UNITED STATES

EXPERIMENTAL MODEL BASIN

NAVY YARD, WASHINGTON, D.C.

VIBRATION TESTS ON U.S.S. HAMILTON AT WASHINGTON NAVY YARD

BY R.T. MCGOLDRICK

EXPERIMENTAL MODEL BASIN
ERECTED 1898
BUREAU OF
CONSTRUCTION AND REPAIR
NAVY DEPARTMENT

DEC. 1933

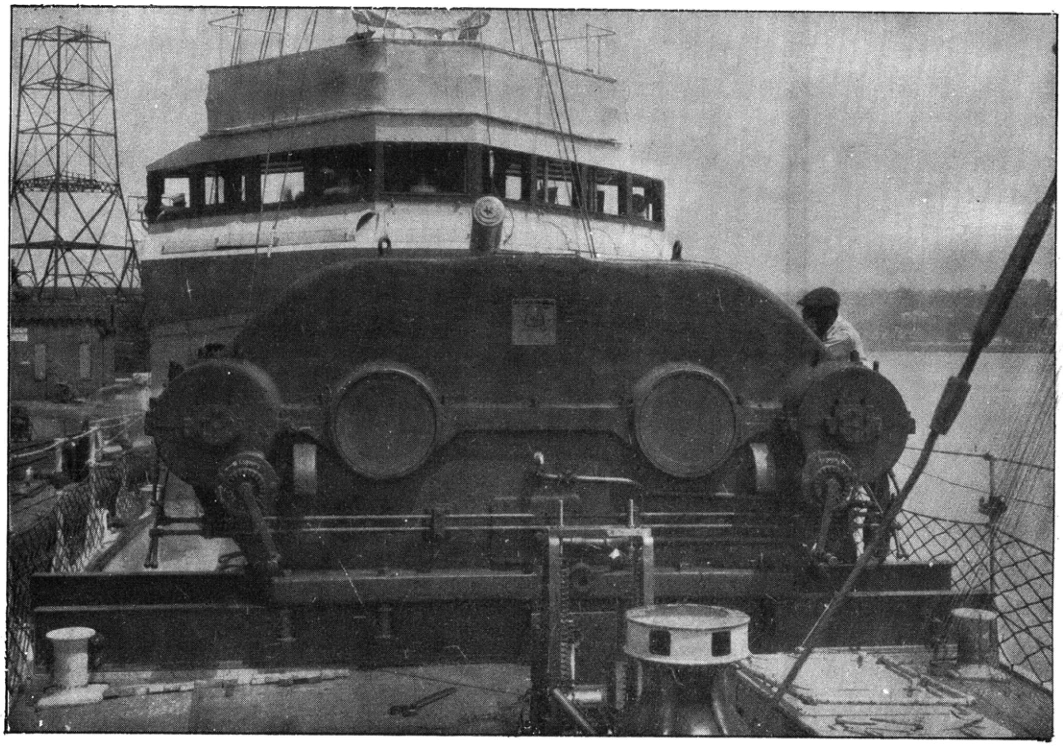
REPORT NO. 372

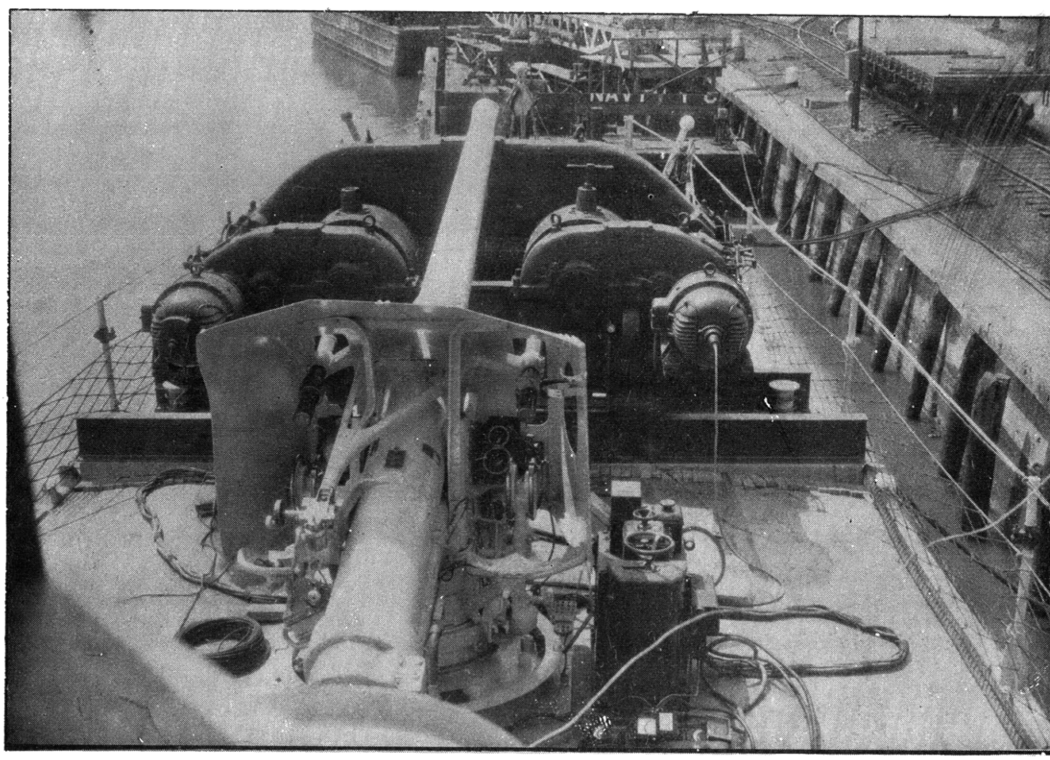
VIBRATION TESTS ON U.S.S. HAMILTON

by R. T. McGoldrick

U.S. Experimental Model Basin
Navy Yard, Washington, D.C.

Dec. 1933 Report No. 372





FIGS. 1 and 1A – VIBRATION GENERATOR ON FORWARD DECK OF HAMILTON

VIBRATION TESTS on U.S.S. HAMILTON

Summary

The hull of the U.S.S. HAMILTON was set in two and three noded vibration by means of a vibration generator placed on the forward deck. The resonant frequencies were determined in two depths of water, one at the dock and the other when anchored in the river about 500 yards from shore.

The average value of the fundamental frequency obtained at the dock was 92.3 vibrations per minute. The average value obtained when anchored in the river was 106.7 vibrations per minute.

Computation by means of Taylor's graphical method gives a fundamental frequency of 116 vibrations per minute. Schlick's empirical formula would require a constant of 1.16×10^5 to check the measured deep water frequency.

A sharp three noded resonance was obtained in deep water at 206 vibrations per minute, but this resonance was not perceptible at the dock.

Object of the Investigation

The object of this experiment was to check theoretically computed values of natural frequencies of ships by actual measurements with a view to determining the unknown damping effect of the surrounding water.

Description of Vibration Generator

The vibration generator built by Losenhausenwerk of Dusseldorf, Germany is shown in the photographs Figs. 1 and 1a and schematically in Fig. 2. The total weight is 49,000 lb. Vibration is set up by two cylindrical eccentric masses which rotate in opposite directions on horizontal shafts. The horizontal components of centrifugal force balance one another whereas the vertical components are additive. The weight of each eccentric mass is 6000 lb. and the eccentricity can be varied from 0 to 11.8 inches. The eccentrics are set in rotation by two motors with two possible gear reductions, one giving a ratio of 6:1 and the other 16:1. The 16:1 ratio is intended to cover a speed range of .9 to 3. Hertz (revolutions per second) and the 6:1 ratio a speed range of 2.5 to 8 Hertz. The motors are designed for 220 volts and are rated at 15 kw. each. As the speed is increased the allowable eccentricity decreases in order that a maximum centrifugal force of 44,000 lb. shall not be exceeded.

Methods of Computing Ship Frequencies

The general method of computing ship frequencies is to estimate either graphically or analytically the out-of-water frequency and then to apply a correction for the so-called virtual mass of the surrounding water. The ship is treated as a beam having non-uniform load distribution and non-uniform section. As the virtual

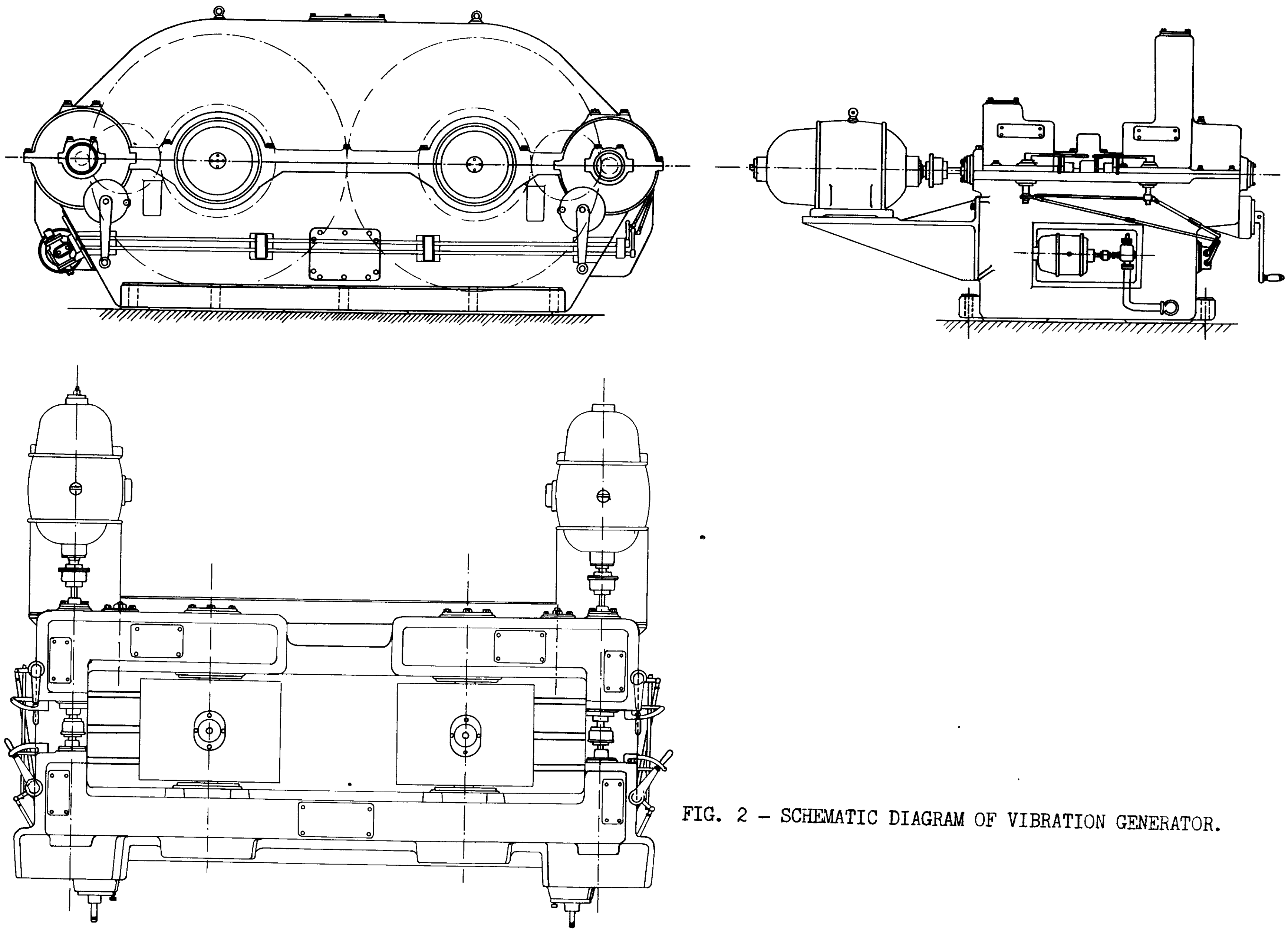
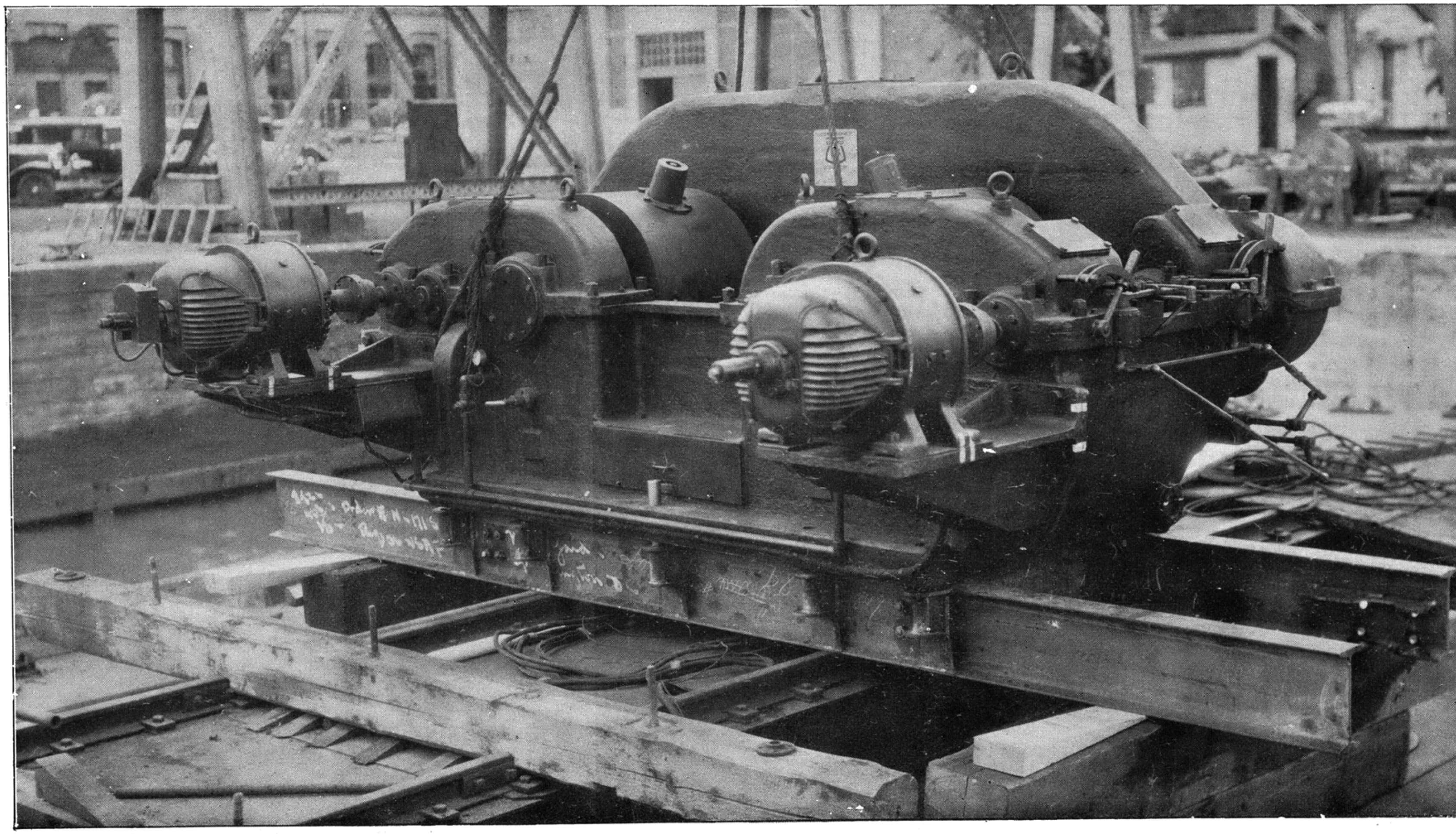


FIG. 2 – SCHEMATIC DIAGRAM OF VIBRATION GENERATOR.



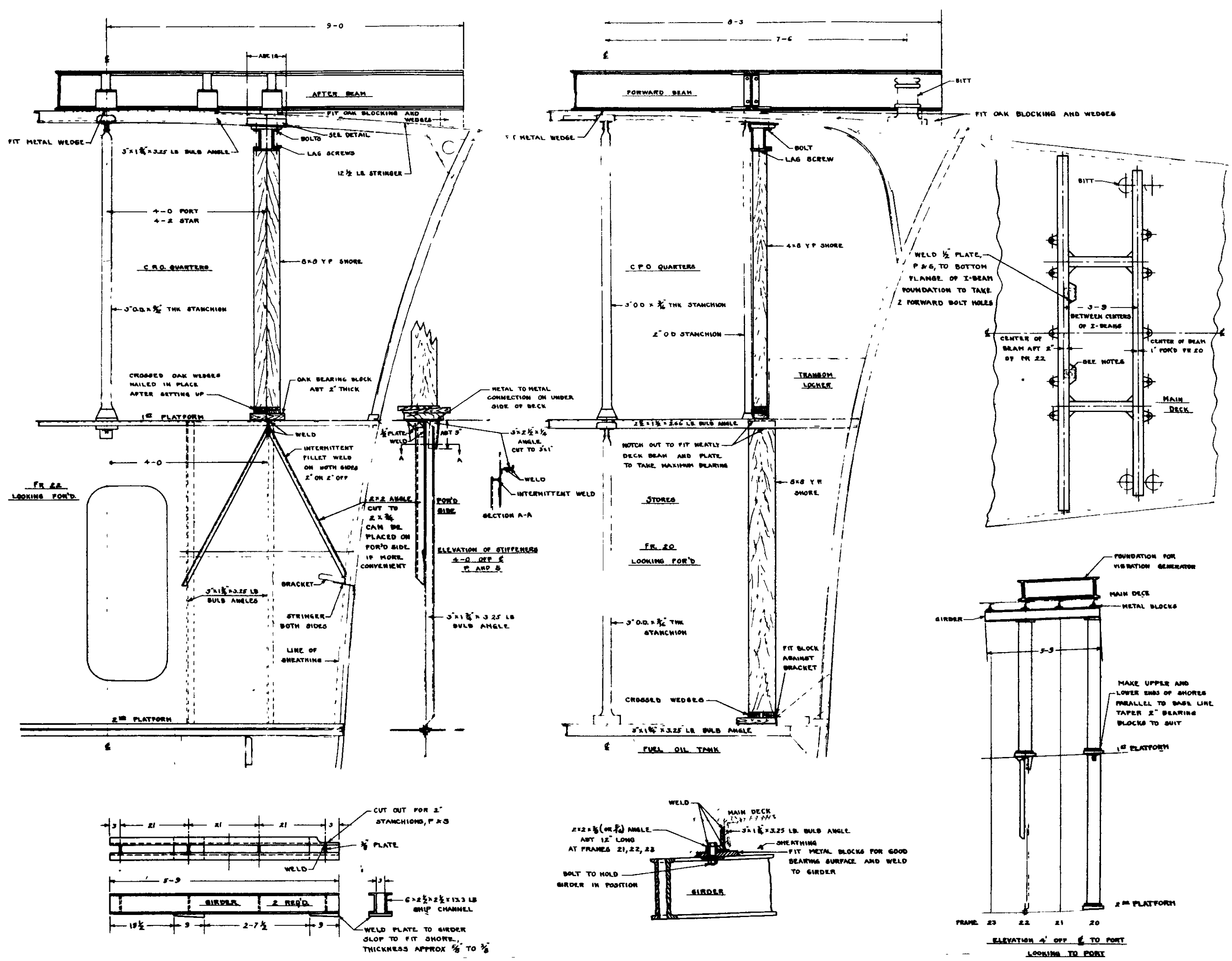


FIG. 4 – FOUNDATION AND SHORING PLAN.

mass may amount to 100% or more of the mass of the ship (see reference 1), various investigators have questioned the value of computing out-of-water frequencies and have preferred to abide by Schlick's empirical formula:

$$n = c\sqrt{I/DL^3}$$

where:

n is the frequency per minute

L is the length in feet

I is the moment of inertia of midship section in ft^2in^2

D is the displacement in tons

c is a constant (1.57×10^5 for destroyers)

(see reference 2).

The only data required are the moment of inertia of the midship section, the displacement, and the length.

Shallow Water Test

The foundation shown in the photograph Fig. 3 was placed on the forward deck of the U.S.S. HAMILTON approximately over frame 22 and wedged firmly in place. Beams were placed under the deck and shored up so as to distribute the load over three frames. The foundation was bolted on one side to these beams and on the other side anchored by U-bolts to the two bitts. The shoring was carried two decks below as shown in Fig. 4 (Wash. Plan A 1087). A calculation showed the stability to be satisfactory with the machine in this location.

The first test was made alongside the dock on May 24th. Power was supplied by the ship's generators. The maximum voltage obtainable was about 120 volts which limited the range of speed over which observations were possible. The first run was made at an eccentricity of 3 degrees, and thereafter the eccentricity was increased in steps of 3 degrees up to 18 degrees. The resonance was determined by noting the speed of maximum amplitude. The power method of determining resonance proved unsatisfactory because of the relatively large internal power consumption of the machine (see discussion of power method at end of report). The average two noded frequency for six different eccentric settings was 92.3 per minute. The range over which vibration was perceptible at an eccentric setting of 18 degrees was 87.6 to 99.6 per minute. The 6:1 gear ratio was used rather than the 16:1 specified in the instructions as it gave better speed regulation at the low voltage available. The following table shows the resonant speeds, centrifugal force, and power consumption at different eccentric settings:

TABLE I

Resonant Speed (rpm by counter)	Eccentricity		Centrifugal Force (lb.)	Power Input (kw.)
	Deg.	Inches		
91.5	3	.31	875	2.0
92.2	6	.62	1775	2.0
94.0	12	1.25	3692	2.0
91.5	15	1.54	4375	2.1
92.1	18	1.85	5320*	2.4

*The maximum centrifugal force obtainable at this speed is 32,000 lb.

No suitable amplitude meter was available for this test but amplitudes were estimated from shore by means of a transit. The double amplitudes at 18 degrees eccentricity were approximately 1 inch at the bow, 7/8 inch at the stern, and 7/32 inch amidships. The nodes occurred at frame 57½ (101 feet from the bow) and frame 125 (219 feet from the bow).

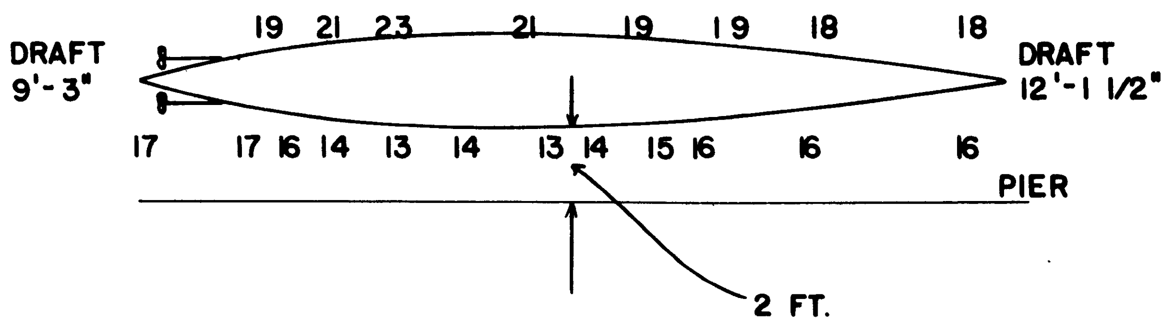


FIG. 5 – VIBRATION TEST – USS HAMILTON, SOUNDINGS – MAY 24, 1:15 P.M.–DEPTH IN FEET

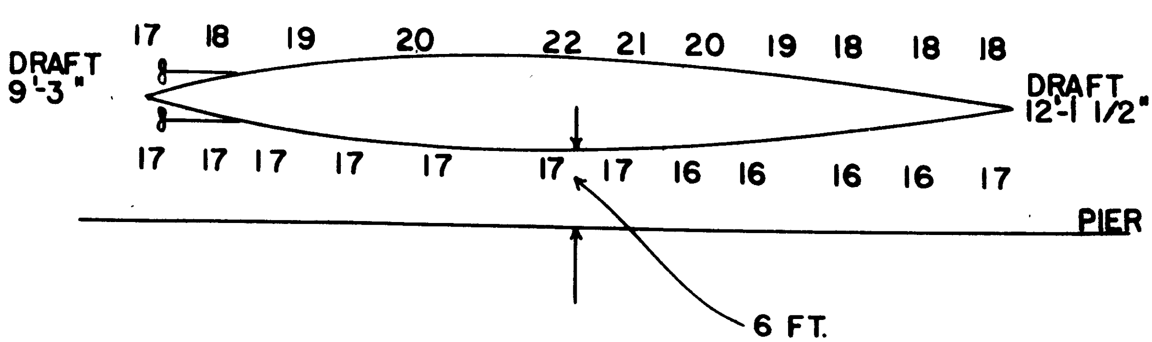
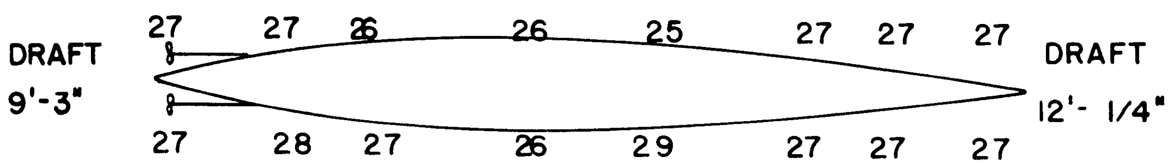


FIG. 6 – VIBRATION TEST – USS HAMILTON, SOUNDINGS – MAY 24, 3:00 P.M.–DEPTH IN FEET

The depth during this test varied from 13 feet to 23 feet averaging about 17 feet (see Fig. 5 and 6). The displacement estimated by correcting the normal weight curve for changes in fuel, water, and ammunition was 1382 tons. The draft was 12'1½" forward and 9'3" aft. The distance of the ship from the dock varied from 2 feet to 6 feet but no check was made to see if the differences in resonant speed were due to these changes.

On May 25th the test was repeated near Quantico on the Potomac River in order to determine to what extent the depth of water affected the resonant frequency. The depths varied from 25 feet to 29 feet averaging about 27 feet (see Fig. 7) and the ship was anchored about 500 yards from either shore. An average resonant frequency of 106.7 vibrations per minute was obtained. This was an increase of 15% over the value obtained at the dock on the previous day. The amplitudes at resonance seemed to be much less for the same eccentric setting than in the previous test although instruments were not available for making actual measurements. The eccentricity was carried as far as 30 degrees without changing the gear ratio. Table II shows the resonant speeds, centrifugal force, and power consumption for the eccentric settings used in this test.



SAME DEPTH ABOUT 500 YDS. ON EACH SIDE

FIG. 7 – VIBRATION TEST–USS HAMILTON, SOUNDINGS NEAR QUANTICO, MAY 25TH–DEPTH IN FEET

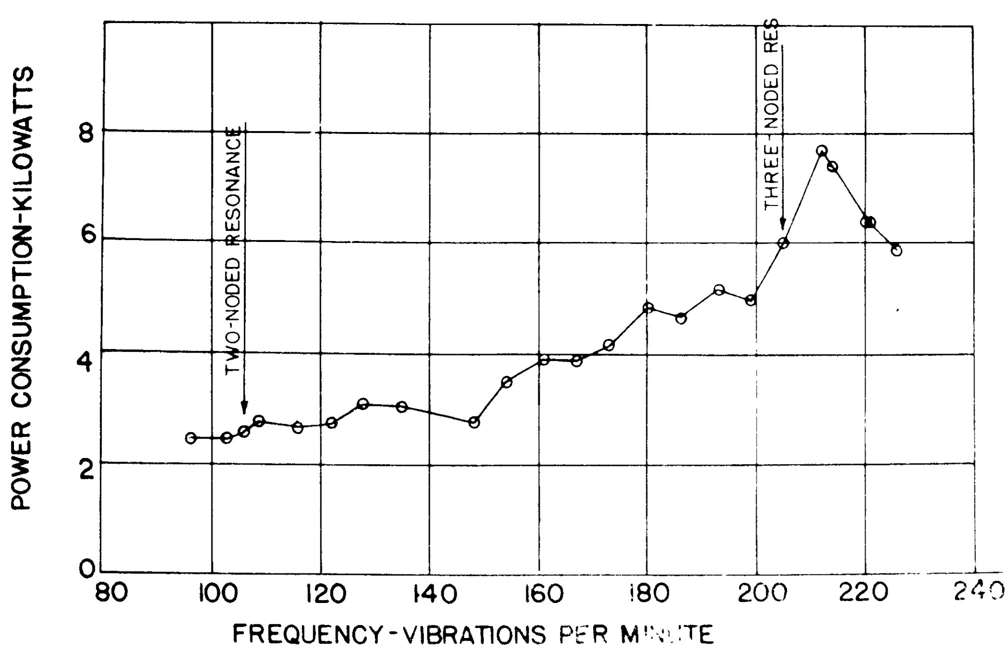


FIG. 7A – VIBRATION TEST – USS HAMILTON, MAY 25TH – POWER VS FREQUENCY

TABLE II

Resonant Speed (rpm by counter)	Eccentricity		Centrifugal Force (lb.)	Power Input (kw.)
	Deg.	Inches		
108	18	1.85	7306	3.9
107	21	2.16	8337	3.8
107	24	2.45	9478	4.0
105.2	27	2.76	10312	4.4
106.5	30	3.05	11712 *	4.8

*The maximum centrifugal force obtainable at this speed is 44,000 lb.

Occurrence of Three Noded Vibration

Reducing the eccentricity to 6 degrees the speed was increased by steps to the highest value that could be obtained, and at 206 rpm three noded vibration appeared. The resonance was quite sharp. The nodes were located at frames 43 (75 feet from the bow), frame 90½ (158 feet from the bow) and frame 147 (257 feet from the bow). The double amplitude at frame 17 was approximately ½ inch (obtained from pallograph records).

At the 6 degree setting power readings were taken over the entire range of speed. The data are given in Table III below, and plotted in Fig. 7a. It will be observed that the peaks of power do not coincide with either the two or three noded resonances.

TABLE III

Speed (rpm)	Power (kw)	Speed (rpm)	Power (kw)
96	2.5	173	4.2
103	2.5	180	4.9
106	2.6	186	4.6
109	2.8	193	5.2
116	2.7	199	5.0
122	2.8	205	6.0
128	3.2	212	7.7
135	3.1	216	5.9
148	2.8	221	6.4
154	3.5	220	6.4
161	4.0	214	7.4
167	3.9		

These speed values include a correction factor of 1.07 applied to the tachometer readings. This factor was obtained by comparison of several tachometer and counter readings made at various speeds. The power readings were taken from the wattmeter and checked against the product of the volt and ampere readings.

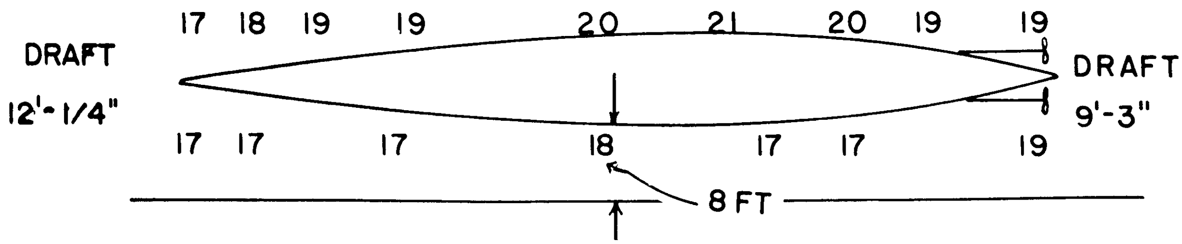


FIG. 8 – VIBRATION TEST – USS HAMILTON, SOUNDINGS – MAY 25, 2:30 P.M.–DEPTH IN FEET

Second Shallow Water Test at Dock

Returning to the dock the test was repeated once more. The depth readings for this test are given in Fig. 8 and are practically the same as on the previous day. With an eccentric setting of 6 degrees power readings were again taken over the whole range of speed. The two noded resonance appeared at 92 rpm which checked with the frequency of 92.2 obtained on the preceding day with the same eccentric setting. Beyond this speed vibration appeared again at 154 rpm and continued up to 244 rpm, the highest speed attainable. While the amplitude appeared to reach a maximum at 205 rpm there was no such indication of a sharp resonance as in the deep water test. This is indicated in Table IV.

TABLE IV

Speed (rpm)	Power (kw)	Remarks	Speed (rpm)	Power (kw)	Remarks
83	1.8	no vibration	167	5.0	slight vibration
92	2.4	vibration	173	5.3	" "
96	2.3	no vibration	180	5.8	vibration
103	2.5	no vibration	186	5.6	"
109	2.7	" "	193	6.3	"
116	3.0	" "	199	6.9	"
122	3.2	" "	205	7.4	max. vibration
128	3.4	" "	212	7.0	vibration
135	3.4	" "	218	8.2	"
141	3.4	" "	225	8.0	"
148	4.1	" "	231	9.0	"
154	4.1	slight vibration	238	8.8	"
161	4.4	" "	244	9.0	"

Computation of Two Noded Frequency

The graphical computation of the two noded frequency is shown in Plates I and II (see reference 4). The approximate method of joining the ends of the shear and bending moment curves was used, and the form of the final deflection curve was found to justify the use of the approximation.

In applying the corrections for shear and rotary inertia in accordance with Taylor's method the following dimensions and ratios for the HAMILTON were used:

length: 310 ft.; beam: 31 ft.; depth 20 ft. 9 inches

$L/D = 15$, $B/D = 1.5$, $L/B = 10$, $D/L = 0.0667$

The total area under the weight curve in Fig. 9 is 267.5 sq. in. corresponding to a total weight of 1382 tons. This represents the displacement during the test and was obtained by correcting the normal weight curve for changes in fuel, water, and ammunition. The displacement estimated from the draft readings was 1422 tons which is about 3% higher.

The midship ordinate on the EY curve (Fig. 10) was 14.25 inches corresponding to end value of 8.86 inches (.608 being the amplitude of the assumed deflection curve at the center relative to the ends). This gives a value for EY at the ends of 1770 ft. tons/inch². Taking E as 13,000 tons/inch² (29,000,000 lb/in²) we get an end deflection value $y = 0.1362$ feet. Since an amplitude of this amount should give us an acceleration equal to "g", we have:

$$4\pi^2 f^2 \times 0.1362 = 32.2$$

Solving for f, we get:

$$f = 2.45 \text{ per sec.} = 147 \text{ per minute}$$

This value for the out-of-water frequency neglects the effects of shear deflection, rotary inertia, and damping of the surrounding water. The shear and rotary inertia factors were taken from the data given by J. L. Taylor (see reference 4). The damping factor was obtained from Taylor's estimate for a destroyer namely that which on the basis of "virtual mass" would amount to 40% of the mass of the ship.

If r = ratio of shear deflection to bending deflection

r' = ratio of rotary kinetic energy to total kinetic energy

r'' = ratio of "virtual mass" of surrounding water to mass of ship,

the frequency obtained from the graphical computation should be divided by $\sqrt{(1+r)(1+r')(1+r'')}$ to obtain the actual frequency. For a ship of the proportions of the U.S.S. HAMILTON, Taylor's data give for r a value of 0.1152 and for r' a value of 0.0323, while r", as indicated above is .40.

Hence by Taylor's method we get a final value for the frequency:

$$n = \frac{147}{\sqrt{1.1152 \times 1.0323 \times 1.40}} = 116$$

as compared with the measured value of 106.7.

If r" the damping factor had been .65 the two values would have agreed. In other words we might say that the experimentally determined value of the "virtual mass" was 65%.

Substituting the proper values in Schlick's formula we have:

$$n = c\sqrt{\frac{35,000}{1382 \times 310^3}} = 106.7$$

and solving for c we find that Schlick's constant for this ship is 1.16×10^5. It is not certain, however, that the depth was sufficient to eliminate the effect of a shallow channel.

Strain Measurements

Strain gage readings taken on the deck plating about amidships at the two noded resonance during the deep water test were as follows:

Eccentricity of masses	Strain Gage Amplitude (double)	Actual Strain, (double)	Bending Stress (lb/in²) ($E=29 \times 10^6$)
21 deg.	4½ s.g. units	.000045 in/in	652
24	5½	.000055	798
27	6	.00006	870
30	7	.00007	1015 *

* If the maximum eccentricity of the machine had been used a stress of 4000 lb/in² could not have been exceeded.

Discussion of the Power Method of Determining Resonance

In using the Spaeth system vibration generator it is intended that resonant frequencies be determined by plotting power-frequency curves and noting the peaks of power consumption (see references 5 and 6). The wattmeter is so connected as to measure the power taken by the armatures only which would include I^2R loss in the armatures, bearing friction, windage, and power taken to maintain vibration. In order to find the latter we must first plot a no load (leerlauf) power curve obtained by clamping the machine to a massive foundation and reading power consumption over the entire speed range with the required eccentricity.

As facilities for clamping this machine to a rigid foundation were not available, a no-load power curve could not be obtained. The power data taken during the vibration test, however, would seem to indicate that the power method cannot be relied upon for determining resonance as there is scarcely any peak in the neighborhood of the two noded resonance and the peak near the three noded resonance occurs at a somewhat higher speed than the actual resonant frequency. This would seem to indicate that the power going into vibration was only a small percentage of the total power taken.

A series of power readings taken after the test at zero eccentricity gave considerably higher values of power consumption over the whole speed range than those obtained during the test except for the peak at about 200 rpm.

This condition might be due to the effect of the centrifugal force in relieving the bearing loads during vibration.

References:

1. J. L. Taylor: "Vibration of Ships", T.I.N.A. 1930, pp. 162–173
2. E. B. Moullin: "The Lateral Vibration of Non–uniform Bars with Application to Ships"
   Cambridge Phil. Soc. Proc. v 24, 1928, pp. 531–53
3. F. H. Todd: "Some Measurements of Ship Vibration", N.E.C.I. 1931
4. J. L. Taylor: "Ship Vibration Periods", N.E.C.I. 1927–8, pp. 143–176
5. R. Bernhard und W. Spaeth: "Rein Dynamische Verfahren zur Untersuchung der Beanspruchungen von Bauwerken"
6. W. Spaeth: "Dynamische Untersuchungen an Schiffen", W.R.H. Vol. XI, No. 5, 1930, pp. 92–93
7. F. M. Lewis: "The Inertia of Water Surrounding a Vibrating Ship" N.A.M.E. Nov. 1930
8. J. J. Koch: "Eine Experimentelle Method zur Bestimmung der Reduzierten Masse des Mitschwingenden Wassers bei Schiffsschwingen" Ingenieur–Archiv IV Band 2 Heft 1933

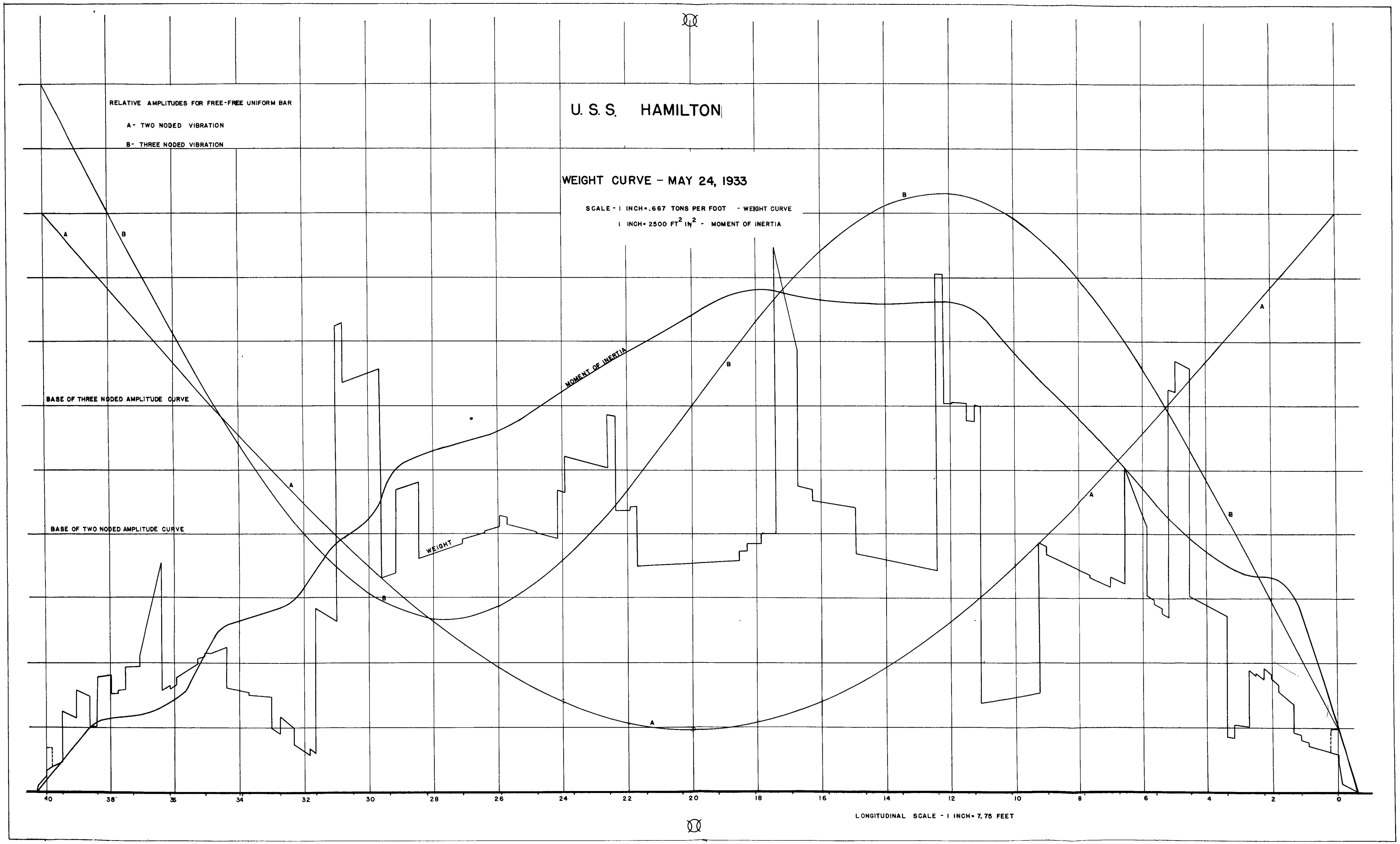

RELATIVE AMPLITUDES FOR FREE-FREE UNIFORM BAR
A- TWO NODED VIBRATION
B- THREE NODED VIBRATION
U. S. S. HAMILTON
WEIGHT CURVE - MAY 24, 1933
SCALE - 1 INCH = .667 TONS PER FOOT - WEIGHT CURVE
1 INCH = 2500 FT² IN² - MOMENT OF INERTIA
BASE OF THREE NODED AMPLITUDE CURVE
BASE OF TWO NODED AMPLITUDE CURVE
MOMENT OF INERTIA
WEIGHT
A
B
40
38
36
34
32
30
28
26
24
22
20
18
16
14
12
10
8
6
4
2
0
LONGITUDINAL SCALE - 1 INCH = 7.75 FEET

PLATE I

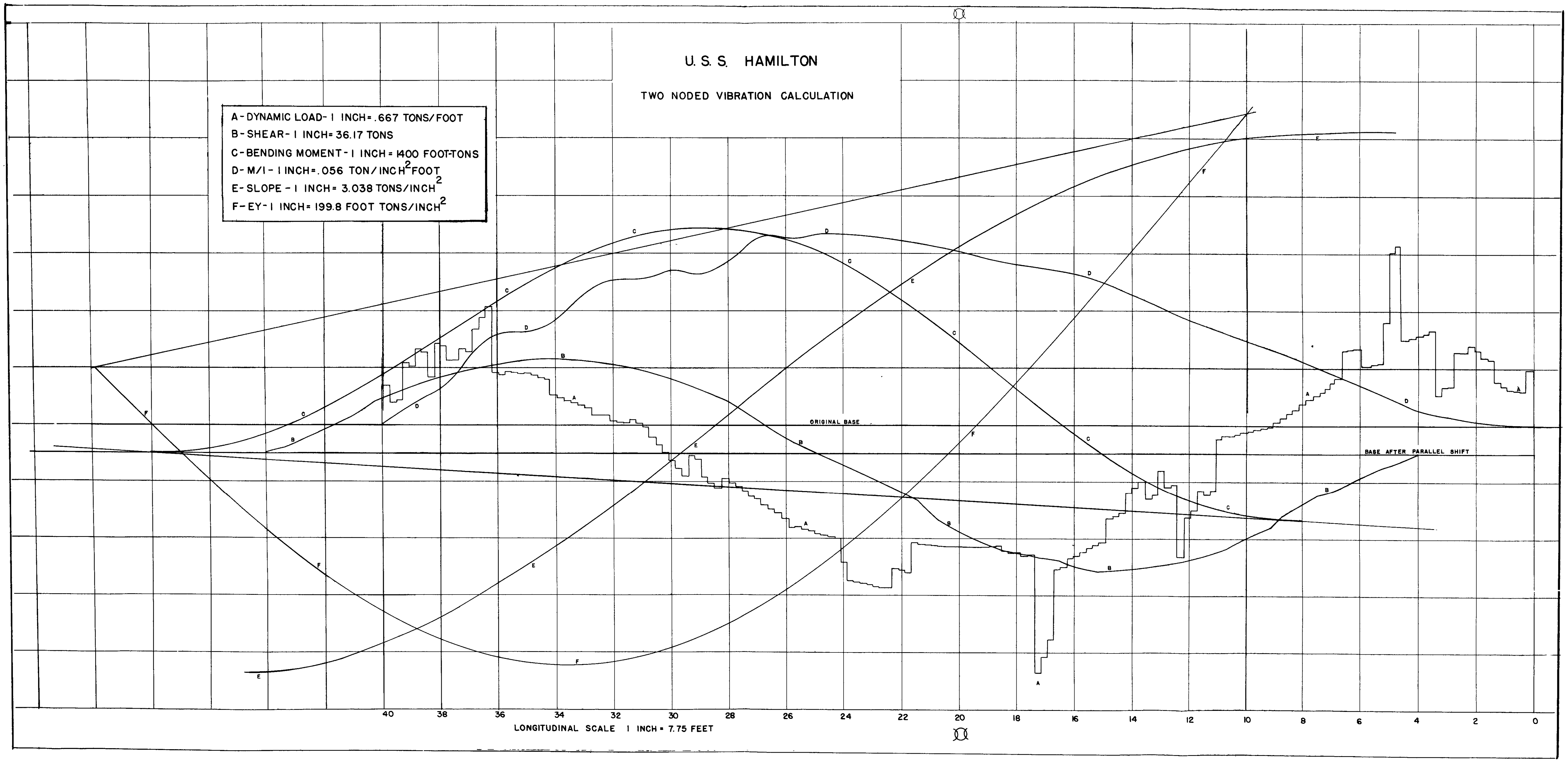

U. S. S. HAMILTON
TWO NODED VIBRATION CALCULATION
A-DYNAMIC LOAD-1 INCH=.667 TONS/FOOT
B-SHEAR-1 INCH=36.17 TONS
C-BENDING MOMENT-1 INCH=1400 FOOT-TONS
D-M/I-1 INCH=.056 TON/INCH² FOOT
E-SLOPE-1 INCH=3.038 TONS/INCH²
F-EY-1 INCH=199.8 FOOT TONS/INCH²
ORIGINAL BASE
BASE AFTER PARALLEL SHIFT
40 38 36 34 32 30 28 26 24 22 20 18 16 14 12 10 8 6 4 2 0
LONGITUDINAL SCALE 1 INCH = 7.75 FEET

PLATE II