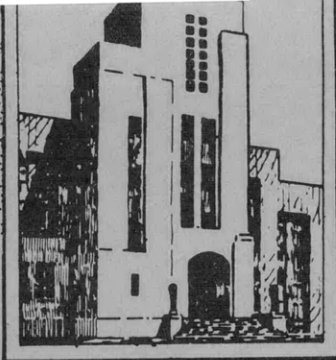


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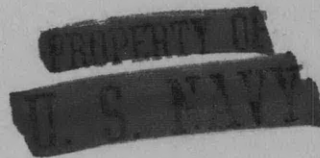
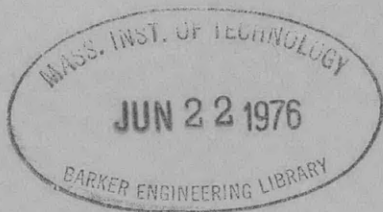
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APPLIED
MATHEMATICS

EFFECTS OF DAMPING ON MODES OF VERTICAL
VIBRATION OF HULL OF USS THRESHER (SSN 593)

by

Ralph C. Leibowitz



STRUCTURAL MECHANICS LABORATORY
RESEARCH AND DEVELOPMENT REPORT

March 1960

Report 1384

11

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TABLE OF CONTENTS

	Page
ABSTRACT	1
INTRODUCTION	1
CALCULATIONS	1
RESULTS	2
DISCUSSION	2
ACKNOWLEDGMENTS	4
APPENDIX – EVALUATION OF PARAMETERS AND DETAILS OF CALCULATION	5
REFERENCES	7

LIST OF FIGURES

	Page
Figure 1 – Vertical Normal-Mode Profiles for Submerged Condition (Determined by UNIVAC)	12
Figure 2 – Vertical Normal-Mode Profiles for Surfaced Condition (Determined by UNIVAC)	12
Figure 3 – Vertical Displacement Response at Selected Stations for a 1-Ton Force Applied at Stern for Submerged Condition (Determined by UNIVAC)	13
Figure 4 – Vertical Displacement Response at Selected Stations for a 1-Ton Force Applied at Stern for Surfaced Condition (Determined by UNIVAC)	13
Figure 5 – Vertical Vibration Amplitude and Phase versus Length for a 1-Ton Force Applied at Stern for Submerged Condition (Determined by UNIVAC)	14
Figure 6 – Vertical Vibration Amplitude and Phase versus Length for a 1-Ton Force Applied at Stern for Surfaced Condition (Determined by UNIVAC)	15

LIST OF TABLES

	Page
Table 1 – Principal Characteristics of USS THRESHER (SSN 593)	8
Table 2 – Mass and Stiffness Data	9
Table 3 – Natural Frequencies and Frequencies Corresponding to Maximum Steady-State Response at Station 0	10
Table 4 – Weight Distribution Data	11

NOTATION

A	Cross-sectional area consisting only of members which take up compressive or tensile loads in bending in sq ft
A'	Cross-sectional area of hull in sq ft
b	Radius of hull (submerged case); half-breadth of the section at the waterline (surfaced case) in ft
C	Lumped damping constant, $C = c\Delta x$ in ton-sec/ft
C'	A coefficient whose value depends on the form of the cross section
c	Damping constant per unit velocity per unit length in ton-sec/sq ft
E	Modulus of elasticity in tension and compression in tons/sq ft
EI	Flexural rigidity of hull in ton-sq ft
G	Modulus of elasticity in shear in tons/sq ft
g	Acceleration due to gravity in ft/sec ²
I	Sectional area moment of inertia in ft ⁴
I_{mz}	Mass polar moment of inertia of a section of hull Δx long taken about a horizontal axis through its center of gravity in ton-sec ² -ft
J	Longitudinal inertial coefficient, applied to correct for fact that motion of water is not confined to transverse planes
K	Shear flexibility factor
KAG	Shear rigidity of beam in tons
L	Length of ship in ft
M	Total mass of hull including virtual mass in ton-sec ² /ft
m	Lumped mass of a section of beam of length Δx (including virtual mass) in ton-sec ² /ft
n	Station number
P	Amplitude of sinusoidal force applied to ship in tons
W_v	Virtual weight per unit length of hull in tons/ft
Δx	Length of element $L/20$ in ft
Y	Absolute single amplitude in mils
ρ	Density of water in tons/ft ³
μ	Mass per unit length in ton-sec ² /ft ²
ω	Natural circular frequency of beam; forcing frequency on beam in rad/sec and cpm

ABSTRACT

The normal modes of vertical flexural vibration of the hull and the steady-state damped response were calculated by means of a digital computer (UNIVAC) for USS THRESHER (SSN 593), for both submerged and surfaced conditions. The methods and data used in making the calculations and the results obtained are presented in this report. Results show that the damping causes appreciable phase changes, at resonance, of the vibration response along the beam; furthermore, the frequencies of peak response and of the corresponding normal modes agree closely for the lower modes, but for the higher modes, the frequency of peak response is greater than that of the corresponding normal mode.

INTRODUCTION

One requirement for designing the propulsion machinery and other equipment to be installed in a ship is a knowledge of the resonance response of the hull. To satisfy this requirement for the design of equipment to be installed on USS THRESHER (SSN 593), the Bureau of Ships¹ requested the David Taylor Model Basin to calculate the normal modes of vibration and the steady-state response to a sinusoidal force of the hull for both submerged and surfaced conditions. The response was computed on the assumption that damping increased with frequency. The calculations were restricted to vertical vibration since experimentally determined and calculated frequencies for submarines show that there is little difference between vertical and athwartship values.

The data used for the calculations are presented in this report. The results are plotted for both submerged and surfaced conditions.* The report discusses the effect of damping and of nonresonant modes on the frequencies corresponding to peak forced response and other interesting features.

CALCULATIONS

The method used for the calculations of normal-mode frequencies and profiles was that outlined in References 3 and 4. For the forced response calculations, the method of References 4 and 5 was used.** In each case the problems were coded for the UNIVAC (digital computer) which was used to carry out the calculations.

The principal characteristics of THRESHER are shown in Table 1.

¹References are listed on page 7.

*The calculations were forwarded to the Bureau of Ships in Reference 2.

**In future calculations at the Taylor Model Basin, the method of lumping the parameters will conform to the procedure outlined in Reference 6 rather than that in Reference 3.

Detailed data on the masses, section moments of inertia, and section areas were obtained from Reference 7 and were supplemented by additional data from the Bureau of Ships. The data were plotted as a function of position along the length of the ship, and smooth curves were faired through the plotted points. The masses and stiffnesses were determined from the curves at 20 or 21 stations along the length. The virtual masses for both submerged and surfaced conditions were calculated as outlined in the Appendix. The data used for the vibration calculations are presented in Table 2.

Table 2 also indicates under P the stations at which a sinusoidal force of 1 ton was assumed to be applied. The steady-state response to this force was calculated assuming a distributed damping $c = 0.03 \mu\omega$ ton/ft per sec/ft, considered a suitable value in Reference 3 on the basis of the available experimental evidence. The calculations were made for values of ω which correspond to the calculated normal-mode frequencies and to 50, 60, 70, 80, 90, and 100 percent of the maximum blade frequency (1000 cpm).

RESULTS

The principal results of the calculations are presented in Figures 1 through 6.

The normal-mode profiles of vertical vibration are drawn for six modes in Figures 1 and 2 for both the submerged and surfaced conditions, respectively. On the basis of experience,⁸ it is believed that not more than six vertical modes will be significant since the length-to-depth ratio of THRESHER is less than 18.

The heaving mode is not shown. It is characterized by the position of the center of gravity, including entrained mass, given in Table 1.

The vertical displacement responses at selected stations to a 1-ton force at Station 0 (the stern) for the submerged and surfaced conditions are shown in Figures 3 and 4, respectively.

The frequencies associated with the maximum amplitudes measured at Station 0 and the corresponding normal-mode frequencies are tabulated in Table 3 for both the submerged and surfaced conditions.

In Figures 5 and 6 are plotted the maximum vertical displacements and phases along the length for a 1-ton sinusoidal force at the stern for both the submerged and surfaced conditions, respectively. The frequencies for which the computations were made include the 12 normal-mode frequencies (whose profiles are shown in Figures 1 and 2) and 50, 60, 70, 80, 90, and 100 percent of the maximum blade frequency. The phase angle ϕ is defined as the angle by which the driving force leads the displacement at a station.

DISCUSSION

The normal-mode profiles shown in Figures 1 and 2 indicate that similar characteristics are exhibited by the submerged and surfaced submarine. A study of Tables 2 and 4 shows that the ratio of the total mass to EI is relatively large in the vicinity of Stations 1

and 2, resulting in considerable local distortion at these stations in the four-, five-, and six-noded mode profiles. For further discussion of similar effects, see Reference 9.

Figures 3 and 4 show that corresponding response curves, obtained from the forced vibration calculations, for the submerged and surfaced cases are generally similar. In the well-defined resonance profiles for Station 0 (Figures 3a and 4a), the amplitude at low frequencies is associated with rigid-body motions. Table 3 shows that the frequencies of peak response and the corresponding normal-mode frequencies agree closely at low frequencies, but at high frequencies the frequency of peak response is greater than the normal-mode frequency. This behavior is at first surprising because, for a single-degree-of-freedom system, damping lowers the resonance frequency.¹⁰ On further examination¹¹ it becomes apparent that there are two opposing influences which affect the frequency of peak response, namely, (1) the effect of damping which tends to shift the peaks to lower frequencies and (2) the contribution, at each resonance frequency, of appreciable response from nearby nonresonant modes. These nearby modes make a proportionally greater contribution to the total response at high frequencies because the resonances are bunched more closely at higher frequencies and the corresponding frequency ratios are closer to unity for the higher frequencies. As a result the peak response is shifted to higher frequencies.

Figures 3 and 4 show that the magnitude and sharpness of the peaks of the resonance curve decrease as the frequency increases. This is due to the assumed increase of damping with frequency or mode number. The figures also show a rapid decrease in response with increase in distance to the point of excitation.

The amplitude and phase profiles obtained from the forced vibration calculations (Figures 5 and 6) show the predominance of the pattern of the nearest normal mode in the lower frequency range and the trend towards concentration of the response at the driving point (Station 0). Alternative explanations for the phenomenon are given on page 12 of Reference 12 and on page 6 of Reference 13. According to Reference 12 the phenomenon occurs because components of the amplitude in each mode are in phase at the stern but do not reinforce each other throughout the ship. In Reference 13 it is argued that strong concentrations of the response near the forcing point are largely the result of the high damping which increases with frequency. Figures 5 and 6 also show that the damping causes appreciable phase changes, at resonance, of the vibration response along the beam.

To check the UNIVAC computations the two-noded vertical flexural frequency was computed for the *surfaced* condition by Schlick's formula.¹⁰ The result was higher than the corresponding UNIVAC computation by about 10 percent.

As a further check, the two-noded vertical flexural frequency was computed for the *submerged* condition from the experimental value for USS NAUTILUS (SSN 571) by the equation for the uniform beam:

$$\omega_{593} = \frac{\sqrt{\frac{I_{593}}{\mu_{593} L_{593}^4}}}{\sqrt{\frac{I_{571}}{\mu_{571} L_{571}^4}}} \omega_{571} = \frac{\sqrt{\frac{1750 \text{ ft}^4}{\left(\frac{266 \text{ ton-sec}^2/\text{ft}}{278.5 \text{ ft}}\right) (278.5 \text{ ft})^4}}}{\sqrt{\frac{1337 \text{ ft}^4}{\left(\frac{256 \text{ ton-sec}^2/\text{ft}}{319.5 \text{ ft}}\right) (319.5 \text{ ft})^4}} (120 \text{ cpm}) = 165.6 \text{ cpm}$$

Here I is the midship section value and μ is the total mass of the hull including the virtual mass divided by the length of the ship. The result is lower than the corresponding UNIVAC computation by about 5½ percent. Published data on the intensity and the natural frequencies and modes of submarine hull vibration are scant. Pertinent data available at David Taylor Model Basin are summarized in Reference 14. The results obtained will serve to check hull vibration theory when corresponding experimental results for THRESHER become available.

ACKNOWLEDGMENTS

The author wishes to express his appreciation to Dr. N.H. Jasper and Dr. W.J. Sette for their critical and constructive review of this report. Mr. A. Rotsko of General Electric Company evaluated the parameters for the submerged condition.

APPENDIX

EVALUATION OF PARAMETERS AND DETAILS OF CALCULATION*

The ship parameters are evaluated here in accordance with the procedure given in Reference 3.

The total length L of the submarine (278.5 ft) was divided into 20 sections, each 13.925 ft long. Station 0 was taken at the extreme aft end.

The total mass (including the virtual mass) of the ship was lumped at 21 stations of the ship; a tabulation of the lumped masses (in ton-sec²/ft) for the submerged and surfaced submarine is given in Table 2. For the *completely submerged condition* the virtual weight per unit length of the hull was calculated using the following expression:^{3,15}

$$W_v = J\rho A' C' \text{ ton/ft}$$

where $\rho = 0.032 \text{ ton/ft}^3$,

$J = 0.79$ (for $L = 278.5 \text{ ft}$ and $B = 31.65 \text{ ft}$),

$b =$ radius of hull in ft,

$A' = \pi b^2 \text{ ft}^2$, and

$C' = 1.0$.

For the *surfaced condition* the virtual weight per unit length of the hull was calculated using the equation:³

$$W_v = \frac{1}{2} (JC' \pi \rho b^2) = \frac{1}{2} J\rho A' C' \text{ ton/ft}$$

where ρ and J have the same values as for the submerged condition,

A' is defined as before,

$J = 0.79$,

b is the half-breadth of the section at the waterline in feet, and

C' is a coefficient depending on form of section.

The lumped virtual weights of the hull are given in Table 4.

For the *submerged condition* the diving planes and fairwater planes were treated as very stiff members, and the associated lumped virtual weights shown in Table 4 were calculated as above with $J = C' = 1$ and $b = 1/2$ the major diameter of the ellipse forming the cross section of the planes.

For the *surfaced condition* only the planes associated with the virtual weights lumped at Stations 1 and 2 remain submerged. Hence only these virtual weights are included in the data tabulated in Table 4.

*Symbols used in this section are defined under Notation.

For the submerged and surfaced conditions, the total lumped masses obtained by dividing the total lumped weights given in Table 4 by $g = 32.2 \text{ ft/sec}^2$ are tabulated in Table 2.

The term $\Delta x/EI$ was calculated as an average value extending between the midpoints of two successive spans. The element length Δx was taken as the distance from the middle of one span to the middle of the next and equal to 13.925 ft for all sections except Stations 0 and 20. For these points, $\Delta x = 6.963 \text{ ft}$ was used. The corresponding moment of inertia I was the average moment of inertia for the span in question. In the calculation of I , only members which take up compressive or tensile loads in hull bending were considered. The effects of frames, floors, stringers, tanks, etc., were neglected. The modulus of elasticity in tension and compression was taken as $1.93 \times 10^6 \text{ ton/ft}^2$.

The term $\Delta x/KAG$ was calculated as an average value between stations with $\Delta x = 13.925 \text{ ft}$ for all sections. Since the hull is circular in cross section, K was taken as $\frac{1}{2}$. The modulus of elasticity in shear G was taken as $0.77 \times 10^6 \text{ ton/ft}^3$. The area A was taken as the average cross-sectional area for the section being considered. In the calculation of the area A , only members which would take up compressive or tensile loads in bending were considered.

The rotary inertia term I_{mz} was neglected since its effect upon hull vibrations is known to be small.⁶

For the forced-vibration calculation, the identical parameters m , $\Delta x/EI$, and $\Delta x/KAG$ are tabulated as for the free-vibration computation. In addition, a lumped damping constant C which increases with frequency^{3,16} is also tabulated (see Table 2) by using the following equation

$$\frac{c}{\mu\omega} = \frac{\frac{C}{\Delta x}}{\frac{m}{\Delta x} \omega} = 0.03$$

Hence

$$\frac{100C}{\omega\Delta x} = \frac{3m}{\Delta x}$$

where C is the lumped damping constant of section Δx in ton-sec/ft,

c is the distributed damping constant in ton-sec/ft²,

m is the lumped mass of section Δx in ton-sec²/ft,

μ is the mass per unit length in ton-sec²/ft²,

Δx is the selected span length (13.925 ft), and

ω is the forcing frequency on the beam in rad/sec.

Steady-state vertical response calculations were made for simple harmonic driving forces located in turn at Stations 0, 4, 5, 6, and 7, respectively. This was accomplished by setting

$P = 1$ ton (see Table 2) at the station of interest (all other P 's = 0), and running a set of calculations for the frequency values in the range of interest.* This was repeated for each station of interest. Separate calculations were made for a 1-ton force located at each of the specified stations (0, 4, 5, 6, and 7) in order to evaluate the effect of the force at each individual location.

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*For $P = 1$ at Station 0, ω ranged from $\omega = 1$ rad/sec to $\omega = 120$ rad/sec in steps of 1 rad/sec. For $P = 1$ at Stations 4 or 5, or 6 or 7, ω ranged from 1 rad/sec to 120 rad/sec in steps of 2 rad/sec. This range was based upon a top shaft speed of 200 rpm which for a five-bladed propeller gives a blade frequency of 1000 cpm or a circular frequency of 100 rad/sec.

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TABLE 1

Principal Characteristics of USS THRESHER (SSN 593)

Length between perpendiculars	278 ft 6 in.
Beam	31 ft 8 in.
Depth	31 ft 8 in.
Design draft (conditions "N" normal diving term)	
Forward at forward perpendicular	22 ft 5 1/8 in. (above molded baseline)
Mean	24 ft 10 3/4 in. (above molded baseline)
Aft at after perpendicular	27 ft 4 3/8 in. (above molded baseline)
Design displacement (submerged)	4332 tons
Design displacement (surfaced)	3745 tons
Longitudinal center of gravity (submerged)	9.63 ft forward midperpendicular
Longitudinal center of gravity (surfaced)	7.37 ft forward midperpendicular
Longitudinal center of gravity, including virtual mass (surfaced)	7.10 ft forward midperpendicular

TABLE 2

Mass and Stiffness Data

n	$W_{\text{submerged}}$ ton-sec ² ft	W_{surfaced} ton-sec ² ft	$\frac{\Delta x^*}{EI} \times 10^8$ $\frac{1}{\text{ton-ft}}$	$\frac{\Delta x}{KAG} \times 10^6$ ft/ton	P^{**}	$\left(\frac{3m}{\Delta x}\right)_{\text{submerged}}$ ton-sec ² ft	$\left(\frac{3m}{\Delta x}\right)_{\text{surfaced}}$ ton-sec ² ft
0 (Stern) ½	0.274	0.373	18.04	32.88	1	0.118	0.160
1 1½	4.129	3.741	8.49	10.96	0	0.885	0.806
2 2½	7.319	5.211	4.01	6.70	0	1.572	1.123
3 3½	6.914	4.803	2.09	4.70	0	1.490	1.035
4 4½	10.083	6.253	1.12	3.53	1	2.160	1.347
5 5½	12.435	8.447	0.679	2.87	1	2.675	1.820
6 6½	14.194	9.571	0.493	2.54	1	3.040	2.062
7 7½	15.186	10.840	0.429	2.13	1	3.260	2.335
8 8½	17.892	11.385	0.553	3.55	0	3.840	2.453
9 9½	16.796	11.932	0.768	2.62	0	3.600	2.571
10 10½	20.703	15.584	0.413	2.22	0	4.450	3.357
11 11½	24.474	19.834	0.372	2.38	0	5.260	4.273
12 12½	18.333	13.437	0.381	2.38	0	3.940	2.895
13 13½	15.927	10.621	0.382	2.38	0	3.425	2.288
14 14½	16.004	11.946	0.382	2.38	0	3.440	2.574
15 15½	18.621	12.362	0.379	2.13	0	4.010	2.663
16 16½	17.960	10.766	0.617	3.73	0	3.860	2.319
17 17½	13.659	8.578	1.23	3.81	0	2.935	1.848
18 18½	9.548	5.632	1.41	4.64	0	2.050	1.213
19 19½	5.035	3.360	2.41	13.91	0	1.080	0.724
20 (Bow)	0.885	0.435	6.01		0	0.380	0.188

* $\Delta x = 13.925$ ft.
**Force P acts alternatively at Stations 0, 4, 5, 6, 7 (see text).

TABLE 3 – Natural Frequencies and Frequencies Corresponding to Maximum Steady-State Response at Station 0

Submerged Condition		Surfaced Condition	
Natural Frequency cpm	Frequency Corresponding to Maximum Steady-State Response at Station 0	Natural Frequency cpm	Frequency Corresponding to Maximum Steady-State Response at Station 0
175	175	207	205
378	378	451	450
562	567	678	678
763	785	906	938
971	1025	1161	---
1130	----	1397	---

TABLE 4 – Weight Distribution Data

n	Keel* Weight tons	Ballast* Weight tons	Virtual Weight of Hull tons	Virtual Weight of Planes tons	Total Weight tons
Submerged Condition					
0 (Stern)	5.93	0.0	2.9		8.83
1	43.75	3.6	32.47	53.13	132.95
2	72.09	32.83	77.63	53.13	235.68
3	94.32	25.53	102.77		222.62
4	125.01	43.51	156.16		324.68
5	161.18	36.38	202.85		400.41
6	183.52	35.41	238.12		457.05
7	170.59	59.63	258.77		488.99
8	188.17	118.62	269.33		576.12
9	148.54	117.71	274.57		540.82
10	311.58	77.97	277.10		666.65
11	443.72	66.19	278.15		788.06
12	229.79	82.02	278.50		590.31
13	160.02	74.68	278.15		512.85
14	195.93	43.47	275.94		515.34
15	220.43	91.16	268.65	19.35	599.59
16	175.74	141.67	254.12	6.78	578.31
17	107.92	109.89	222.00		439.81
18	74.59	60.87	171.98		307.44
19	60.45	0.60	101.08		162.13
20 (Bow)	10.54	0.0	17.97		28.51
Total	3183.81	1221.74	4039.21	132.39	8577.15
Surfaced Condition					
0	10.67		1.33		12.00
1	55.33		12.00	53.13	120.46
2	87.33		27.33	53.13	167.79
3	107.53		47.13		154.66
4	132.87		68.47		201.34
5	172.00		100.00		272.00
6	184.00		124.20		308.20
7	205.53		143.53		349.06
8	220.40		146.20		366.60
9	233.53		150.67		384.20
10	347.80		154.00		501.80
11	484.00		154.67		638.67
12	278.00		154.67		432.67
13	187.33		154.67		342.00
14	231.33		153.33		384.66
15	248.93		149.13		398.06
16	209.33		137.33		346.66
17	164.20		112.00		276.20
18	104.47		76.87		181.34
19	67.53		40.67		108.20
20	8.67		5.33		14.00
Total	3740.78		2113.53	106.26	5960.57
<p>*Submerged keel weight includes hull weight (hull, machinery, cargo, ballast) less salt water, fresh water, oil, and lead ballast.</p> <p>Surfaced hull weight includes hull structure, machinery, liquid and dry cargo, and liquid and lead ballast.</p>					

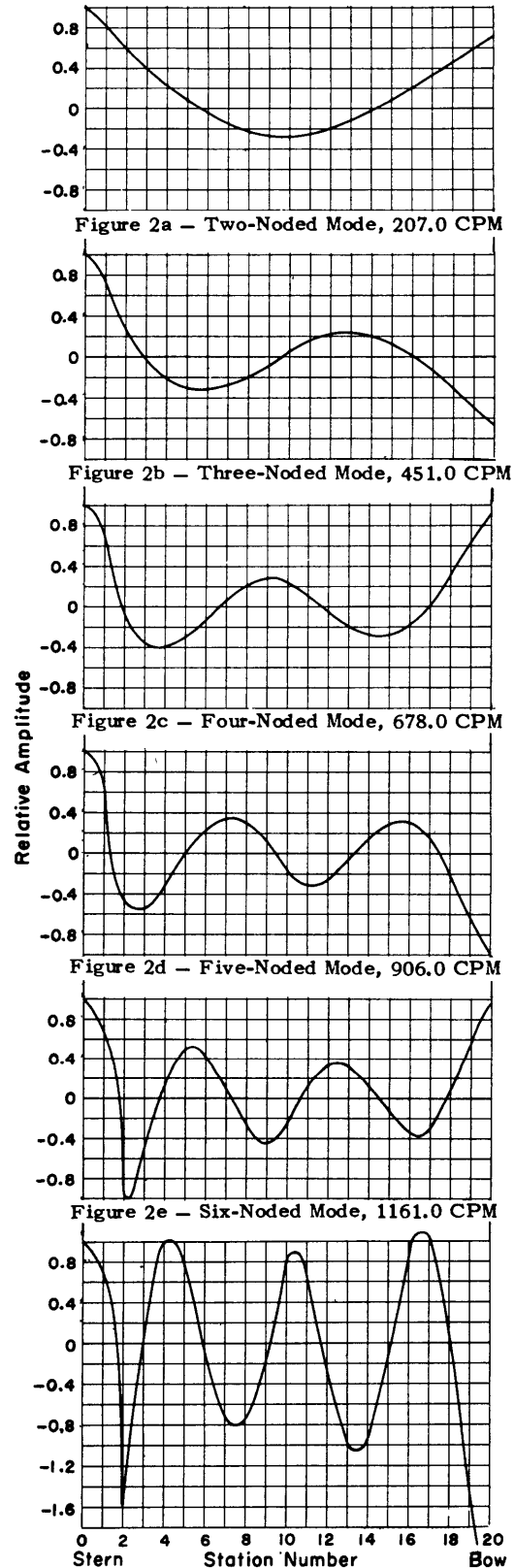
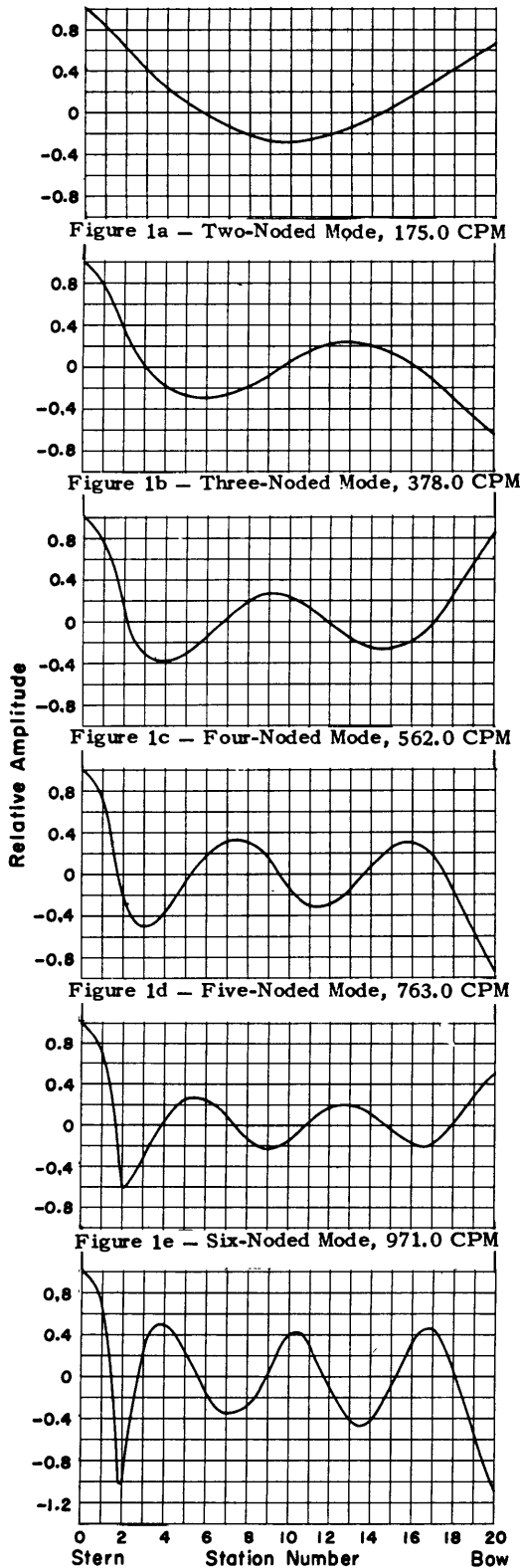


Figure 1 - Vertical Normal-Mode Profiles for Submerged Condition (Determined by UNIVAC) Figure 2 - Vertical Normal-Mode Profiles for Surfaced Condition (Determined by UNIVAC)

Taylor Model Basin Numbers Stations $n = 0$ (Stern), 1, 2, ... 20 (Bow);
 Bureau of Ships Numbers Stations $n = 0$ (Bow), 1, 2, ... 20 (Stern).

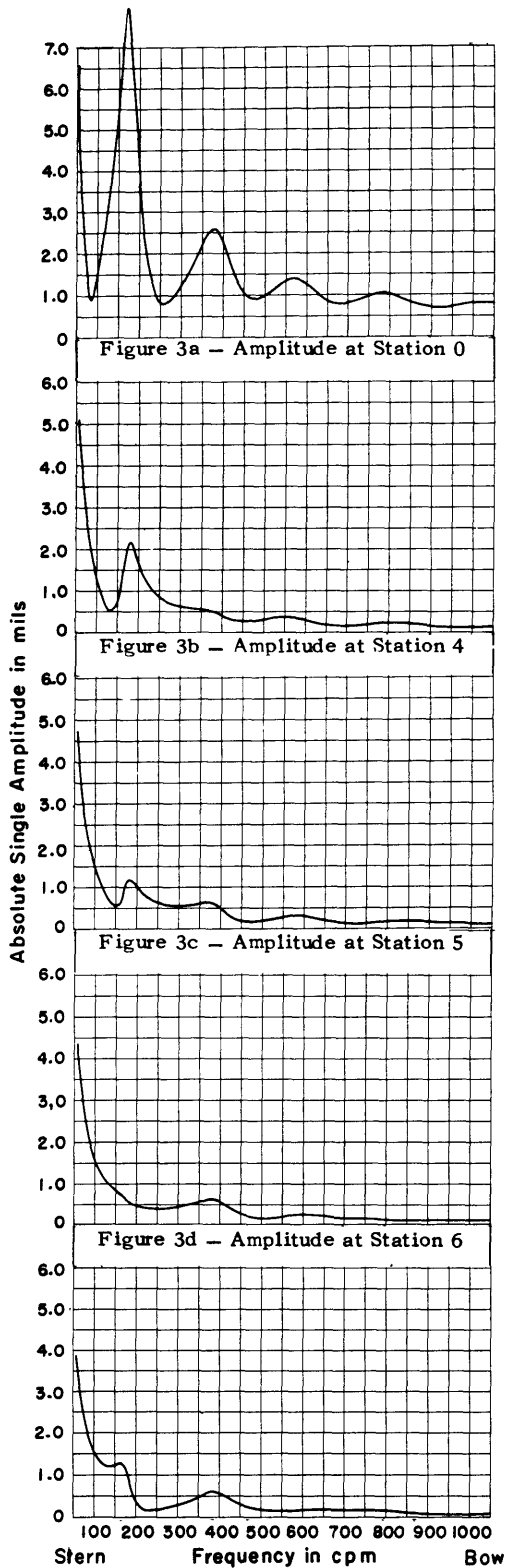


Figure 3 – Vertical Displacement Response at Selected Stations for a 1-Ton Force Applied at Stern for Submerged Condition (Determined by UNIVAC)

Taylor Model Basin Numbers Stations $n = 0$ (Stern), 1, 2, --- 20 (Bow);
 Bureau of Ships Numbers Stations $n = 0$ (Bow), 1, 2, --- 20 (Stern).

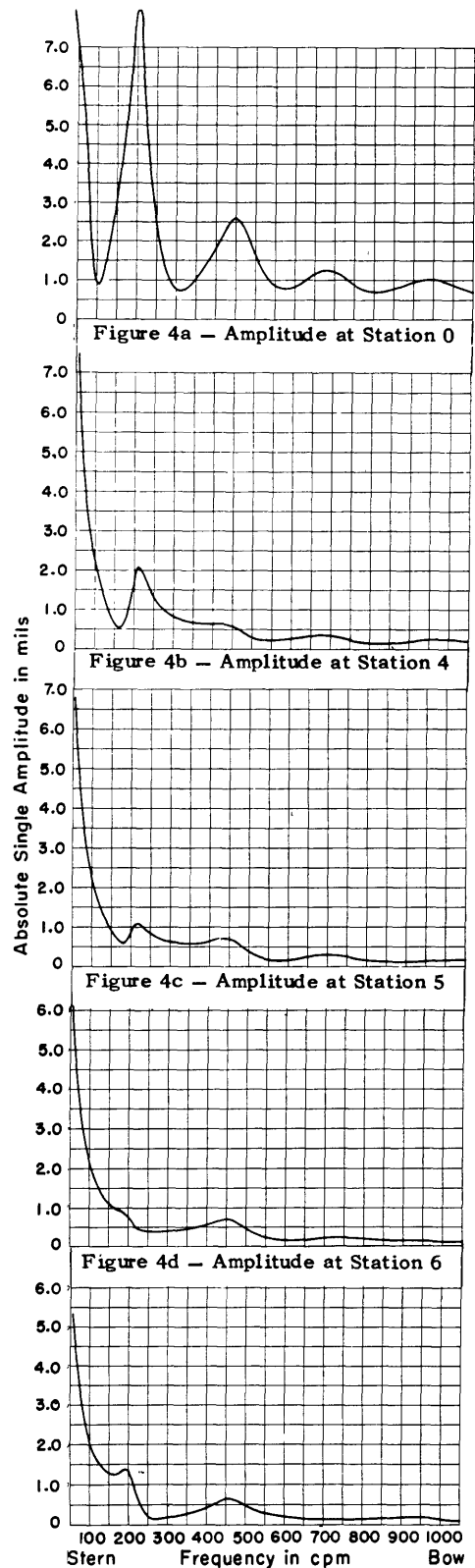


Figure 4 – Vertical Displacement Response at Selected Stations for a 1-Ton Force Applied at Stern for Surfaced Condition (Determined by UNIVAC)

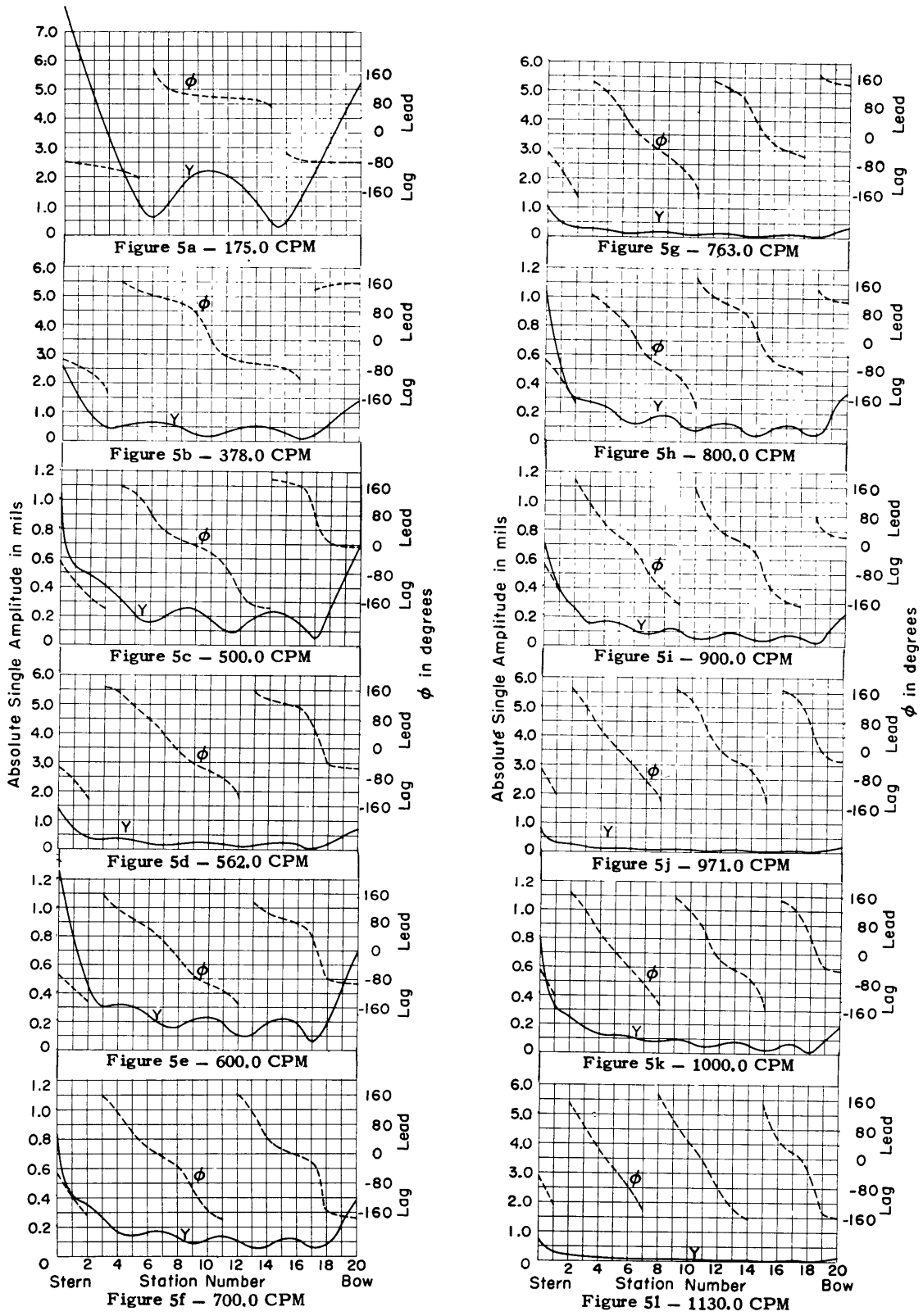


Figure 5 – Vertical Vibration Amplitude and Phase versus Length for a 1-Ton Force Applied at Stern for Submerged Condition (Determined by UNIVAC)

Taylor Model Basin Numbers Stations $n = 0$ (Stern), 1, 2, --- 20 (Bow);
 Bureau of Ships Numbers Stations $n = 0$ (Bow), 1, 2, --- 20 (Stern).

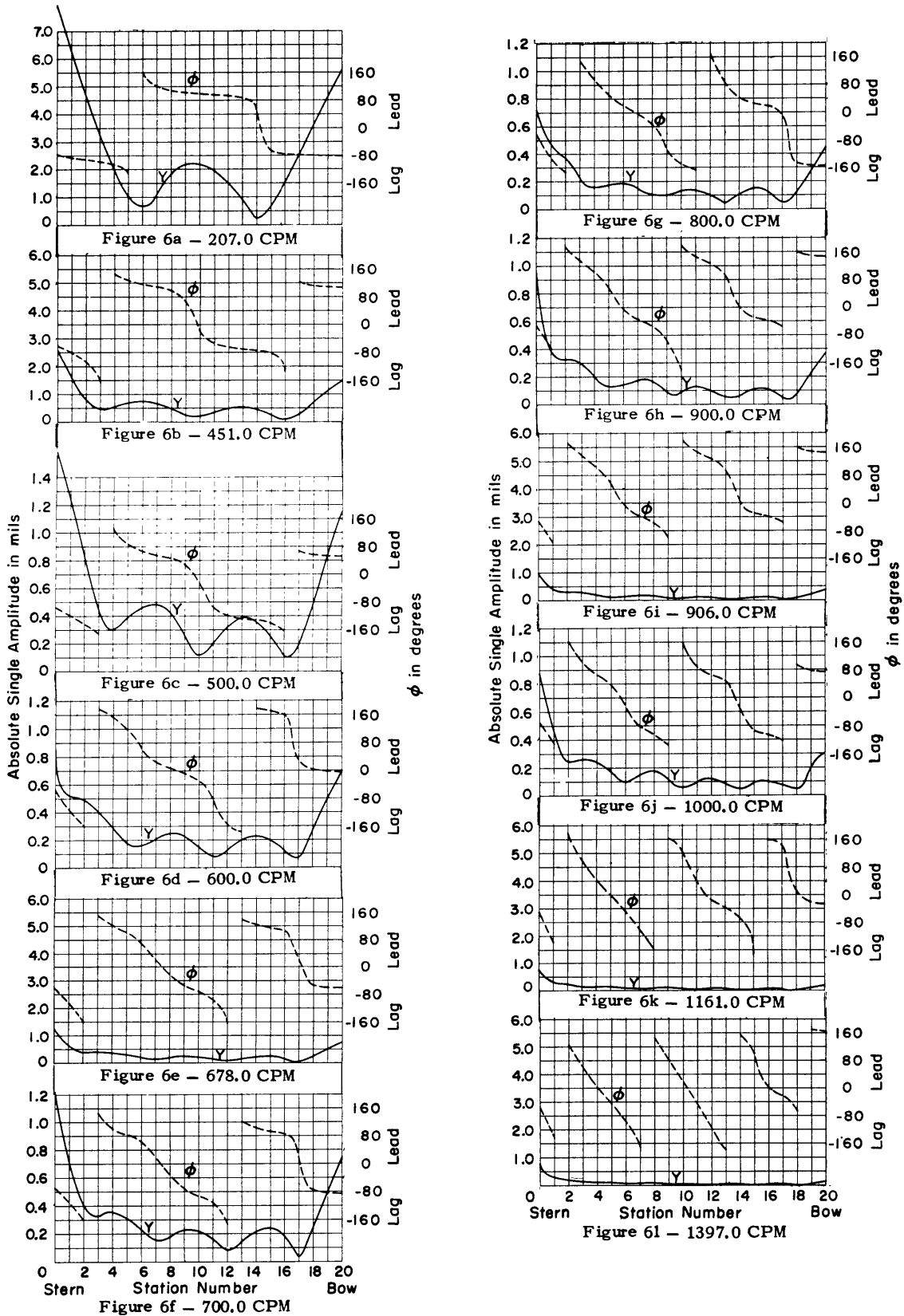


Figure 6 – Vertical Vibration Amplitude and Phase versus Length for a 1-Ton Force Applied at Stern for Surfaced Condition (Determined by UNIVAC)

Taylor Model Basin Numbers Stations $n = 0$ (Stern), 1, 2, --- 20 (Bow);

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