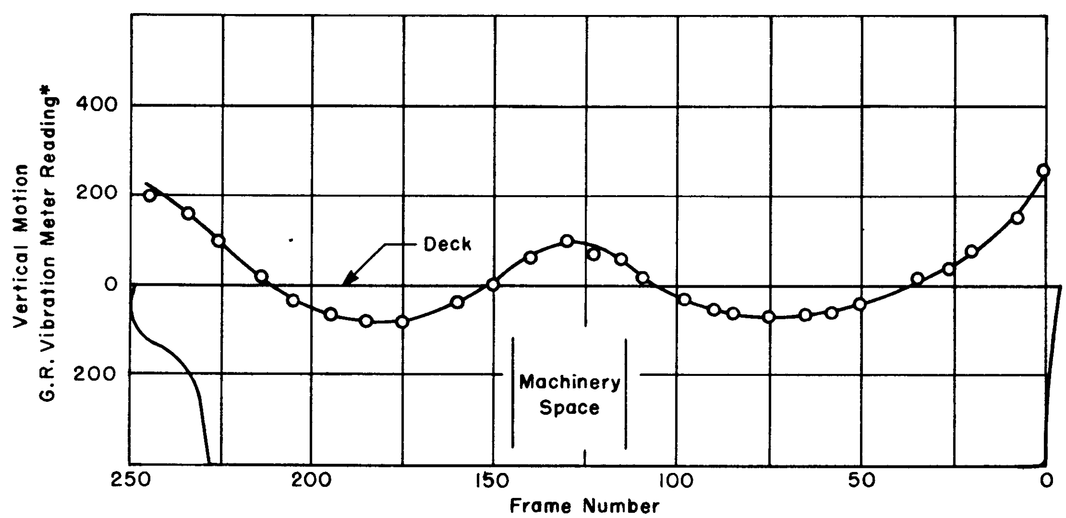


Figure 3 - Profile of the Amplitudes of Vertical Vibration at 565 CPM Measured along the Centerline of the Main Deck

During this test the vibration generator was mounted on the main deck at the bow to produce vertical vibration. The exciting force produced by the vibration generator was 390 pounds. The displacement of the vessel during this test was 1447 tons.

* See the footnote on page 4.

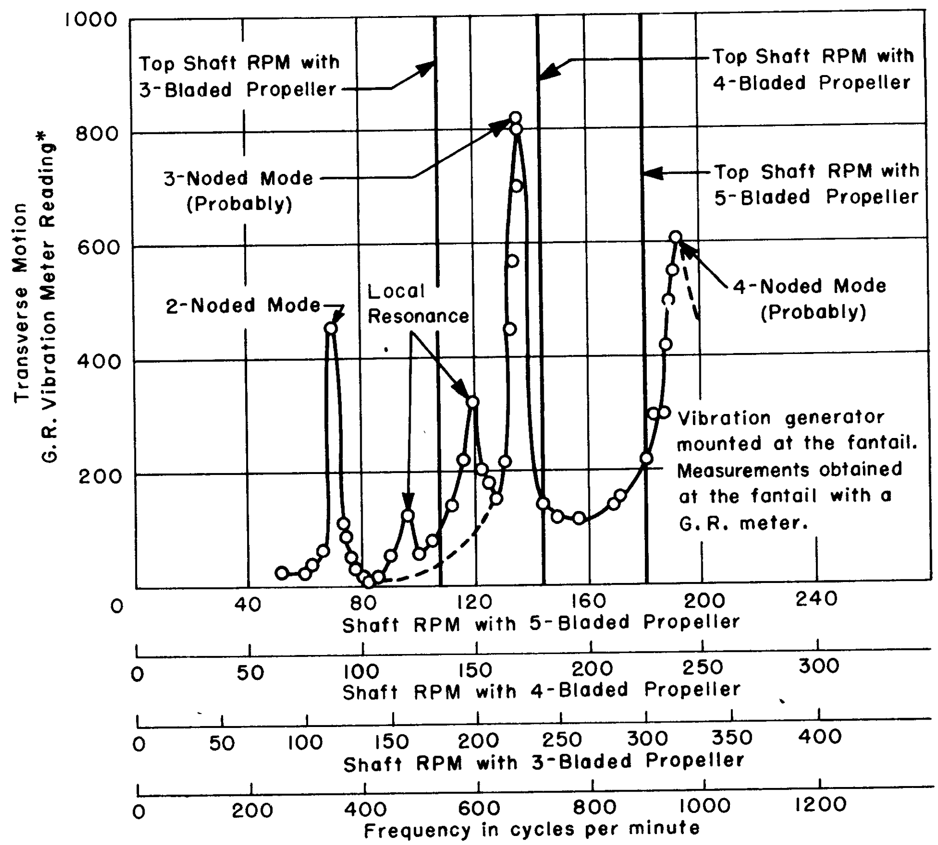


Figure 4 - Amplitude-Resonance Curve for Athwartship Vibration Measured on the Main Deck at the Fantail

During this test the vibration generator was mounted on the main deck at the fantail to produce athwartship vibration. The exciting force in pounds produced by the vibration generator was 4.41 $(CPS)^2$. The displacement of the vessel during the test was 1487 tons.

A quick calculation employing Schlick's empirical formula (4) for determining the 2-noded vertical flexural frequency of ship hulls in water substantiates this assumption. The formula is

$$N = C\sqrt{\frac{I}{DL^3}}$$

where N is the number of vibrations per minute,

C is Schlick's empirical coefficient, with values ranging from 1.28×10^5 to 1.57×10^5, according to the class of ship,

I is the area moment of inertia of the midship section in $\text{feet}^2\ \text{inch}^2$,

D is displacement of the ship in long tons, and

L is the overall length of the ship in feet.

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* The General Radio vibration meter indicates root-mean-square amplitude. The true single amplitude in microinches is obtained by multiplying the meter reading by $\sqrt{2}$.

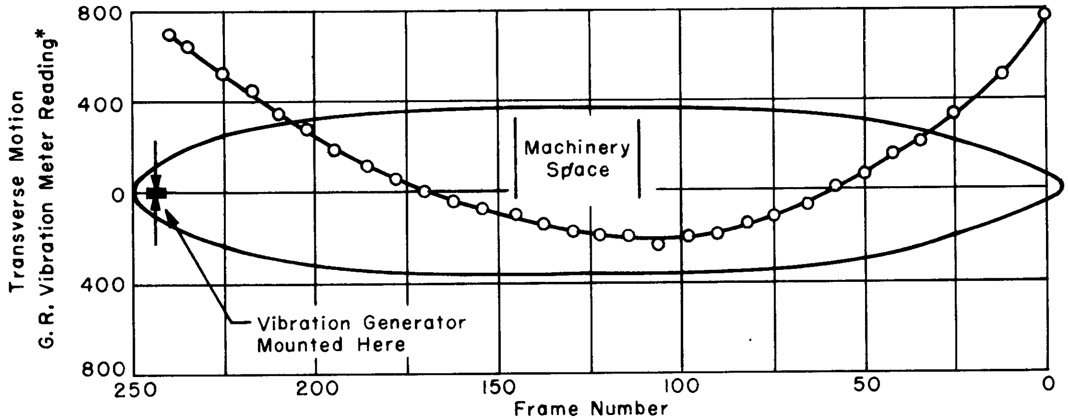

Figure 5 - Profile of the Amplitudes of Athwartship Vibration at 350 CPM Measured along the Centerline of the Main Deck

During this test the vibration generator was mounted at the fantail to produce athwartship vibration. The exciting force produced by the vibration generator was 150 pounds. The displacement of the vessel during this test was 1487 tons.

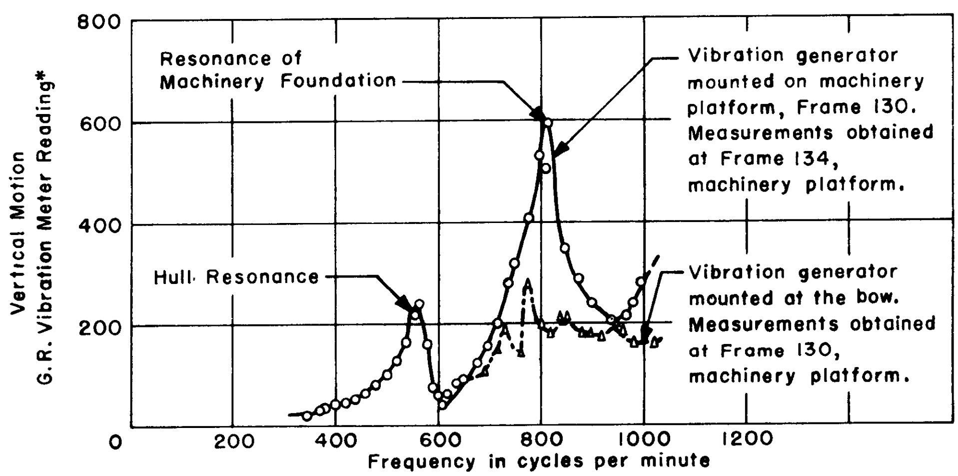

Figure 6 - Amplitude-Resonance Curves for Vertical Vibration Measured on the Turbogenerator Foundation

During these tests the vibration generator was mounted on the machinery platform at Frame 134 and at the bow to produce vertical vibrations. The exciting force in pounds produced by the vibration generator was 4.41 $(CPS)^2$.

If, in the calculation for the PONTCHARTRAIN, 1.3×10^5 is used as Schlick's empirical coefficient for this type of ship, the solution is as follows:

$$N = 1.3 \times 10^5 \sqrt{\frac{80,000}{1470(245)^3}} = 250 \text{ CPM}$$

which agrees very closely with the 240 CPM found with the vibration generator.

* See the footnote on page 6.

Critical frequencies of the hull in athwartship vibration were found at 350, 680, and 960 CPM, as shown in Figure 4. Amplitudes of athwartship vibration at 350 CPM were measured along the centerline of the main deck at various intervals, and it was found that this frequency is the 2-noded horizontal flexural frequency of the hull, as shown in Figure 5.

A vertical resonance of the turbogenerator foundation was found at 820 CPM, as shown in Figure 6. No resonances of the foundation were observed in the athwartship direction below 1000 CPM.

DISCUSSION OF RESULTS

If the present 3-bladed propeller is replaced by a 4-bladed propeller, the 3-noded horizontal frequency of the hull will then fall within the range of propeller-excited frequencies. It is roughly estimated that this critical frequency would occur at 150 shaft RPM under normal displacement. However, at the top speed of 180 shaft RPM there would be no disturbance due to athwartship vibration, since this speed would occur at a point of minimum amplitude on the amplitude-resonance curve of the hull; see Figure 4. The objectionable 2-noded horizontal mode at 100 shaft RPM would occur below the normal operating range with a 4-bladed propeller.

The effect of a 4-bladed propeller on the vertical vibration of the hull would be that the objectionable 4-noded vertical frequency of the hull at 180 shaft RPM would occur at approximately 130 shaft RPM and that a multinoded resonant frequency of the hull would be excited at full power. However, since the magnification of this multinoded mode at the machinery location is much less than that of the present 4-noded mode, and since the force per blade with a 4-bladed propeller would be less than it is with the 3-bladed propeller, the amplitudes produced at full power with a 4-bladed propeller would quite possibly be much less than those which are now encountered. It is probable that the vibration of the fantail would still be considerable at full power.

A 5-bladed propeller would decrease the impulse per blade still further than a 4-bladed propeller, but a vertical resonance of the turbogenerator foundation might possibly be encountered at 165 shaft RPM. However, since this speed with a 5-bladed propeller would occur at a point of minimum amplitude on the amplitude-resonance curves of the hull, in both the vertical and the athwartship directions, as shown by Figures 2 and 4, it is possible that the vibration of the turbogenerator foundation would not be as serious as it is at present.

The objectionable 4-noded vertical critical frequency of the hull would occur at a speed of approximately 120 shaft RPM with a 5-bladed propeller, and the 2-noded horizontal critical hull frequency would occur at 60 shaft RPM, well below the operating range. At full power, a 4-noded horizontal frequency of the hull would be approached. The lower curve in Figure 2 indicates that at full power vertical vibration of the hull would not be serious, but the upper curve in Figure 2 indicates that the fantail would still vibrate appreciably.

CCNCLUSIONS

From Figures 2 through 6 it can be concluded that an increase in the number of blades would be beneficial. The objectionable 4-noded vertical and 2-noded horizontal hull frequencies would occur at lower shaft speeds, and at such speeds both the steady thrust and thrust variation would be less. The increase in the number of blades would also decrease the impulse per blade, thus decreasing appreciably the amplitudes at the critical frequencies of the hull. In view of the possibility of a machinery-foundation resonance with a 5-bladed propeller, a 4-bladed propeller should be tried first. Satisfactory performance with a 4-bladed propeller would obviate the necessity of trying 5 blades. However, if performance is considered unsatisfactory with 4 blades, then 5 blades should be tried. If additional local resonances are produced with a 4- or 5-bladed propeller, local structural changes will be necessary.

Although the afterbody and sternpost design of vessels of this type is not discussed in this report, it is assumed that the question of tip clearance and the desirability of rake or skewback would be considered before a 4- or 5-bladed propeller was finally adopted.

REFERENCES

(1) BuShips letter QS/OWASCO (332), QS/PONTCHARTRAIN, of 23 April 1945 to the Taylor Model Basin.

(2) Commandant, Mare Island, letter PG/S68(390-742309) of 2 May 1945 to SupShips, San Pedro.

(3) "Construction and Operation of the Losenhausen 440-Pound Vibration Generator," by E.O. Berdahl, TMB Report 539, April 1945.

(4) "Vibration Tests on USS HAMILTON (DD141) at Washington Navy Yard," by R.T. McGoldrick, EMB Report 372, December 1933.