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AN EVALUATION OF A DAMPED VIBRATION ABSORBER IN ROUGH SEAS, (U)  
JUN 78 P A FARRELL, G LONG  
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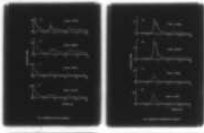
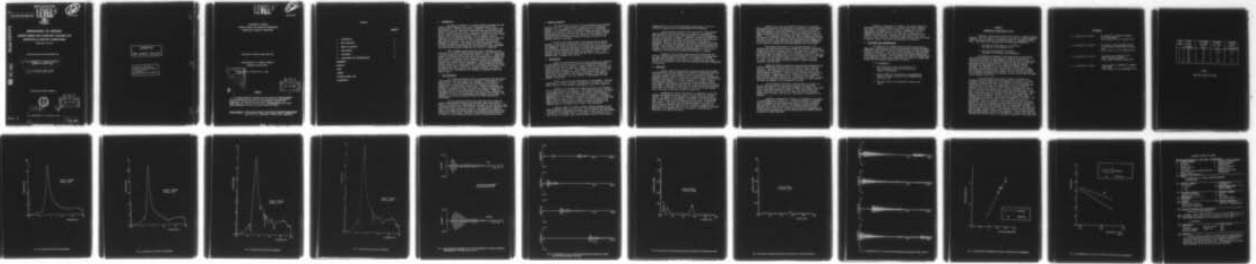
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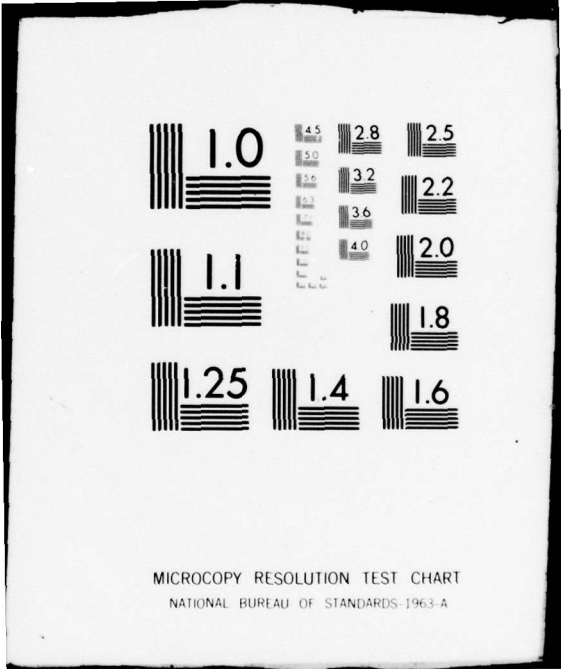
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Structures Technical Memorandum 280

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AN EVALUATION OF A DAMPED VIBRATION  
ABSORBER IN ROUGH SEAS

10  
P.A. Farrell and G. Long

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SUMMARY

↙ A damped vibration absorber has been fitted to HMAS Balikpapan and the responses of the ship and of the absorber were measured in rough seas. The effectiveness of the absorber is shown to be essentially the same as in calmer seas. Recommendations for tuning future absorbers on the basis of sea trials are made. ↗

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**CONTENTS**

**Page No.**

1.	INTRODUCTION	1
2.	TEST CONDITIONS	1
3.	METHOD OF ANALYSIS	2
4.	TEST RESULTS	2
5.	DISCUSSION	3
6.	CONCLUSIONS AND RECOMMENDATIONS	5

**REFERENCES**

**APPENDIX**

**TABLE**

**FIGURES**

**DOCUMENT CONTROL DATA**

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## 1. INTRODUCTION

As stated in reference 1, a damped vibration absorber has been fitted to HMAS Balikpapan and an initial assessment made of its characteristics. The data presented in reference 1 were measured while the ship was vibrated firstly by a mechanical vibrator and then by the action of slight seas (sea state 1) and these two methods of excitation did not cause the amplitude of the absorber to exceed  $\pm 20$  mm ( $\pm 0.8$  ins) which is 16% of its design amplitude. For the reasons listed in reference 4 the absorber was designed to use commercial air springs and these springs are non-linear i.e. their spring-rate depends on the amplitude of vibration. Thus whilst this previous test indicates a satisfactory behaviour of the absorber at small amplitudes ( $< \pm 20$  mm) it still remained to demonstrate its effectiveness at larger amplitudes. This could best be done by conducting a sea trial in rough seas.

HMAS Balikpapan sailed from HMAS Moreton (Brisbane) on the 14th March and arrived at HMAS Waterhen (Sydney) on the 16th March after a voyage through seas which varied from state 4 to state 1. During this voyage the vibrations of the ship and of the absorber were recorded for various values of absorber tuning (achieved by altering the pressure in the absorbers air-springs) and the resulting records were subsequently analysed using digital Fourier techniques. These analyses showed that although operating at larger amplitudes the absorber functions correctly and the tuning curves of reference 1 are still applicable; however it is recommended that fine tuning of the absorber be based on sea trials. The results also indicated that the absorber was operating within its design limits.

## 2. TEST CONDITIONS

The Landing Craft-Heavy (LCH) was instrumented with four servo-accelerometers and their outputs were recorded on a four-channel, F.M. tape-recorder. One accelerometer was attached to the absorber at a distance of 0.94 metres from the pivot and measured the vertical acceleration of the absorber at this point. Another accelerometer was mounted on a bulkhead at frame 70 to give the vertical acceleration of the ship, adjacent to the accelerometer on the absorber. These two accelerometers were mounted such that errors due to their rotation were minimised. The last two accelerometers were mounted in the bridge - one measuring the vertical acceleration whilst the other measured the longitudinal.

All the records were taken whilst the ship was in water of depth of approximately 200 metres and the ship's mass varied from 522 tonnes on the first day to approximately 512 tonnes on the last day. Five air-spring pressures were used and these, together with the subjective sea state at the time, are given in table 1. The pressures are given in PSIG as the permanently-mounted gauges which measure these pressures are calibrated in these units. Similarly all accelerations have been given in terms of  $g$ , the acceleration due to gravity (9.8 metres/sec./sec.)

### 3. METHOD OF ANALYSIS

The time history from each of the accelerometers was recorded unfiltered on an F.M. tape-recorder in analogue form for subsequent analysis. The accelerometer on the absorber gives its acceleration relative to an inertial frame of reference whereas the present analysis requires the vibration relative to the ship. Therefore before analysis the acceleration measured on the ship, adjacent to the absorber, was subtracted from the absorber acceleration, giving the motion of the absorber relative to the ship.

The data were analysed using a dual-channel digital Fourier analyser. Each record was digitised by this analyser at the rate of 25.6 samples per second and stored in 128 blocks, each of 512 points with 50% redundancy. Each block was scaled by the Hanning weighting function and its frequency spectrum determined. The means of these 128 spectra are shown for the roughest sea conditions in figures 1 and 2. The vertical scale in these figures gives the RMS value of acceleration as measured over a bandwidth of 0.075 Hz.

### 4. TEST RESULTS

Results are presented only for the four roughest sea conditions as the calmest sea did not excite the hull fundamental bending mode sufficiently for accurate analysis. Figure 1 shows the acceleration of the LCH at frame 70 as a function of frequency for each of the four sea states analysed (see table 1). The response of the vessel in the rigid body pitch mode is deleted for clarity. The pressure shown beside each curve in this figure is the indicated pressure in the upper air springs (the pressure in the lower springs is approximately 14 PSIG greater in each case).

Figure 2 gives the same information for the absorber. Here the acceleration is that of a point on the absorber 0.94 metres aft of the pivot, relative to the adjacent accelerometer on the LCH at frame 70.

Figures 3 to 6 present the transfer functions of the absorber for the four roughest sea conditions. The transfer function is the ratio of the displacement of the absorber at its centre of percussion to the displacement of the ship at the same frame and so gives the dynamic magnification of the absorber as a function of frequency. The dynamic magnification at the absorber's resonance (the height of the single large peak in each of figures 3 to 6) is equal to half the inverse of the damping ratio of the absorber and from this simple relationship the dampings shown on the figures and in table 1 were determined.

Figure 7 shows the largest transient response measured on the ship and absorber during the sea trial. The acceleration data measured were converted to displacement and the results scaled to give the response at frame 68, the absorber centre of percussion. The response of the ship, shown in the upper curve, has also been filtered to show only the



displacement in the two node bending mode (approximately 3.4 Hz) to facilitate comparison with the displacement of the absorber.

All the figures presented so far have been for data obtained from the absorber and from the ship in the region of the absorber. HMAS Labuan is a sister ship of HMAS Balikpapan but unlike Balikpapan it is not fitted with an absorber. Reference 2 presents vibration data measured on the bridge of Labuan during an earlier sea trial and so to facilitate comparison, data were also measured on the bridge of Balikpapan. Figure 3 shows typical fore-and-aft acceleration histories measured on the bridge of Balikpapan when the upper spring pressure was 68 PSIG. These histories have been digitally filtered to show only the response in the two node bending mode (approximately 3.4 Hz) as it is vibration in this mode that the absorber is designed to suppress.

Figures 9 and 10 present respectively the fore-and-aft and vertical acceleration spectra measured on the bridge of Balikpapan when the upper spring pressure was 68 PSIG. This pressure is near the optimum (see next section) and these data are presented in the range 0-50 Hz as an indication of the rough sea vibration environment experienced on the bridge.

## 5. DISCUSSION

Figure 1 shows the acceleration spectra measured near the bow on Balikpapan for the four roughest sea conditions encountered during the sea trial. Since the sea conditions varied for each of the spring pressures examined a direct comparison of the amplitudes of the measured responses cannot be made. It is not even correct to compare ratios of the heights of the peaks at the two-node frequency (~ 3.4 Hz) to those at the three-node frequency (~ 6.5 Hz) as the relative heights of these peaks depend on the frequency distribution of the wave energy and this is not constant. Reference 3 states that the optimal tuning of the absorber occurs when the two peaks in the region of 3.4 Hz are of equal height. This is the case for 68 PSIG as shown in figure 1(b).

Comparing this figure with figure 3 of reference 3 shows that although the frequency tuning is correct (the two peaks are of equal height), the damping in the absorber is too small as indicated by the ratio of the height of these peaks to the height of the trough between them. This is borne out by the damping shown in table 1 as calculated from figure 4. This shows that the absorber damping for 68 PSIG is 4.6% of critical whereas equation A1.8 of reference 3 indicates the damping should be of the order of 10% of critical, depending on the exact mass ratio. The Appendix contains a discussion of the problem of determining the correct mass ratio.

Figures 3 to 6 show the transfer functions of the absorber. These transfer functions are essentially the responses of single degree-of-freedom systems to input displacements at the end of the springs. Hence the damping of that single DOF system can be determined from the simple relationship given in section 3 and these dampings are given in table 1. It is apparent that although the number of dampers attached to the absorber was unaltered during the test, the resulting damping varied with the sea state. The commercial shock-absorbers used as dampers are such that in the roughest seas when the amplitude (and hence velocity) of the absorber is greatest the resulting damping rate is the least.

The maximum displacement of the absorber depends on the exciting force and the sea trial was intended to check that the maximum excursion was within the design limits. The lower curve of figure 7 shows a time history of the displacement of the absorber at its centre of percussion (near the centre line of the air springs) relative to the ship. This curve shows the maximum excursion of the absorber to be of the order of  $\pm 80$  mm ( $\pm 3$  ins) which is within the design limit of  $\pm 127$  mm ( $\pm 5$  ins). This particular transient showed the largest amplitude recorded on the sea trial.

Figure 11 presents acceleration time histories measured on the bridge of HMAS Labuan (an LCH without an absorber) during a previous sea trial. Again the histories have been filtered to show only the response in the fundamental hull mode and they show that the LCH has little damping in this mode. Figure 8 shows equivalent histories for an LCH fitted with a damped absorber, the presence of which is apparent from the beats in these latter curves. A comparison of figures 8 and 11 shows that the absorber (although not ideally tuned, as stated earlier) increases the damping in this fundamental hull mode.

The fore-and-aft acceleration spectrum shown in figure 9 is for the same conditions as that shown in figure 1(b) except that whereas the former is for acceleration measured on the bridge, the latter is for acceleration measured at frame 70. Both curves exhibit twin peaks in the region of 3.4 Hz but in figure 1(b) they are of equal height while in figure 9 they are not. This point is discussed in the Appendix.

Reference 1 describes the tuning of an absorber and its figure 3 is reproduced here as figure 12. It shows the variation of absorber natural frequency with spring pressure and the four cases considered here are plotted onto this figure. The agreement between the points obtained from a rough sea trial and the curves obtained from small amplitude tests is very good, showing that possible non-linearities due to air spring characteristics do not greatly alter the absorber natural frequency at larger amplitudes.

Reference 1 also gives (as figure 8) two curves showing the recommended air spring pressure as a variation with ship's loading. This figure is reproduced here as figure 13 together with the point representing the combination of 522 tonnes and 68 PSIG. Of the combinations covered by the sea trial this one is considered to be closest to correct tuning. Although this point does not lie on the curve CD it does imply that this curve, rather than the curve AB, should be used for tuning. This point was unresolved in reference 1.

#### 6. CONCLUSIONS AND RECOMMENDATIONS

The operation of a damped vibration absorber was tested in rough seas. These seas did not cause the absorber to exceed its design amplitude and the absorber natural frequency varied with spring pressure as shown in reference 1. The action of the absorber was such that hull vibration in the two-node vertical bending mode decayed rapidly after each transient excitation. The measurements show that the effectiveness of the absorber in rough seas is essentially the same as in calmer seas except that the resultant damping rate is reduced.

It is recommended:

1. that curve CD of figure 8 in reference 1 be used in preference to curve AB for initial tuning of the absorber,
2. that the damping of the absorber be approximately doubled, the exact increase to be determined from sea trials, and
3. that fine tuning of the absorber be based on sea trials.

## APPENDIX

### MATHEMATICAL IDEALISATION OF SHIP

Reference 3 describes the design and tuning of a damped vibration absorber. For that analysis, the ship was idealised as a single degree-of-freedom system for vibration at frequencies near its fundamental bending mode. This idealisation incorporates two simplifying assumptions:

1. that the hull mode shape is not altered by the addition of the absorber, and
2. that the hull mode shape is constant throughout the frequency range considered.

The formulae of reference 3 give expressions for calculating the optimum absorber tuning and damping in terms of the mass ratio (equations A1.10 and A1.8 respectively) and also show that the mass ratio may be calculated from the mass distribution of the ship only when the mode shape is known (equation A1.4). This mode shape is usually determined by experiments on the ship before the absorber is even designed. The characteristics of the absorber are then based on this mode shape but as shown in point 1 above, this is not strictly correct, and so although the formulae of reference 3 may be used for the design and preliminary adjustment of the absorber the fine tuning should be based on the results of sea trials.

If the mode shape of the ship is constant throughout the frequency range of interest then the response at any point on the ship is always a constant proportion of the response at any other point. A comparison of figures 1(b) and 9 shows that at 3.16 Hz the acceleration of the bridge is 1.07 times the acceleration of the bow (frame 70) whereas at 3.68 Hz the ratio is 0.59. A direct consequence of this is that the tuning of the absorber which makes the twin peaks equal at one point of the ship will not necessarily make them equal elsewhere. Where, then, on the ship should the criterion of reference 3 (equal peaks for optimal tuning) be applied? The main source of vibrational energy dissipation is the absorber (some energy is dissipated in the ship's hull but the damping ratio is so small (0.5% of critical) that this is negligible). Hence the absorber is most effective when the vibrational energy of the ship is transferred to the absorber as quickly as possible. This occurs when the frequency spectrum of the displacement response measured on the ship at the absorber centre of percussion (near frame 68) has twin peaks of equal height. This absorber tuning will also give twin peaks of equal height in the frequency spectrum of the relative displacement response measured anywhere on the absorber. Figure 2 shows that this condition was not met by any of the absorber tunings used in the sea trial.

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Craft-Heavy.  
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SEA STATE	SHIPS LOADING (TONNES)	TOP SPRING PRESSURE (PSIG)	ABSORBER NAT. FREQ. (HZ.)	ABSORBER DAMPING (% CRITICAL)
3	522	78	3.55	5.5
4	522	68	3.40	4.6
1	512	65	-	-
2	518	64	3.40	7.1
2	518	61	3.30	6.3

TABLE 1

CONDITIONS DURING SEA TRIAL

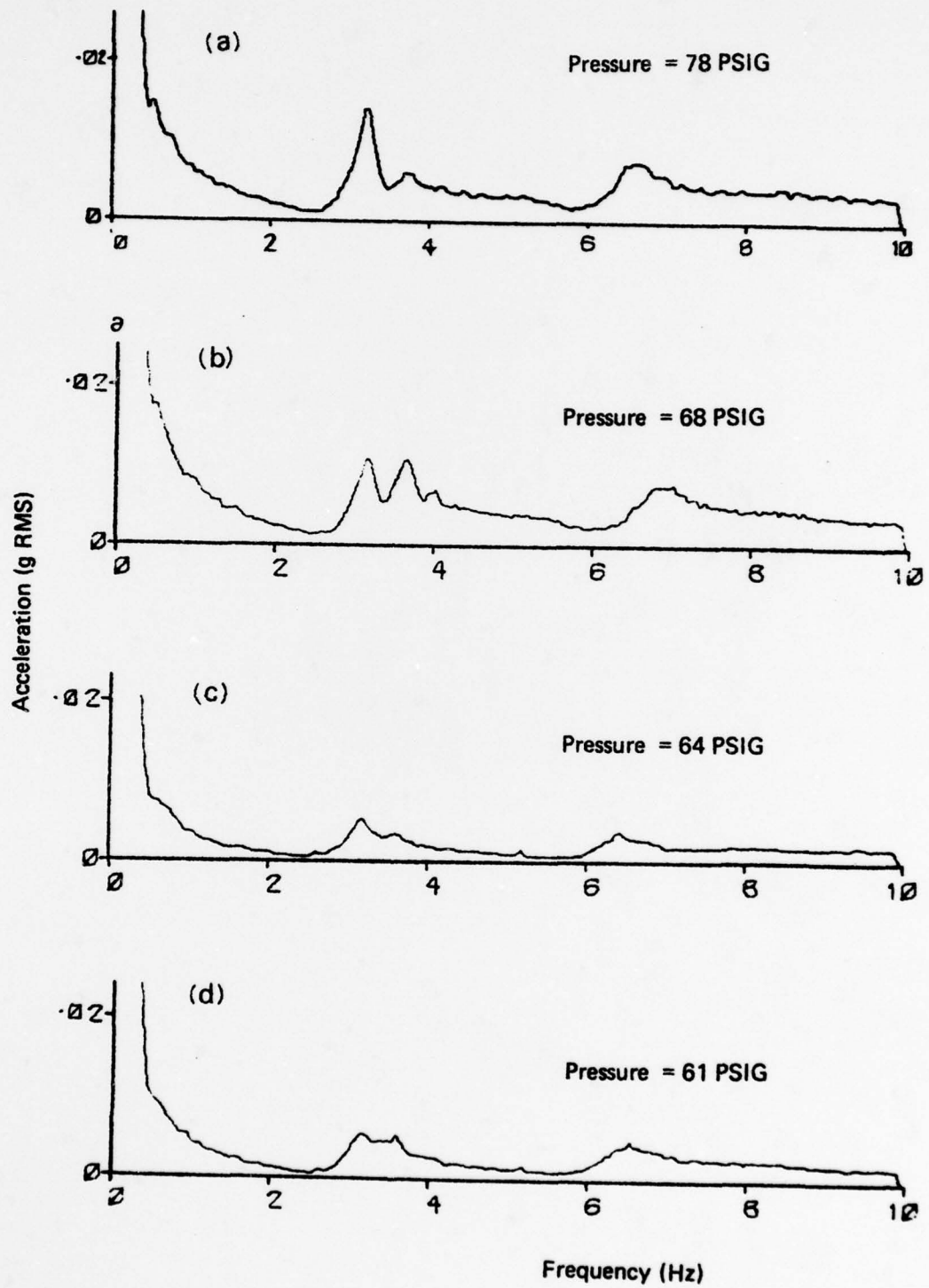


FIG. 1 RESPONSE OF SHIP AT FRAME 70

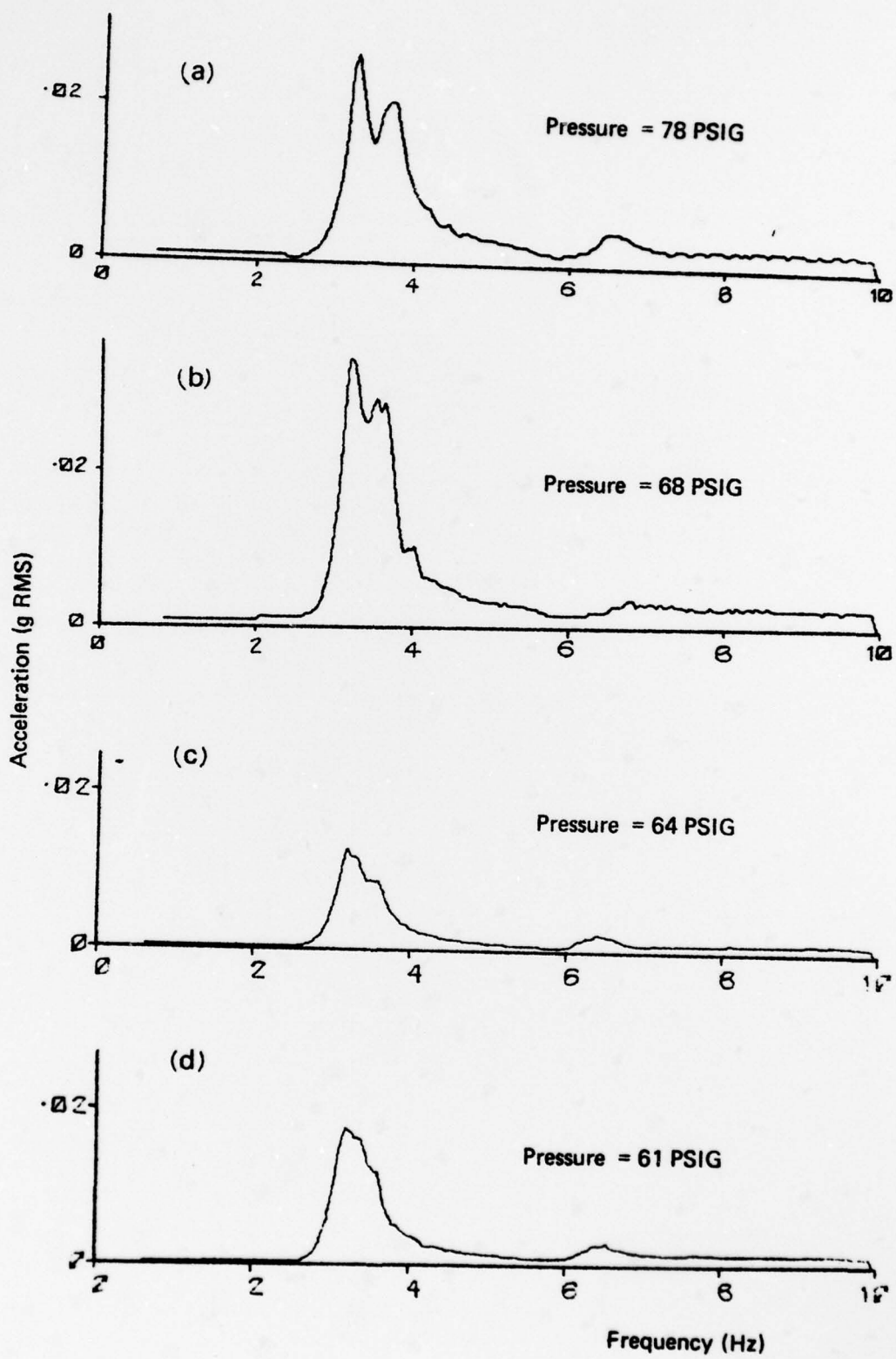


FIG. 2 RESPONSE OF ABSORBER AT FRAME 70



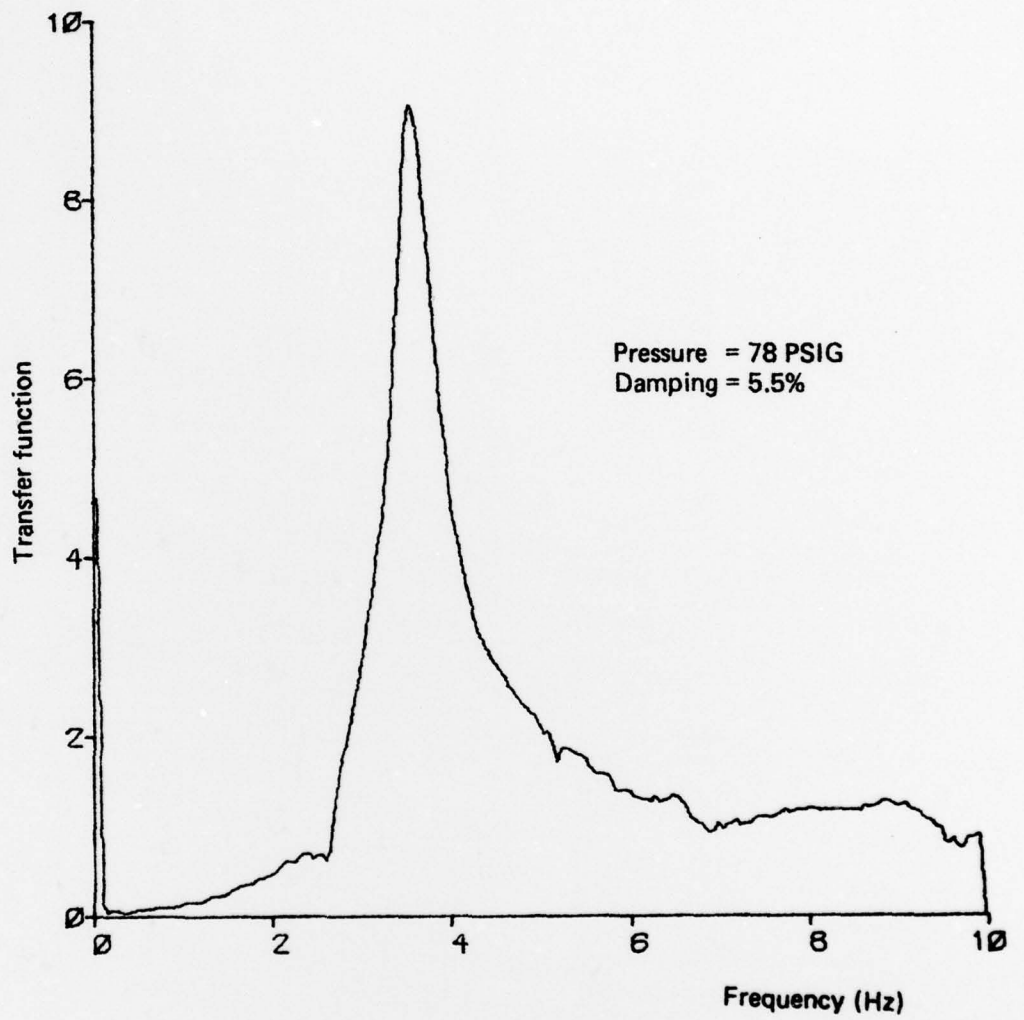


FIG. 3 TRANSFER FUNCTION OF ABSORBER

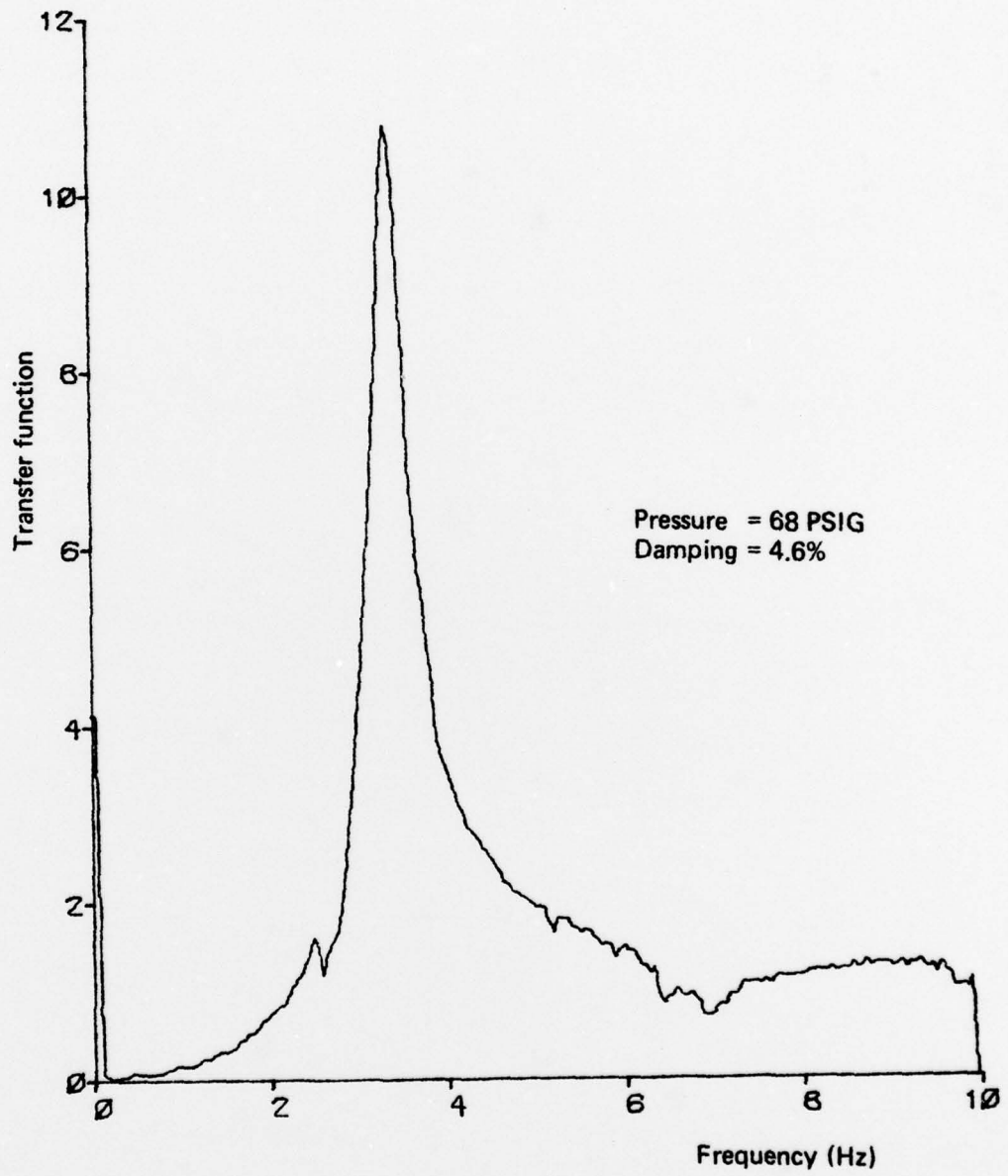


FIG. 4 TRANSFER FUNCTION OF ABSORBER

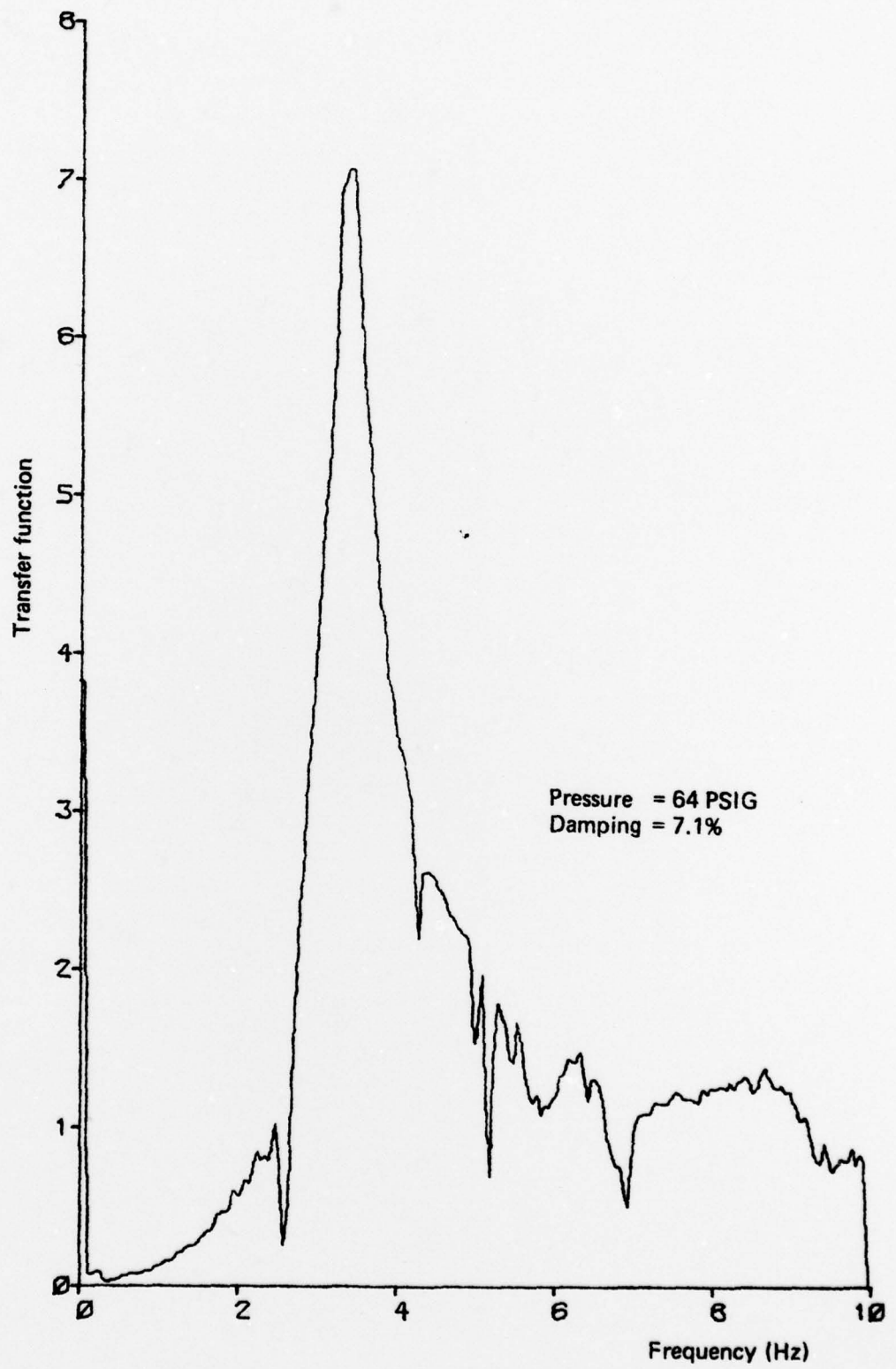


FIG. 5 TRANSFER FUNCTION OF ABSORBER

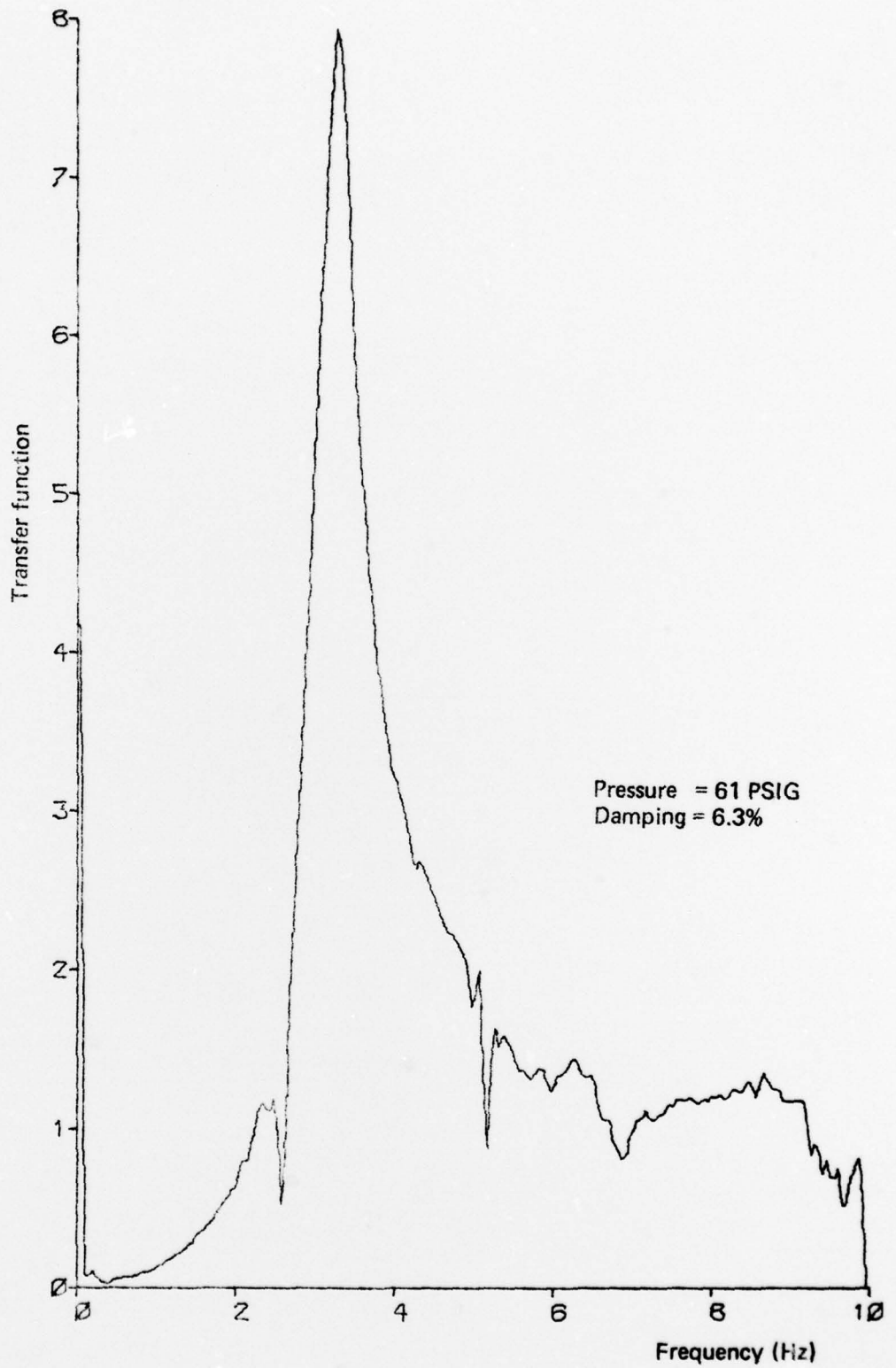
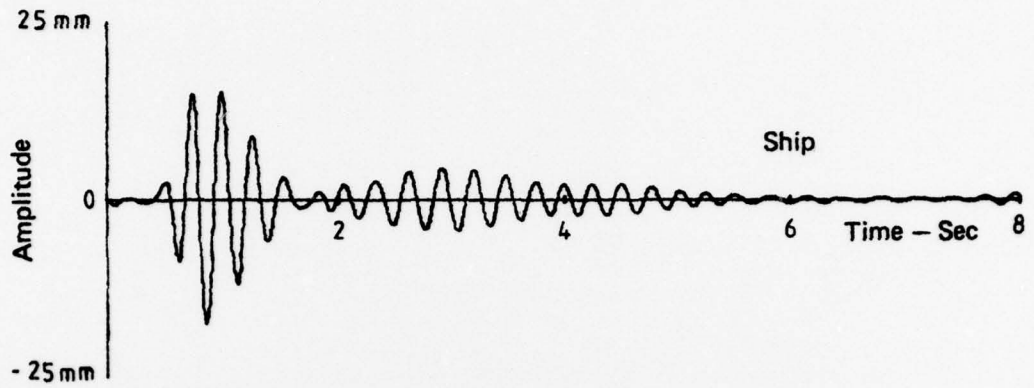


FIG. 6 TRANSFER FUNCTION OF ABSORBER



Top spring pressure 68 PSIG  
amplitudes at frame 68

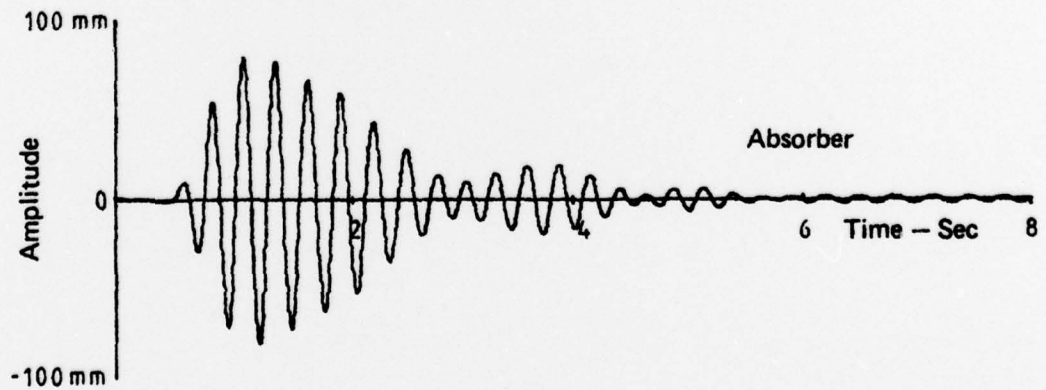


FIG. 7 DISPLACEMENT RESPONSE OF SHIP AND ABSORBER TO LARGE TRANSIENT  
(RESPONSE ONLY IN BAND 1.5Hz to 5.5Hz)

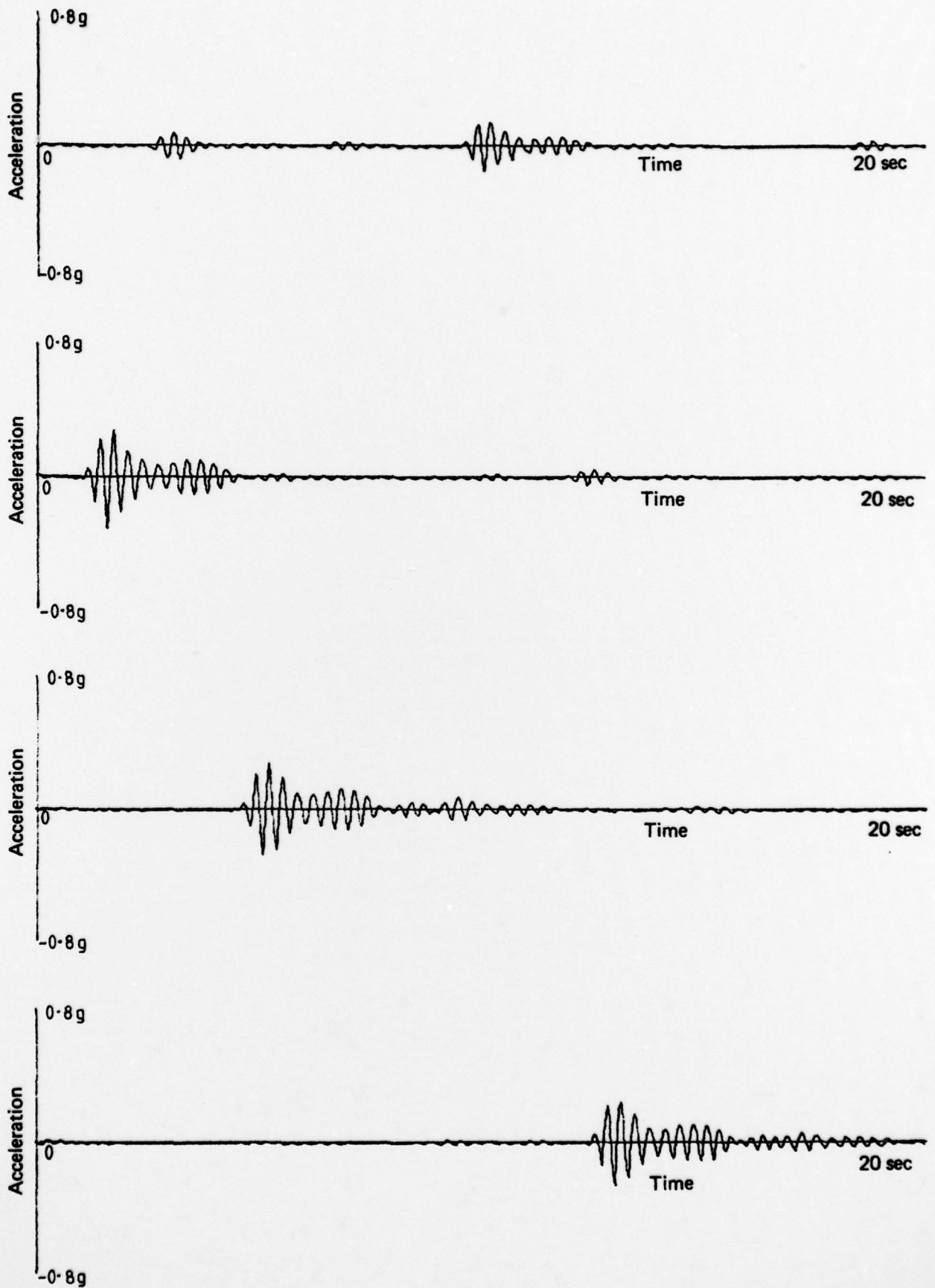


FIG. 8 FUNDAMENTAL HULL VIBRATION MEASURED ON BRIDGE OF HMAS BALIKPAPAN (PRESSURE = 68 PSIG)

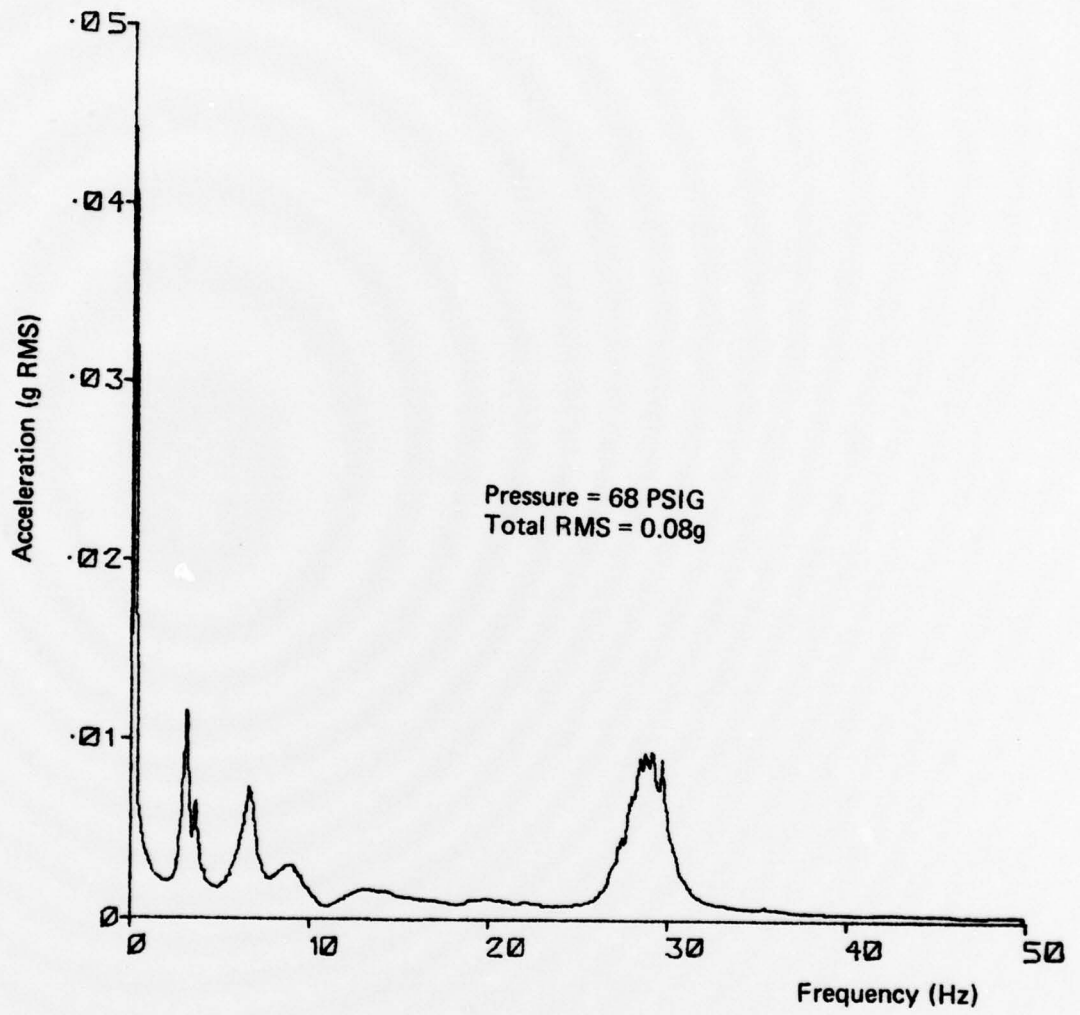


FIG. 9 FORE-AND-AFT RESPONSE MEASURED ON BRIDGE OF HMAS BALIKPAPAN

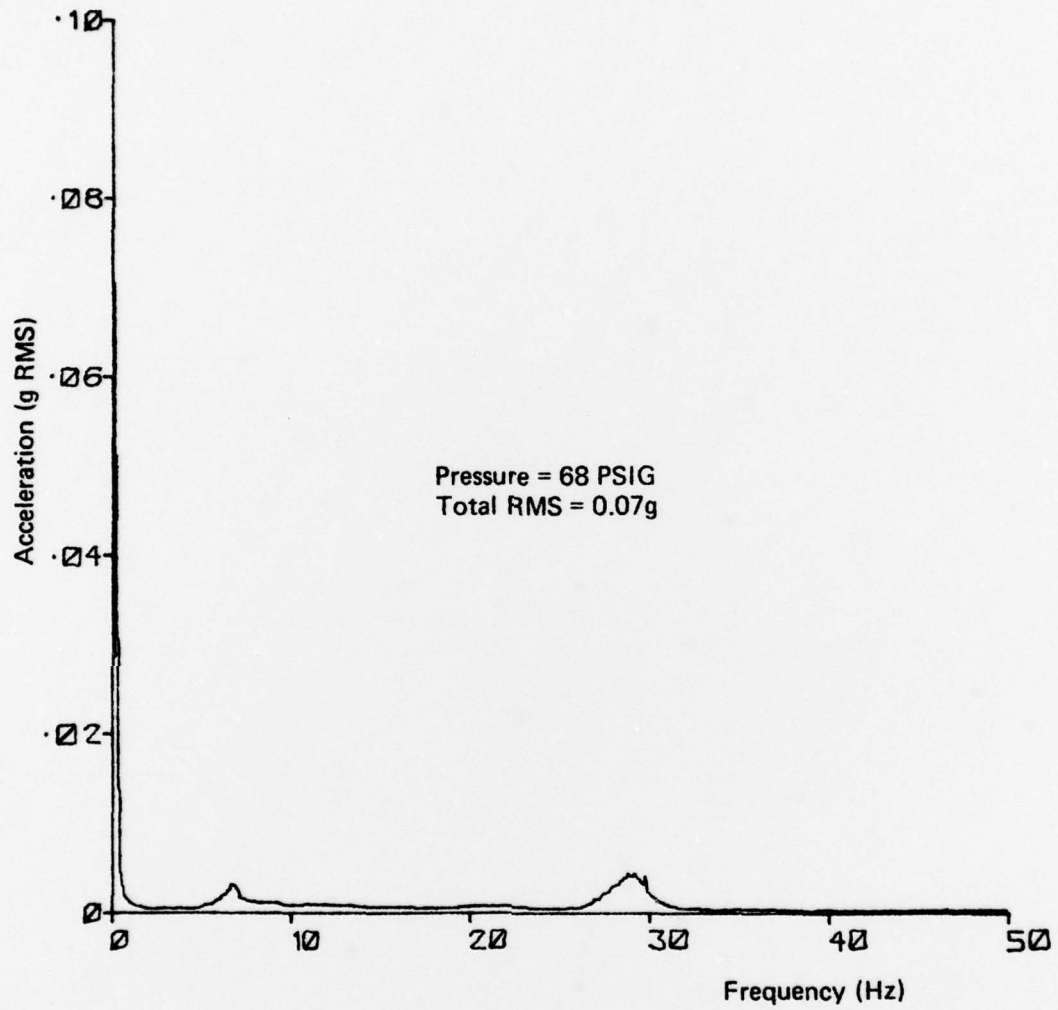


FIG. 10 VERTICAL RESPONSE MEASURED ON BRIDGE OF HMAS BALIKPAPAN



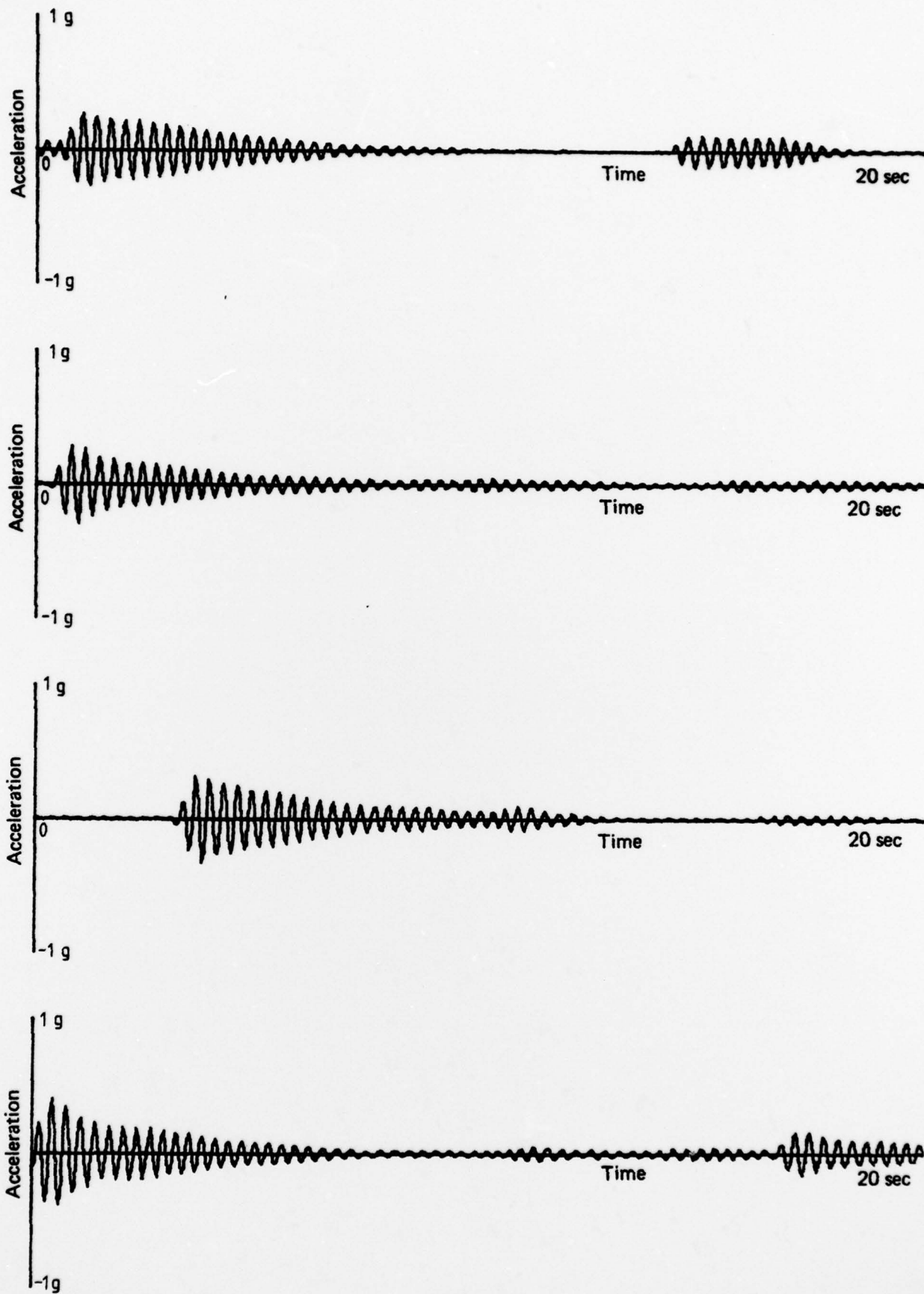


FIG. 11 FUNDAMENTAL HULL VIBRATION MEASURED ON BRIDGE OF HMAS LABUAN

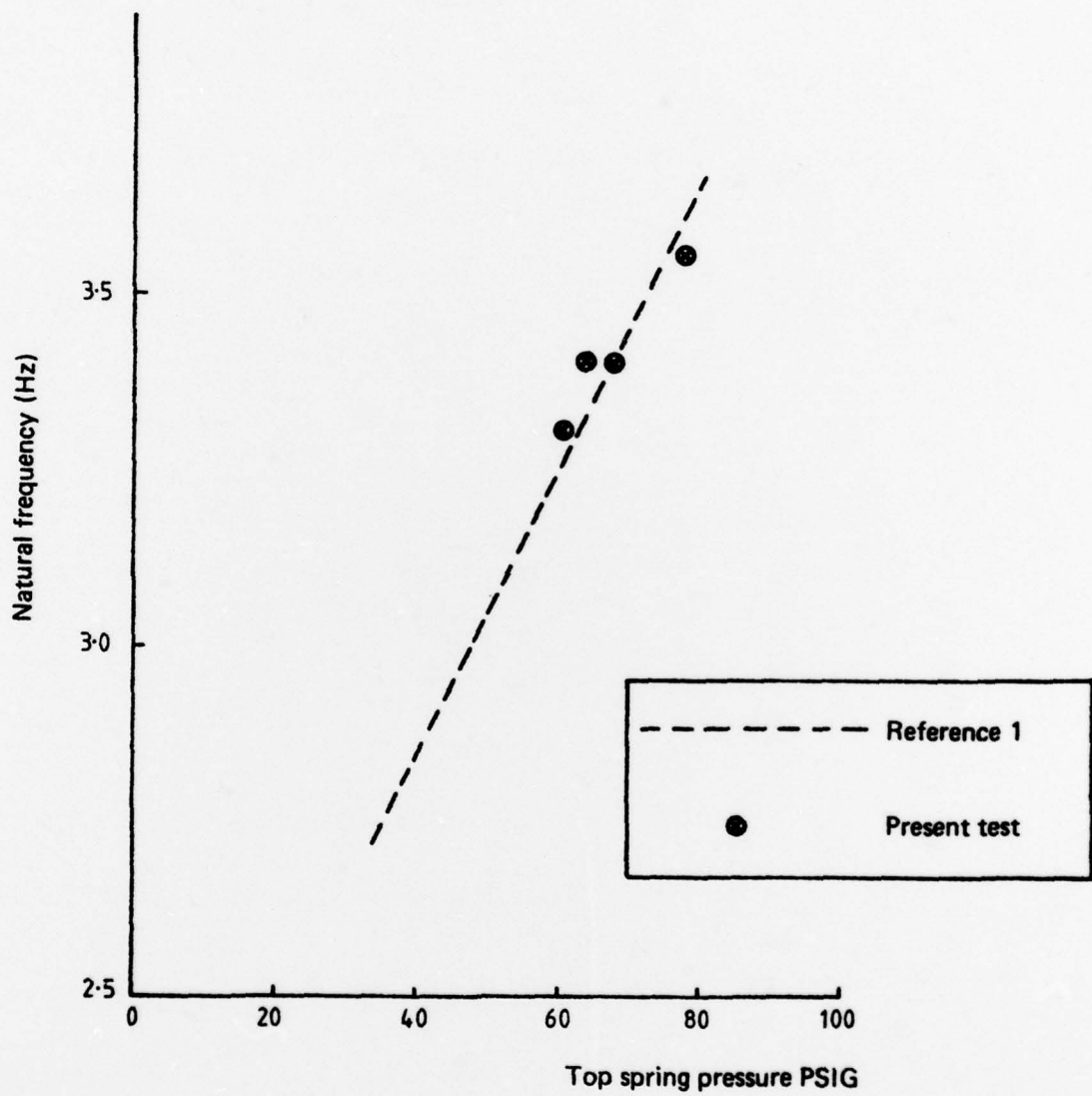


FIG. 12 VARIATION OF ABSORBER NATURAL FREQUENCY WITH PRESSURE

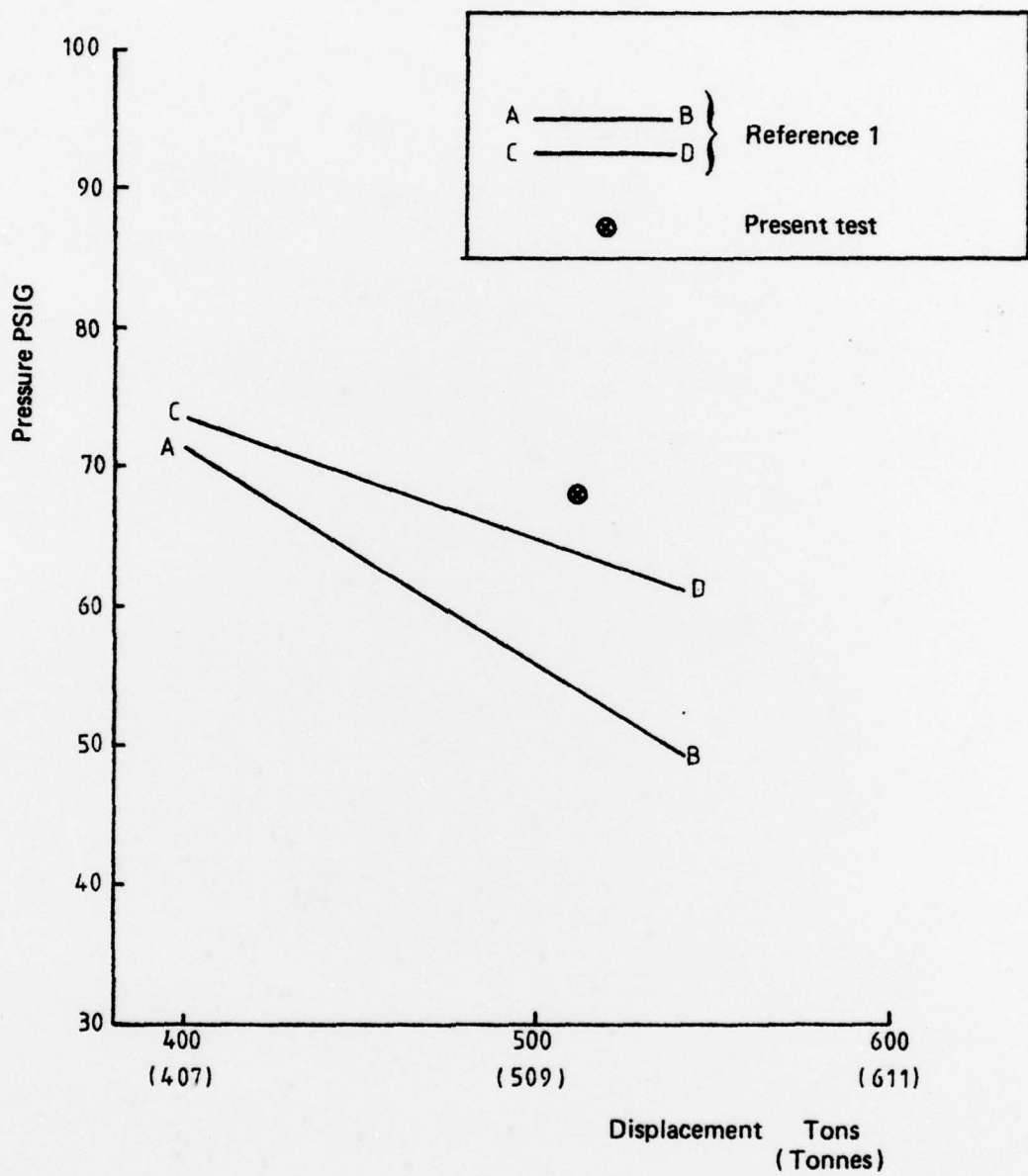


FIG. 13 RECOMMENDED AIR SPRING PRESSURE VERSUS SHIP DISPLACEMENT

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b. Document Series and Number: Structures Technical Memorandum 280	b. Title in isolation: UNCLASSIFIED							
c. Report Number: ARL-Struc-Tech-Memo-280	c. Summary in isolation: UNCLASSIFIED							
3. TITLE: AN EVALUATION OF A DAMPED VIBRATION ABSORBER IN ROUGH SEAS.								
4. PERSONAL AUTHORS: P.A. Farrell and G. Long	5. DOCUMENT DATE: June 1978							
7. CORPORATE AUTHOR(S): Aeronautical Research Laboratories	6. TYPE OF REPORT AND PERIOD COVERED:							
9. COST CODE: 23 2610	8. REFERENCE NUMBERS a. Task: NAV 76/18 b. Sponsoring Agency:							
10. IMPRINT: Aeronautical Research Laboratories, Melbourne	11. COMPUTER PROGRAM(S) (Title(s) and Language(s):							
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13. ANNOUNCEMENT LIMITATIONS (of the information on this page): No Limitation.								
14. DESCRIPTORS: Vibration. Vibration Damping. Vibration Meters.	15. COSATI CODES: Ship Structural. 2011 Components. 1310 1402							
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