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VIBRATION-GENERATOR SURVEY OF THE USS NIAGARA (APA87)

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[redacted]

BY NORMAN H. JASPER

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VIBRATION-GENERATOR SURVEY OF THE USS NIAGARA (APA87)

by

Norman H. Jasper

ABSTRACT

The natural modes of vibration of the hull of the USS NIAGARA (APA87) were determined by using a vibration generator. Five modes of vertical vibration, three modes of athwartship vibration, and one mode of torsional vibration were defined. The experimental natural frequencies of vibration are compared with the natural frequencies calculated on the basis of theoretical considerations. Values of the damping constant are calculated on the basis of the experimental data. It is found that the ratios of the frequencies of the first three modes of vertical and transverse vibration to the frequency of the fundamental mode are approximately as the numbers 1, 2, and 3.

INTRODUCTION

The vibration tests described in this report were authorized by the Bureau of Ships in Reference (1).* The primary purpose of the tests was the determination of the natural modes of vibration of the hull of the USS NIAGARA (APA87), a U.S. Maritime Commission Design S4-SE2-BD1. A knowledge of the modes of vibration is very helpful in analyzing the motion of a ship and of shipboard equipment that are subjected to shock loading. Also, the data obtained from the tests were to be used to check the validity of the several methods available for calculating the natural modes of vibration of ships.

It was the intention during this series of experiments to define as many vertical modes of vibration as could be identified and to determine, in addition, the principal transverse and torsional modes. The vessel's displacement was 5500 tons, corresponding to a mean draft of 12 ft 11 in. The depth of the water at the test site was 140 ft. The test was conducted in Chesapeake Bay off Patuxent, Maryland, during April 1948.

GENERAL CONSIDERATIONS

A relatively small harmonic exciting force, when applied to a massive elastic structure such as a ship, can produce appreciable vibratory motion of the structure. The amplitude of this vibratory motion depends

* Numbers in parentheses indicate references at the end of this report.

principally on the magnitude of the exciting force, the point and frequency of application, and the magnitude and character of damping forces.

The motion of the ship may, in general, be resolved into its normal modes of vibration. When the vessel is vibrating in a normal mode, the effectiveness in that mode of any applied force varies directly as the relative ordinate of the corresponding amplitude profile at the point of application of the force. For a beam, the effect of applying a force at a node is zero, although this force has an effect in any mode which does not have a node at the point of force application, this effect being again proportional to the relative ordinate of the corresponding amplitude profile.

The most effective positions of load application for purposes of vibration-generator tests are at the fore and aft extremities of the vessel; here maximum effectiveness is obtained for all modes of hull vibration. It is necessary to provide a stiff connection between the vibration generator and the main structure of the ship girder in order to minimize bothersome local resonance effects.

For tests of the APA87 a TMB vibration generator was placed on the former gun foundation at the fantail, at Frame 159 1/2, on the upper deck. Owing to its location, the vibration generator, when put into operation, produced horizontal forces which were applied at a considerable distance from the axis of torsion. These forces in turn produced both a twisting moment and a horizontal force. With this arrangement of the vibration generator both torsional and transverse vibrations are excited, and for purposes of the present tests it becomes necessary to separate the effects of the two types of vibration. After these tests were completed, the vibration generator was set up for vertical excitation in such a manner as not to produce twisting moment. The separation may be effected on the basis of the following considerations.

Amplitudes of transverse vibration measured at the torsional axis of the vessel are not affected by torsional vibration. The torsional axis is assumed to be at the intersection of the vertical centerline plane with the neutral plane of the ship girder. The position of this axis was calculated for several stations along the ship and is plotted in Figure 1. It is seen that the position of the axis is not far from the location of the first platform deck over the greater portion of the length of the vessel.

In order to determine the torsional amplitudes of vibration, measurements of vertical motion were made perpendicular to a horizontal plane through the calculated neutral axis. The amplitude of linear motion divided by the distance of the point of measurement from the neutral axis gives the angular motion in radians. It was found that the amplitudes of transverse

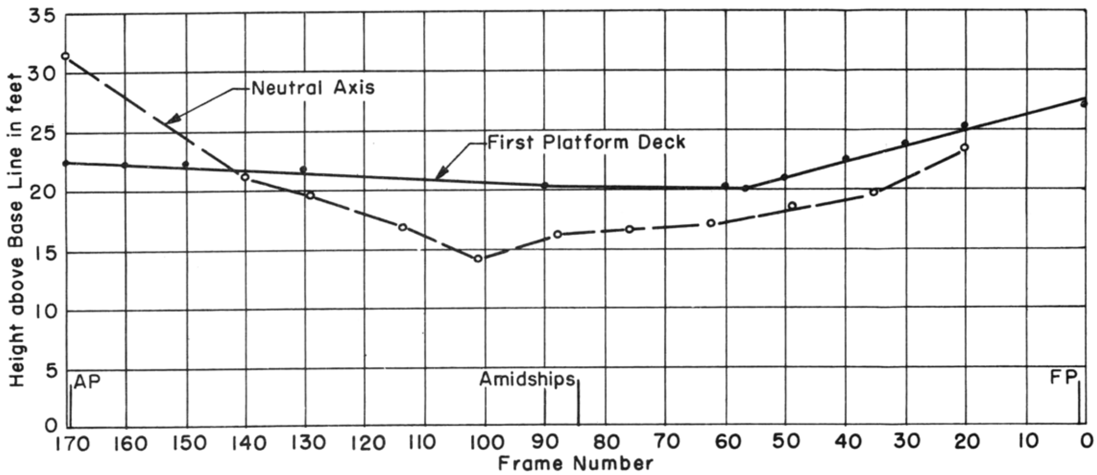

Figure 1 - Location of the Neutral Axis of the APA87

vibration were negligible when the vessel vibrated in its fundamental torsional mode. Also the amplitudes of torsional vibration were negligible when the vessel was vibrating in the flexural modes of vibration.

Experience has shown the necessity of having the depth of water equal to or greater than six times the draft of the vessel in order practically to eliminate the effect of the sea bottom on the vibration characteristics. The depth of water at the test site was ample for this purpose.

TEST PROCEDURE

Initially, the vibration generator, a detailed description of which is given in Reference (2), was installed so as to excite the transverse and torsional modes of vibration; see Figure 2. Amplitudes of vibration were

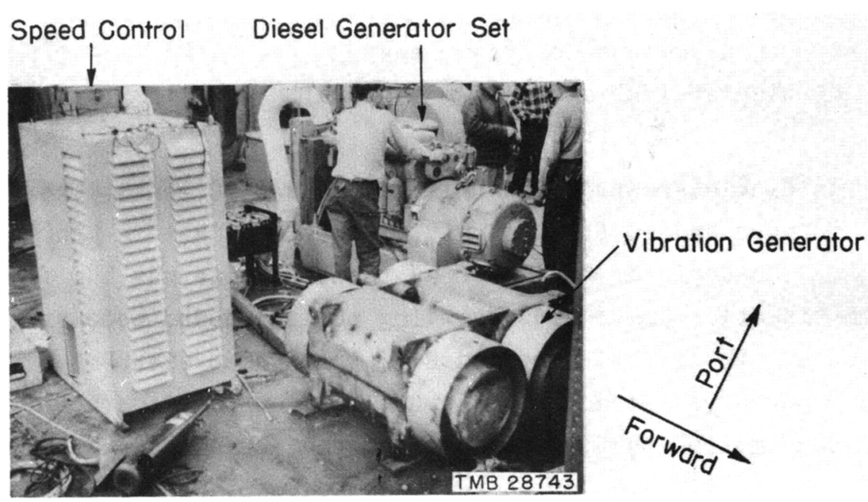

Figure 2 - Installation of TMB Medium Vibration Generator on the APA87

measured at several stations over a range of frequencies from 50 to about 1000 cpm. The power consumed by the vibration generator was measured at each of the test speeds. The frequencies at which the various vibration resonances occurred were determined as accurately as practicable, and the amplitudes at each one of these resonances were measured along the length of the vessel by means of a General Radio vibration meter. Check measurements were made at several stations by means of TMB pallographs. Care was taken to make the measurements in such a manner as to permit separation of torsional and transverse motions. The method used is outlined in the preceding section. In this manner the first three modes of transverse vibration and the fundamental mode of torsional vibration were obtained.

The vibration generator was then oriented to excite the vertical modes of vibration. The range of frequencies from 50 to 800 cpm was covered in a manner similar to that described previously.

The power required for the vibration generator was supplied by a 200-v d-c diesel-generator set. The vibration-measuring instruments used were the General Radio vibration meter, TMB pallographs, and Shure crystal accelerometers, the output of which was integrated and then amplified and recorded on a Brush magnetic-type oscillograph.

TEST RESULTS

The ship tested is shown in Figure 3. Its main characteristics are as follows:

Length overall	426 ft
Length on load waterline	400 ft
Length between perpendiculars	400 ft
Beam, molded	58 ft
Depth, molded to upper deck	37 ft
Displacement at 15-ft 6 in. draft	6740 tons
Mean Draft, full load	15 ft 6 in.

The displacement during this test was 5500 tons. The draft was 9 ft 4 in. forward and 16 ft 6 in. aft, corresponding to a mean draft of 12 ft 11 in. The depth of the water at the test site was 140 ft. The modes of vertical vibration up to and including the six-noded mode are plotted in Figures 4 to 8. The first three modes of transverse flexural vibration are plotted in Figures 9, 10, and 11. Figure 12 shows the character of the vertical vibratory motion that exists between the indicated natural modes of vibration. The exciting force as well as its point of application is

(Text continued on page 10.)

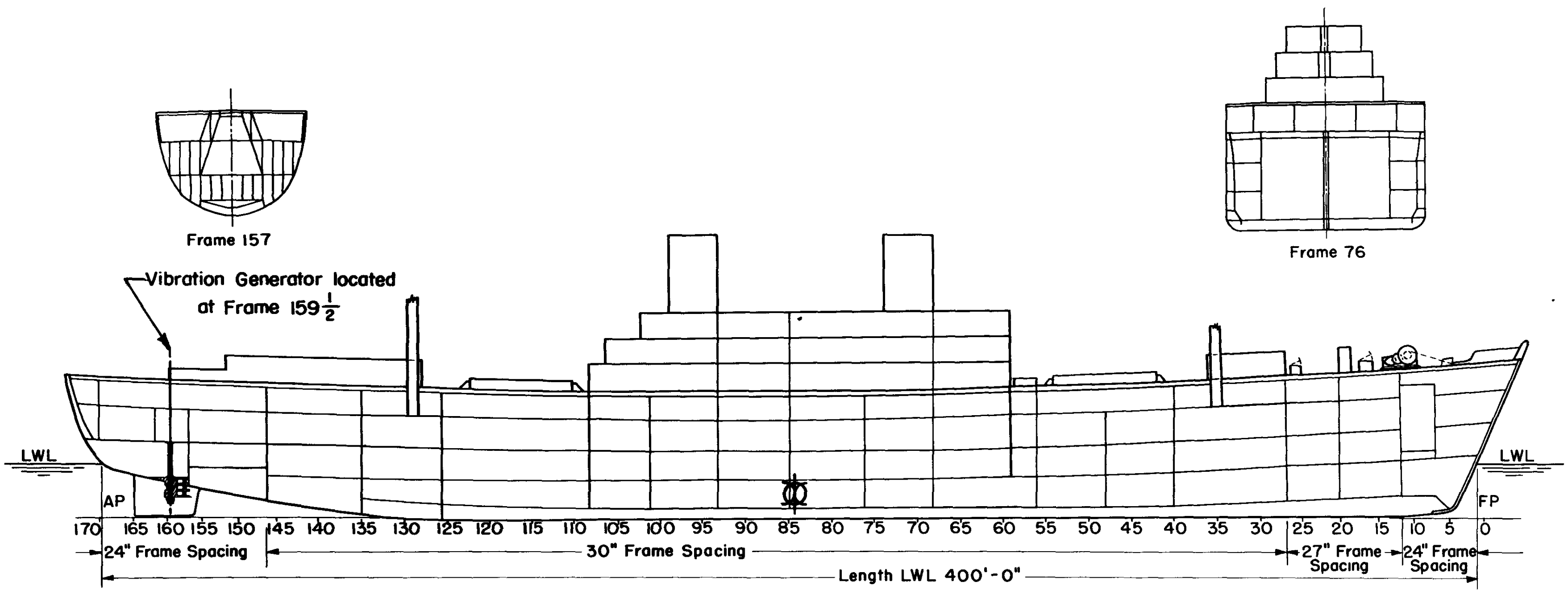

Figure 3 - Inboard Profile and Sections USS NIAGARA (APA87) U.S. Maritime Commission Design S4-SE2-BD1

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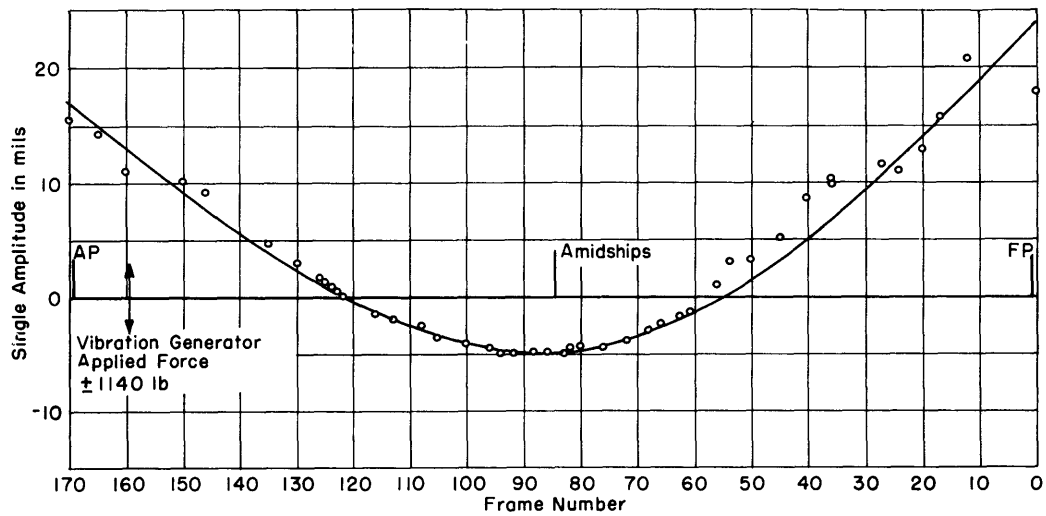

Figure 4 - Two-Noded Mode of Vertical Flexural Vibration
of the APA87 at a Frequency of 110 CPM

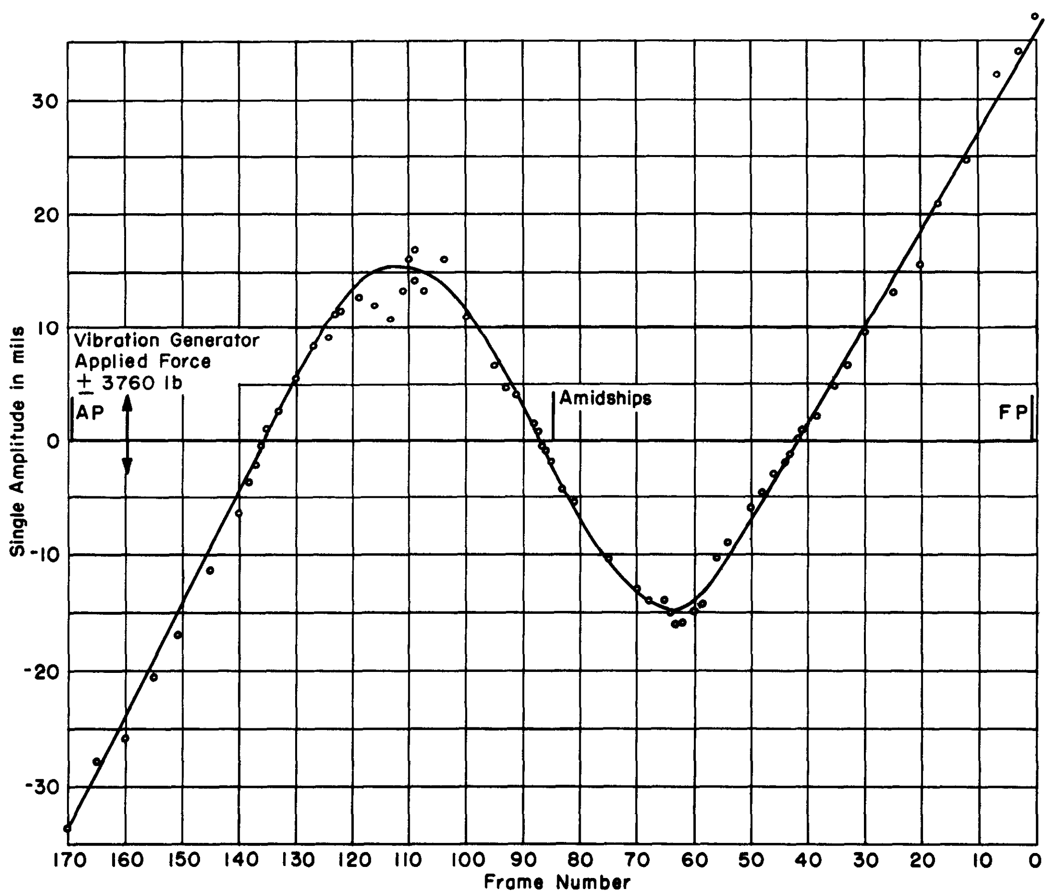

Figure 5 - Three-Noded Mode of Vertical Flexural Vibration
of the APA87 at 200 CPM

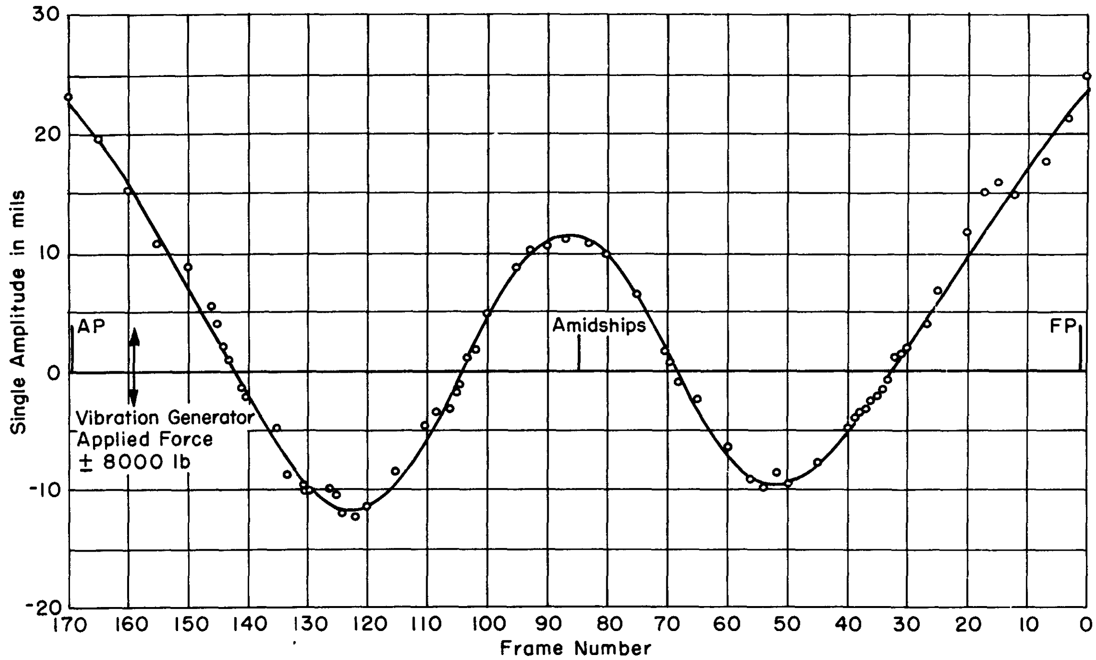

Figure 6 - Four-Noded Mode of Vertical Flexural Vibration of the APA87 at 292 CPM

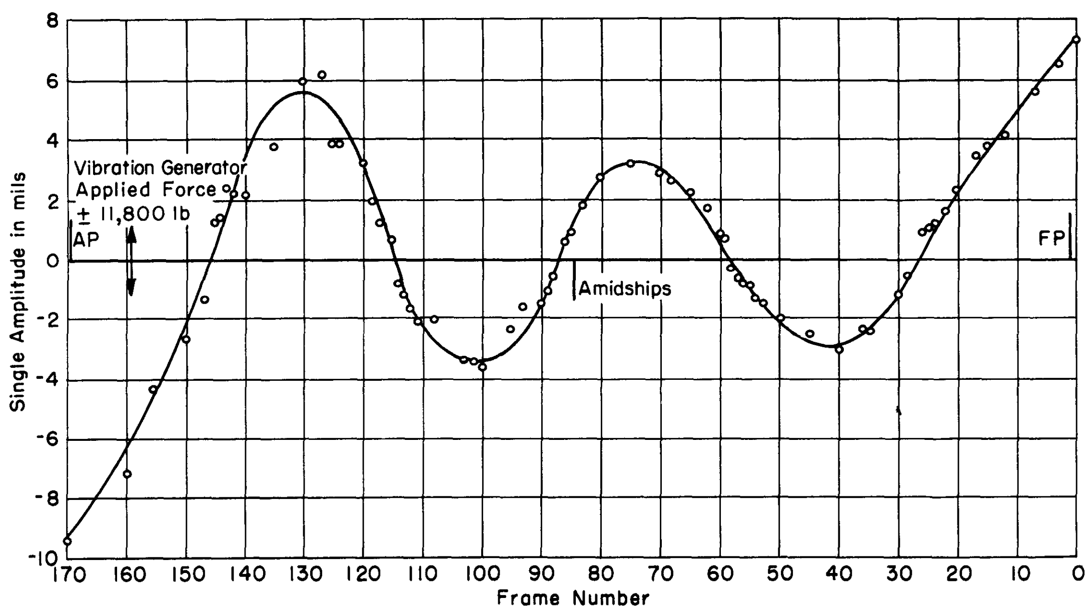

Figure 7 - Five-Noded Mode of Vertical Flexural Vibration of the APA87 at 355 CPM

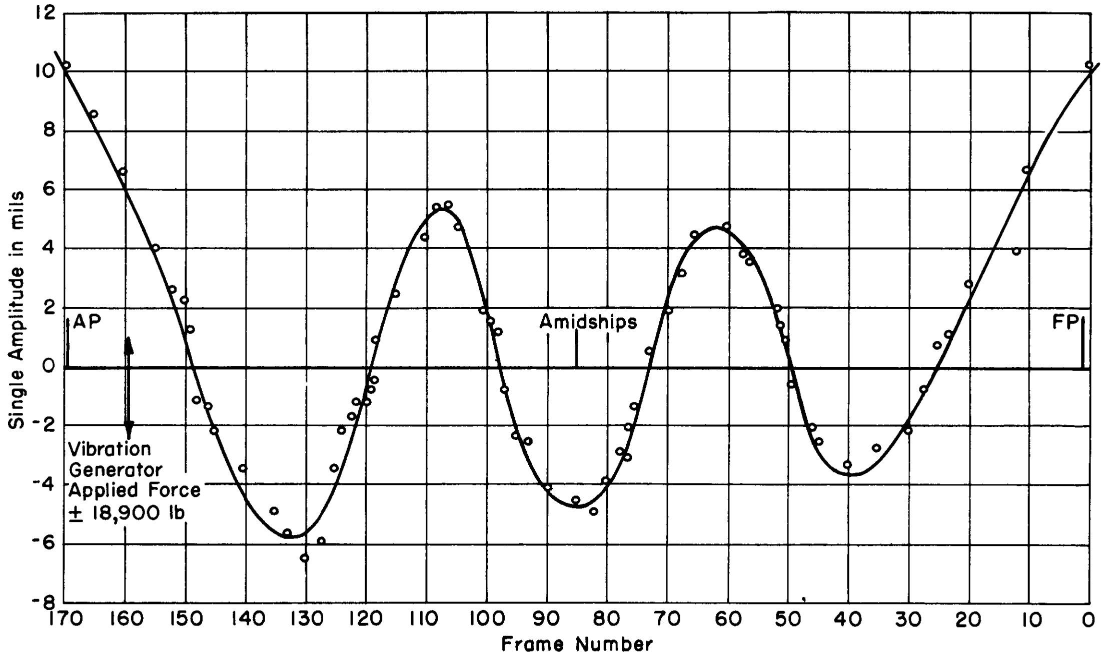

Figure 8 - Six-Noded Mode of Vertical Flexural Vibration of the APA87 at 448 CPM

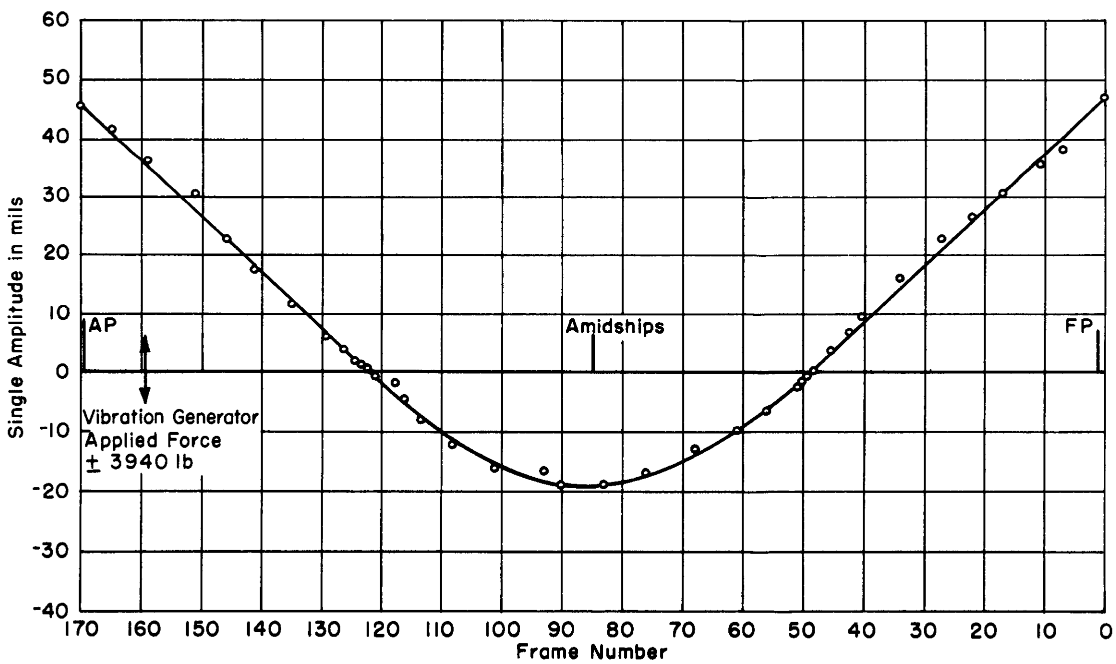

Figure 9 - Two-Noded Mode of Transverse Flexural Vibration of the APA87 at 190 CPM

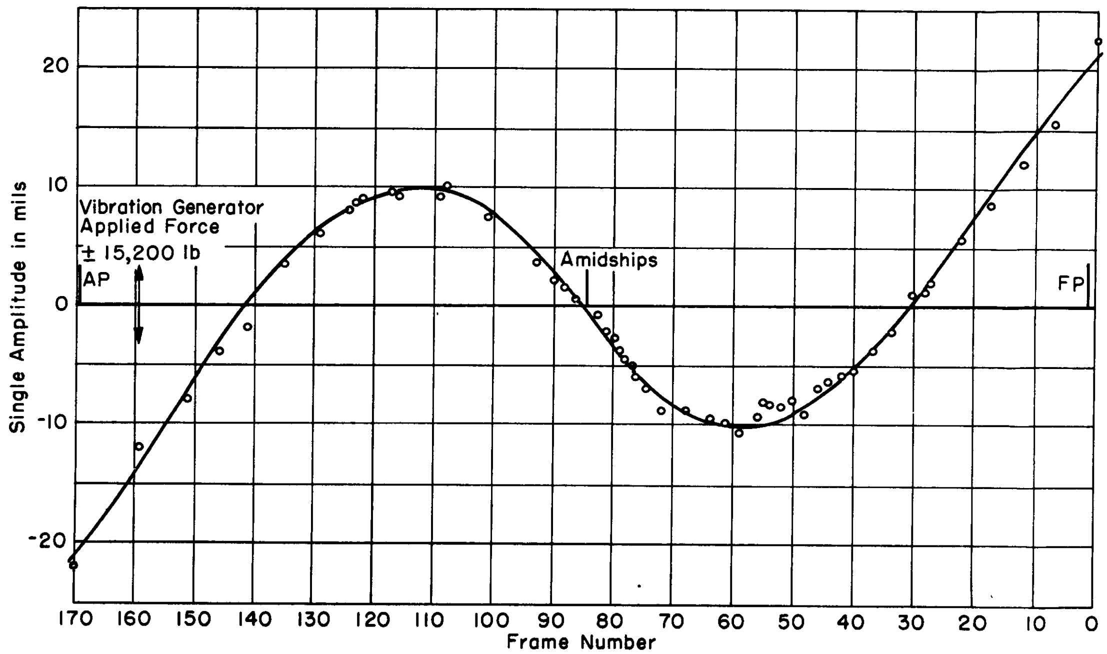

Figure 10 - Three-Noded Mode of Transverse Flexural Vibration of the APA87 at 402 CPM

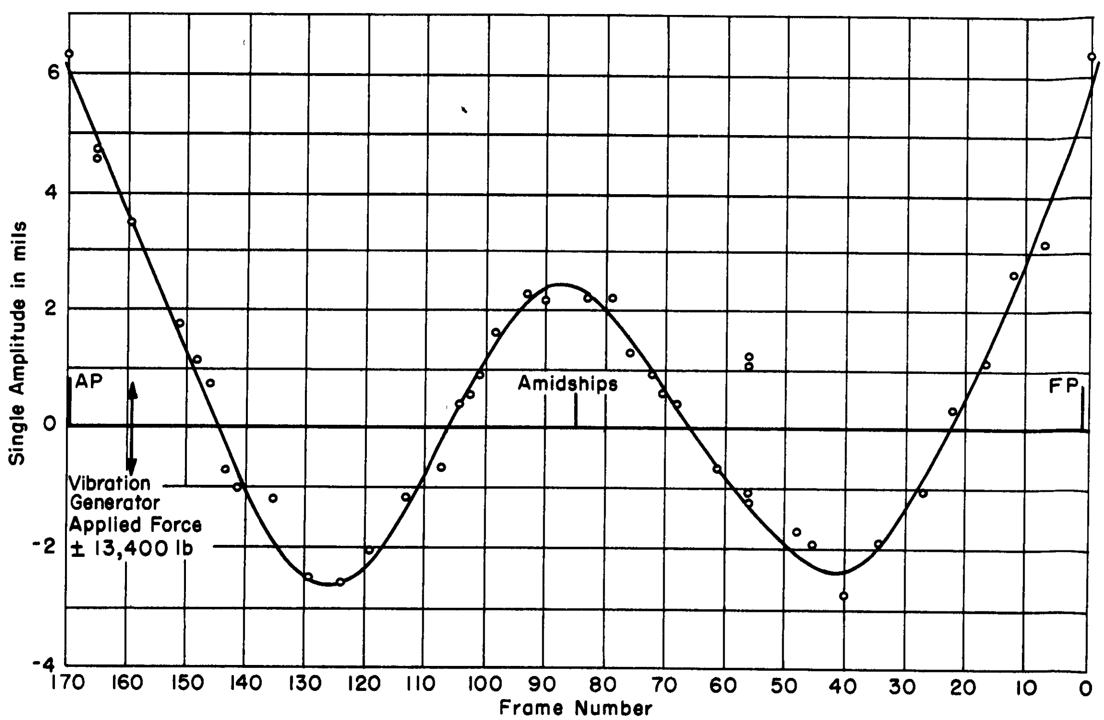

Figure 11 - Four-Noded Mode of Transverse Flexural Vibration of the APA87 at 585 CPM

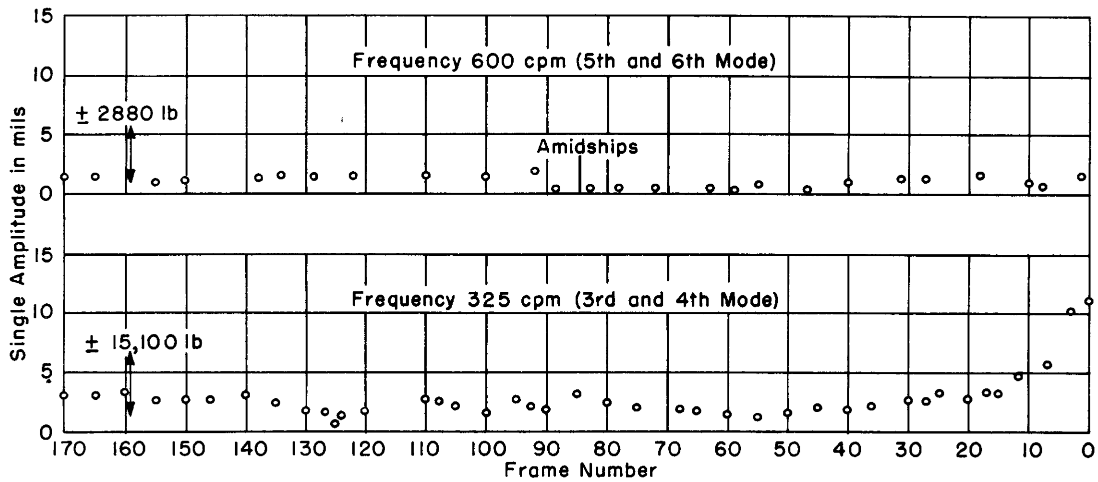

Figure 12 - Amplitudes of Vertical Flexural Vibration of the APA87

indicated on the figures. Figure 13 shows a plot of amplitudes of vertical vibration measured at Frame 3 over the range of frequencies covered. Figure 14 is a similar plot of amplitudes of transverse vibration at Frames 1, 12, and 27. It is interesting to note that definite resonances were not observed at frequencies above 600 cpm, an indication that the vessel does not act as a beam in this higher frequency range, and consequently that any calculations based on beam theory cannot be expected to apply in this range. It is believed that the ship structure, when excited by high-frequency forces, vibrates in many local modes, approaching an infinite-degree-of-freedom system. The fundamental mode of torsional vibration is illustrated in Figure 15.

For purposes of comparison, a plot of the natural modes of flexural vibration of a free-free uniform bar is presented in Figure 16. Table 1 gives the salient characteristics of the several modes of vibration.

An estimate was made of the damping associated with the hull vibration, based on the results of this series of tests. For the purpose of this estimate the damping was assumed to be of the viscous type and the damping coefficient to be of constant value throughout the vessel's length. With these assumptions we may, for any mode of vibration, at resonance, equate the work done by the vibration generator to the work dissipated through damping; see Reference (3).

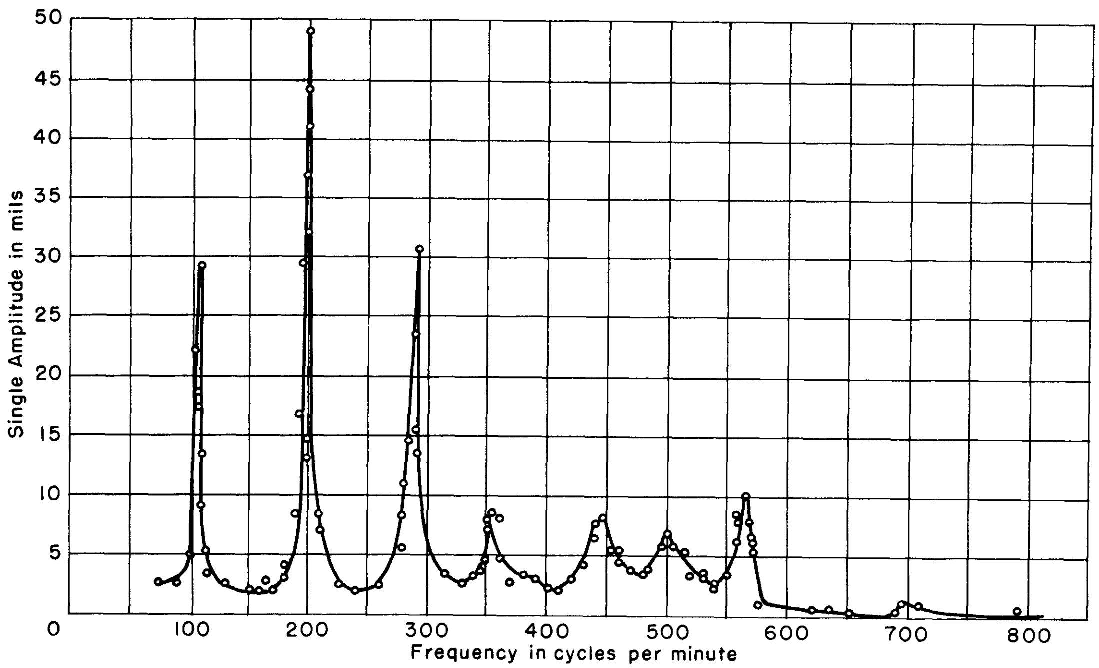

Figure 13 - Resonance Curve for Vertical Vibration of the APA87

The vibration generator was installed at Frame 159 1/2.

The eccentricity setting of the vibration generator was 75° and the force applied was 0.094 $(CPM)^2$, in pounds.

Measurements were made at Frame 3 at the centerline of the main deck.

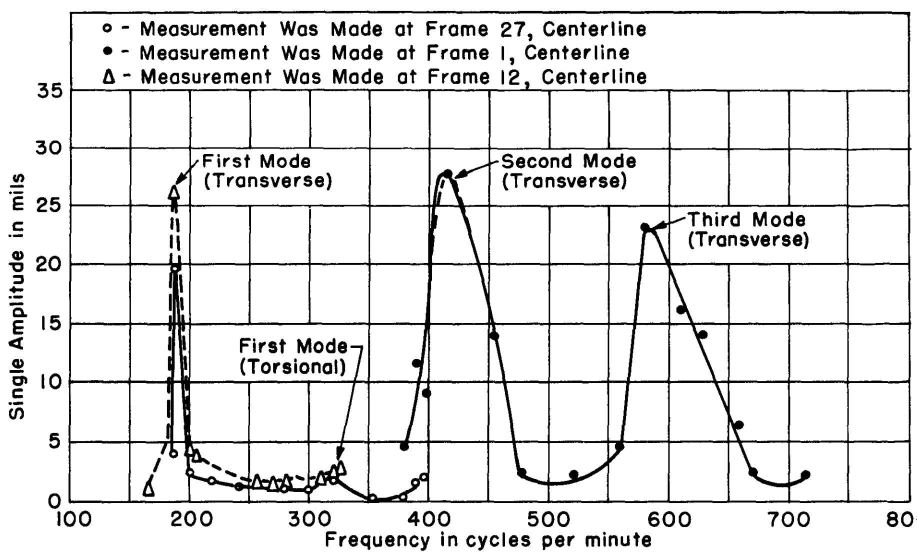

Figure 14 - APA87 Resonance Curve for Transverse and Torsional Vibration

The eccentricity setting of vibrator was 90°; the exciting force was 0.109 $(CPM)^2$, in pounds.

Measurements were made with TMB pallographs.

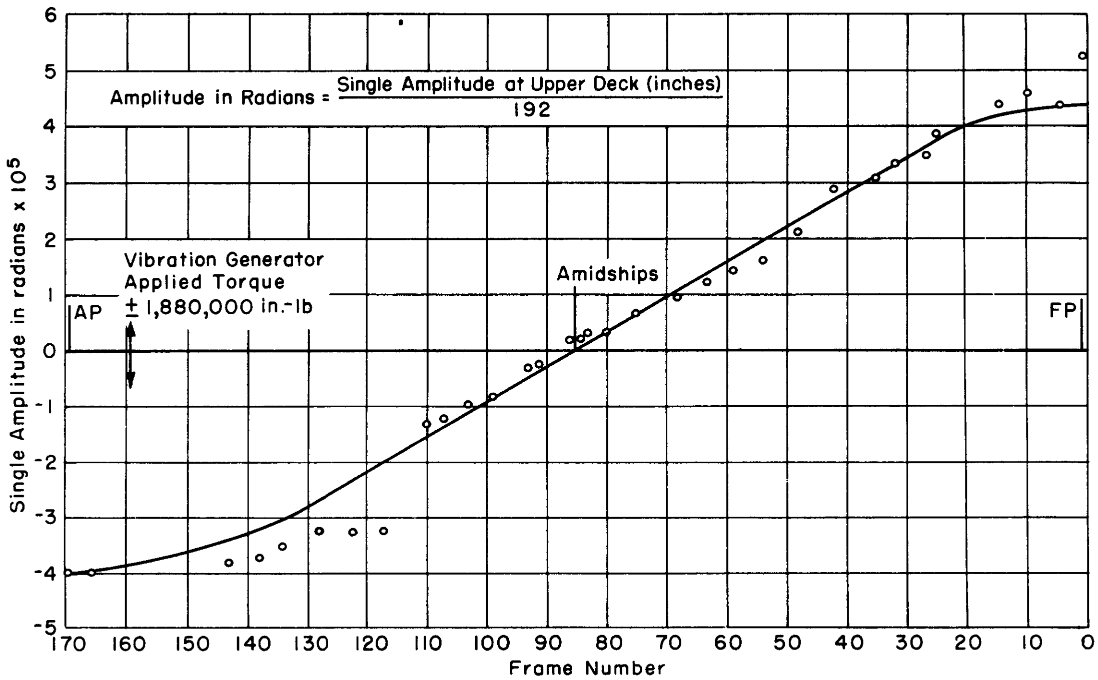

Figure 15 - Fundamental Mode of Torsional Vibration of the APA87 at 322 CPM

TABLE 1

Data Obtained from Vibration-Generator Surveys on APA87

Type of Beam	Type of Vibration	Number of Nodes	Natural Frequency of Vibration CPM	Distance of Node from Amidships Per Cent of Length						Damping Constant lb-sec/in³	
				Aft of Amidships			Forward of Amidships			Based on Projected Area*	Based on Area of Wetted Surface
Free-Free Bar	Flexural	2				27.6	27.6				
		3			36.7	0	0	36.7			
		4			40.5	14.8	14.8	40.5			
APA87	Vertical	2	110			22.4	18.9			0.020	0.0032
		3	200		31.8	1.4	—	26.7		0.024	0.0037
		4	292		35.8	12.1	10.4	32.5		0.048	0.0075
		5	355	38.1	18.8	1.7	—	16.7	36.6	0.182	0.0290
		6	448	39.4	21.4	8.1	7.3	22.0	37.2	0.162	0.0250
	Transverse	2	190			23.1	23.0			0.003	0.0022
		3	402		33.6	0.2	—	35.5		0.017	0.0118
		4	585		37.5	13.3	11.9	38.5		0.047	0.0324
	Torsional	1	322	Damping constant $C = 467 \frac{\text{in-lb-sec}}{\text{Radian-in}^2}$							0.007
Steel Model	Flexural Vertical									0.066	0.018

* These data are based on the projection of the submerged part of the vessel on the plane normal to the vibratory motion.

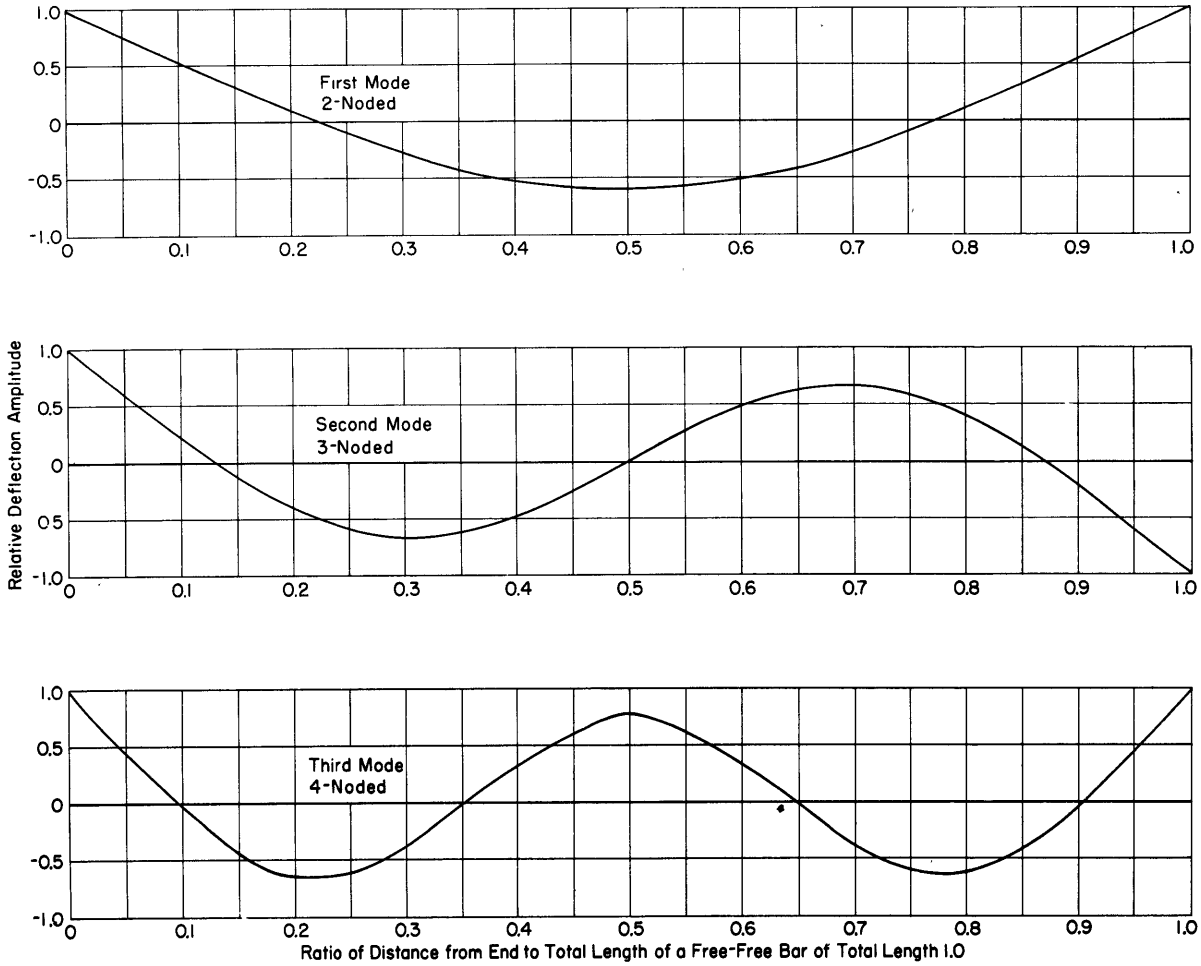

Figure 16 - Natural Modes of Flexural Vibration of a Free-Free Uniform Bar

Work done by vibration generator per cycle = $\pi P_0 y_0$

Work dissipated by damping per cycle = $\int_0^l \pi c' \omega y^2 dx$

where P_0 is the force amplitude applied by the vibration generator,

y_0 is the amplitude at the point at which P_0 is applied,

c' is the damping constant per unit length,

x is the distance from the end of the ship,

y is the amplitude at any point x, and

ω is the circular frequency.

Therefore

$$\pi P_0 y_0 = \pi c' \omega \int_0^l y^2 dx$$

$$c' = \frac{P_0 \, y_0}{\omega \int_0^l y^2 dx}$$

If it is assumed that this damping is primarily external, then a suitable parameter to use in connection with it is the wetted-surface area. Upon dividing c' by the wetted surface per unit of length, a value of c is obtained which gives the effective resistance per unit velocity per unit area of wetted surface. This value can be compared with the value obtained from any other similar tests.

For purposes of comparison, the damping constant was calculated from model test data obtained previously and reported in Reference (4). The rectangular model was made of furniture steel and displaced 3034 lb. Its principal dimensions were as follows: length 18 ft., beam 27 in., depth 17 1/2 in., draft 16 in. The damping constant calculated on the basis of the model test was 0.018 lb-sec/in³. This is of the same order of magnitude as the damping constants derived from the APA87 tests. The damping existing in the fundamental torsional mode of vibration was evaluated in a manner similar to that used for the linear vibrations. The effective damping constant is 467 $\frac{\text{in-lb-sec}}{\text{radian-in}^2}$, that is, the dimensions are in inch-pounds of torque per radian per second of angular velocity per square inch of wetted surface. In order to compare the torsional damping constant with the damping constant for the linear vibrations the following assumption was made:

The cross section of the ship was replaced by a semicircular section whose cross-sectional area was the same as that of the ship. With this assumption the equivalent damping constant per unit velocity of the periphery of the semicircular cross section was calculated to be 0.007 lb-sec/in³. This value of the damping constant is comparable to that obtained for the flexural vibrations. On the basis of these and later tests it appears that, for the lower modes of vibration of ships, the damping factors are of the same order of magnitude. The value of the damping constant is given in Table 1. The maximum vibratory velocity experienced during the APA87 tests was 1 in. per sec. At such low values of velocity, viscous flow should obtain. The magnitude of the hydrodynamic drag and friction forces involved at these velocities is very much less, by a ratio of about one or two hundred to one, than the experimental damping values. It must therefore be concluded that most of this damping is contributed by internal damping of the ship itself and that the wetted surface, which has been used in the foregoing analysis, is not a rational parameter. An inspection of the values given for the damping constant in Table 1 shows that the damping increases appreciably with the higher modes of vibration; this additional damping may be supplied by friction of riveted joints and by hysteresis losses in the steel structure incident to local vibration.

If it is now assumed that the damping is internal rather than external, we may choose as a logical parameter the maximum internal strain energy of the system. An analysis based on the kinetic and strain energies for the first three modes of vertical flexural vibration showed that, for these modes, the energy loss per vibratory cycle is approximately proportional to the maximum strain energy in the ship girder. This energy loss per cycle was found to be approximately 13 per cent of the maximum strain energy. If further analysis of these data and of similar data obtained from tests of other vessels substantiates this pattern, we will have a rational method for treating the damping forces associated with the vibration of ships.

COMPARISON OF CALCULATED AND EXPERIMENTAL NATURAL FREQUENCIES OF VERTICAL HULL VIBRATION

Table 2 shows the natural frequencies as calculated by Prohl's method (5) and by the Rayleigh-Ritz method and as determined experimentally.

TABLE 2

Comparison of Calculated and Experimental Values of Natural Frequencies of Vertical Hull Vibration

These data are based on a displacement of 5500 tons.

Mode	Natural Frequency of Vibration, cycles per minute		
	Experimental	Rayleigh-Ritz Method, Considering Bending Only	Prohl Method, Considering Bending and Shear
2 noded	110	105	100
3 noded	200	214	196
4 noded	292	476	325

An inspection of Table 2 shows that the influence of shear forces must be taken into consideration if any accuracy is to be expected in determining the frequency of the higher modes. It is seen from Table 1 that the ratios of the frequencies of the first three modes of vibration to the frequency of the fundamental mode are 1.0, 1.83, and 2.66 for the vertical vibration and 1.0, 2.12, and 3.07 for the athwartship vibration. It is believed that the factors 1, 2, and 3 applied to the frequency of the fundamental mode of vibration of any conventional ship will give more accurate values of the frequencies of the first three modes of vibration than analytic methods which consider flexure alone. This deduction is substantiated by tests made on ships quite different in form from the APA87; for example, destroyers.

CONCLUSIONS

1. A survey of vibrations made with a vibration generator is an accurate and informative method of determining the natural modes of vibration of ship hulls after the vessel is completed. The vibration amplitude profiles are equivalent to a plot of influence factors for the particular vessel tested; by means of these factors it is possible to predict vibration amplitudes due to any given applied vibratory forces.

2. The first three modes of flexural vibration may be determined with fair accuracy by applying the factors 1, 2, and 3 to the frequency of the fundamental mode, which may be obtained by means of shock excitation of the vessel or by standard calculation, such as Schlick's formula.

3. Nearly all of the damping associated with hull vibration is contributed by internal damping of the ship itself.

4. Resonance frequencies of the hull as a beam were not observed to exist above 600 cpm for the APA87.

PERSONNEL

The tests were conducted by S. Davidson, E.L. Cecil, V.S. Hardy, and Norman H. Jasper. The frequency values given in Table 2 were calculated by Dr. A.N. Gleyzal, E.V. Adams, and A.R. Welch. All are members of the Vibration Branch, Structural Mechanics Department of the Taylor Model Basin. The investigation was made under the direction of Mr. Jasper.

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(1) BuShips CONFIDENTIAL letter S60-(2) (332) of 25 February 1948 to TMB.

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

(3) Den Hartog, J.P., "Mechanical Vibrations," McGraw-Hill Book Co., 1940, pp. 14-19.

(4) McGoldrick, R.T., "A Study of Ship Hull Vibration," TMB Report 395, February 1935.

(5) Adams, E.J., and Welch, A.R., "Calculation of Flexural Critical Frequencies of Ship Hulls by Prohl's Method," TMB Report 582, July 1947