NAVAL SHIP RESEARCH AND DEVELOPMENT CENTER
Bethesda, Maryland 20034

EVALUATION OF HULL VIBRATORY (SPRINGING) RESPONSE OF GREAT LAKES ORE CARRIER M/V STEWART J. CORT

by

Milton O. Critchfield

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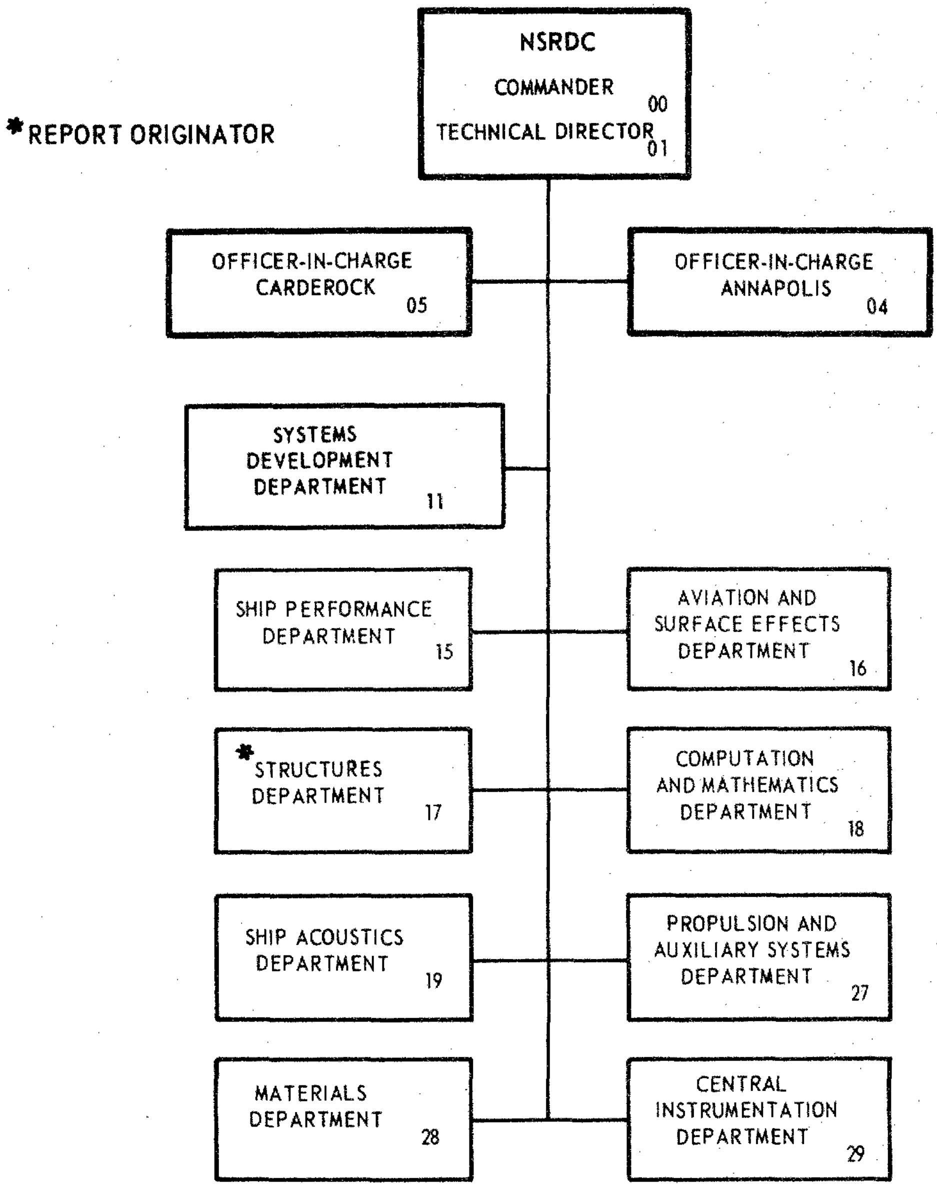

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NOTATION

c	Distance from beam neutral surface to outer free surface
E	Young's modulus of elasticity
L	Ship length
N	Number of peak-to-peak record variations
y_i	Amplitude of i^{th} record peak or trough
y_{max}	Maximum record value
x	Distance along beam from midpoint
$\delta_{\boxtimes}$	Midpoint to bow or stern deflection
$\sigma_{\boxtimes}$	Midship bending stress
$\sqrt{E}$	Root mean square of peak record values
rms	Root mean square of instantaneous record values

ABSTRACT

The hull vibration (springing) of the Great Lakes ore carrier STEWART J. CORT was evaluated on the basis of accelerometer and strain gage data collected during a regular round trip voyage between Burns Harbor, Indiana, and Taconite Harbor, Minnesota, in August 1972. Springing was recorded when in the loaded condition, CORT encountered 4- to 6-ft seas, 60 deg to the bow in Lake Superior. The maximum first mode deflection profile and midship bending stress were determined from a direct analysis of this springing record. A spectrum analysis of the same record produced flexural frequencies of vibration for the first three vertical modes and what appeared to be the first horizontal mode, rms deflection profiles for the first three vertical modes, and the first mode rms midship bending stress. An important finding of the investigation was that the springing deflections of the CORT involved larger contributions than expected from the second flexural mode of vibration. Although the contribution of the second mode to midpoint bending stress is theoretically zero, its contribution to bending stresses at other points merits consideration. An evaluation of first vibratory mode damping by means of ten anchor drop tests was unsuccessful, partly because CORT underwent springing while at zero speed in a 1-ft sea. The springing appeared to interfere with the anchor-drop-induced ship response.

ADMINISTRATIVE INFORMATION

The work reported herein was sponsored jointly by the U. S. Coast Guard under Military Interdepartmental Purchase Request Z-70099-3-30384 and the Naval Ship Systems Command (NAVSHIPS) under Task Area SF 43.422.315, Project F 43 422, Program Element 62754N. Preparation of this report was done under Work Unit 1730-118.

INTRODUCTION

The longitudinal bending strength of Great Lakes ore carriers has been under investigation since 1965 in a program implemented by the Hull Structure Committee of the Society of Naval Architects and Marine Engineers (SNAME) in cooperation with numerous government, regulatory, and industrial organizations in the United States and Canada. Hull bending stresses have previously been measured and the associated wave environments determined for one U.S. ore carrier, EDWARD L. RYERSON, and three Canadian ore carriers.

In 1965, the first year of the program, stress recordings from the RYERSON revealed the presence of an unusual low-cycle vibratory response of the hull girder. This phenomenon consisted of a series of first mode bending vibrations that gradually increased and then decreased in amplitude and resulted in stresses of up to 15,000 psi peak to peak.[1] This phenomenon, apparently recorded for the first time at significant levels on RYERSON, is commonly referred to as "springing." Springing is generally considered to be caused by the existence of sufficient energy in the wave spectrum at frequencies which—when combined with ship speed—result in an encounter frequency coincident with the first mode bending frequency of the ship.[1] Actually, the encounter frequency does not remain constant at the first mode frequency due to fluctuations in wave conditions and—to some extent—to ship speed. This results in a near resonant response consisting of first mode vibrations which cyclically grow and diminish in amplitude with time.

At present, springing is not of particular concern for naval ships because wave encounter frequencies are below their first mode bending frequencies. However, a sufficient increase in either hull flexibility, speed, or both in any future ships would have the effect of bringing wave encounter and first mode frequencies into coincidence.

From a theoretical standpoint, a resonant first mode vertical flexural response of the hull girder (springing) is generally not the only type of wave-excited resonant condition which could conceivably occur for naval ships. Other possibilities include horizontal (athwartship) and torsional hull girder resonant responses. Furthermore, wave energy may also conceivably set up other types of resonant conditions, e.g., athwartship bending of catamaran hulls and the vibration of structural subassemblies on future high-performance ships. Suffice it to say that all possible wave-induced resonant conditions for future ship configurations are not as yet foreseen. It would therefore appear that research into the springing of conventional monohulls should constitute a basic starting point for understanding the general problem of wave-excited resonances.

Accordingly, several discussions were held with the U. S. Coast Guard which has been sponsoring research for a number of years on the effect of springing on the longitudinal strength of bulk carriers. The Coast Guard subsequently awarded a contract to the Naval Ship Research and Development Center (NSRDC) for an evaluation of the springing-related hull vibration of a new ore carrier, the M/V STEWART J. CORT on the Great Lakes. Sponsorship of this task was shared by NAVSHIPS because of possible future application to naval ships. The 1000-ft-long CORT is the longest ore carrier to operate on the Great Lakes since enlargement of the locks at Sault Ste. Marie. Its principal characteristics and hull form data are given in Appendix A.

The initial objectives of the NSRDC dynamic response evaluation were (1) determination of underway deflections associated with springing and (2) determination of first mode hull

[1] Seaway Stresses Observed aboard the Great Lakes Bulk Ore Carrier EDWARD L. RYERSON (1965–1968)," SNAME T&R Bulletin 2-18 (May 1971).

damping. This information was to be obtained by utilizing a system of accelerometers along the CORT deck and was intended to supplement the hull stress and other data being recorded by the on-board automatic instrumentation system installed and monitored by Teledyne Materials Research Company.[2] Later in the data analysis phase of the NSRDC program, the objectives were broadened to include a determination of the modal composition of the data records and the extent of participation of the various flexural vibratory modes during springing.

The hull vibration evaluation program was accomplished in coordination with the U.S. Coast Guard, Bethlehem Steel Corporation, and Teledyne Materials Research Company during a regular round trip of CORT between Burns Harbor, Indiana, and Taconite Harbor, Minnesota, from 13 to 17 August 1972. A summary of the basic findings of this program will be included in the 1972 season report now in preparation.[3]

ON-BOARD INSTRUMENTATION SYSTEM

Hull vibration and anchor drop measurements were performed by using the following instrumentation system. Five accelerometers with magnetic bases were positioned at the starboard and port bow, quarter point, midpoint, and stern of CORT, as indicated in Figure 1. The accelerometers were placed on the main deck except for the two bow gages which were located one deck below in the windlass room. Initially, an accelerometer was also used at the three-quarter point, but it was later moved to the stern when the stern gage became inoperative. Cables from the accelerometers were strung along the main deck wireway, up the stairway to the forward upper deck, and vertically downward through two hatches at the extreme bow into the forward machinery room where the data recording system was located. In the case of the two bow accelerometers, it was only necessary to drop the cables through the windlass room hatch to the forward machinery room.

Signals from the five accelerometers were recorded on both paper and magnetic tape along with a midship strain signal, as illustrated in the instrumentation layout of Figure 2. Midship strain was monitored from the Teledyne Material Research Company instrumentation system so that data recorded by the accelerometers could be time correlated and compared with those based on strain-gage measurements. The monitored midship strain signal represents an average between starboard and port bow gage signals. The port bow, stern, and midpoint accelerometer signals were filtered on board by using a bandpass of 0.2 to 2 Hz. Amplifier

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[2] "Instrumentation of M/V STEWART J. CORT–Lake Trials Data," Teledyne Materials Research Technical Report E-1419(a), published by U. S. Coast Guard under Project 723511 (22 Dec 1971).

[3] "Instrumentation of M/V STEWART J. CORT–1972 Season," Teledyne Materials Research Technical Report E-1419(i), to be published by U. S. Coast Guard (in preparation).

gains were set as indicated in Figure 2. The oscilloscope and switch box were used to sample the various channels of data in the process of setting amplifier gains. The use of an oscilloscope facilitated an on-board qualitative data analysis. The instrumentation system was set up on the starboard side of the machinery room.

A list of the instrumentation used on board CORT is given in Appendix B by name and model number together with a few important specifications for the instruments.

UNDERWAY HULL VIBRATION EVALUATION

During the trip from Burns Harbor to Taconite Harbor in the ballast condition (13 to 15 August 1972), the seas were generally calm with wave heights less than 1 ft. Vibration measurements were performed for a 1-hr period when light springing developed in eastern Lake Superior several hours after CORT left Whitefish Bay.

On the return trip from Taconite Harbor in the loaded condition, CORT encountered 4- to 6-ft seas at 60 deg to the bow approximately 150 miles east of Taconite Harbor on the evening of 15 August 1972. Low amplitude springing was first noted at about 2000 hr. By 2300 hr the flexing of the ship associated with springing became clearly visible when looking aft from the bow, and it continued until at least 0130 hr on 16 August. In addition to vertical vibrations, CORT also experienced horizontal (athwartship) vibrations.

Accelerometer and strain gage signals were recorded between 2347 hr on 15 Aug and 0045 hr on 16 August. The signals recorded during this 1-hr period were found to be the largest recorded at any time during the round trip. An 18-min segment of this 1-hr record, the portion containing the largest signal amplitudes, was singled out for analysis and will be the subject of study and discussion for the remainder of this report. This record segment was subjected to a direct analysis which resulted in a determination of maximum deflection and stress values and a spectrum analysis from which rms values were obtained. More specifically, the direct analysis of the record segment (that is, a measurement of peak-to-peak signal amplitudes) provided (1) the maximum hull deflection profile for the first vertical flexural mode and (2) the maximum midship bending stress associated with the first mode. The spectrum analysis of the data using a fast Fourier transform analyzer led to a determination of (1) the bending frequencies for the first three vertical vibration modes and a likely value for the first horizontal mode, (2) root mean square (rms)* deflection profiles for the first three vertical bending modes, and (3) rms midship bending stress for the first mode. Finally, as a check on the validity of the maximum and rms values respectively based on direct and spectrum analyses, these values were correlated by using statistical theory for Rayleigh distributions.

*As used here, rms is defined in Appendix D.

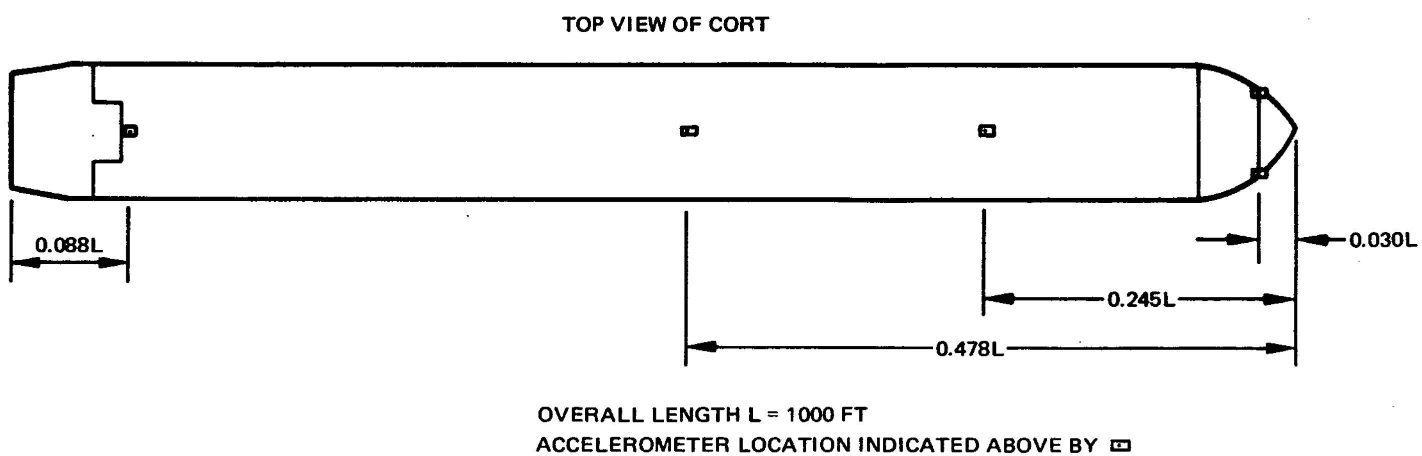

Figure 1 – Accelerometer Gage Points on Main Deck of CORT

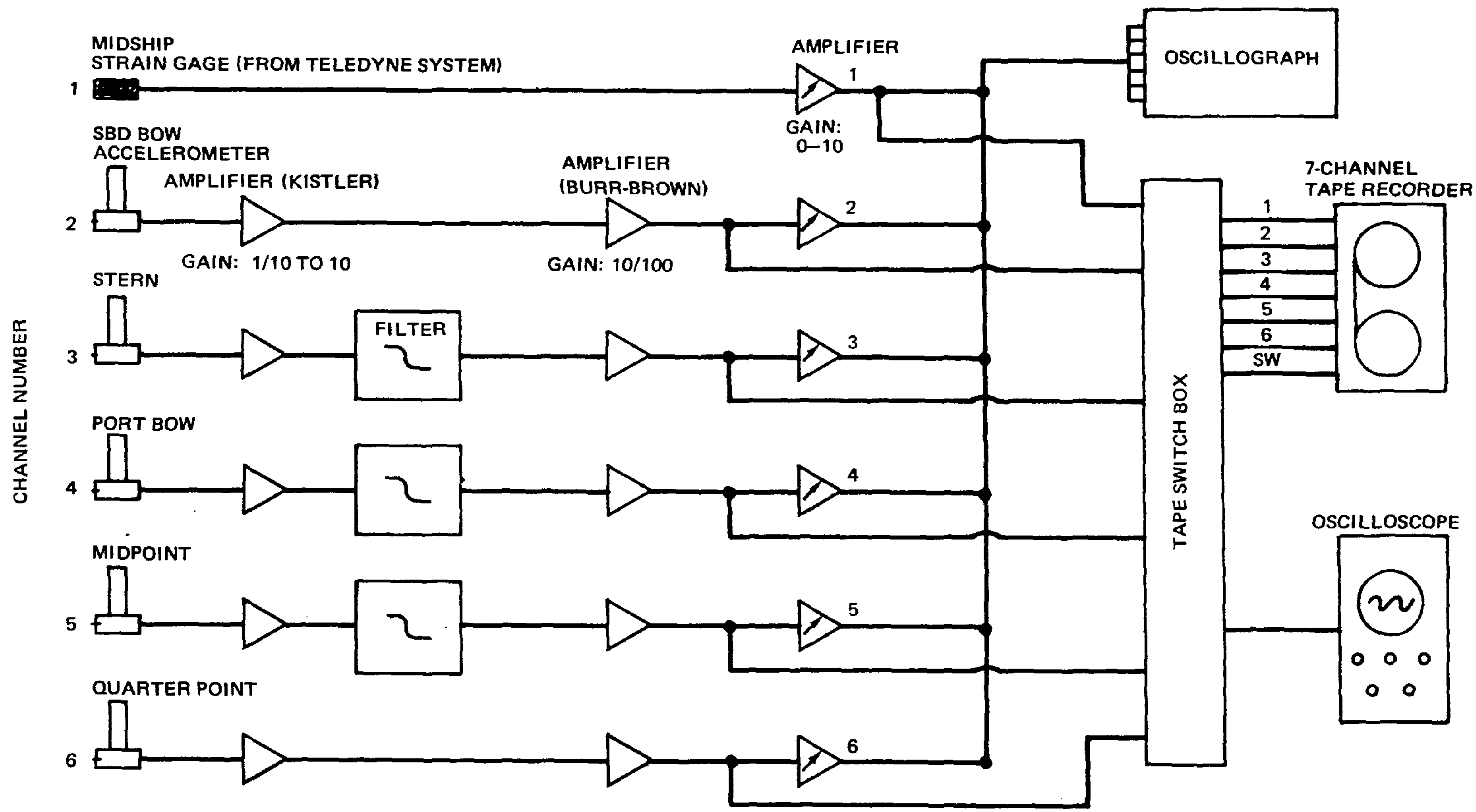

Figure 2 – On-Board Instrumentation System

A summary of the underway vibration measurements performed during the entire trip is given in Appendix C.

DIRECT ANALYSIS AND RESULTS

The six data channels (five accelerometer and one strain channel) of the 18-min maximum response record were filtered by using the instrumentation system illustrated in Figure 3; this system is located in the NSRDC Ship Acoustics Department and has the capability for filtering two channels at a time, as indicated by Channels A and B of Figure 3. The limits on the bandpass filter were set at 0.2 and 0.4 Hz to eliminate flexural modes and noise above the first mode frequency; previous sea trial measurements had indicated that this frequency was in the range of 0.3 to 0.4 Hz.

The maximum response portion of the 18-min record is shown in Figure 4 in the unfiltered and filtered conditions for two of the six data channels, midship stress and starboard bow acceleration. Note that the unfiltered stress signal primarily reflected the presence of the first flexural mode whereas the unfiltered bow accelerometer signal indicated the presence of higher frequency components as well.

Maximum Hull Deflection Profile (First Mode)

First-mode double amplitude acceleration values* were obtained for the gage points along the CORT deck (see Table 1) by applying appropriate calibration signals to all of the filtered accelerometer signals. Double-amplitude hull deflections, also given in Table 1, were then computed by dividing these accelerations by the square of the frequency, 0.3120 Hz, identified as the first mode bending frequency from a spectrum analysis of the data as discussed later.

Figure 5 is a plot of the maximum first mode deflections obtained by halving the values given in Table 1. Note that the largest first-mode contributions to the CORT deflection profile were approximately 5 in. peak to peak at the bow and stern and approximately 2 in. at the midpoint. The very small quarter-point deflection value shows that it was near a nodal point, as expected.

Maximum Midship Bending Stress (First Mode)

A maximum bending stress of 2240 psi single amplitude was determined from the midship strain gage record of Figure 4. For purposes of comparison, a midship stress was also obtained from the CORT deflection profile of Figure 5 by using two analogous approaches, as discussed in the following.

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*Actual or rms acceleration, deflection, or stress values not specifically designated in this report as double amplitude or peak to peak are single amplitude.

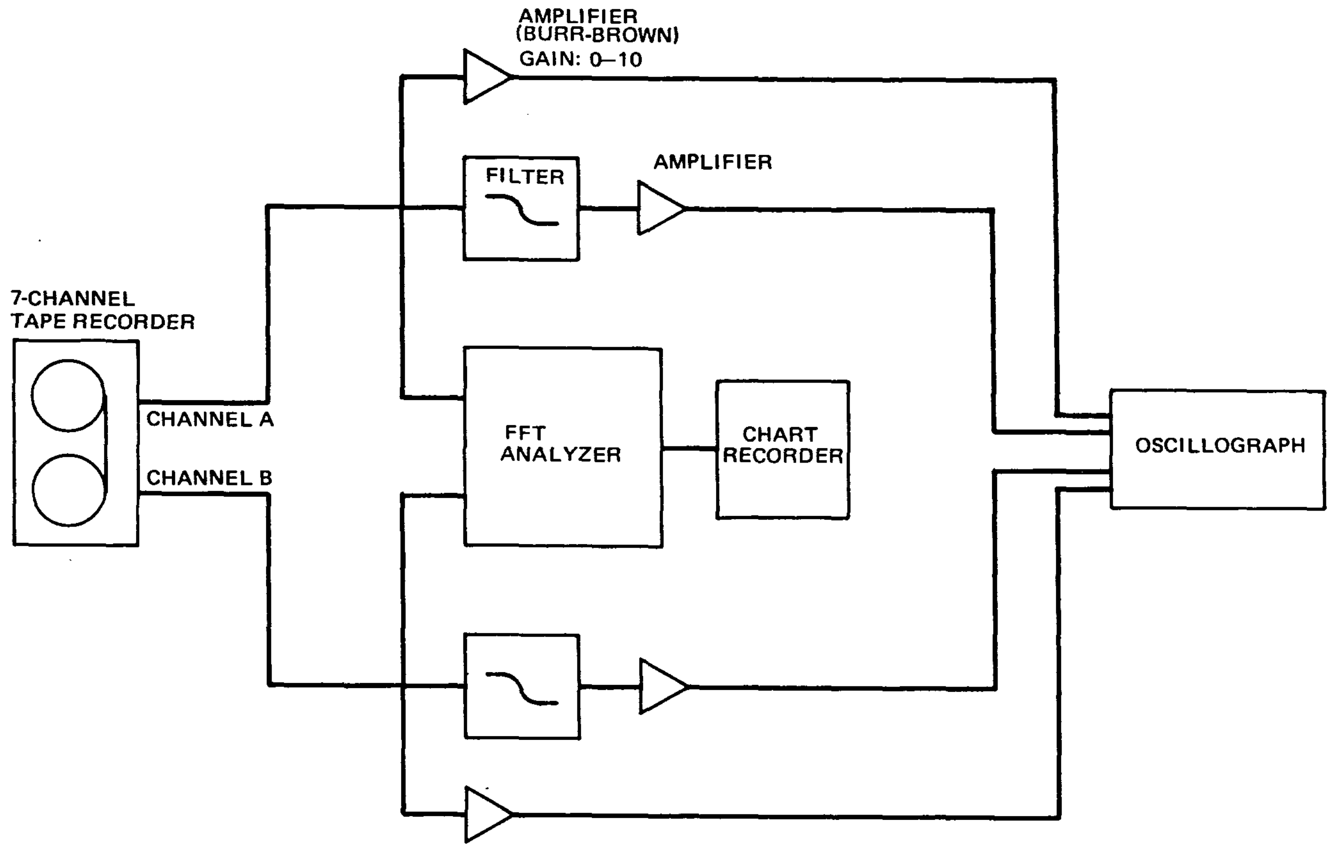

Figure 3 – Laboratory Filtering System

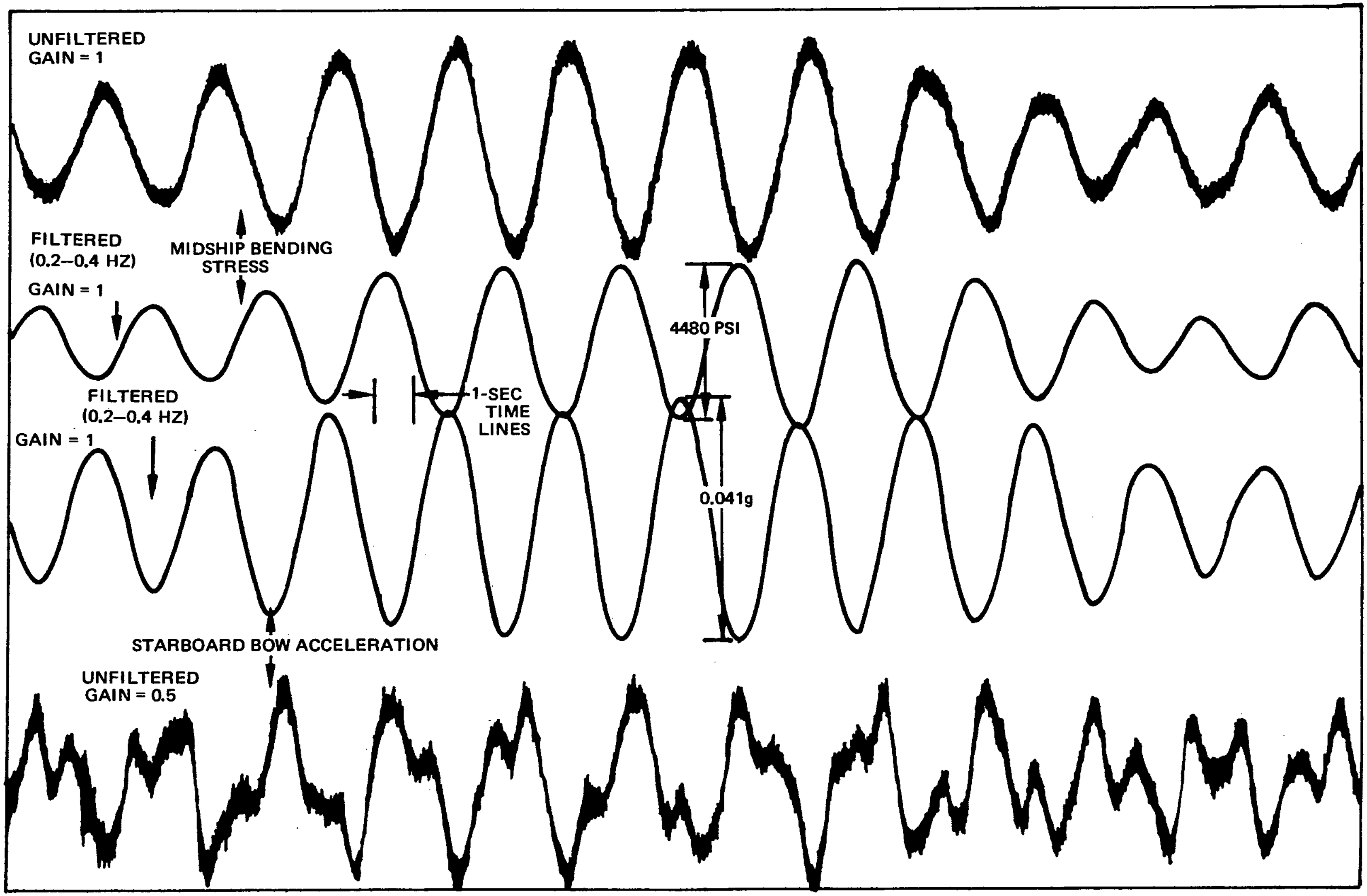

Figure 4 – Maximum Springing Record

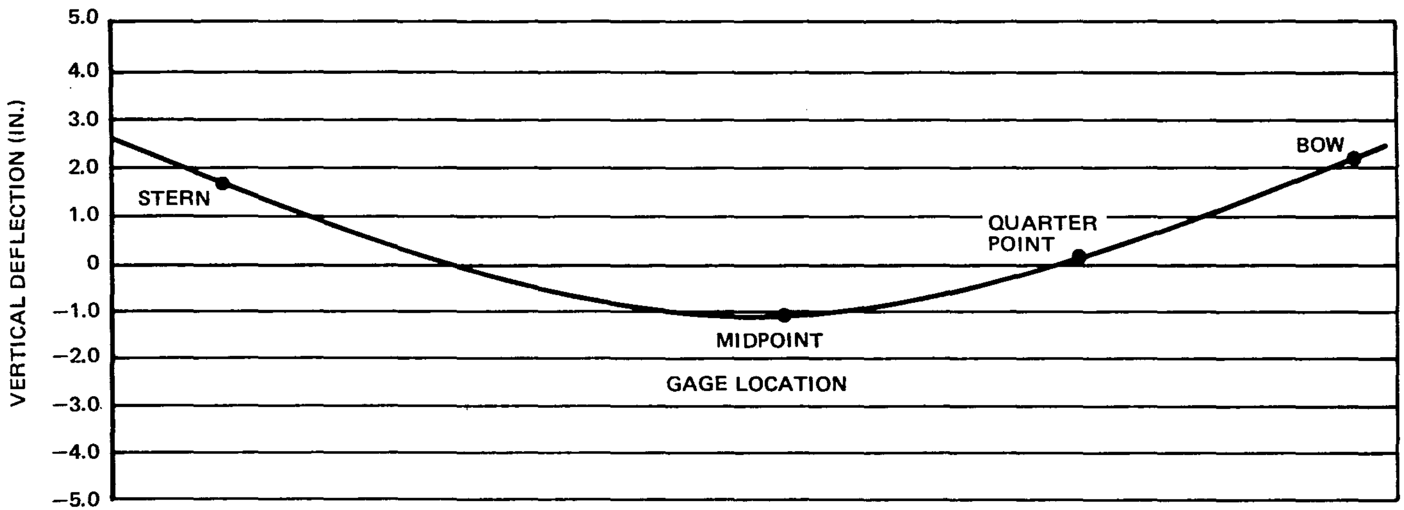

Figure 5 – Maximum Hull Deflection Profile–First Mode

TABLE 1 MAXIMUM FIRST MODE VERTICAL HULL ACCELERATIONS AND DEFLECTIONS

(Values are double amplitude)

Gage Point	Stbd Bow (0.030 L*)	Port Bow (0.030 L)	Average Bow (0.030 L)	Quarter Point (0.245 L)	Midpoint (0.478 L)	Stern (0.912 L)
Acceleration, g	0.041	0.048	0.045	0.003	0.021	0.034
Deflection, in.	4.10	4.66	4.38	0.30	2.10	3.40
*L = CORT length						

TABLE 2 – VERTICAL BENDING MODE FREQUENCY RATIOS

Ship Name	Ship Type	First Mode / First Mode	Second Mode / First Mode	Third Mode / First Mode
STEWART J. CORT	Ore Carrier	1	2.31	3.94
C. A. PAUL	Ore Carrier	1	2.35	3.73
NIAGARA	Transport	1	1.83	2.65
CHARLES R. WARE	Destroyer	1	2.08	3.30
PERE MARQUETTE	Car Ferry	1	2.00	3.10
OLD COLONY MARINER	Dry Cargo	1	1.89	2.77
NORTHAMPTON	Cruiser	1	2.11	2.99
STATEN ISLAND	Ice Breaker	1	1.93	2.57
Average		1	2.06	3.13

TABLE 3 – CORT FREQUENCY RATIO COMPARISON WITH SHIPS IN GENERAL AND EULER BEAM THEORY

Source of Ratio	First Mode / First Mode	Second Mode / First Mode	Third Mode / First Mode
Ratio Averages from Table 2	1	2.06	3.13
STEWART J. CORT	1	2.31	3.94
Euler (Uniform) Beam Theory	1	2.76	5.40

Vedeler[4] assumed that the distribution of bending moment in ore carriers can be described by a parabolic curve. Making use of this and other assumptions (see Appendix E), Vedeler developed a formula for midship stress $\sigma_{\boxtimes}$ in terms of midpoint to bow or stern deflection $\delta_{\boxtimes}$:

$$\sigma_{\boxtimes} = 10.08 \frac{\delta_{\boxtimes} \, Ec}{L^2} \tag{1}$$

where E is Young's modulus of elasticity,

c is the distance from the beam neutral surface to the outer free surface, and

L is the total ship length.

An analogous formula, derived on the basis of the Euler Bernoulli beam theory, is

$$\sigma_{\boxtimes} = 10.99 \frac{\delta_{\boxtimes} \, Ec}{L^2} \tag{2}$$

The derivation of this formula is presented in Appendix E.

As seen in Figure 5, the midpoint to stern deflection (3.75 in.) was slightly greater than the midpoint to bow value (3.55 in.). Thus, for the purpose of computing $\sigma_{\boxtimes}$, the average of these values (3.65 in.) was chosen to represent $\delta_{\boxtimes}$. Substitution of $\delta_{\boxtimes} = 3.65$ in., $E = 30 \times 10^6$ psi, c = 300 in., and L = 12,000 in. into Equations (1) and (2), gave midship stresses of 2300 and 2510 psi, respectively. These stresses agreed with the strain-based value of 2240 psi to within 2.7 and 12.1 percent, respectively.

As evidenced by the stress results above, the Vedeler formula (Equation (1)) appears to be superior to the analagous formula (Equation (2)) for predicting midship bending stress from deflection data, at least for CORT.

SPECTRUM ANALYSIS AND RESULTS

A spectrum analysis was also performed on the six channels (five of accelerometer data and one of stress data) of the maximum springing record. This analysis was accomplished on a time/data fast Fourier transform (FFT) analyzer in the NSRDC Ship Acoustics Department. The analysis utilized the following FFT analyzer settings: frequency range, 0 to 10 Hz; resolution, 512 spectral or frequency lines in the specified range; and 16 ensemble averages. Two data channels were processed at a time as illustrated in Figure 3. Since the output of the FFT analyzer is in the form of a mean square value spectrum for each input data channel, the operation of the FFT on the six-channel springing record resulted in five acceleration mean square value spectra and one midship stress spectrum.

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[4] Vedeler, A., "Longitudinal Strength of Great Lakes Bulk Carriers," The American Ship Building Company, Paper presented at the Great Lakes Section, SNAME Meeting (4 Feb 1965).

Vertical and Horizontal Bending Frequencies

Figures 6–10 show the acceleration spectra obtained for the starboard and port bow, quarter point, midpoint, and stern responses when the acceleration mean square value spectra were divided by the frequency bandwidth of 0.0195 Hz (10 Hz/512 spectral lines. The midship bending stress spectrum shown later in Figure 11 was obtained in a similar manner.

It is seen from the spectra of Figures 6 to 10 that the largest acceleration responses were at frequencies of 0.3120, 0.7215, and 1.229 Hz. The lowest value, 0.3120 Hz, was identified as the first mode frequency for vertical bending vibrations by observing that it corresponded in the data records to bow and stern motions which were in phase but which were both out of phase with the midpoint motions.

The frequencies of 0.7215 and 1.229 Hz were determined to be those of the second and third vertical bending modes. This conclusion was based on the comparison given in Table 2 of the ratios of these values to that of the first mode frequency (0.7215/0.3120 and 1.229/0.3120) and the ratios of second to first mode and third to first mode vibration frequencies for a number of other ships.[5] It is seen from Table 2 that the CORT frequency ratios were in close agreement with those of another ore carrier, the C. A. PAUL. Furthermore, it is of interest to note in Table 3 that the CORT frequency ratios fell approximately midway between the average of the values in Table 2 and the corresponding values associated with the vibration of a uniform free-free beam. This is as expected in view of the relative uniformity with which the mass and stiffness are distributed over the CORT length.

The midship stress spectrum of Figure 11 revealed the first and second vertical bending frequencies but showed no trace of the third mode frequency. The very small response at 0.7215 Hz is as expected since the ship midpoint is a point of inflection for the second mode, resulting in zero moment and therefore zero bending stress.

Another response of small magnitude was evident in Figure 11 at 0.9165 Hz. This response is considered to be associated with the first horizontal bending mode because (1) horizontal (athwartship) vibrations were experienced during the recording of data and were estimated to be occurring at 0.9 Hz; (2) horizontal vibrations should not excite the vertically mounted accelerometers (Figures 6 to 10 did not show any response in the vicinity of 0.9 Hz); and (3) on the basis of the Johnson formula,[5] the first mode frequency was estimated at 0.85 Hz (see Appendix F).

The small response at 0.9165 Hz in Figure 11 apparently represents the amount of signal that passed from the midship strain bridge despite the fact that the starboard and port

[5]McGoldrick, R.T., "Ship Vibration," NSRDC Report 1451 (Dec 1960).

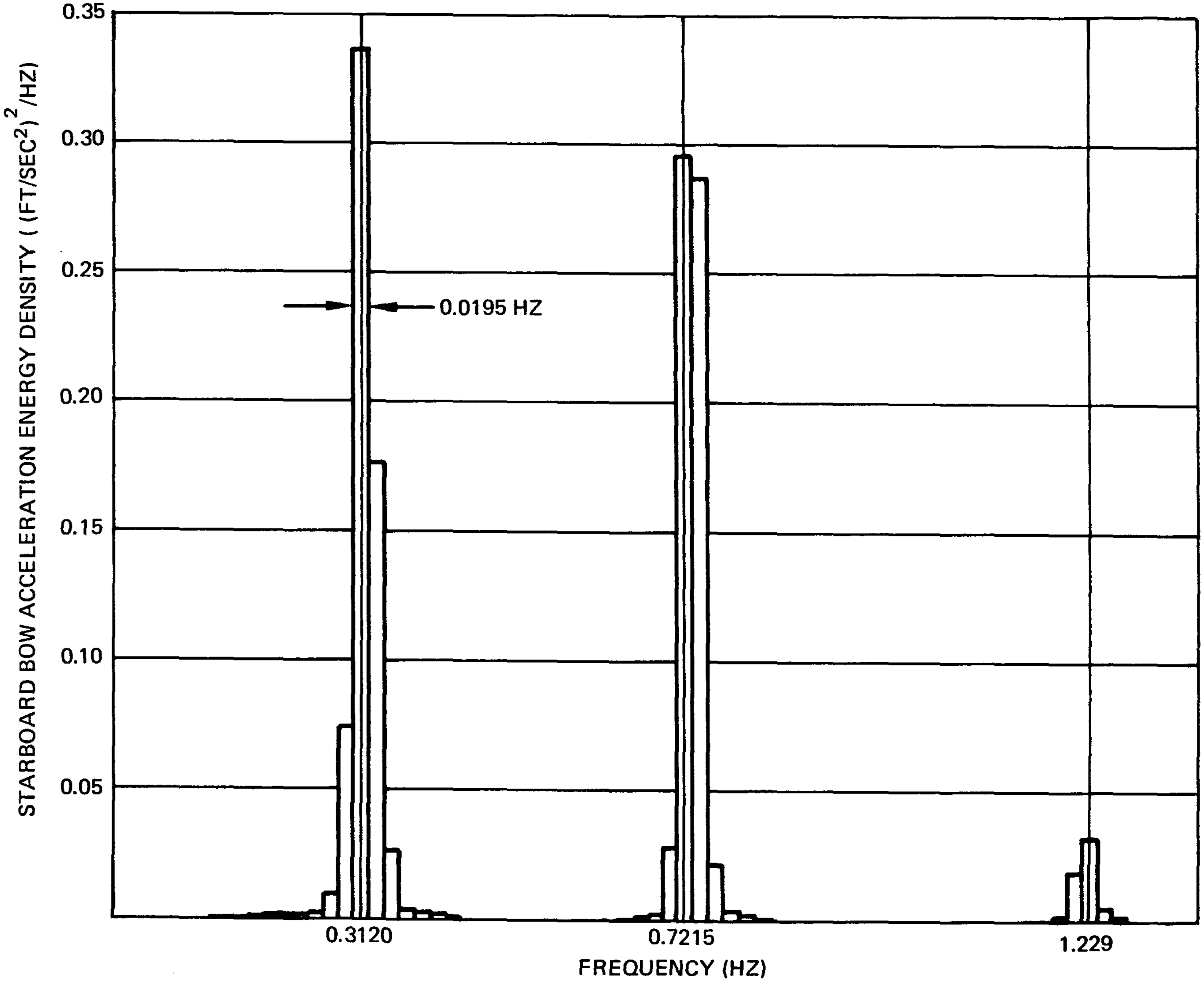

Figure 6 — Starboard Bow Acceleration Spectrum

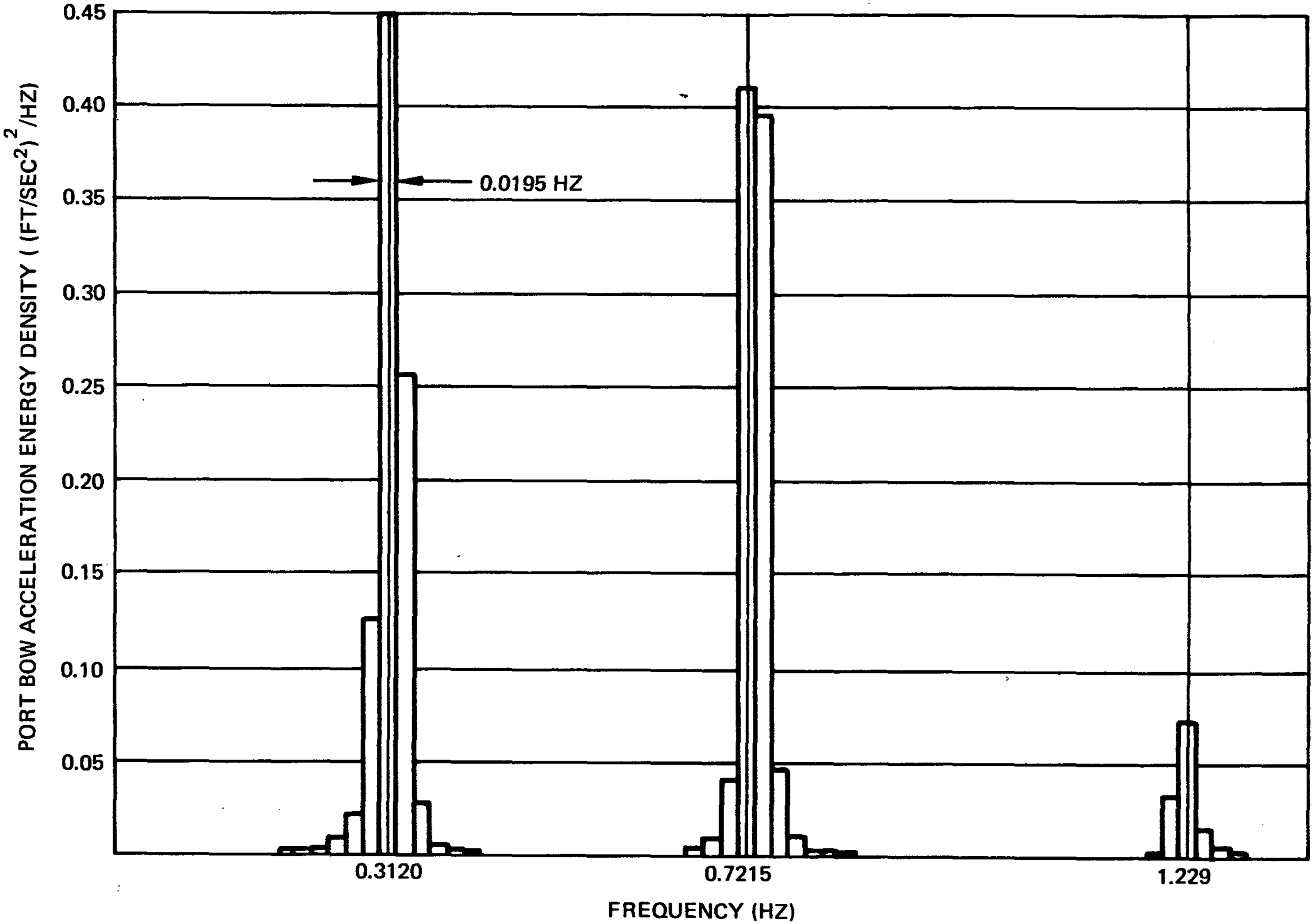

Figure 7 – Port Bow Acceleration Spectrum

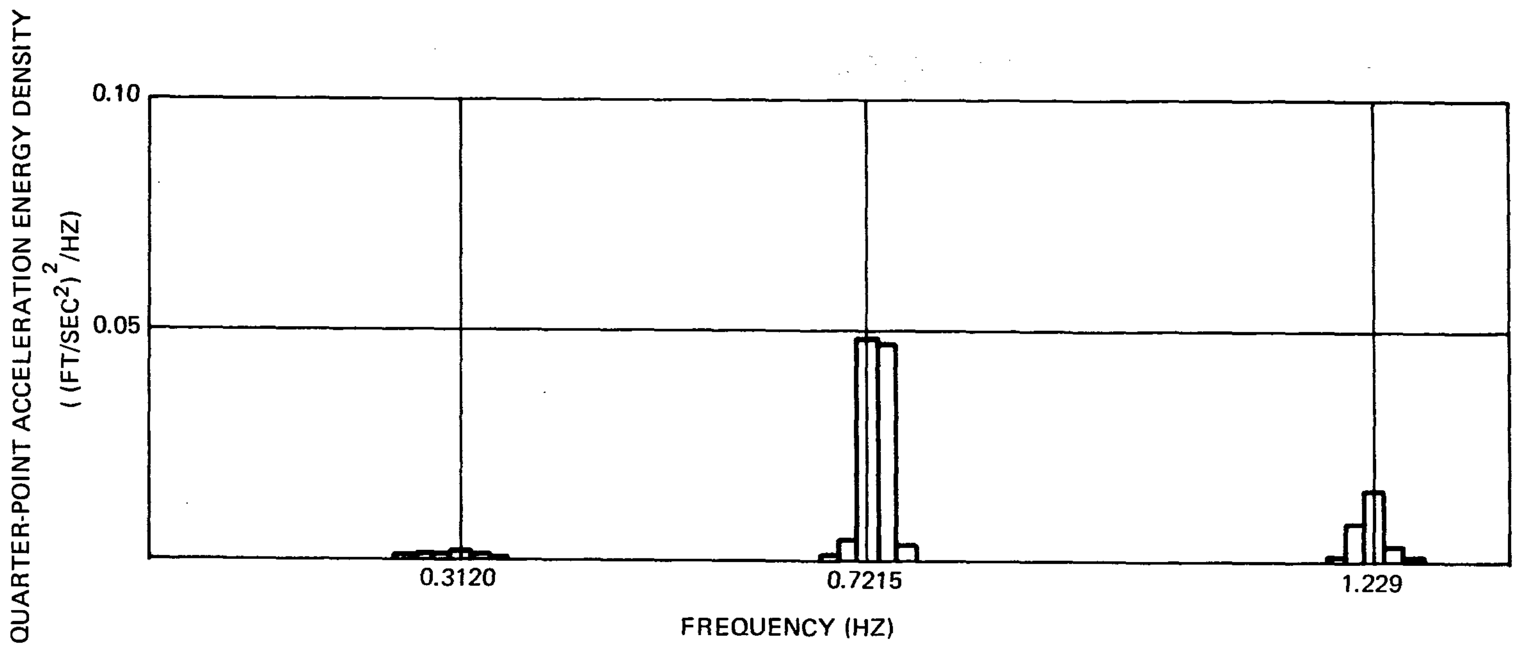

Figure 8 — Quarter-Point Acceleration Spectrum

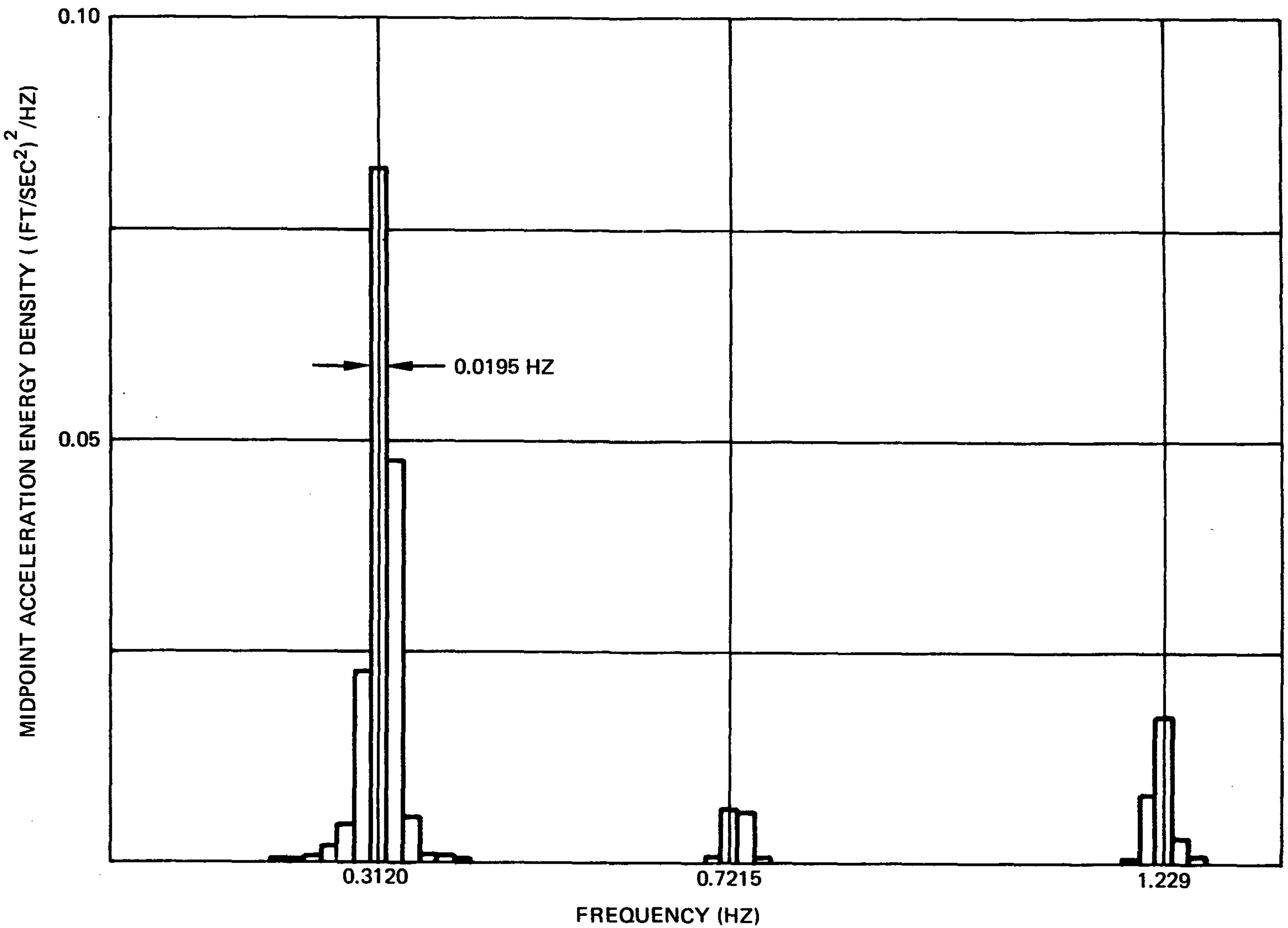

Figure 9 – Midpoint Acceleration Spectrum

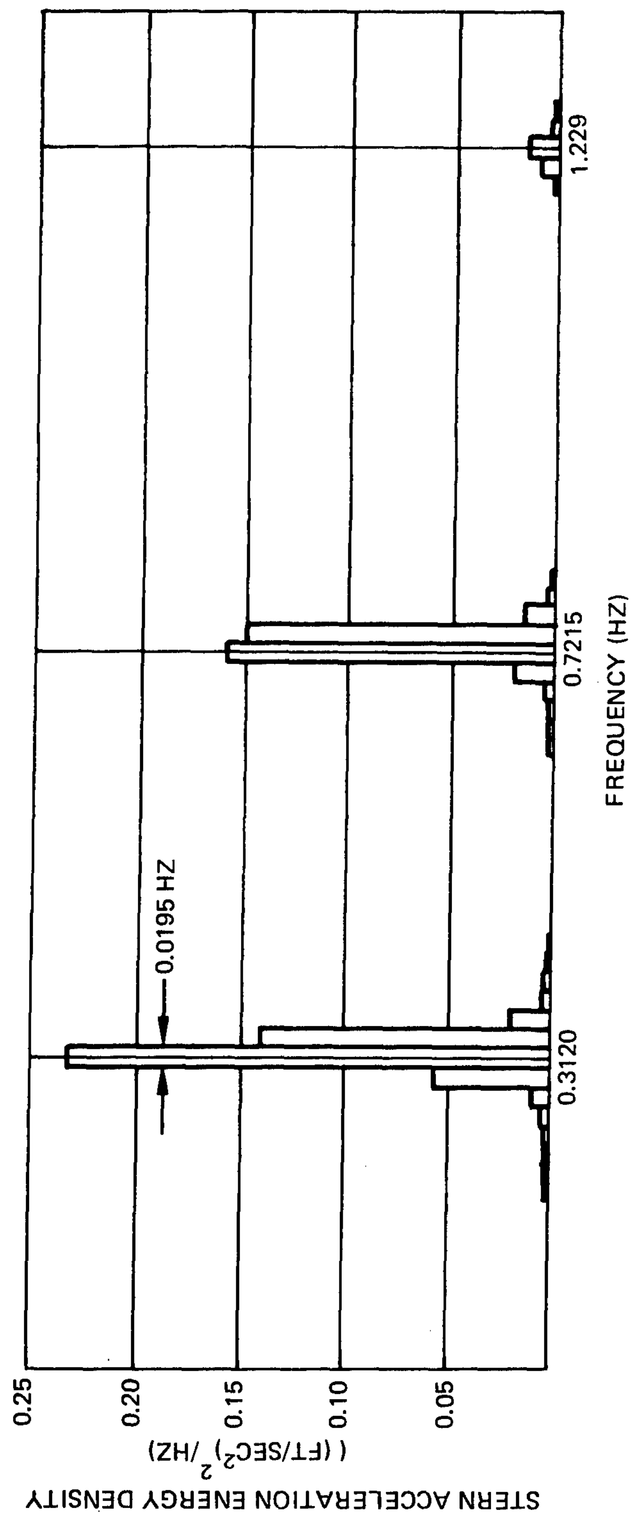

Figure 10 – Stern Acceleration Spectrum

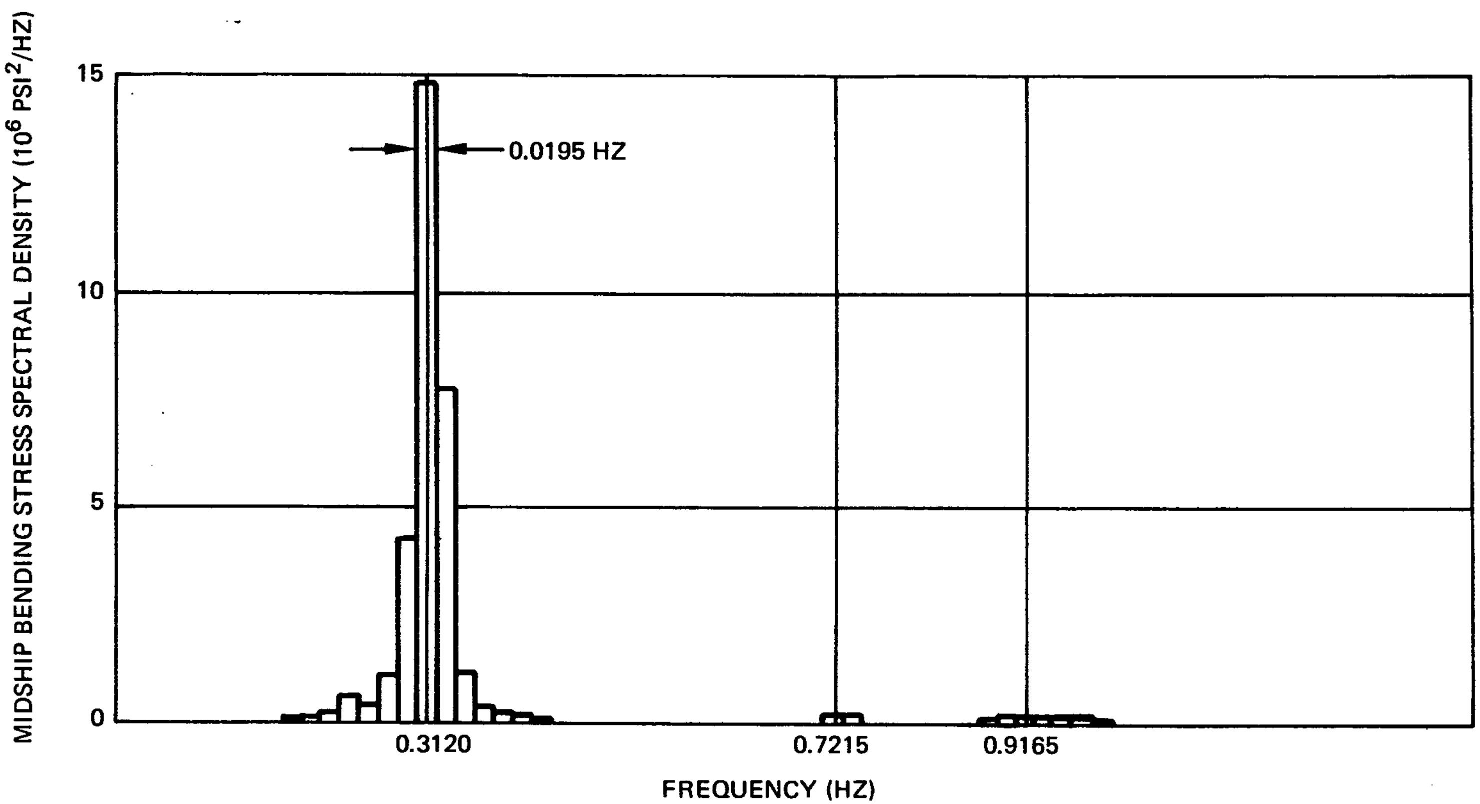

Figure 11 – Midship Bending Stress Spectrum

bow gages were wired to cancel any strain contribution due to horizontal bending. Therefore, the magnitude of the 0.9165-Hz response in Figure 11 has no validity or physical meaning. For this reason, it cannot be established whether or not the horizontal vibrations observed represent springing in the horizontal direction.

RMS Hull Deflection Profiles

A value of mean square acceleration associated with each response frequency and gage location was obtained by computing the area under each response peak in Figures 6 to 10. An rms value of deflection was then determined for each response frequency and gage point by dividing the square root of a given mean square acceleration by the square of the appropriate peak response frequency. The resulting rms deflections are listed in Table 4; the bow deflections shown there were obtained by averaging the starboard and port bow values.

Figure 12 is a plot of the rms deflection profiles for the first three vertical bending modes based on the values of Table 4. In plotting these values, it was necessary to introduce the phasing appropriate to each mode since Table 4 provided only the deflection magnitudes.

RMS Midship Bending Stress

By taking the square root of the area under the 0.3120-Hz peak of Figure 11, the rms stress associated with the first mode response was found to be 790 psi.

It is interesting to note that the third mode response that had been observed in the acceleration spectra of Figures 6 to 10 was not present in the stress spectrum of Figure 11. This indicates that the accelerometer instrumentation was apparently more effective than the strain gages in picking up the third mode. The low second mode response is attributed to the presence of an inflection point in the vicinity of the midpoint of the ship for the second mode.

CORRELATION OF MAXIMUM AND RMS VALUES

The first mode rms deflection values of Table 4 may be used to predict maximum deflection values for the 18-min data record under consideration. A comparison between these predicted maxima and those measured directly from the record (one-half of the deflections in Table 1) provides a check on the accuracy of the first mode rms values.

The acceleration spectra of Figures 6 to 10, from which the rms deflections were obtained are narrow band and therefore should follow a Rayleigh distribution. Since the Rayleigh distribution applies, the maximum record value y_{max} may be related to the root mean square of the peak record values $\sqrt{E}$ and the number of peak-to-peak record variations N

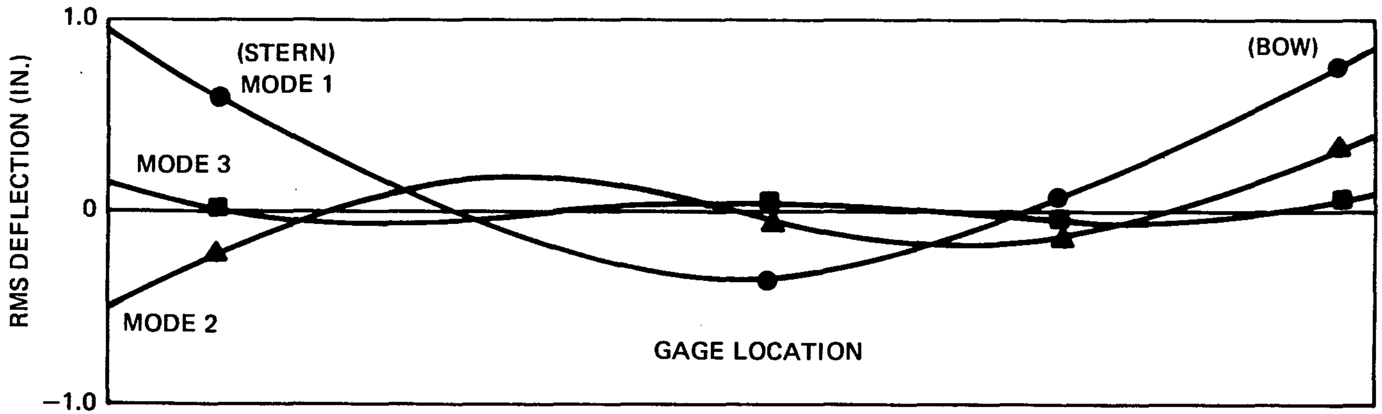

Figure 12 – RMS Hull Deflection Profiles–Modes 1 to 3

by the expression given by Longuet-Higgins[6] (see page 254):

$$y_{max} = \sqrt{E}\sqrt{\log_e N} \tag{3}$$

where $\sqrt{E} = \sqrt{\sum_{i=1}^{2N} y_i^2/2N}$ (see Appendix D) and y_i is the amplitude of the i^{th} record peak or trough. The quantity $\sqrt{E}$ may also be related to the rms record values by

$$\sqrt{E} = \sqrt{2}\ \text{rms} \tag{4}$$

for the narrow-band record under consideration. Inserting Equation (4) into Equation (3),

$$y_{max} = \sqrt{2}\ \text{rms}\sqrt{\log_e N} \tag{5}$$

Table 5 compares measured and predicted maximum first mode deflections y_{max} for gage locations along the CORT deck. The predicted values were obtained by substituting the rms values of Table 4 and an N value of 335 into Equation (5). The differences between measured and predicted maxima varied between 13 and 20 percent.

A similar comparison was made for midship bending stress by employing the same procedure used for deflections. The predicted maximum of 2700 psi was 20.5 percent larger than the measured maximum of 2240 psi. The percentage differences found above between measured and predicted maximum values are considered to be within acceptable bounds for the type of statistical predictions made.

HULL DAMPING EVALUATION

A series of anchor drop tests was conducted during the CORT round trip voyage in order to determine the first flexural mode hull damping in the ballast and the loaded conditions. Appendix C summarizes the test conditions for the ten tests performed, five in the ballast and five in the loaded condition.

The tests were carried out as follows. An anchor was lowered to the waterline and stopped. The windlass brake was released, permitting the anchor to drop. After the one shot marker appeared on the anchor chain, indicating that a 90-ft section of chain was out, the windlass brake was applied to stop the anchor as quickly as possible. When both anchors were dropped simultaneously, the same procedure was applied for each.

During each test, the output of the five accelerometers shown in Figure 1 and the Teledyne midship strain gage bridge were recorded on magnetic and paper tape using the

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[6] Longuet-Higgins, M.S., "On the Statistical Distribution of the Heights of Sea Waves," J. Marine Res., Vol. 2, pp. 245–266 (Dec 1952).

TABLE 4 – VERTICAL RMS HULL DEFLECTIONS

Gage Point	Mode 1 in.	Mode 2 in.	Mode 3 in.
Bow	0.754	0.327	0.065
Quarter Point	0.035	0.119	0.043
Midpoint	0.349	0.042	0.037
Stern	0.597	0.221	0.035

TABLE 5 – MEASURED VERSUS PREDICTED MAXIMUM FIRST MODE DEFLECTIONS

Gage Point	RMS Deflection in.	Maximum Deflection		Difference percent
		Predicted in.	Measured in.	
Bow	0.754	2.57	2.19	+17.4
Quarter Point	0.035	0.12	0.15	+20.0
Midpoint	0.349	1.19	1.05	+13.3
Stern	0.597	2.04	1.70	+20.0

instrumentation system in Figure 2. As illustrated in Figure 2, the port bow, midpoint, and stern signals were filtered on board by using a bandpass of 0.2 to 2 Hz.

The magnetic tapes for Tests 6–10 (Appendix C) were filtered in the Ship Acoustics Department using the instrumentation system in Figure 3 and a bandpass of 0.2 to 0.4 Hz. Figures 13 and 14 are reproductions of the filtered and unfiltered records for Tests 6 and 7. The figures show the recorded starboard bow accelerometer and strain gage signals beginning with a few cycles before the anchor was released and ending a few cycles after the anchor was stopped by application of the windlass brake.

The paper tapes were reviewed for a general type of attenuated oscillation signal which could serve as the basis for a logarithmic decrement and thereby a damping determination. It became apparent from this review that a reliable damping assessment could not be made. The fundamental obstacle was the presence of significant seaway-induced first mode vibrations during, before, and after the actual anchor drop (see Figures 13 and 14). This first mode vibration exhibited a period of 3 sec and the characteristic beating phenomenon associated with springing. Therefore, it appears that springing occurred (see Figures 13 and 14) while the CORT was stationary and being readied for the anchor drop tests. The amplitudes of accelerations measured from the anchor drop records were in the range of 0.001 to 0.05 g.

The following additional observations during the review of the filtered anchor drop records may further explain why a reliable damping assessment was not possible.

1. Some of the responses were no greater than the amplitude of the acceleration preceding or following the anchor drops (see Figure 13).

2. It appeared that for many of the tests, the only effect of the braking was to temporarily disrupt the general motions that already existed due to prevalent sea conditions. This observation is brought out most clearly by the strain gage signals in Figure 13.

3. Finally, in the few tests in which some type of attenuated oscillation was exhibited, it was not possible to determine the portion due to the characteristic vibrations (increasing and decreasing amplitude) associated with springing and the portion correctly attributable to hull damping (see Figure 14).

SUMMARY AND CONCLUSIONS

This report deals with the findings of underway hull vibration and damping evaluations carried out on the ore carrier STEWART J. CORT during a round trip voyage between Burns Harbor, Indiana, and Taconite Harbor, Minnesota, during 13 to 17 August 1972.

Underway accelerometer and strain gage data were collected on both legs of the trip with the CORT in both the loaded and ballast conditions. The largest amplitude springing

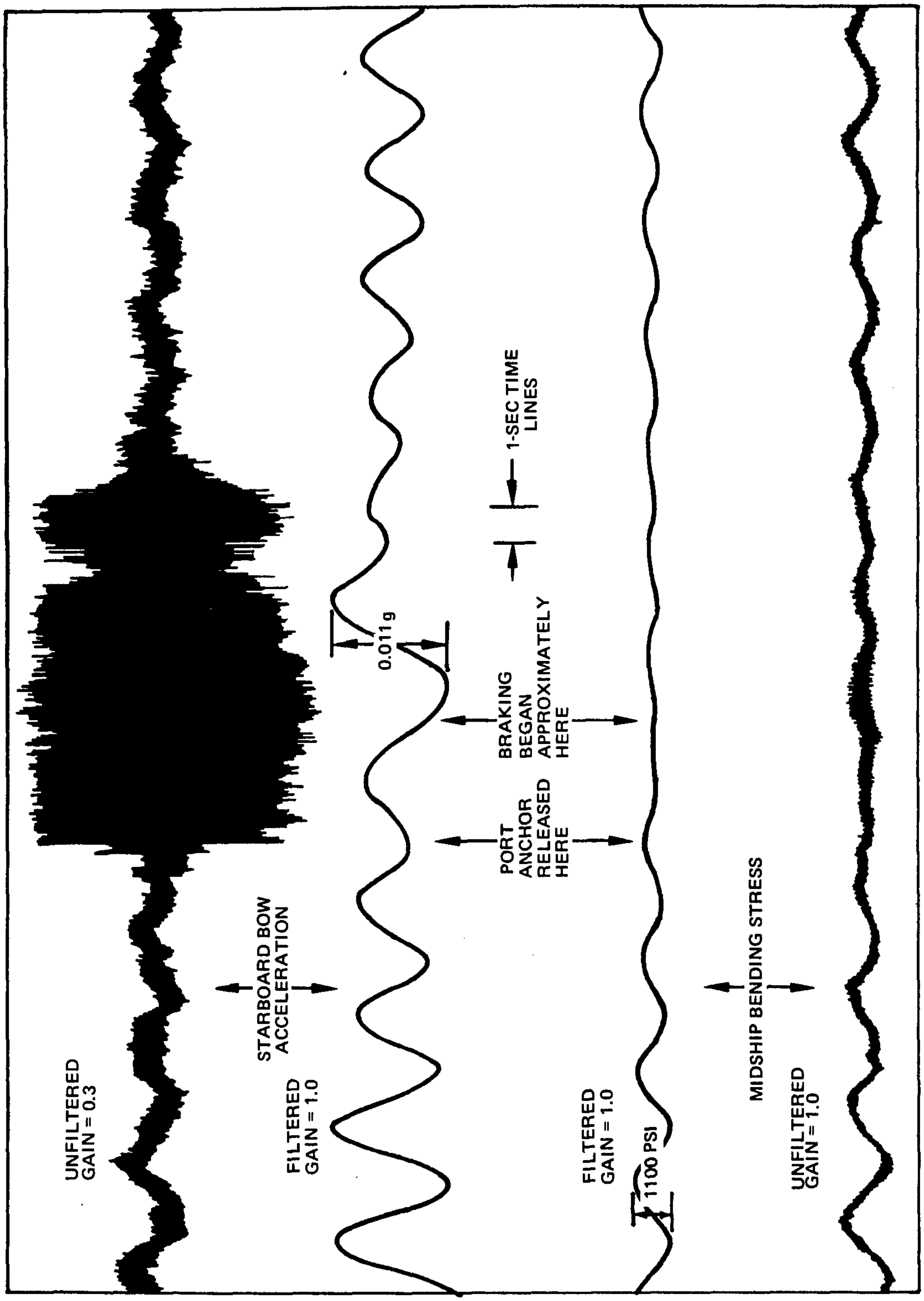

Figure 13 – Response Record for Anchor Drop Test 6

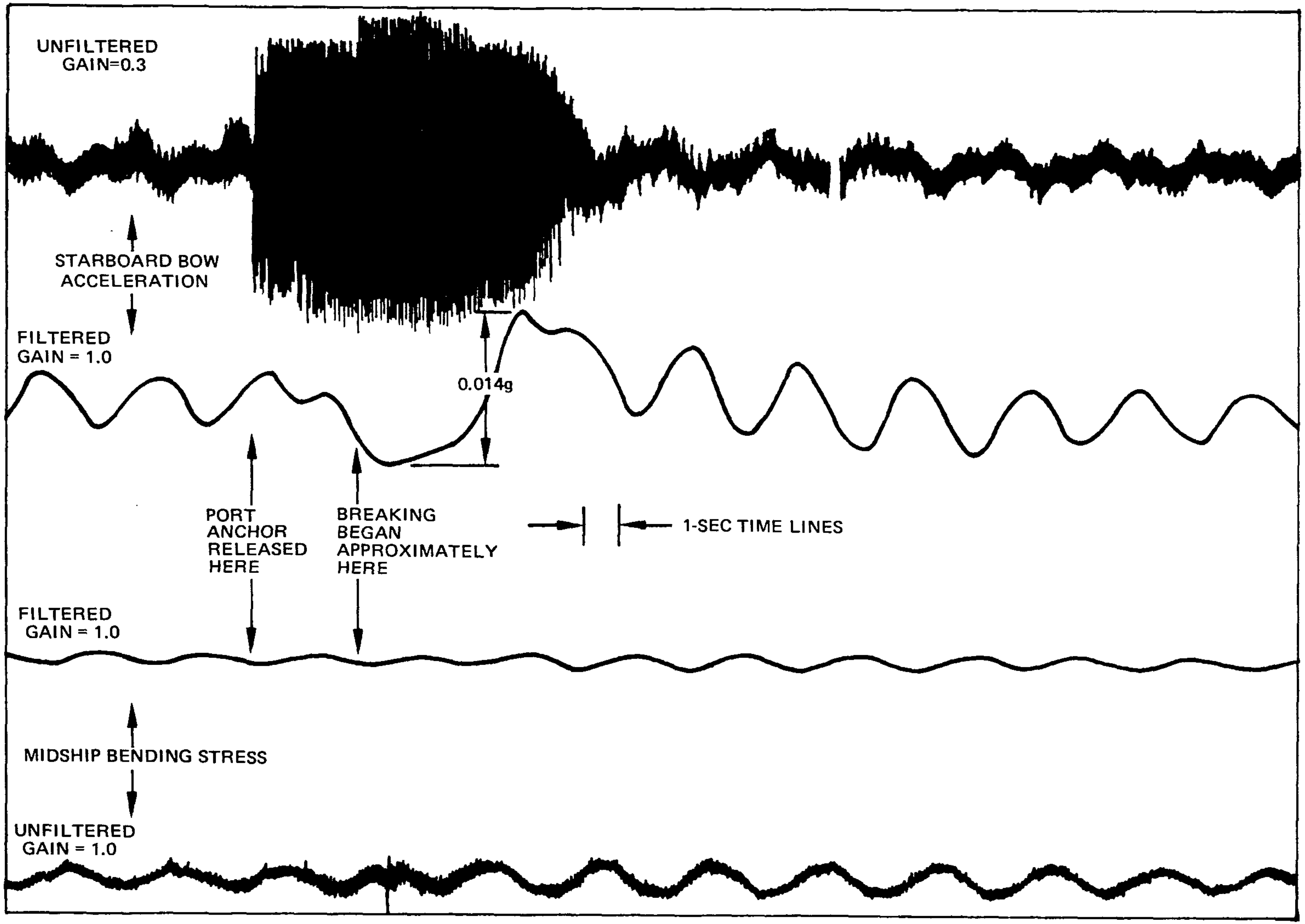

Figure 14 – Response Record for Anchor Drop Test 7

data were recorded in Lake Superior 150 miles east of Taconite Harbor during a 1-hr period when CORT (in the loaded condition) encountered 4- to 6-ft seas at 60 deg to the bow. The maximum hull deflection profile for the CORT first flexural vibratory mode was obtained (Figure 5) from a direct analysis of this 1-hr data record. The contribution of the first mode to the CORT deflection profile was approximately 5 in. peak to peak at the bow and 2 in. at the midpoint.

Midship bending stresses of 2300 and 2510 psi were predicted from the CORT deflection profile and Equations (1) and (2); these agreed with the strain-based value of 2240 psi within 2.7 and 12.1 percent, respectively. Therefore, with 2240 psi as a reference value, it appears that for such predictions the Vedeler formula, Equation (1), is superior to that of the analogous formula, Equation (2), derived on the basis of the Euler Bernoulli beam theory. Furthermore, this suggests that Equation (1) can be expected to permit the computation of midship stress from deflections and vice versa with reasonably good accuracy, at least for CORT.

A spectrum analysis of the accelerometer records produced the vibration frequencies for the first, second, and third modes of CORT of 0.3120, 0.7215, and 1.229 Hz, respectively. This determination was based in part on the close agreement between ratios of second to first mode frequencies and third to first mode frequencies for CORT and another ore carrier, C. A. PAUL (see Table 2).

The rms hull deflection profiles associated with the above frequencies were also determined (Figure 12). An important finding was that the second mode rms deflection profile involved deflections that were larger than expected in relation to the first mode profile. However, although the data of the present study indicated a sizeable second mode component in the deflection profile of CORT during springing, the extent to which this finding applies to bulk carriers or ships in general remains to be determined. Lastly, the unexpectedly large second mode rms deflection profile did not contribute to the midship bending stress because the ship midpoint is an inflection point. The second mode, however, will augment the first mode stresses at other points along the CORT deck.

Maximum values of deflection and stress were predicted from rms values based on statistical theory for narrow-band spectra. It was found that the predicted maxima agreed with the measured maxima to within 13 to 20.5 percent; this agreement is considered satisfactory for this type of statistical prediction.

In addition to the first and second mode responses at 0.3120 and 0.7215 Hz in the midship stress spectrum (Figure 11), a small response was also observed at 0.9165 Hz. It was concluded that this response corresponds to the horizontal (athwartship) vibrations which were experienced during the data recording period and which were estimated to occur at 0.9 Hz. The level of athwartship vibrations was no doubt considerably larger than indicated by this spectrum because the midship strain gages were wired together to cancel out strains related to athwartship motion. Accordingly, it is not possible to establish whether or not the horizontal vibrations observed represent springing in the horizontal direction.

A reliable evaluation of the CORT first mode damping could not be obtained from the anchor drop tests primarily because throughout these tests, CORT was observed to be springing while at zero speed.

ACKNOWLEDGMENTS

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APPENDIX A

CORT PRINCIPAL CHARACTERISTICS AND HULL FORM DATA

CHARACTERISTICS

Length Overall, ft – in.	1000–0
Length between Perpendiculars, ft – in.	988–6
Length on Waterline, ft – in.	998–0
Beam Molded, ft – in.	104–7 1/4
Depth at Side, Molded, ft – in.	49–0
Camber (straight line), in.	0
Tumble Home, in.	0
Deadrise, in.	0
Bilge Radius, in.	15
Maximum Draft, Summer, Great Lakes, MLD, ft – in.	27–10
Displacement, long tons (at SLL MLD)	74,400
Light Ship Weight, long tons	15,510*
Deadweight, Total, long tons, at 27 ft – 10 in.	58,890
Break Horsepower, Normal	14,800
Shaft Horsepower, Normal	14,000
Revolutions per Minute at Normal Power	900
Propeller Diameter, ft–in.	18.0
Number of Blades	4
Gross Tonnage (est.)	33,000
Net Tonnage (est.)	30,000
Crew Accommodations	33
Operating Ballast Drafts, ft – in.	9–0 fwd, 20–6 aft
Total Cubic, ft^3	1,647,705
Ballast Cap. Total (all tanks except FP), long tons	38,872
Section Modulus as built, HTS, $in^2 - ft$	92,169

HULL FORM DATA

Displacement Length (LBP) ft – in.	988–6
Beam Molded, ft – in.	104–7 1/4
Design Draft, Molded, ft – in.	25–9
Displacement, Molded, Fresh Water, long tons	68,330
Length-Beam Ratio	9.45
Beam-Draft Ratio	4.06
Length of Entrance, ft	160.0
Length of Parallel Middlebody, ft	736.0
Length of Run, ft	92.5

Run – Entrance Ratio	0.578
Block Coefficient (LBP)	0.924
Prismatic Coefficient (LBP)	0.924
Midship Coefficient (LBP)	0.999
Water Plane Coefficient (LBP)	0.975
Displacement – Length Ratio $\Delta/(LBP/100)^3$	70.74
Longitudinal Center of Buoyancy, feet forward midships (LBP)	4.45
Wetted Surface, square feet	150,462
Designed Sea Speed, mph at 25 ft – 9 in.	16.00
Designed Sea Speed (VK), knots	13.89
Speed-Length Ratio $VK/\sqrt{LBP}$	0.442

*No crew, stores, fuel.

APPENDIX B

INSTRUMENTATION LIST

ON-BOARD	LABORATORY
Servoaccelerometer/Kistler Amplifier System, Model 305T/515T Linearity Deviation (accuracy) = 10^{-5}g	Burr-Brown Amplifier, Model 1631
Burr-Brown Amplifiers **Model 1631:** Gain 0–10 (adjustable), frequency response, dc to 5 kHz Model 1632: Gain 10/100, frequency response, dc to 10 kHz	Kron-Hite Filter, Model 330 A Bandpass 0.04–2000 Hz (adjustable) Consolidated Electrodynamics Corp Oscillograph, Model 5–124 Paper speed – 0.25 in/sec
Kron-Hite Filter, Model 3550 Low Pass 0.2 – 2 Hz	Tektronix Oscilloscope, Type 323
Bell & Howell Oscillograph, Model 5-134 Paper speed – 0.1 in/sec	
Tektronix Oscilloscope, Type 323	Lockheed Tape Recorder, Model 417 Recording speed 1 7/8 in/sec, FM frequency bandwidth 0 to 625 Hz (±0.5 dB over the frequency range)
Lockheed Tape Recorder, Model 417 Recording speed 1 7/8 in/sec, FM frequency bandwidth 0 to 625 Hz (± 0.5 dB over the frequency range)	Hewlett Packard Oscillator, Model 3310 A FFT Analyzer, Model TD 1923-C (Manufactured by Time/Data, a subsidiary of General Radio)

APPENDIX C

MEASUREMENTS TAKEN DURING THE EVALUATION

TABLE C.1 – SUMMARY OF UNDERWAY VIBRATION MEASUREMENTS

Time and Date	Location miles	Condition	Wave Height ft	Rel. Heading deg	Ship Speed rpm	Remarks
1622–1722, 8/14	70 W of Whitehead Bay	Ballast	<1	–	16	–
1536–1625, 8/15	40 E of Taconite Harbor	Loaded	1–2	0	–	–
2051–2134, 8/15	125 E of Taconite Harbor		–	45	–	Noticeable rolling
2347, 8/15–0045, 8/16	175 E of Taconite Harbor		4–6	60	15.7	Springing and horizontal Vibrations (0.9 Hz)
0821–0850, 8/16	310 E of Taconite Harbor		4–6	15	16.4	–
1240–1305, 8/17	150 N of Burns Harbor	Loaded	1	0	–	Swells

TABLE C.2 – SUMMARY OF ANCHOR DROP MEASUREMENTS

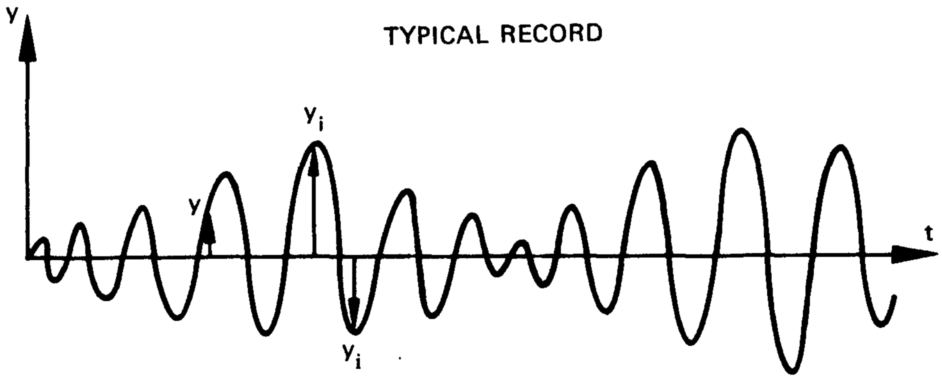
(Ship speed was zero for the anchor drop tests.)

Test No.	Time & Date	Location	Condition	Water Depth ft	Approximate Draft Fwd ft in.	Approximate Draft Midpoint ft in.	Approximate Draft Aft ft in.	Wave Height ft	Relative Heading deg	Anchor Dropped	Length of Chain out from Windlass	Distance/ Time to Stop Anchor	Remarks
1	1115, 8/14	Whitefish Bay	Ballast	130	18–6	21–4	24–10	<1	–	Stbd	1 Shot (90 ft)	30 ft	Hit bottom
2	1125, 8/14	Whitefish Bay		130	18–6	21–4	24–10		–	Stbd	1 Shot (90 ft)	20 ft	–
3	1140, 8/14	Whitefish Bay		130	18–6	21–4	24–10		–	Both	Port: 1 shot + 11 links Stbd: 1 shot + 20 links	Port: 11 ft Stbd: 20 ft	–
4	0718, 8/15	Taconite Harbor		750	19–11	20–7	21–7		–	Both	Port: 1 shot + 48 links Stbd: 1 shot at WL	Port: 48 ft Stbd: 30 ft within 2 sec of each other	–
5	0725. 8/15		Ballast		19–11	20–7	21–7	<1	–	Both	Port: 2 shot at WL Stbd: 1 shot 20 ft below WL	Port: 30 ft Stbd: 50 ft within 1 sec	Began braking at 2 shot mark
6	1405, 8/15		Loaded		–	–	–	1–2	0	Port	Not quite 1 shot	1–2 sec	–
7	1415, 8/15				–	–	–	1–2		Port	Not quite 1 shot	1–2 sec	–
8	1425, 8/15	Taconite Harbor		750	–	–	–	1–2		Both	Both 1 shot	Within 1 sec	Perfect braking coordination
9	0623, 8/17	Northern Lake Michigan		630	–	–	–	1		Both	Both 1 shot	1 sec	–
10	0632, 8/17	Northern Lake Michigan	Loaded	630	–	–	–	1	0	Both	Both 1 shot	1 sec	–

APPENDIX D

DEFINITIONS OF STATISTICAL PARAMETERS

TYPICAL RECORD



y_i — peak record displacements about reference line

y — instantaneous record displacement about reference line

Expected or Mean square of value of y_i

$$E = \overline{y_i^2} = \frac{\sum_{i=1}^{2N} y_i^2}{2N}$$

Root mean square value of y_i:

$$\sqrt{E} = \sqrt{\overline{y_i^2}}$$

where N = number of cycles variations in record.

Temporal mean square value of y:

$$\overline{y^2} = \lim_{T \to \infty} \frac{1}{T} \int_0^T y^2(t)\, dt$$

Root mean square value of y:

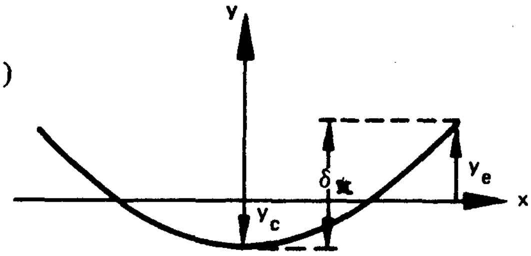
$$\text{rms} = \sqrt{\overline{y^2}}$$

APPENDIX E

MIDSHIP BENDING STRESS FORMULAS

FORMULA DERIVATION BASED ON EULER BERNOULLI BEAM THEORY

For a free-free uniform beam undergoing small deflection vibrations in its fundamental mode, the deflection y at any point along its length (see adjacent sketch) is related to the midpoint deflection y_c by the expression[1]



$$y = y_c \left(C_1 \cos \frac{mx}{L} - C_2 \cosh \frac{mx}{L} \right) \qquad \text{(E1)}$$

where L is the beam length and m, C_1, and C_2 are fundamental constants respectively equal to 4.73 radians, 1.1532 and 0.1532.

Bending stress σ is related to the deflection y by the expression

$$\sigma = Ec \frac{d^2y}{dx^2} \qquad \text{(E2)}$$

where E is Young's modulus of elasticity and c is the distance from the beam neutral surface to the outer free surface.

The substitution of Equation (E1) into (E2) yields

$$\sigma = \frac{Ecy_c m^2}{L^2} \left(- C_1 \cos \frac{mx}{L} - C_2 \cosh \frac{mx}{L} \right) \qquad \text{(E3)}$$

Evaluating Equations (E1) and (E3) at x = L/2 and x = 0, respectively, after inserting the fundamental constants stated above, we have

$$y(x = L/2) = y_e = - 1.66\, y_c \qquad \text{(E4)}$$

and

$$\sigma\,(x = 0) = \sigma_{\boxtimes} = - 29.228 \frac{y_c\, Ec}{L^2} \qquad \text{(E5)}$$

Referring to the sketch above, the beam midpoint to end point deflection is

$$\delta_{\boxtimes} = -y_c + y_e \qquad \text{(E6)}$$

Substituting (E4) into (E6),

$$\delta_{\boxtimes} = -2.66\, y_c \qquad \text{(E7)}$$

Finally, inserting $y_c = -\delta_{\boxtimes}/2.66$ from (E7) into (E5), the following formula for midship bending stress is obtained:

$$\sigma_{\boxtimes} = 10.99 \frac{\delta_{\boxtimes} Ec}{L^2} \tag{E8}$$

THE VEDELER MIDSHIP STRESS FORMULA

The Vedeler formula for midship bending stress is based on the following assumptions:[4]

1. Bending moment is maximum at the midship point.
2. Bending moment has a parabolic distribution within the region $-0.45L < x < 0.45L$ (see sketch).
3. Bending moment is zero in the regions $-0.5L < x < -0.45L$ and $0.45L < x < 0.5L$.
4. Moment of inertia is constant along the entire length of the ship.

APPENDIX F

PREDICTION OF FIRST HORIZONTAL (ATHWARTSHIP) BENDING MODE FREQUENCY

The Johnson formula[5] is given by

$$\frac{f_H}{f_V} = \sqrt{\frac{I_H \times M_V}{I_V \times M_H}}$$

where f_H is the first mode horizontal bending frequency,

f_V is the first mode vertical bending frequency,

I_H is the midship section area moment of inertia for bending in the horizontal plane,

I_V is the midship section area moment of inertia for bending in the vertical plane,

M_V is the total mass for vertical vibration (ship mass plus added mass), and

M_H is the total mass for horizontal vibration (ship mass plus added mass).

Substituting $f_V = 0.31$ Hz, $I_H/I_V \approx 3$ (based on rough calculation), and $M_V/M_H = 2.39$ (this ratio was computed based on the formulas and procedure of Bruck[7] and the CORT characteristics of Appendix A) into the Johnson formula above, an estimate of the first mode horizontal bending frequency is computed to be $f_H = 0.85$ Hz.

[7] Bruck, H.A., "Procedure for Calculating Vibration Parameters of Surface Ships," NSRDC Report 2875 (Oct 1968).

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82	C	1962	ZALOUMIS ANGELOS
83	C	1962	ZALOUMIS ANGELOS
84	C	1962	ZALOUMIS ANGELOS
85	C	1962	ZALOUMIS ANGELOS
86	C	1962	ZALOUMIS ANGELOS
87	C	1962	ZALOUMIS ANGELOS
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13. ABSTRACT

The hull vibration (springing) of the Great Lakes ore carrier STEWART J. CORT was evaluated on the basis of accelerometer and strain gage data collected during a regular round trip voyage between Burns Harbor, Indiana, and Taconite Harbor, Minnesota, in August 1972. Springing was recorded when in the loaded condition, CORT encountered 4- to 6-ft seas, 60 deg to the bow in Lake Superior. The maximum first mode deflection profile and midship bending stress were determined from a direct analysis of this springing record. A spectrum analysis of the same record produced flexural frequencies of vibration for the first three vertical modes and what appeared to be the first horizontal mode, rms deflection profiles for the first three vertical modes, and the first mode rms midship bending stress. An important finding of the investigation was that the springing deflections of the CORT involved larger contributions than expected from the second flexural mode of vibration. Although the contribution of the second mode to midpoint bending stress is theoretically zero, its contribution to bending stresses at other points merits consideration. An evaluation of first vibratory mode damping by means of ten anchor drop tests was unsuccessful, partly because CORT underwent springing while at zero speed in a 1-ft sea. The springing appeared to interfere with the anchor-drop-induced ship response.

14 KEY WORDS	LINK A		LINK B		LINK C	
	ROLE	WT	ROLE	WT	ROLE	WT
Springing						
Carrier (Ore, Bulk)						
M/V STEWART J. CORT						
Great Lakes						
Hull Vibration						
Response (Vibratory, Dynamic, Transient)						
Vibration Measurements (Evaluation)						
Sea Trial, Full Scale Tests						
Longitudinal Strength						
Frequencies, Modes						
Stresses, Deflections, Accelerations						
Hull Damping						
Spectrum Analysis						
Spectra (Narrow Band)						
RMS Values						