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SEA TRIALS OF A DAMPED VIBRATION ABSORBER ON H.M.A.S. BALIKPAPA--ETC(U)

JAN 78 G LONG, P A FARRELL

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SEA TRIALS OF A DAMPED VIBRATION ABSORBER ON H.M.A.S. BALIKPAPAN

G. Long and P. A. Farrell

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(6) SEA TRIALS OF A DAMPED VIBRATION ABSORBER ON H.M.A.S. BALIKPAPAN,

(10) G. Long and P. A. Farrell

(9) Technical memo,

(11) Jan 78

Summary

Sea trials of a damped vibration absorber installed on a 550 tonne ship are described. The absorber was designed to increase the damping in the fundamental vertical bending mode of the ship. Vibration levels, as a result of wave action, were reduced by a factor of approximately six.

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1. INTRODUCTION

The L.C.H. is a landing craft approximately 43 metres long of maximum displacement 550 tonnes, and is designed to carry up to three 50 tonne vehicles. In service, as a result of its blunt bow design (see Fig. 1), severe impact loads are experienced in heavy seas. These loads excite the lower frequency hull vertical bending modes and cause not only severe discomfort to the crew but also damage to the ship's structure and equipment.

This problem has been investigated by staff from the Aeronautical Research Laboratories and in reference 1 it was proposed to reduce the vibration by installing a damped vibration absorber. This absorber is essentially a 10 tonne steel mass supported on air springs and fitted in the bow of the ship (see Fig. 2). The design of this absorber is described in more detail in reference 2.

The absorber was fitted to H.M.A.S. Balikpapan and sea trials conducted to check its effectiveness. These trials consisted of vibration tests in calm water to confirm the design characteristics of the absorber, followed by sea trials.

The variation of ship natural frequency with displacement, and the variation of absorber natural frequency with air spring pressure were measured by exciting the ship with a mechanical vibrator. The effect of absorber tuning and damping on the ship's response to forced vibration were also measured. From these data a graph of recommended air spring pressure versus ship's displacement was derived. This graph enables the operator to obtain the optimum absorber effectiveness for any specified ship's loading.

Only one value of ship's loading was tested in open sea conditions. For this loading, the optimum absorber tuning and several variations from the optimum were assessed. The data obtained were analysed to provide power spectral density estimates of the vibration environment in the frequency range zero to ten hertz.

2. VIBRATION TESTS IN CALM WATER

The vibration absorber was designed to be fitted as far forward in the ship as possible in order to be most effective. Absorber tuning is carried out by varying the air spring pressure, and absorber damping is adjusted by varying the number of damper units. In order to assess the absorber design, to determine if its natural frequency range and damping adjustments are adequate, steady state vibration tests were carried out in calm water. Steady state tests were also conducted to determine the ship's vibration characteristics. These tests are described in the following sections.

2.1 Variation of ship natural frequency with loading

These tests were conducted in calm water (depth approximately 15 m. (50 ft.)) and the various ship loadings used are given in Table 1. The ship was excited by the A.R.L. low frequency vibrator and the ship amplitude measured over a range of excitation frequencies. A typical result of such a test is given in Fig. 3 where the ship amplitude (measured at frame 72) per unit applied force is plotted as a function of frequency. The natural frequency is indicated by the large peak in the ship's response.

In Fig. 4 the variation of ship natural frequency with loading is presented for the range of loadings given in Table 1. Over this range the variation is close to linear.

2.2 Variation of absorber natural frequency with air pressure

To provide data for absorber tuning it is necessary to measure the variation of absorber natural frequency with air spring pressure. This was done by exciting the ship with the mechanical vibrator and measuring the response of the absorber relative to the ship as described in reference 2. Since the mechanical vibrator was limited in power capacity all the damper units were removed from the absorber to ensure that the absorber amplitudes were adequate.

In Fig. 5 the absorber natural frequency is plotted as a function of air spring pressure. In the range of pressures investigated the variation is essentially linear. The effect on natural frequency of adding dampers to the absorber should be negligible if the dampers behave in an ideal way. In practice the dampers increase the stiffness of the system and hence increase the natural frequency. This is only a small effect of the order of 2% and the corrected curve is shown in Fig. 5. This corrected curve is used to determine the air spring pressures to obtain a selected absorber frequency.

The absorber damping is varied by adding or removing dampers. Since there are 12 damper units good control of damping is achieved in this way. The variation of absorber damping with number of dampers is shown in Fig. 6. For these tests the dampers were added 4 at a time starting at the position closest to the pivot and working outwards.

The effectiveness of the dampers varies with the square of the radius from the pivot.

2.3 Effect of absorber on ship response to forced vibration

Having established the ship's natural frequency and the characteristics of the absorber, it is possible to tune the absorber to have the optimum effect on the ship's response. For the various values of ship's loading used the absorber frequency was tuned to be approximately 2% lower than the ship's frequency.

The ship was again vibrated in calm water using the A.R.L. mechanical vibrator and the response measured, both with the absorber in operation and locked. In Fig. 7 the results of such a test are presented. Two curves are drawn, one for the absorber locked and one for the absorber correctly tuned. The former curve peaks sharply at the natural frequency of the ship which indicates that the ship is very lightly damped. The latter curve is relatively flat over the frequency range of interest. The difference in peak values of the two curves is an indication of the reduction in vibration amplitude under continuous excitation, as a result of fitting the absorber. In this figure the reduction is of the order of seven to one. Similar sets of curves were obtained for each of the ship's loadings investigated.

In Fig. 8 the effect of absorber tuning on its effectiveness is indicated. Three curves are presented, each for a different value of air spring pressure. Increasing pressure, which improved the tuning, significantly reduced the ship's response to forced vibration. When the absorber was close to the optimum tuning the mechanical vibrator was not powerful enough to obtain an adequate response from the ship-absorber system. As a result the absorber amplitude was lower than would be desired and its natural frequency was affected slightly by non-linearity in the shock absorbers. The effect of such non-linearities is to increase the absorber frequency at low amplitudes of vibration.

This effect will not reduce the absorber effectiveness but creates difficulties in determining the air spring pressures to obtain the optimum tuning. In Fig. 8, for example, it is considered that for larger absorber amplitudes the optimum top spring pressure should be approximately 448 kPa (65 Psig) and not 400 kPa (58 Psig) as indicated. This is discussed in more detail later.

In Fig. 9 the effect on ship's response of varying the absorber damping from approximately 4 per cent to 7 per cent of critical damping is indicated. It is clear that the latter value is closer to the ideal. (Ideal damping and tuning will result in both peaks in Fig. 9 having the same height).

2.4 Selection of absorber tuning for each value of ship's displacement

From the data presented in Figs. 4 and 5 it is possible to obtain a graph of optimum absorber tuning versus ship's displacement. This is presented in Fig. 10. Also shown in this figure are the actual pressures used for tests on 5 loading conditions. At the time of testing it was considered that these values were close to the optimum; however it is possible that these too were affected by non-linearities in the damper units and that higher spring pressures may be needed for larger absorber amplitudes. This problem can only be resolved by exciting the ship with a larger exciter or by more detailed analysis of rough-sea trials. Since further sea trials are proposed, the latter alternative will be used.

3. SEA TRIALS OF ABSORBER

The vibration tests conducted in calm water were adequate to check the absorber design and provide a basis for optimum absorber tuning. The next series of tests was designed to assess the absorber's performance under transient conditions at sea.

During the test period very rough sea conditions were not experienced. Hence, in order to obtain some operational data, the ship was ballasted to provide the conditions which were known to cause the worst hull vibration. The bow of the ship was high in the water and strong impacts between waves and the flat bottom of the ship occurred frequently. These impacts caused maximum absorber amplitudes of the order of 40 mm (1.57 ins) peak-to-peak, which is approximately 16% of the maximum allowable stroke. Throughout the series of tests sea and wind conditions did not vary greatly. Prior to the sea trial the ship natural frequency was measured and the absorber spring pressures adjusted to 496 kPa (72 Psig) and 593 kPa (86 Psig) in the top and bottom springs respectively. The response of the combined ship-absorber system was also checked.

At sea, the ship and absorber response were measured by accelerometers at frame 68 and recorded on a magnetic tape recorder for subsequent analysis. The ship was steered directly into the waves in order to maximise the vibration input. Four tests were carried out. These were:-

1. Measurement of response with absorber locked.

2. Measurement of response with absorber correctly tuned, pressures 496 kPa (72 Psig) and 593 kPa (86 Psig).

3. Measurement of response with absorber detuned, pressures 427 kPa (62 Psig) and 524 kPa (76 Psig).

4. Measurement of response with absorber detuned, pressures 358 kPa (52 Psig) and 455 kPa (66 Psig).

The vibration records obtained were analysed in two ways to determine the absorber effectiveness. The first method is based on the Random Decrement approach, reference 3, and produces from a random signal an approximation to the impulse response of the system. In this instance this is equivalent to the response of the ship to an impulse from a single wave. Since the method is based on a large number of averages a good estimate of the impulse response is obtained. This impulse can be analysed by standard methods to obtain the frequencies and damping of the ship-absorber system.

The second method of analysis is based on a Fourier analysis of the acceleration response, from which the frequency response of the system can be obtained. This analysis again involves an averaging process and provides data from which valid comparisons of the absorber effectiveness can be made.

3.1 Random Decrement Analysis

Since the frequency range of interest is in the region of the two-node mode at approximately 3.6 Hz, the recorded data were filtered by a band-pass filter, so that only data in the frequency range 1.5 Hz to 5.5 Hz were used. The Random Decrement signatures of the data for the four sets of test conditions are reproduced in Figs. 11 - 14. In these figures the top diagram is the Random Decrement signature, which is equivalent to a time history of the displacement response of the ship to a single impulse. The centre diagram is a plot of the same response on a logarithmic scale and the lower diagram is the power spectrum of the signature. A more detailed description of how these figures are derived is given in Appendix A2.

In Fig. 11 the Random Decrement signature for the ship with the absorber locked is shown. After 5 seconds the amplitude of vibration is still large indicating that there is little damping in the ship. The magnitude of the damping can be estimated directly from the Random Decrement signature or from the graph of log-amplitude. If the damping were constant it would be possible to draw a straight line through every peak of the log-amplitude plot. In this instance the damping varies with amplitude but a mean estimate gives the damping as 1.3% approximately. This is of the order of twice the damping measured in calm water vibration tests. It is thought that this increase in damping is a non-linear characteristic of the ship, i.e. the ship damping increases as amplitude increases. In Fig. 11 the power spectrum indicates that only one frequency is present at 3.5 Hz.

In Fig. 12 similar graphs are presented for the ship and the correctly tuned absorber. The vibration amplitude rapidly decays to a low value and beats are apparent at this lower vibration level. The log-amplitude plot shows that the damping is non-linear and appears to increase as amplitude decreases. This is a consequence of beating between the two frequencies occurring in the response. These two frequencies are evident in the power spectrum graph. It is not practicable to estimate a damping factor from such a response, but a direct comparison with Fig. 11 shows that the absorber reduces the vibration response significantly.

Figures 13 and 14 present the results for the ship-absorber system for two other tuning values. Neither response is as good as that in Fig. 12 for the correctly tuned system, but is better than that shown in Fig. 11. Both show evidence of beats between two close frequencies.

In Fig. 15 a direct comparison of two time histories of vibration is made. One time history is for the ship alone and the other for the ship and correctly tuned absorber. Clearly the absorber significantly reduces the vibration level of the ship and between wave impulses the ship vibration decays rapidly to negligible values. With the absorber inoperative the ship vibration rarely stops.

3.2 Fourier Analysis

As described above the acceleration of the ship at frame 68 was recorded on an analogue magnetic tape recorder for each of the four absorber conditions listed. These records were then analysed on a digital Fourier analyser to give the frequency response plots shown in Figures 16 - 19. (More details of this analysis are given in Appendix A1).

Fig. 16 shows the averaged frequency spectrum (as a proportion of the acceleration due to gravity) of the ship with the absorber locked.

There are three distinct peaks in the response at 0.25, 3.5 and 7 Hz. The first gives the average R.M.S. acceleration at frame 68 of the ship pitching as a rigid body. The peak at 3.5 Hz gives the averaged R.M.S. acceleration of the ship vibrating in its two-node vertical mode (see ref. 1) whilst the third peak gives the averaged response in the three-node vertical mode. The height of each of these peaks is a measure of the ship response at that frequency to the sea state experienced. The absorber is designed to reduce the response of the two-node mode of the ship only. (N.B. the frequency of this mode varies with ship's loading and water depth and so the frequency of 3.5 Hz is relevant only to this set of conditions).

Fig. 17 shows the averaged frequency spectrum of the ship when the absorber is correctly tuned. The two-node mode response peak at 3.5 Hz is no longer apparent and is replaced by two much smaller peaks at 3.35 and 3.8 Hz approximately. The peaks at 0.25 and 7 Hz are still present and although they are slightly greater in magnitude than before, this is because the sea state has increased slightly. The absorber has effectively reduced the ship response in the two-node mode by a factor of approximately six.

Figs. 18 and 19 show the absorber effectiveness when its natural frequency is altered from the optimum by decreasing the air spring pressure by 69 kPa (10 Psi) and 138 kPa (20 Psi) respectively. Comparing these figures with Figs 16 and 17 shows that although the detuned absorber is less effective it still significantly reduces the ship's vibration in the two-node mode.

4. DISCUSSION OF RESULTS

The aim of the tests was to check the absorber design and to assess its effectiveness in reducing wave induced vibration. The tests reported in section 2.1 have confirmed that the absorber has a sufficiently variable natural frequency to cater for all the allowable ship loadings. The air spring pressures required are not excessive, even for the highest natural frequency required, and the number of dampers used (12 units) should provide a good safety factor in the event of one or two units failing in service. It is considered that the absorber design is adequate, based on these data.

A more difficult problem is to define a criterion with which to assess the absorber effectiveness in rough seas. Since the ship and absorber behave as a two-degree-of-freedom system in the neighbourhood of the two-node vertical mode, beats occur in the impulse response (see Fig. 12).

These make it difficult to apportion a value of damping to the combined ship-absorber system. As shown in Fig. 12, the log amplitude curve indicates that the damping appears to increase over the first 1 second of response and then decrease again after approximately two seconds. This is as a direct result of beating and indicates an energy transfer from the ship to the absorber and then back again. In Fig. 20 this effect is shown more clearly. Fig. 20 compares the theoretical impulse response of a two-degree-of-freedom system with the theoretical impulse response of a single-degree-of-freedom system with 5% damping. The upper trace in Fig. 20 is the single-degree-of-freedom response and the lower traces the theoretical responses of the ship and the absorber. The ship response drops rapidly from the initial peak while the absorber amplitude builds up over a number of cycles before decaying. This build up of response of the absorber is due to the energy transferred from the ship. The effect of this transfer is to reduce the ship amplitude more rapidly than if the ship behaved as a 5% damped single-degree-of-freedom system. It is this effect which makes it difficult to estimate an equivalent single-degree-of-freedom damping for the ship-absorber system.

In Fig. 21 another method of comparison has been tried. In this figure the maximum potential energy of the ship only in each half cycle of oscillation is plotted as a function of time. This energy is proportional to the square of the maximum amplitude in each half cycle. Three systems are compared, the two-degree-of-freedom system and a 5% and a 10% single-degree-of-freedom system. The slope of the curve is a measure of the instantaneous damping. The figure shows that the damping of the two-degree-of-freedom system starts with a low value but rapidly increases to much more than 10%. After 6.5 half cycles this system is more effective than the 5% damped system and after 12.9 half cycles it has dissipated as much energy as the 10% damped system. Thus, for a major portion of the decay time the instantaneous damping is greater than 5% and even exceeds 10%. Since the basic damping in the ship, measured in calm water is of the order of 0.6% it is clear that the absorber increases the damping significantly. The improvement of approximately 6 times indicated in section 3.2 is considered to be conservative in some respects. It is felt that this method of estimating the energy dissipated is a reliable method of assessing the absorber effectiveness.

These results, together with the data in sections 3.1 and 3.2 indicate that the absorber is effective in rapidly dissipating the vibration energy in the ship caused by wave motions.

5. CONCLUSIONS

The vibration absorber effectively reduces the vibration levels on the L.C.H. by a factor of approximately 6. The rate of decay of vibration is significantly increased and the ship is free of vibration for long intervals between wave impulses.

Further tests of the absorber operation in rough seas remain to be carried out to validate the system completely.

The tests undertaken so far confirm that the absorber used has largely met the design requirements.

Appendix A

Analysis of sea trial data

A1. Fourier analysis

The acceleration time histories, recorded as the ship was excited by wave action, were analysed using a special purpose digital Fourier analyser. The analogue signals were digitised at 25.6 samples per second into 32 blocks of 512 points each, with 50% redundancy. The frequency spectra of these blocks were calculated using the analyser and averaged to give the mean spectrum. (N.B. The data was first multiplied by a Hanning weighting function to reduce the effects of side lobes).

The analyser determines these spectra with a bandwidth of 0.05 Hz but the use of the Hanning weighting function increases the effective bandwidth to 0.075 Hz. Thus the RMS value of response at any frequency on these curves defines the RMS value of acceleration (scaled by the acceleration due to gravity) at that frequency, measured with a bandwidth of $\pm$ .0375 Hz about that frequency.

A2. Random Decrement analysis

In this approach the analogue data were digitised at a rate of 100 samples per second, since the Random Decrement procedure requires a sampling rate of approximately 30 times the frequency of interest. Each data record was adjusted to have zero mean value, scaled to have a maximum amplitude of unity and then double-integrated to provide a displacement-time history. A Fourier analysis of these data would reveal large responses at 0.25, 3.5 and 7 Hz. Since an analysis of the 3.5 Hz data only is required the data were filtered by a band pass digital filter which had a centre frequency of 3.5 Hz and upper and lower cut off frequencies of 5.5 Hz and 1.5 Hz respectively. The filter shape was of the "1-cosine" type.

The Random Decrement signatures of these filtered displacement time histories were calculated giving 512 point (5.12 seconds) records of the impulse response. This response may be analysed in either the time domain or the frequency domain by standard methods

In order to obtain a good Fourier transform of the random decrement signature it is necessary that the signal analysed should have decayed to zero. Otherwise spurious frequencies may be indicated due to side lobes. To overcome this problem the signatures shown in Figs. 11-14 were multiplied by an exponential factor before the Fourier transform was calculated. The value of the exponential was such that it reduced the signal level to one tenth of its value in 4 seconds. The power spectrum curves displayed in Figs. 11-14 therefore have had the damping in each of the modes artificially increased. The increase in damping is easily allowed for in any analysis, and the improvement in the resolution of the power spectrum curve justifies the use of such a technique.

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1. G. Long & P. A. Farrell — Interim report on vibration on landing craft heavy
A.R.L. T.M./STRUC 233, 1975.

2. G. Long & P. A. Farrell — The design and installation of a damped vibration absorber
A.R.L. Structures Note 440, 1977.

3. H. A. Cole — On-line failure detection and damping measurement of aerospace structures by Random Decrement signatures.
NASA CR-2205, 1973.

TABLE 1

DETAIL OF SHIP LOADINGS

Lightship	341.38	341.38	341.38	341.38	341.38	341.38
Crew & Effects	2.0	2.0	2.0	2.0	2.0	2.0
Absorber	12.0	12.0	12.0	12.0	12.0	12.0
B'Head 28 Steel	2.0	2.0	2.0	2.0	2.0	2.0
No.1 WB (P&S)						
No.3 Void (P,S &C.L)						
No.2 Void (C.L)						
Daily Service	0.27	0.27	0.27	0.27	0.27	0.27
Lub. Oil	1.4	1.4	1.4	1.4	1.4	1.4
Extended Range	0.5	0.5	0.5	0.5	0.5	0.5
No.2 WB (P&S)	98.4	98.4	70.43		19.90	
No.4 WB (Fwd P&S)	64.77					
No.4 WB (Aft P&S)						
No.5 FW (C.L.)	8.0	8.0	6.62	6.7	6.54	6.7
No.5 WB (P&S)	37.1	31.8	24.58	24.9	20.52	21.2
Fuel Oil	18.76	18.34	17.15	16.38	14.86	15.4
Bilge Water		2.0	5.0	5.0	5.0	5.0
Calculated Weight	586.58	518.09	483.33	412.53	426.37	407.85
Draft Fwd	5'0½"	4'7"	4'1¼"		2'11¾"	
Draft Aft	8'2"	7'5"	7'3¾"		7'3-7/8"	
Trim between marks	3'1½"	2'10"	3'2½"		4'4⅛"	
Mean Draft	6'7¼"	6' 0"	5'8½"		5'1⅜"	
Displacement	604	532	500		438	
Natural Frequency	3.24Hz	3.34Hz	3.43Hz	3.6Hz	3.54Hz	3.61Hz

N.B. Figures are in tons.

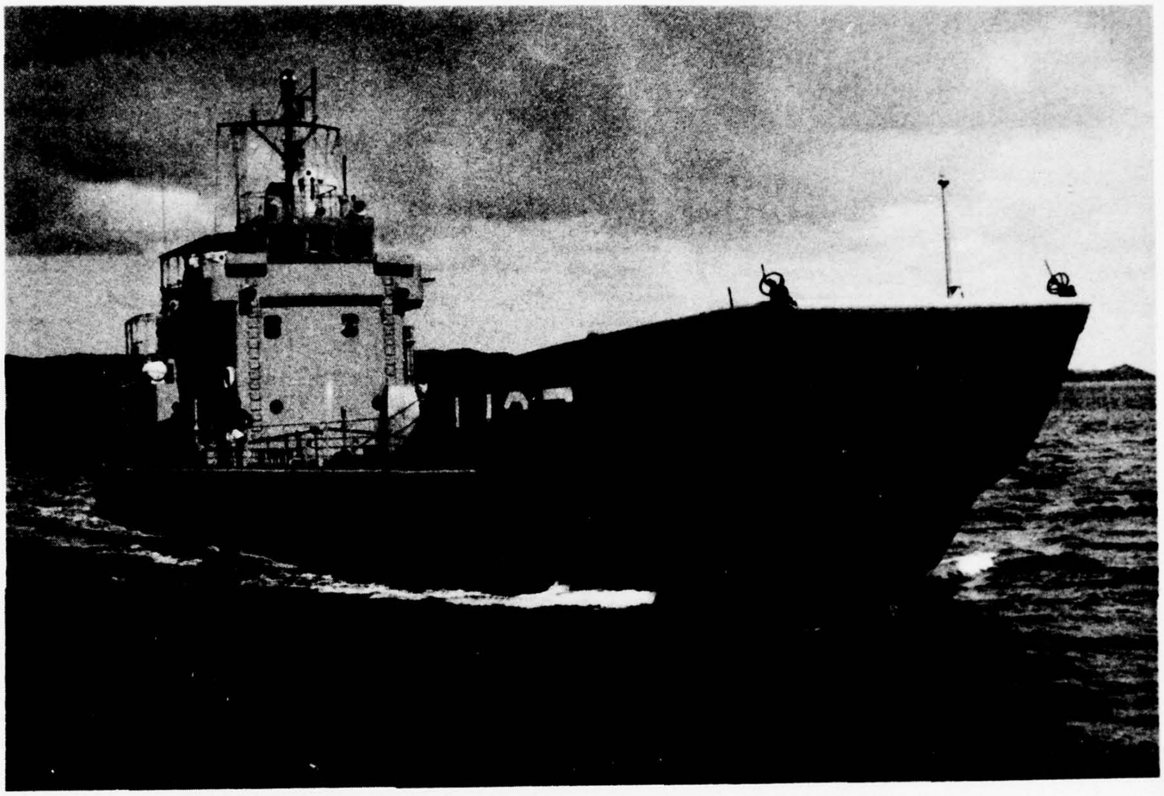
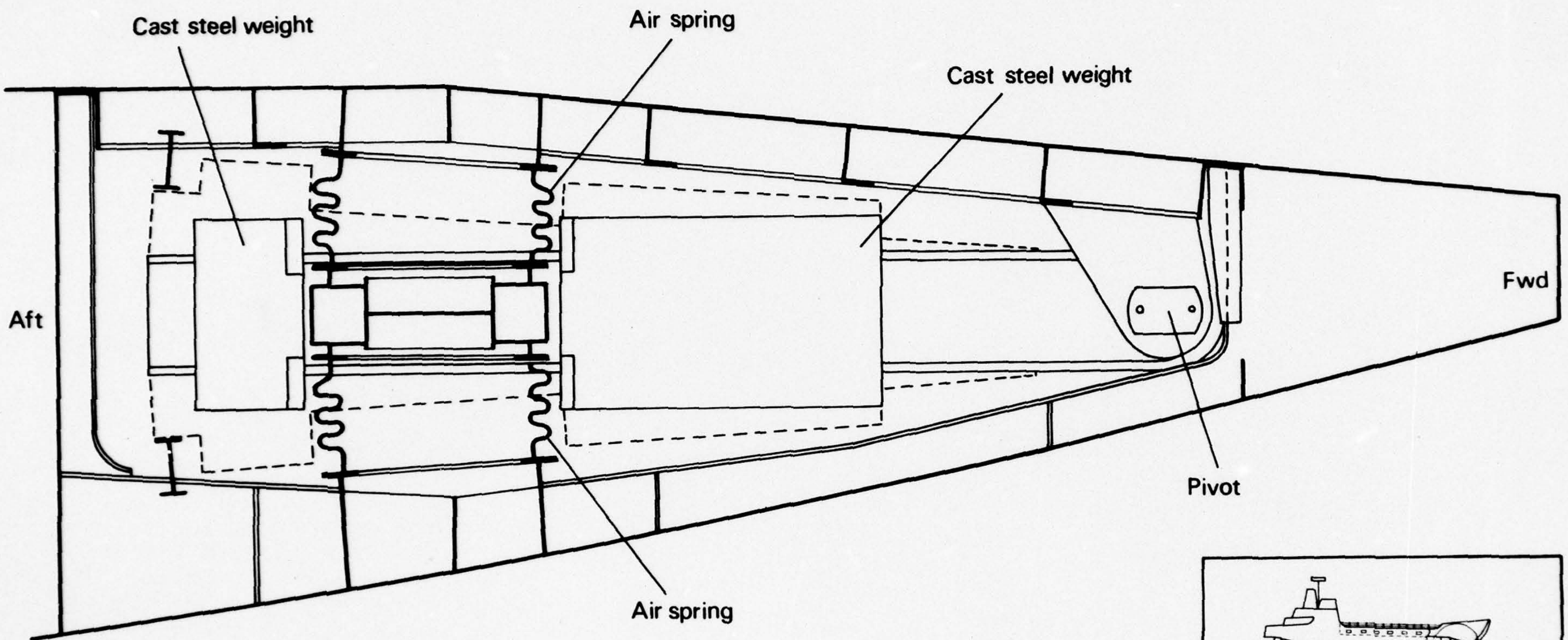
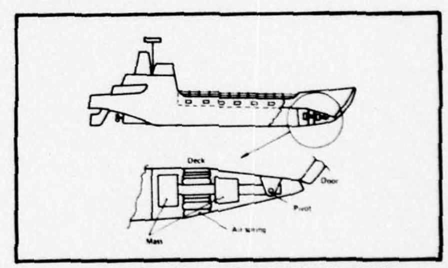


FIG. 1 LANDING CRAFT





POSITION OF ABSORBER IN SHIP

FIG. 2 ELEVATION THROUGH SHIP CENTRELINE, SHOWING ABSORBER LOCATED IN BOW

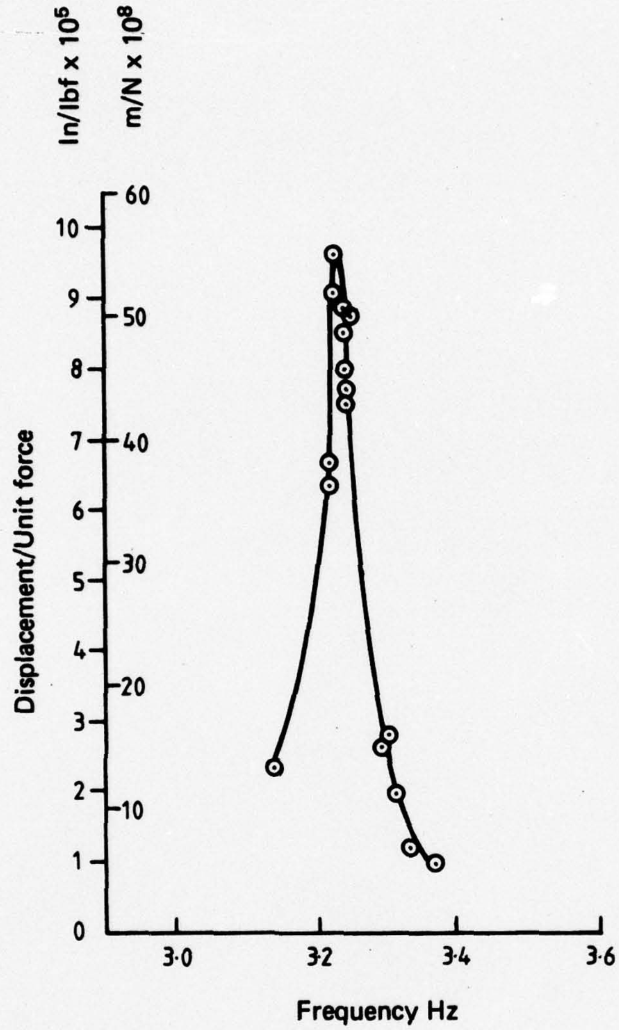


FIG. 3 VARIATION OF SHIP RESPONSE WITH FREQUENCY

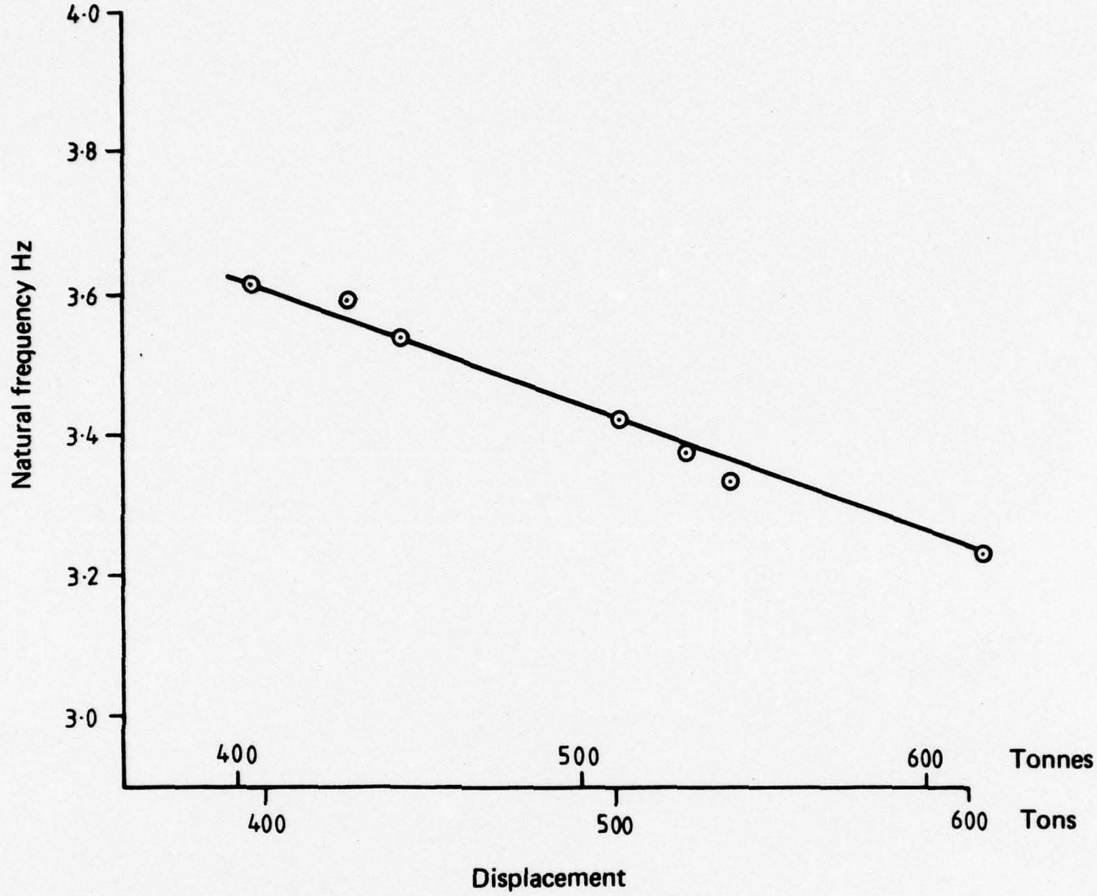


FIG. 4 VARIATION OF SHIP NATURAL FREQUENCY WITH DISPLACEMENT

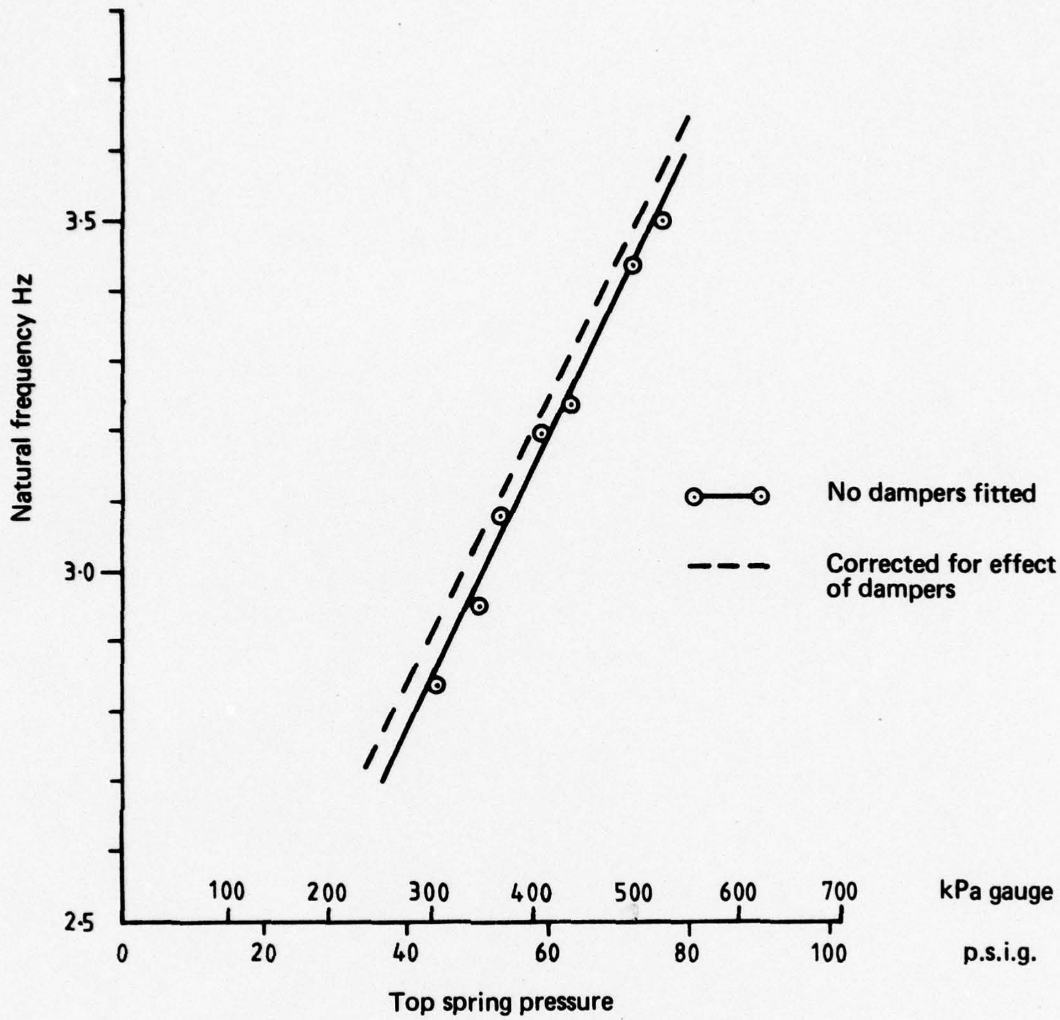


FIG. 5 VARIATION OF ABSORBER NATURAL FREQUENCY WITH PRESSURE

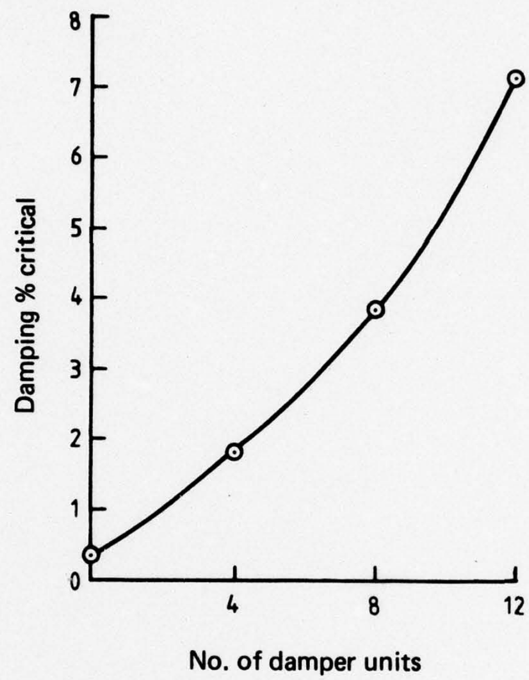


FIG. 6 VARIATION OF ABSORBER DAMPING WITH NUMBER OF DAMPERS

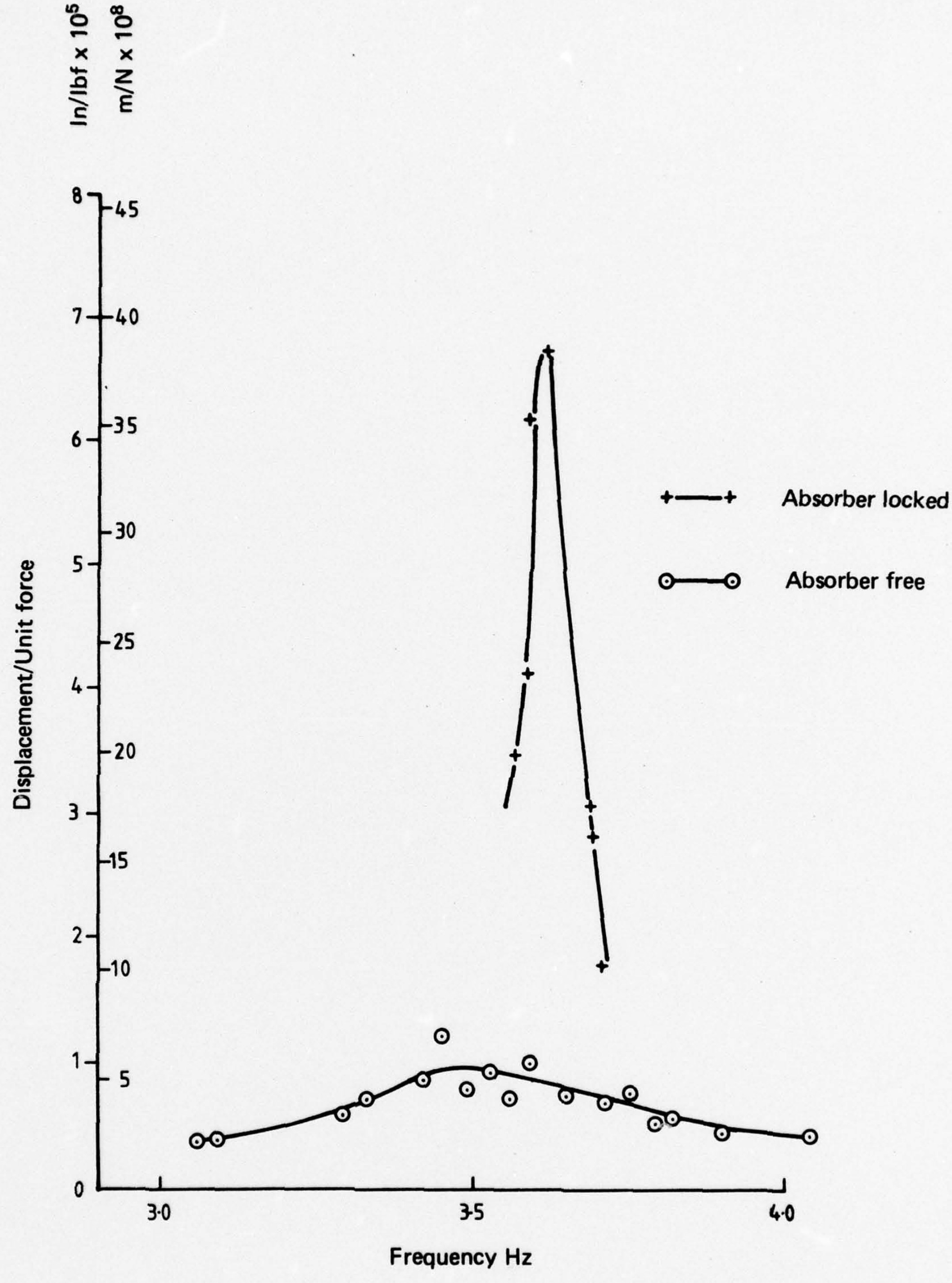


FIG. 7 EFFECT OF ABSORBER ON SHIP RESPONSE

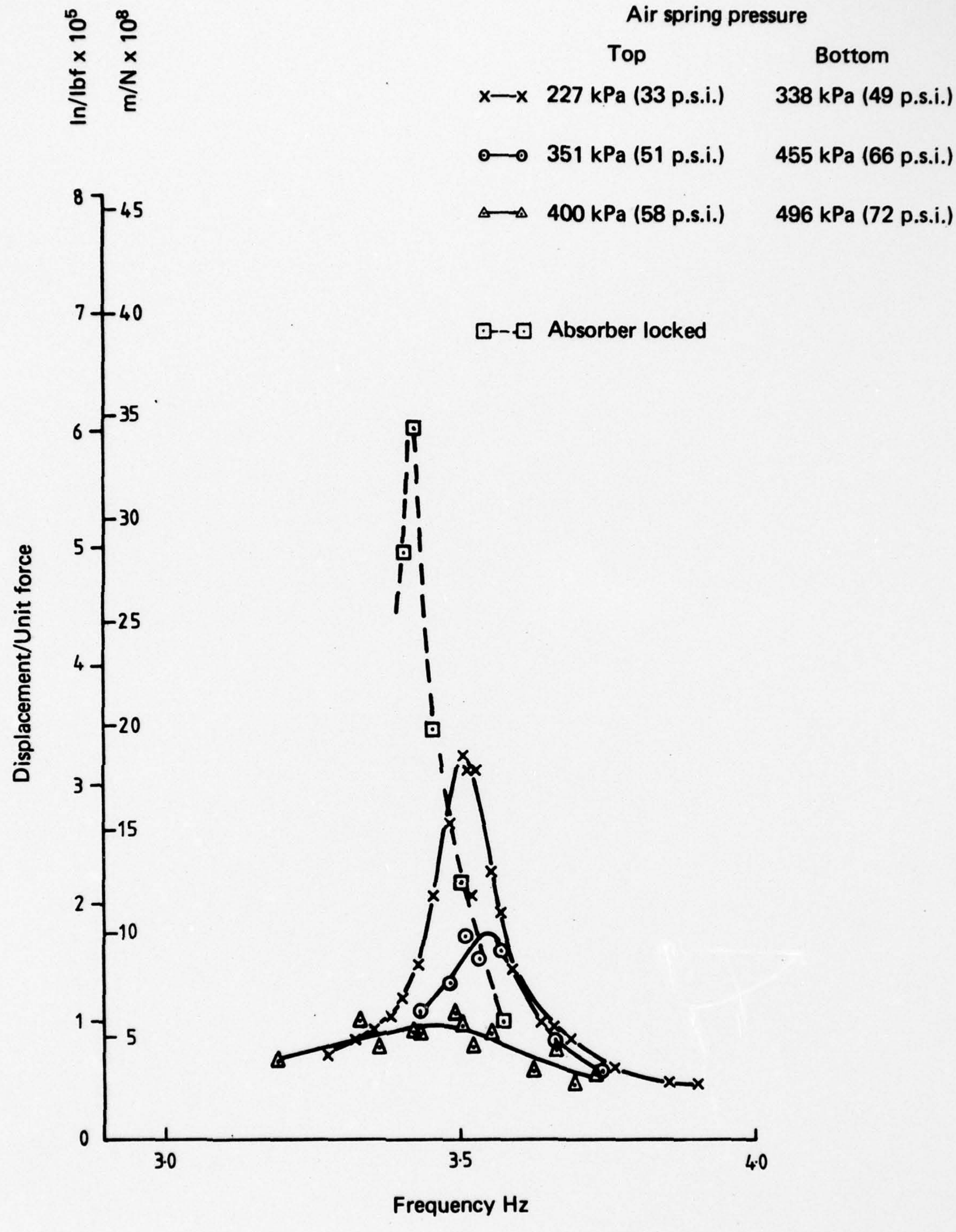


FIG. 8 EFFECT OF ABSORBER TUNING ON SHIP RESPONSE

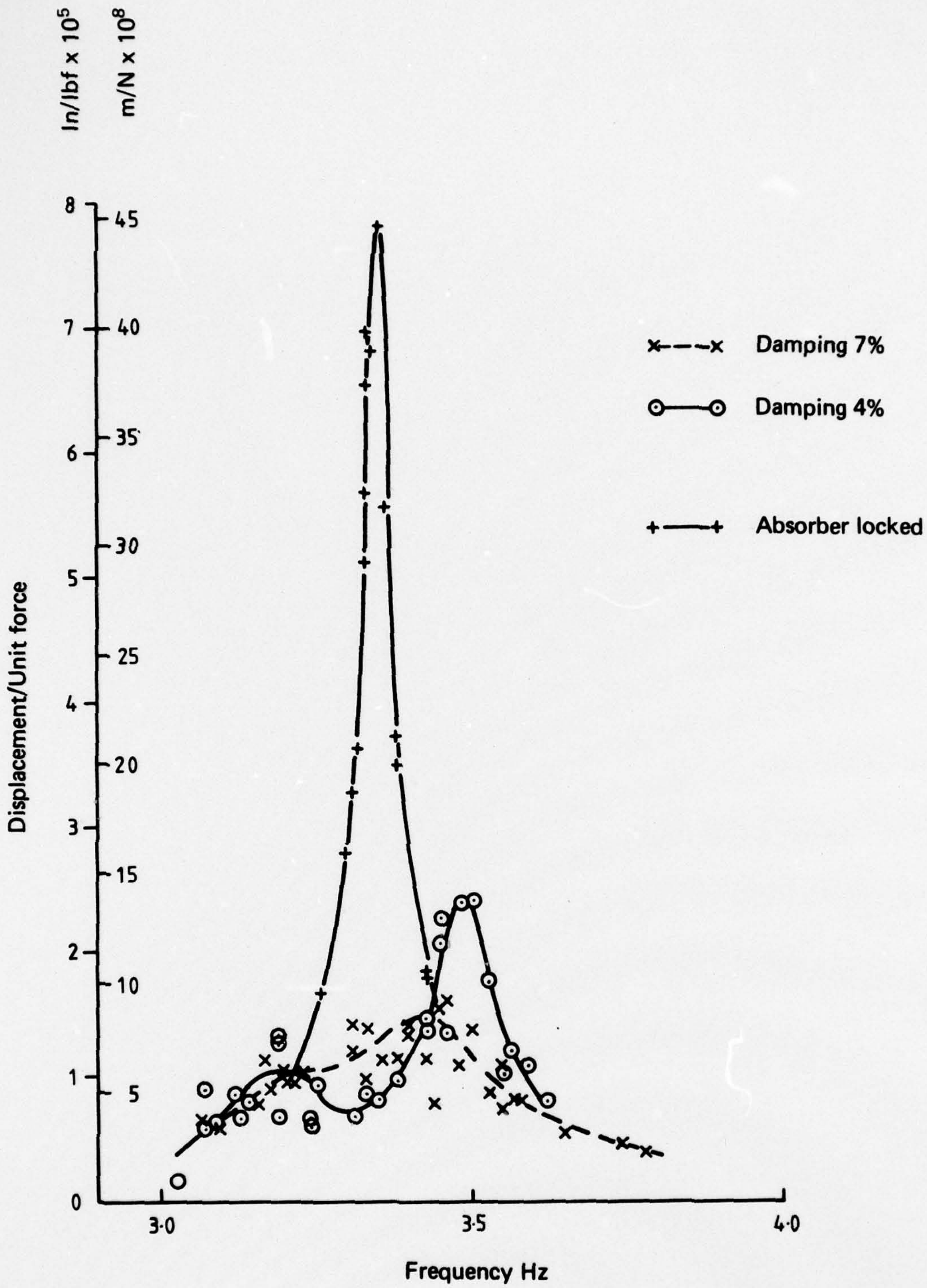


FIG. 9 EFFECT OF ABSORBER DAMPING ON SHIP RESPONSE

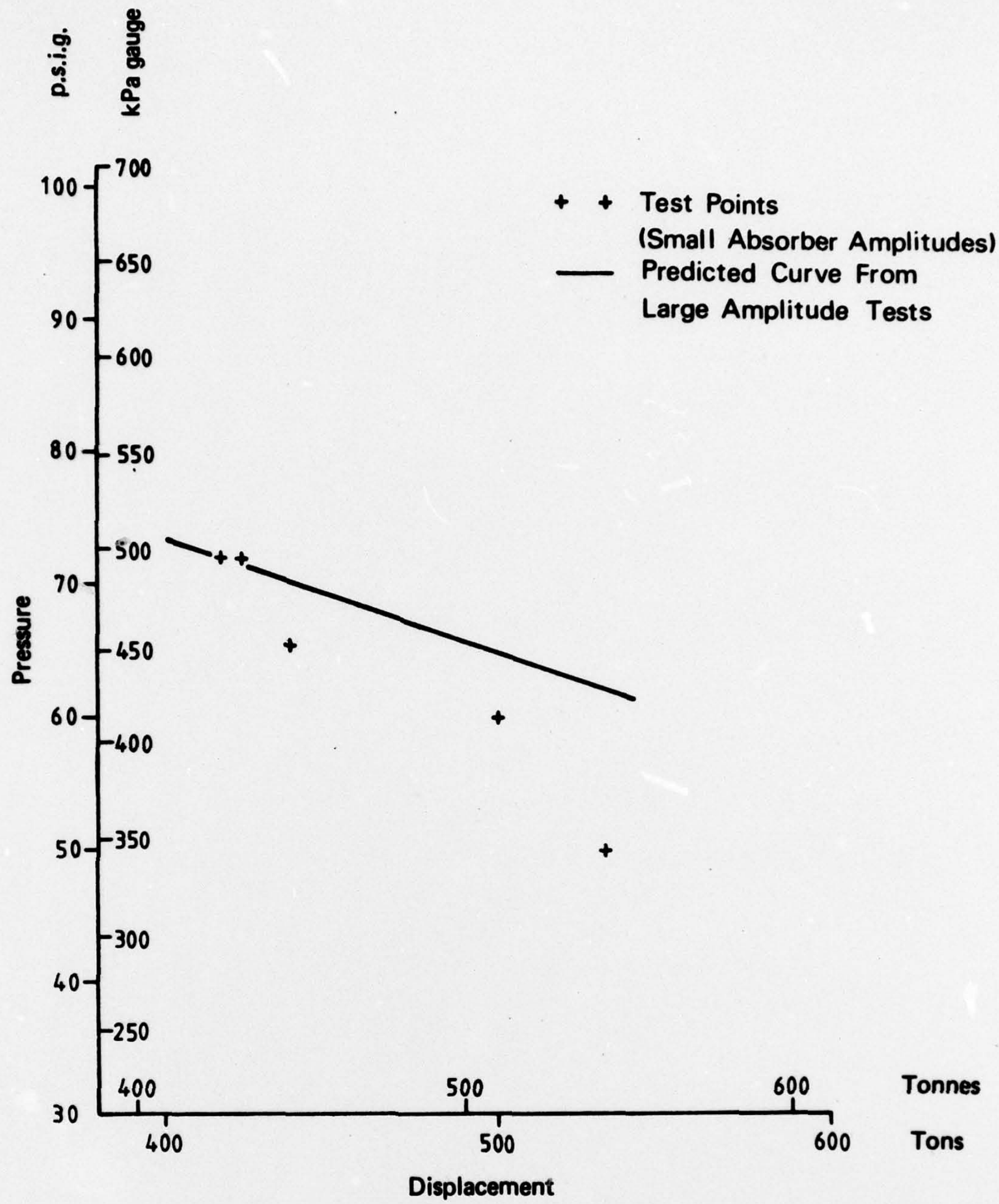


FIG. 10 RECOMMENDED AIR SPRING PRESSURE VERSUS SHIP DISPLACEMENT

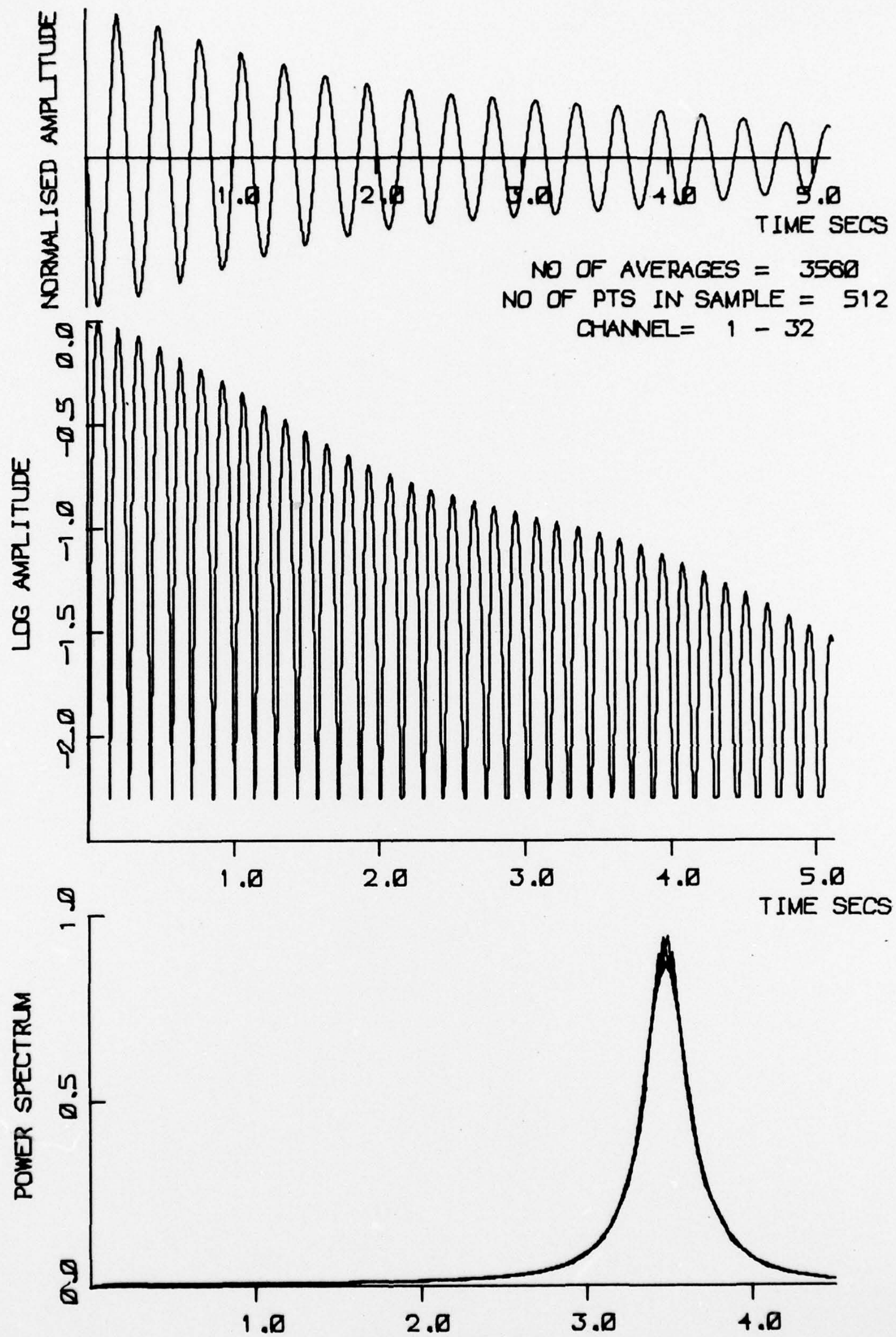


FIG. 11 RANDOM DECREMENT ANALYSIS (absorber locked)

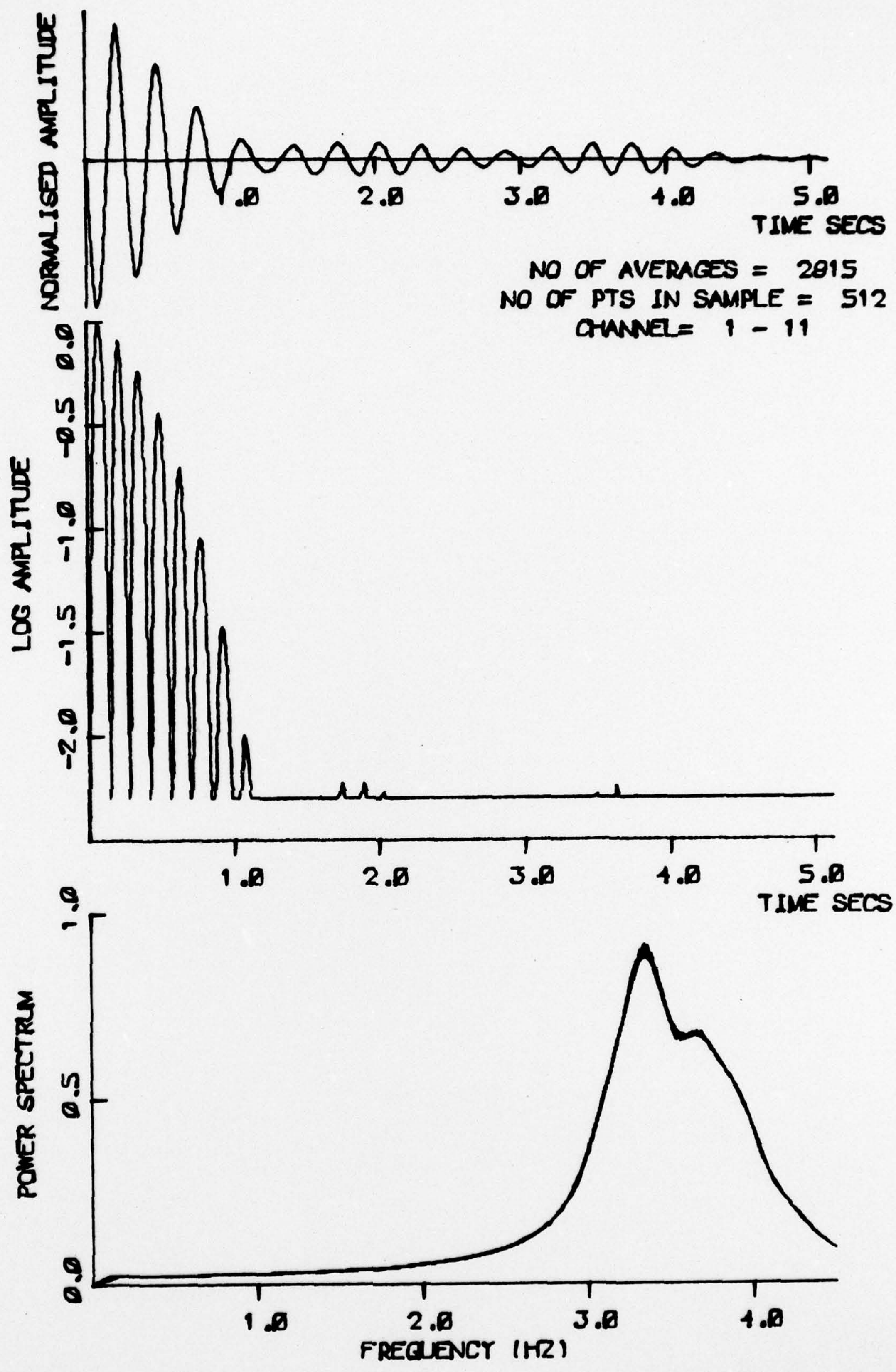


FIG. 12 RANDOM DECREMENT ANALYSIS (absorber correctly tuned)

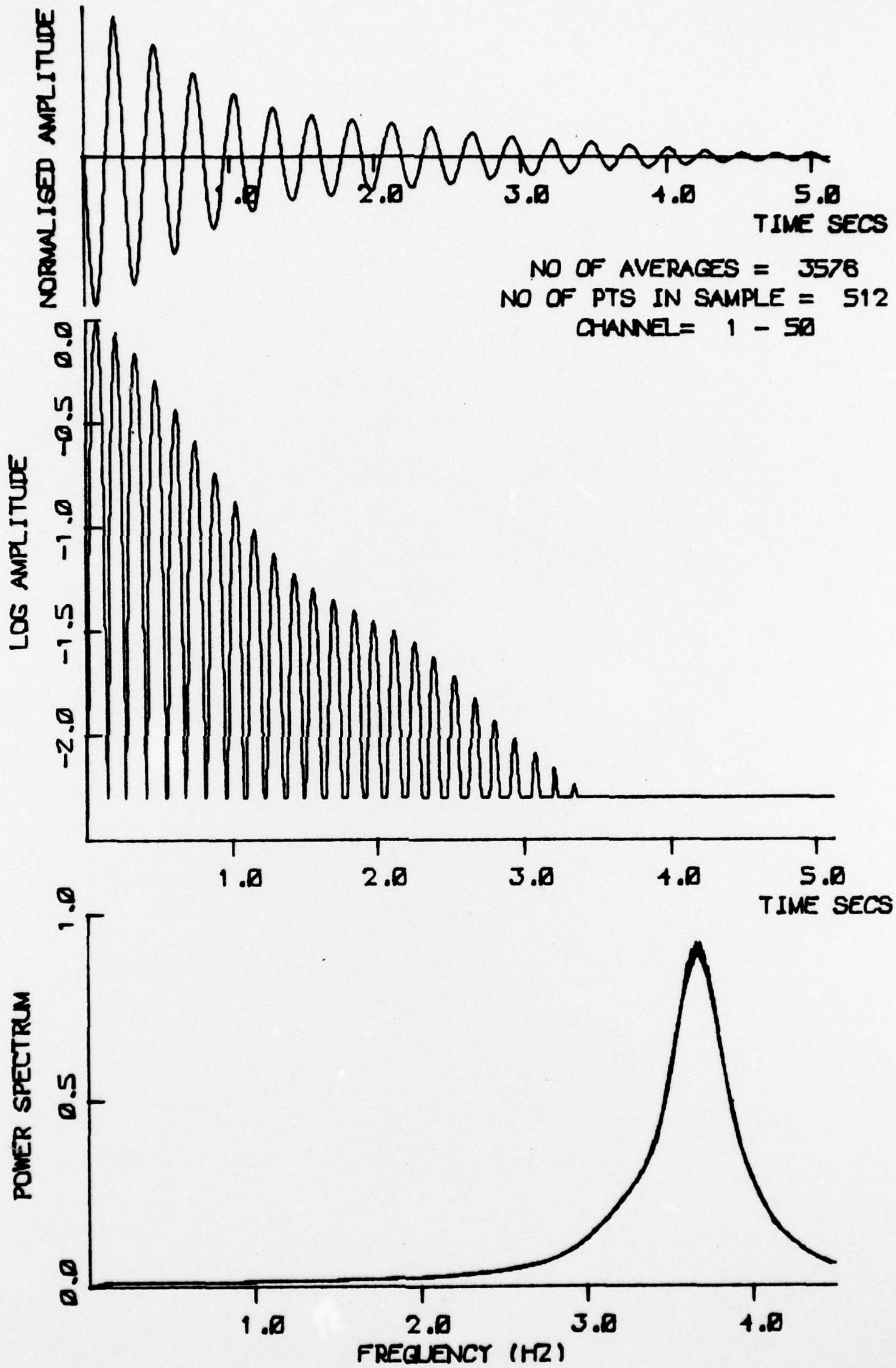


FIG. 13 RANDOM DECREMENT ANALYSIS (absorber detuned 10 psi)

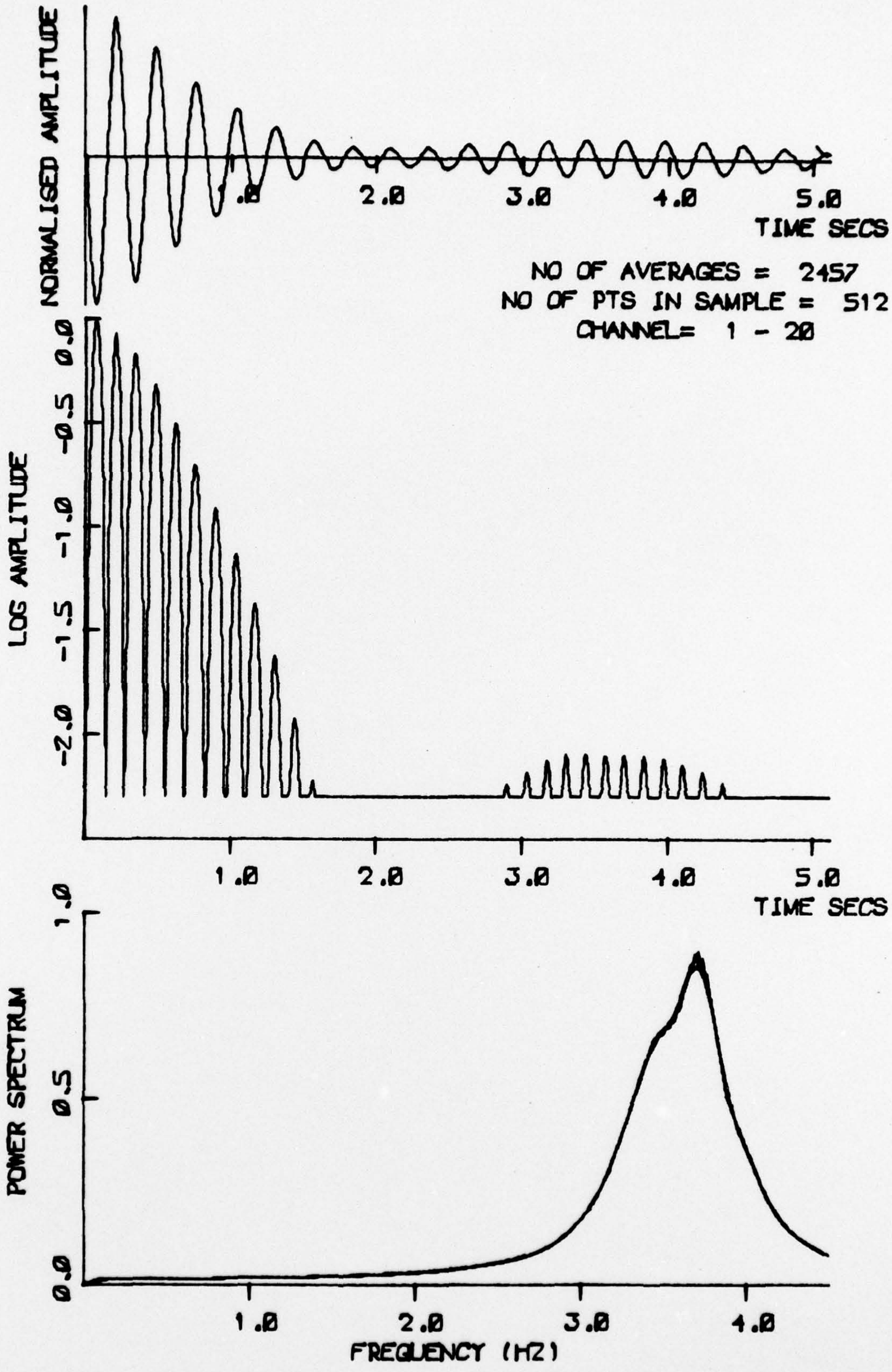


FIG. 14 RANDOM DECREMENT ANALYSIS (absorber detuned 20 psi)

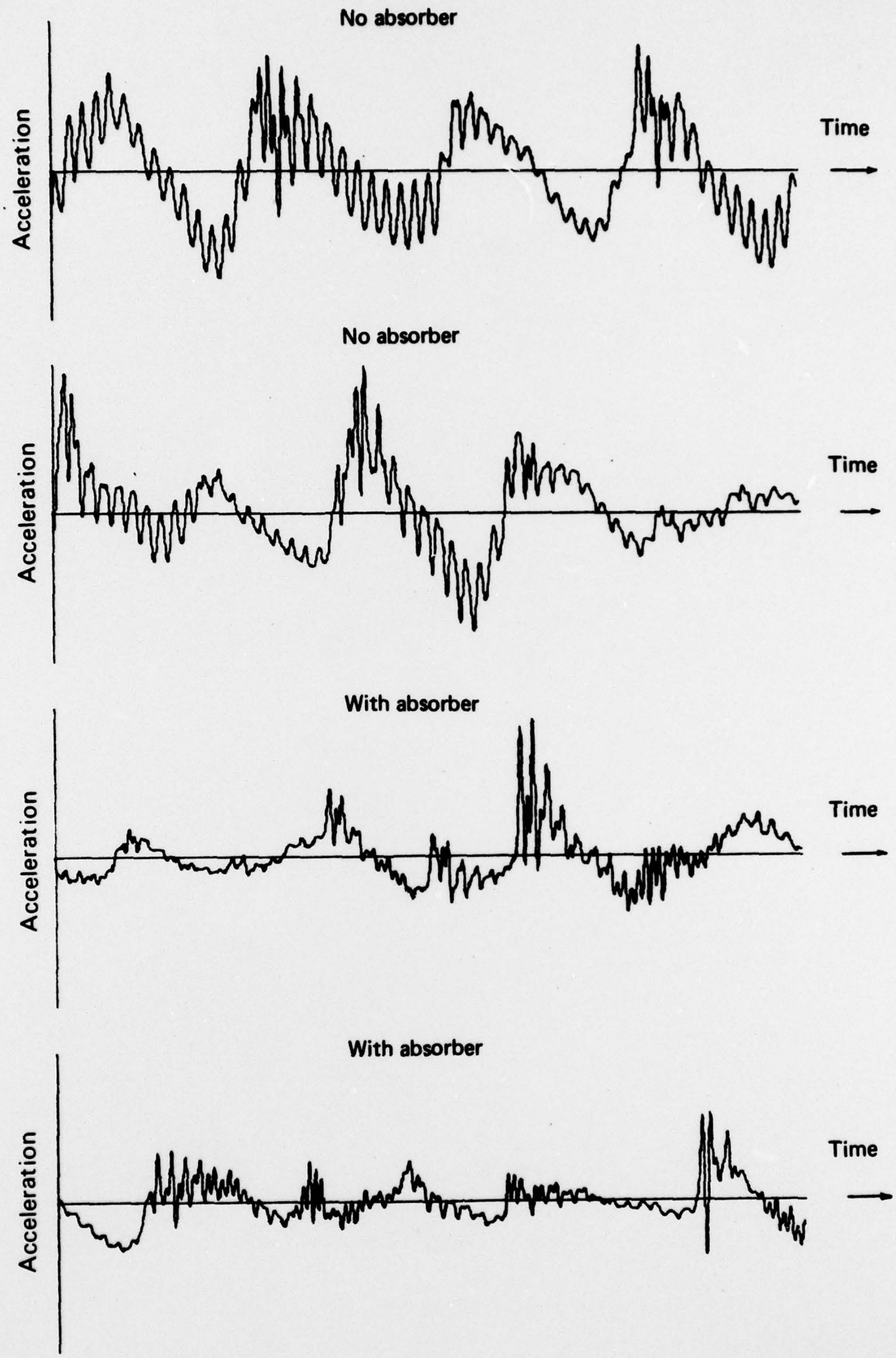


FIG. 15 COMPARISON OF SHIP RESPONSE WITH AND WITHOUT ABSORBER

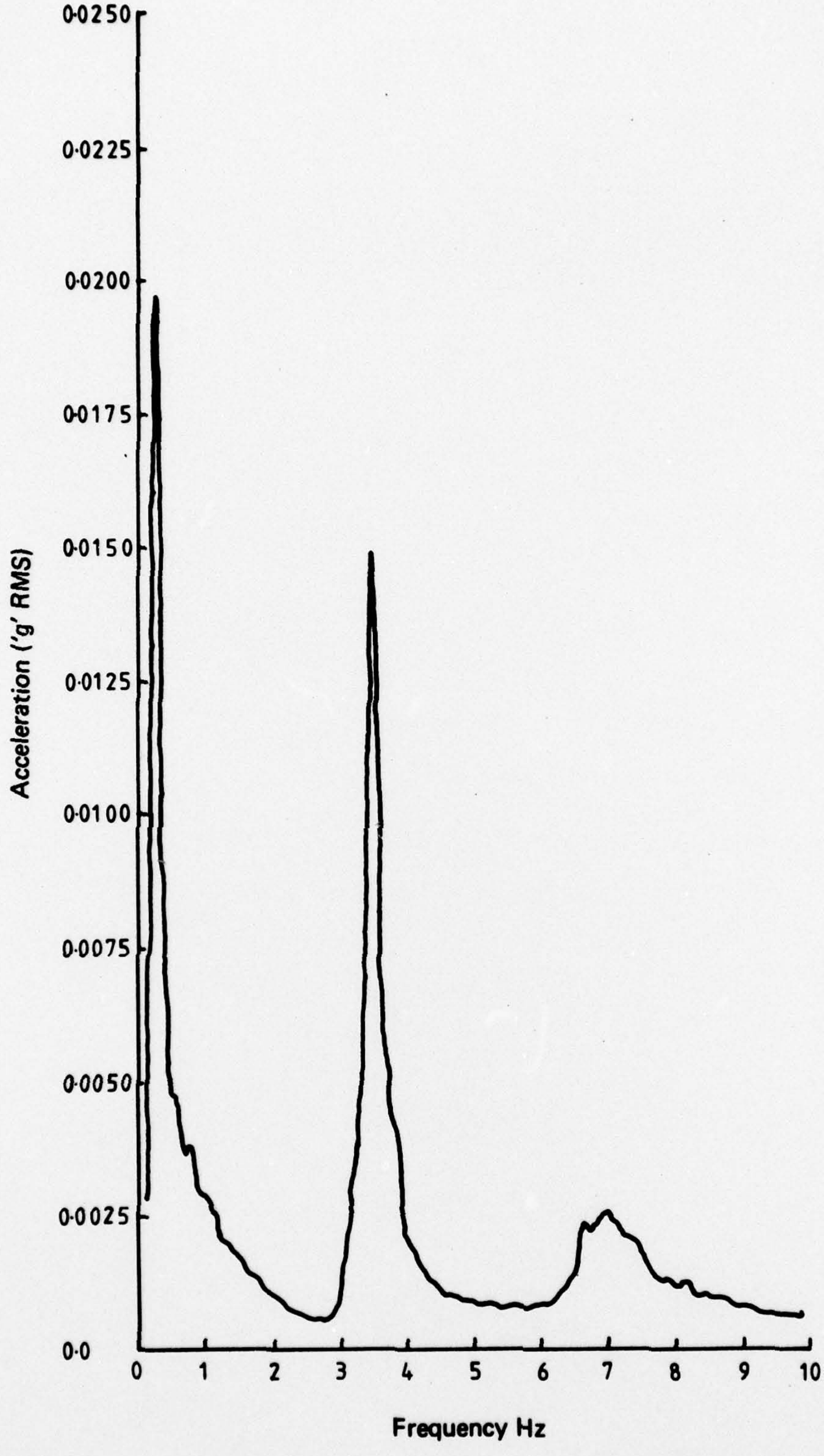


FIG. 16 FREQUENCY SPECTRUM OF RESPONSE (absorber locked)

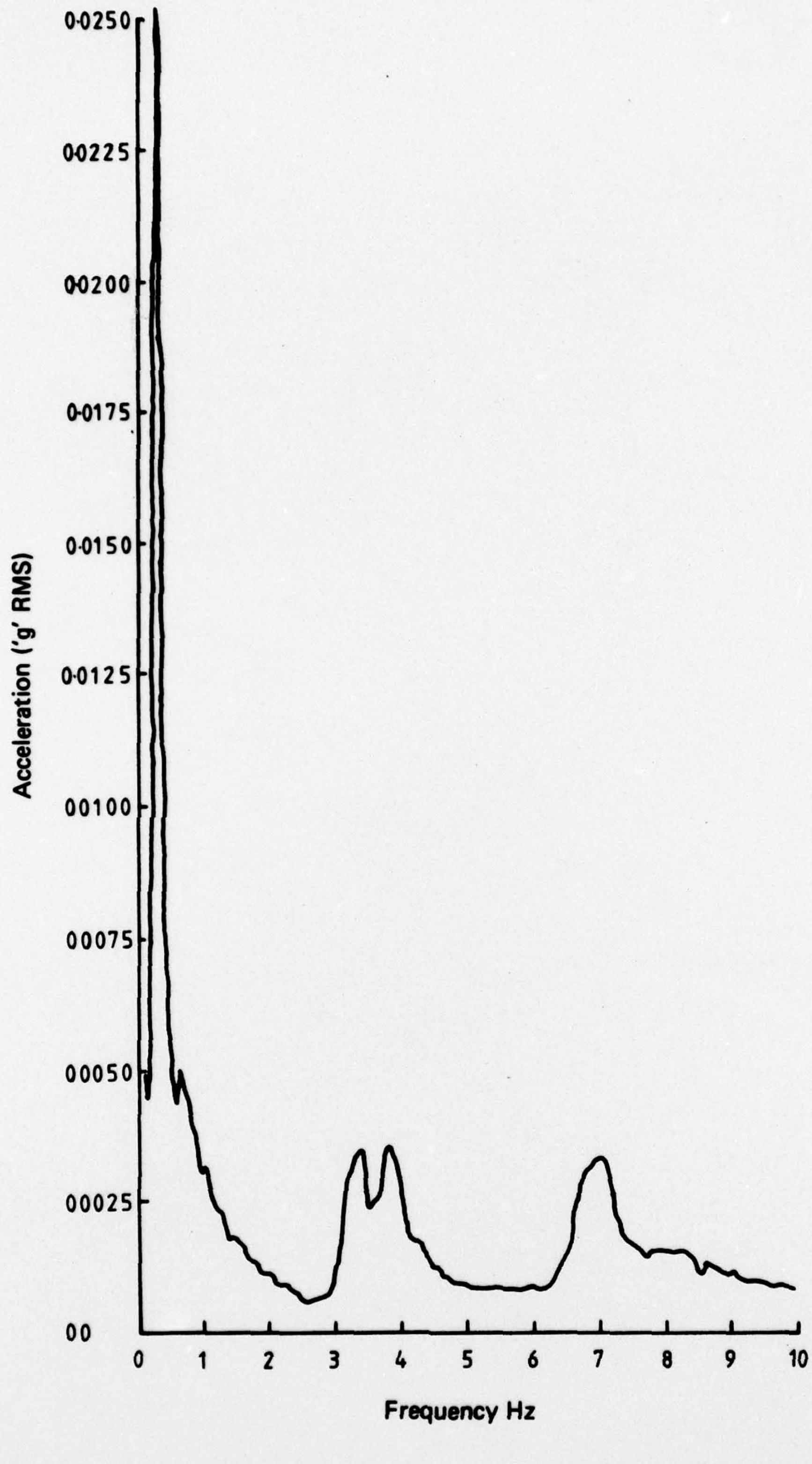


FIG. 17 FREQUENCY SPECTRUM OF RESPONSE (absorber correctly tuned)

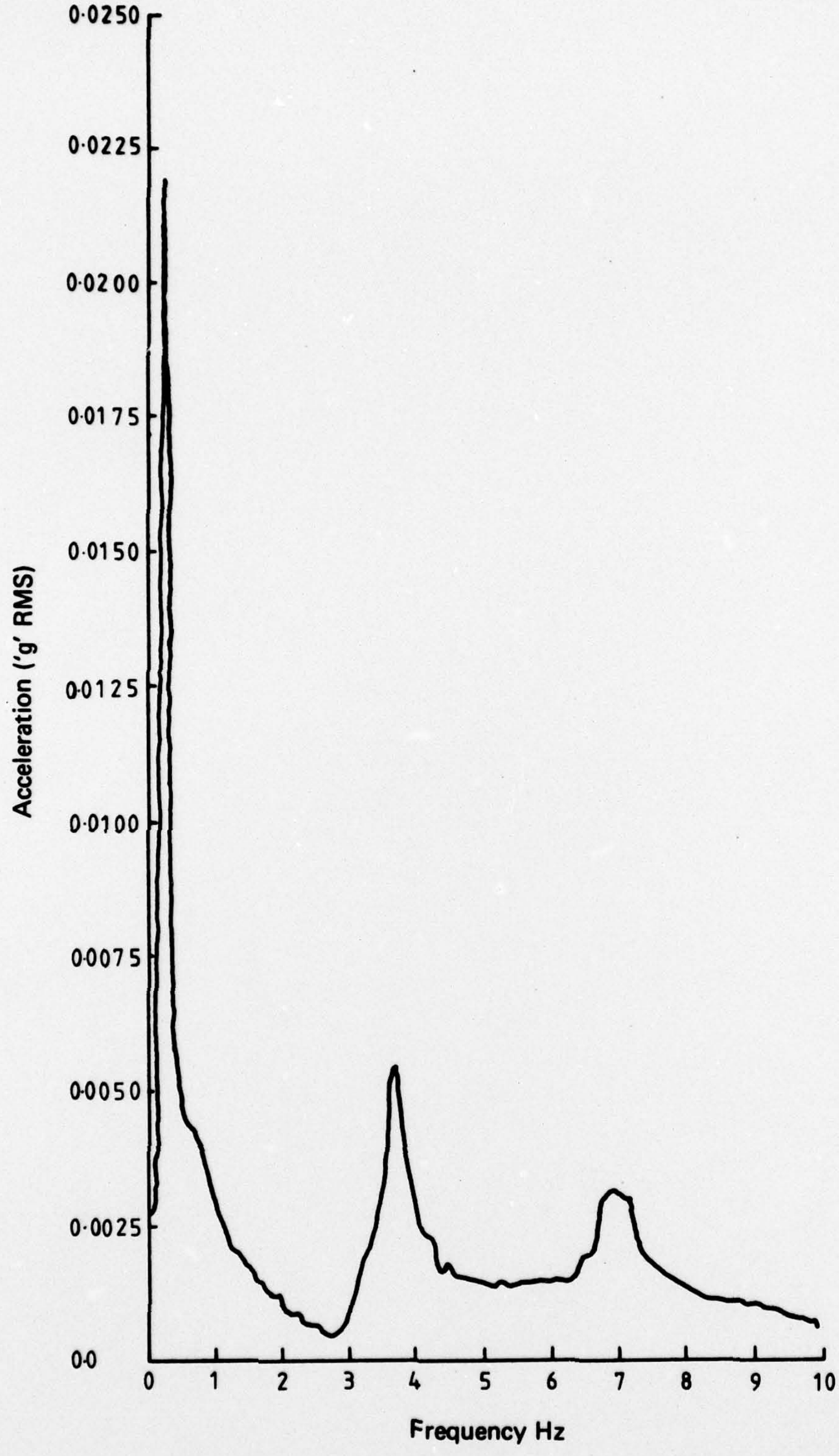


FIG. 18 FREQUENCY SPECTRUM OF RESPONSE (absorber detuned 10 p.s.i.)

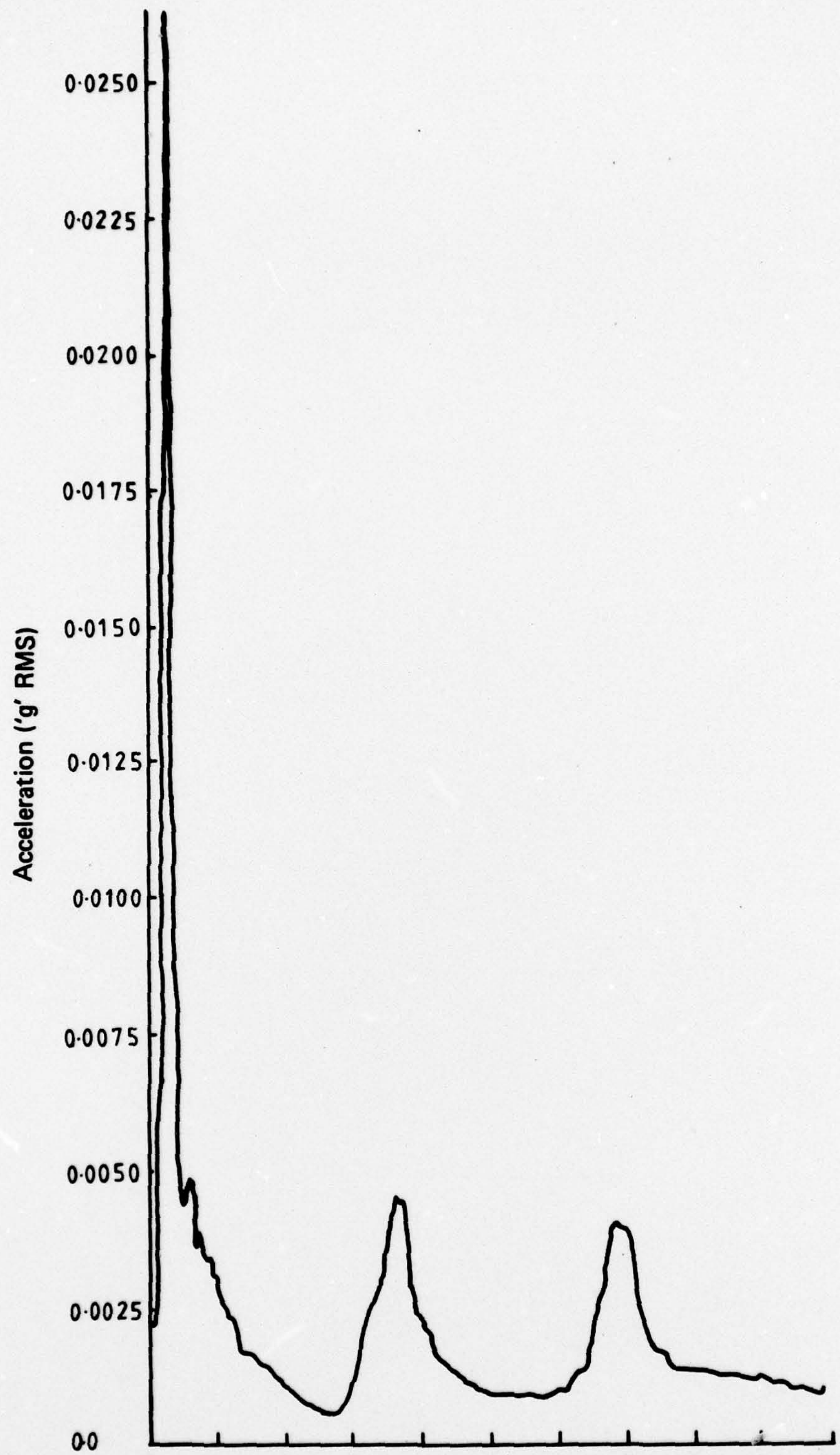


FIG. 19 FREQUENCY SPECTRUM OF RESPONSE (absorber detuned 20 p.s.i.)

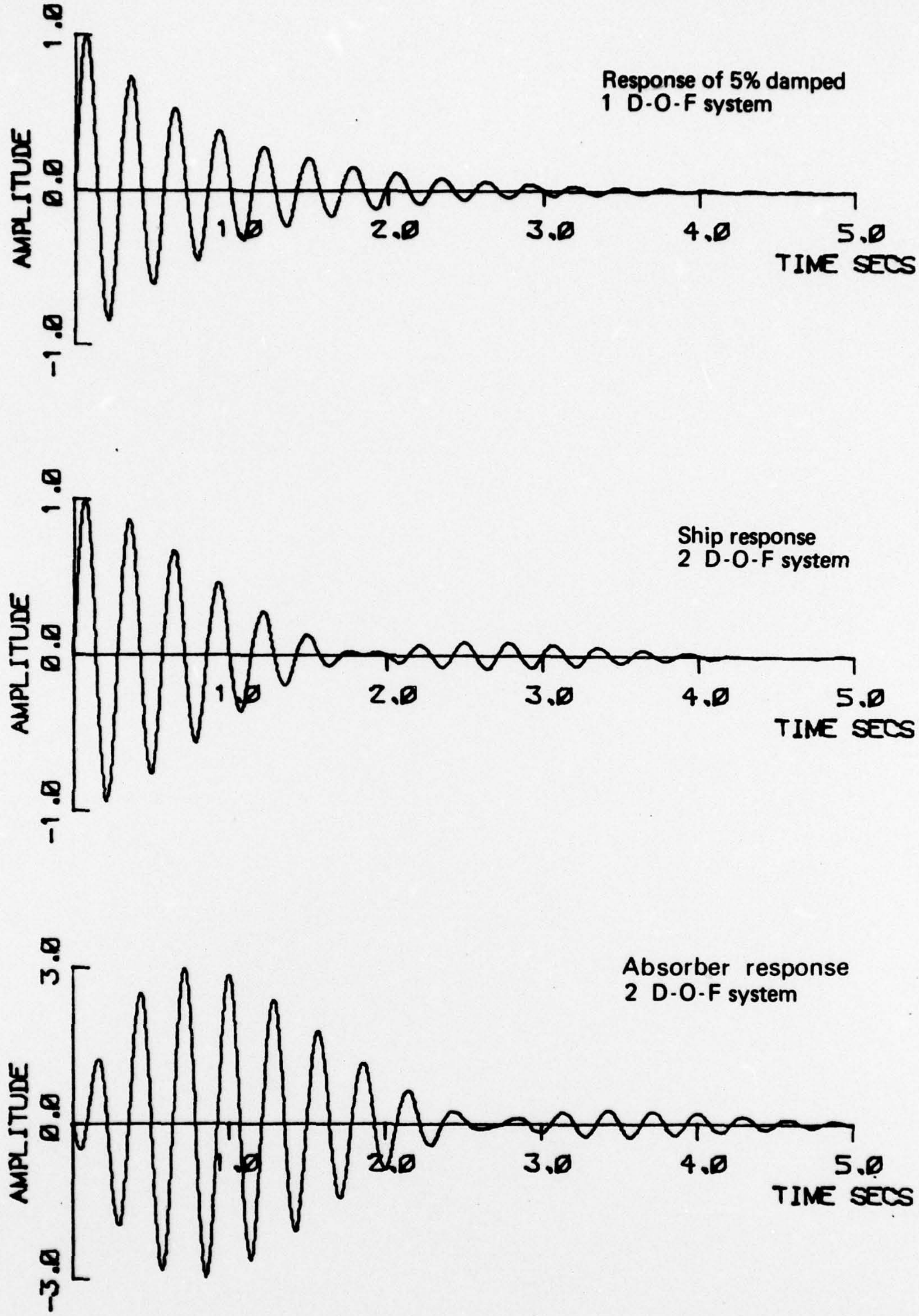


FIG. 20 COMPARISON OF THEORETICAL IMPULSE RESPONSES

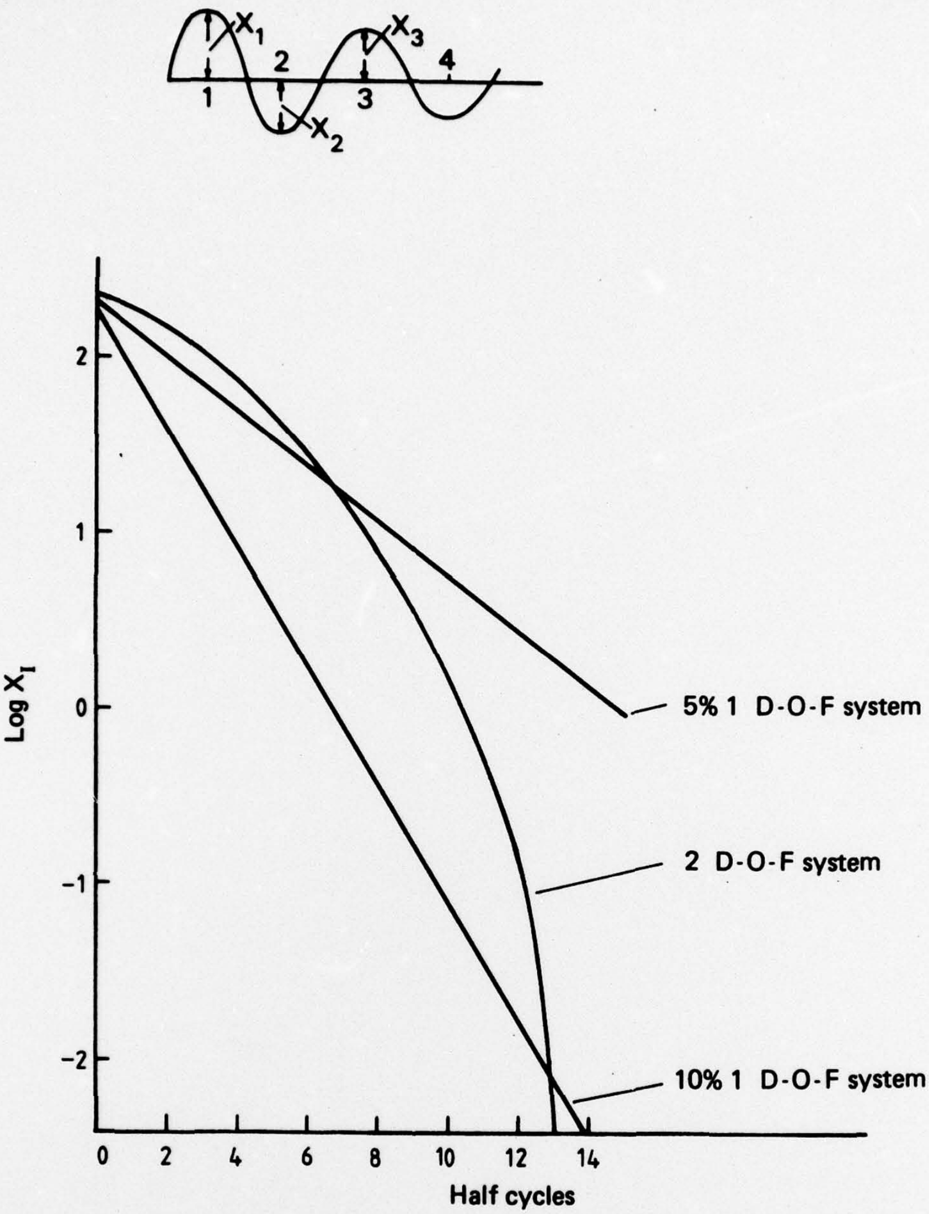


FIG. 21 COMPARISON OF ENERGY ABSORBED IN THREE SYSTEMS

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