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VIBRATION TEST ON SHAFT BOSSING

OF

U. S. S. HAMILTON

AT

NORFOLK NAVY YARD, JUNE 11, 1931

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U. S. EXPERIMENTAL MODEL BASIN

NAVY YARD, WASHINGTON, D. C.

JULY, 1931.

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VIBRATION TEST ON SHAFT BOSSING
of
U. S. S. HAMILTON
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NORFOLK NAVY YARD, JUNE 11,1931

SUMMARY

The shaft and bossing were set in forced vibration by mounting a vibration generator on the end of the shaft in place of the propeller. The frequency of the periodic force was varied until resonance was obtained. The force was applied in different planes by jacking over the shaft. Resonance occurred on both shafts at a frequency of 1710 vibrations per minute. Maximum amplitude occurred when the exciting force was perpendicular to the plane of symmetry of the bossing.

GENERAL

The purpose of this test was to find the natural frequency of vibration of the new propeller shaft bossings arising from unbalanced forces on the propellers in order to predict whether a condition of resonance would arise under operating conditions. Since the maximum shaft speed is about 450 RPM, the periodic unbalanced condition with propellers of three blades would occur at a frequency of around 1350 per minute. If the natural frequency is well above this value no excessive vibration is to be expected. The safe limit depends on the

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sharpness of the resonance curve and this is determined by the damping conditions. The greater the damping forces the broader is the resonance curve and the smaller the amplitude at resonance, so that, in addition to determining the resonant frequency, it becomes equally important to determine the sharpness of resonance. This is analogous to selectivity in tuned electrical circuits where the resistance corresponds to the damping force.

In a forced vibration the power required to maintain vibration is all expended in overcoming the damping resistance and is dissipated in the form of heat. The average power P is given by the equation:

$$P = \frac{A^2 w^2 l}{2}$$

where A is the amplitude, l the damping resistance per unit velocity, and w the circular frequency (2π x frequency). Since the amplitude decreases rapidly after resonance is passed maximum power is required at resonance. Hence the power-frequency curve gives not only the resonant frequency but also an indication of the damping.

APPARATUS AND METHOD OF TESTS

The vibration generator used for determining the resonance curve is shown on the attached photographs. It consists essentially of two parallel shafts geared together so that they rotate in opposite directions. On each end of each shaft is a pair of weights which can be set at any desired angle. When the two weights are opposite one another their common center of gravity falls on the axis and the machine is balanced so that there is no external force applied regardless of the speed. But, if one of the weights is held fixed and the other moved around the circumference, the common center of gravity moves out from the axis and a centrifugal force equal to mrw^2 arises, where m is the combined mass of the two weights, r the distance of the common center of gravity from the axis, and w the angular velocity. According to the manner in which the weights are set the machine can produce vertical, tilting, or torsional oscillations. Since for vertical oscillations all pairs of weights are acting together the total force is 4 times the value given above.

A motor is mounted on each axle of the generator and a wattmeter is connected in so as to give the power taken by the armatures only. The wattmeter thus gives the power required to maintain the vibration plus that taken to overcome friction in the bearings. The latter is slight and is measured by clamping the machine to a rigid foundation and making a power-frequency curve. From this the curve obtained during a vibration test can be corrected so as to give only the power required to maintain vibration.

This machine is designed for a maximum centrifugal force of 430 lb. and a maximum speed of 3600 RPM.

For the purpose of this test a special carrier was made by means of which the vibration generator could be mounted on the shaft in place of the propeller while the vessel was in dry dock. The design of this carrier is shown on Norfolk Plan N-10133, Washington No. Misc.127. Only one keyway was cut instead of two as shown on the drawing.

Photographs, Norfolk Nos. 599-31 and 600-31, prints of which were forwarded to the Navy Yard, Washington, and the Bureau of C&R with Commandant, Norfolk letter DD141/S43-1(5-MD) of 19 June 1931, show the exciter mounted on the end of the propeller shaft.

Wedges were first driven in between the shaft and the bearing so that the vibration would be transmitted directly to the bossing. The generator was mounted on the table of the carrier and the weights were offset 10° for vertical vibration, in other words vibration in a plane perpendicular to the base of the machine regardless of the inclination of the machine to the horizontal. In order to determine the angle of inclination the plane thru the axis of the bossing was taken as the plane of reference. The shaft was jacked over so that the base of the vibration generator was parallel to this plane (as nearly as could be judged by eye). From this position rotation was measured in stages of 30° by scribing on the end of the bossing and attaching a pointer to the key on the shaft.

CONDUCT OF TESTS

With the machine in the initial position on the shaft and with a 120 volt d. c. line from the ship connected to the control box terminals a run was made up to a speed of 46 Hertz (1 Hertz = 1 cycle per second). Vibration could easily be detected in the neighborhood of 30 Hertz but the amplitude was exceedingly small. The weights were then offset 20° whereupon a very distinct vibration could be felt at a speed of 28.5 Hertz (1710 RPM). The safety limits of the machine would not permit increasing the eccentricity. No satisfactory method of measuring the amplitude could be devised within the time allowed for the test, but at the end of the propeller shaft it appeared to be of the order of 1/8 inch.

Because of the smallness of the amplitude the power taken by the machine was less than 5 watts. This made it impossible to obtain a power-frequency curve and hence to determine the resonant frequency by the power method. There seemed, however, to be fairly sharp resonance. The resonant speed was determined merely by feeling the vibration at the end of the shaft.

At maximum amplitude vibration could be felt all over the stern of the ship. The vibration of the rudder could easily be seen and vibration could be felt along the entire length of the bossing. It was reported that the vibration could be felt in the engine room.

Owing to the inadequacy of the power readings and the lack of an accurate method of measuring the amplitude, the test boiled down to a determination of the resonant frequency and the plane of maximum vibration.

RESULTS

The resonant frequency was found to be 28.5 Hertz (1710 per minute) for both shafts. Resonance occurred at the same frequency regardless of the inclination of the machine, but maximum amplitude occurred when the exciting force was perpendicular to the axis of the bossing, and minimum amplitude when parallel to the axis.

On the port side vibration could be detected at about 26 Hertz and continued to about 32 Hertz. On the starboard side the resonance curve seemed to be broader and vibration was felt at 24 Hertz. Removal of the wedges around the shaft did not affect the results.

CONCLUSIONS

In conclusion it should be pointed out that this test was run under several conditions which would not exist at sea. The absence of the propellers increases the natural frequency. It would be preferable to devise a means of mounting the vibration generator with the propeller in place. A rough estimate shows that the resonant frequency would have been about 20% lower or approximately 1370 with the propellers in place.

In dry dock the inertia and damping effects of the water are not present resulting in a higher natural frequency.

The fact that the vessel was resting on blocks also probably increased the frequency since the stern of the ship was vibrating as a cantilever beam whose length was shortened by the keel blocks.

With a heavier machine vibration could have been detected over a much greater frequency range and a better idea of the sharpness of resonance could have been obtained by making a power-frequency curve. On the other hand, the problem of mounting such a machine, especially without removing the propellers, would have been much more difficult.

Photographs

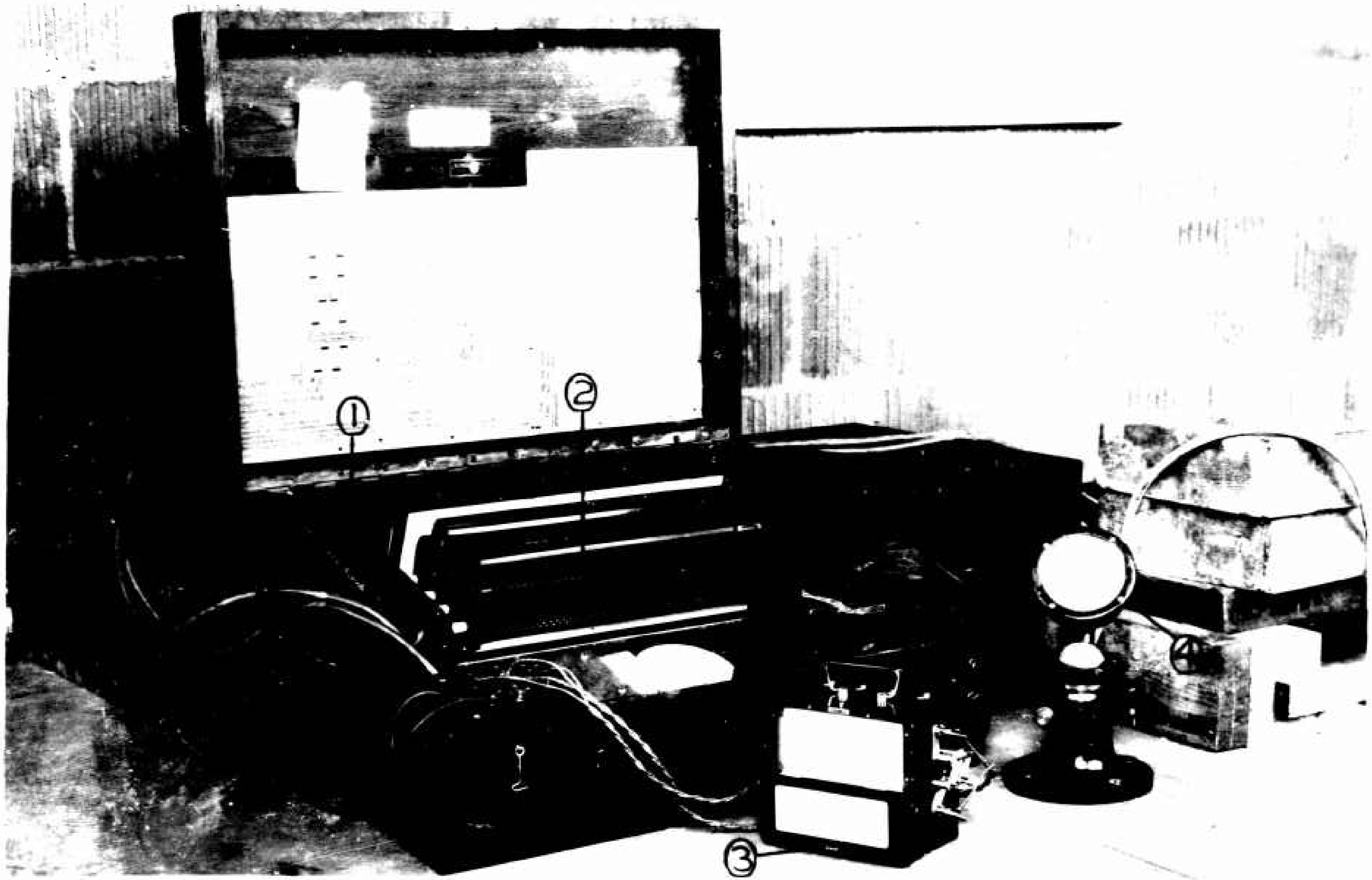
1. Vibration Generator - Control Apparatus.

2. Vibration Generator with Control Apparatus.

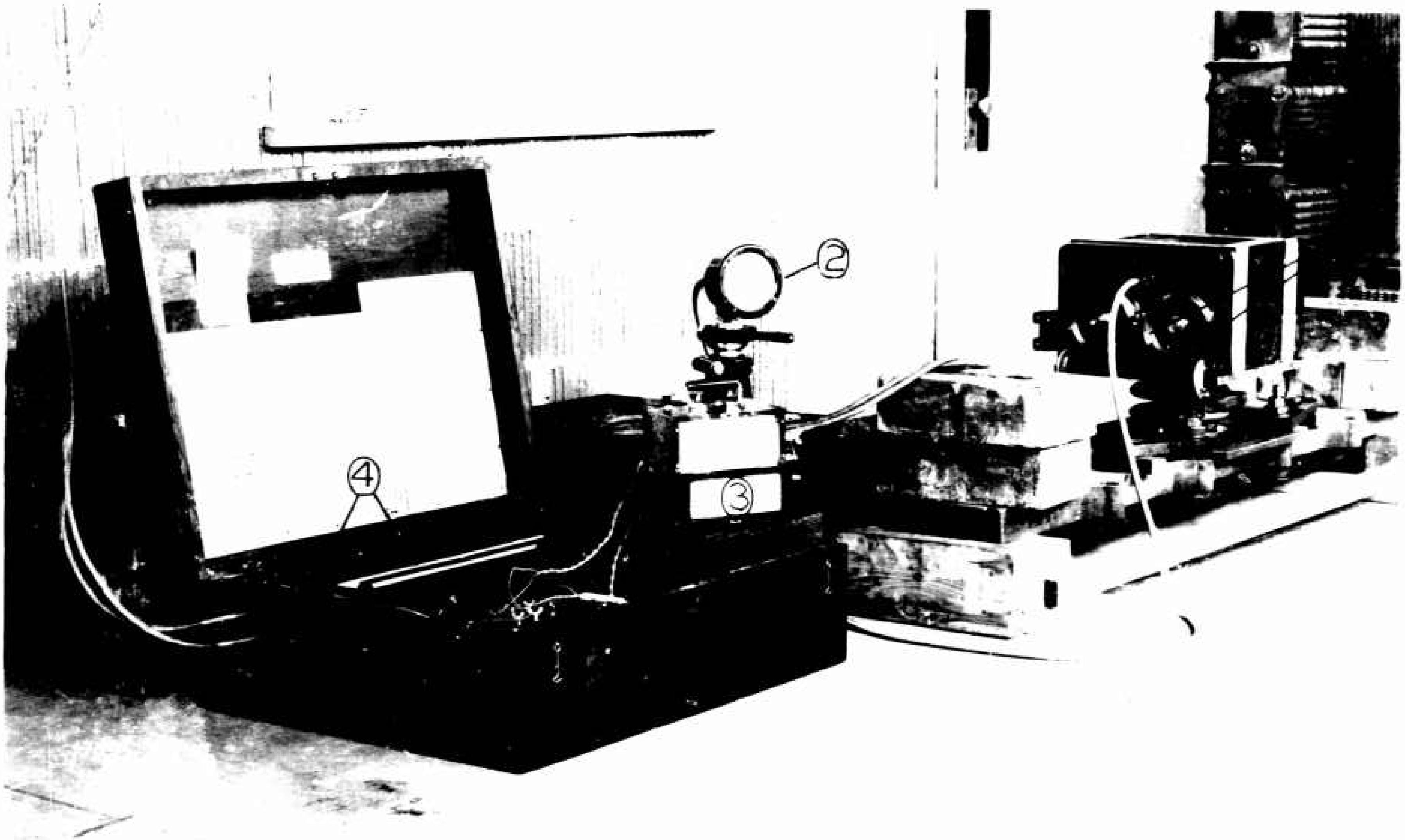
3. Vibration Generator.

4. U.S.S. HAMILTON - Experimental Shaft Bossing, Port Side, Showing Mechanical Vibration Apparatus in Place.

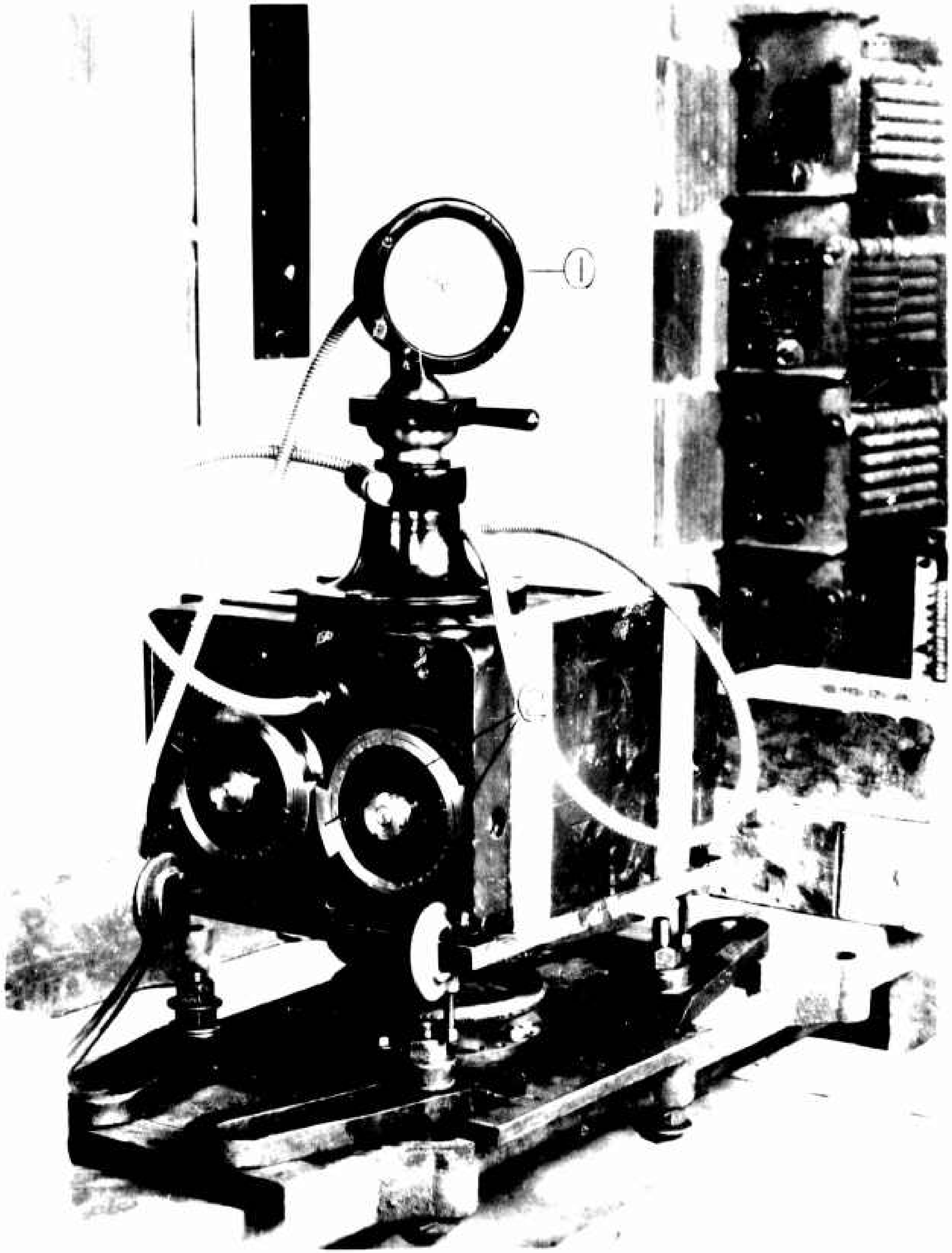
5. U.S.S. HAMILTON - Conducting Vibration Test on Port Shaft and Bossing with Mechanical Vibration Exciter, Manufactured by Losenhausenwerk, Dusseldorf, Germany.



VIBRATION GENERATOR - CONTROL APPARATUS
(1) Field rheostat (3) Wattmeter
(2) Armature rheostat (4) Tachometer



VIBRATION GENERATOR WITH CONTROL APPARATUS
(1) Weights (3) Wattmeter
(2) Tachometer (4) Rheostats



VIBRATION GENERATOR
(1) Tachometer (2) Adjustable weights


U.S.S. HAMILTON (DD 141)
EXPERIMENTAL SHAFT BOSSING PORT SID
SHOWING MECHANICAL VIBRATION
APPARATUS IN PLACE.
NOTE: SHAFT HELD RIGIDLY TO BOSSING
BY WEDGES.
NFK. NAVY YD.
PORTS. VA.
11 JUNE 1931.


TEST ON
WITH MECHANICAL
SHAFT
MANUFACTURED
DUSSELDORF