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THE DESIGN AND INSTALLATION OF THE LCH VIBRATION ABSORBER

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PART 1 VIBRATION IN LCHs AND ITS SOLUTION

1. Introduction

1.1 The LCHs were originally designed for the Department of the Army by the 'Australian Shipbuilding Board' (ASB), now called the Shipbuilding Division of the Department of Transport and in the Specification No 130, the ASB stated 'The Primary Role is the transhipment of cargo from ships offshore to water terminals and underdeveloped beaches. The secondary role is the transport of cargo on coastal and inter island voyages and along inland waterways. The craft is a functional type landing craft with a limited seagoing capacity for use in relatively sheltered coastal and inter island waters'.

1.2 Once in service with the Army, excessive vibration began causing problems immediately mainly to the equipment on the mast and bridge (January 1973). The binnacle for mounting the compass to the wheelhouse top fractured and was stiffened with a reinforcing ring, mast yard arms fractured and were stiffened with additional braces and the air horn, originally mounted on shock mounts, actually came adrift and is now mounted on the wheelhouse top rather than on the mast.

1.3 By the time all 8 LCHs came into service, these problems were clearly evident in all craft and minor repairs/modifications were carried out locally to solve each particular problem, some of which were still of a recurring nature. Some more serious failures were reported in February 1974 when fractures in welds in hull plating were discovered and locally rectified. These structural failures, together with a report of excessive rolling by an LCH (supposedly beyond the angle of vanishing stability), formed the basis for an operational restriction which was placed on the craft to restrict their operations to coastal waters unless there were 'overriding operational considerations' or they were escorted.

1.4 In July 1974, an LCH lost a bow door while on a mission to Lord Howe Island and an improved method of restraining the bow door in the closed position was evolved. Even though the operational restrictions were theoretically still in force, in January 1975 while discussing bow door modifications, it was decreed that 'LCHs will be required to undertake ocean passages for the next few years regardless of whether or not they were designed for this role'.

2. Preliminary Investigation

2.1 In February 1975, the Aeronautical Research Laboratories in Melbourne (ARL) were asked to help investigate the problem of severe vibration in these craft. In March 1975, the fact that these vessels had not been designed for ocean voyages in heavy sea states was again stressed and it was agreed that the limitations imposed on the LCHs should be reinforced because of this severe vibration. On 6 May 1975, the first vibration measurements were made onboard HMAS LABUAN and established that wave action on the bow caused the ship to vibrate in the 2 node and 3 node modes of vibration (see Figure 1) at frequencies of 3.6 and 6.9 Hz respectively and that the 2 node mode, because it was extremely lightly damped, was the mode most liable to cause the structural damage.

2.2 The initial ARL report also stated that the levels recorded were high for the excellent sea conditions prevalent at the time and the vibration levels likely to be experienced in rough seas should give rise to serious concern over the safety of the vessel. The report recommended that the operation of these craft in rough seas be restricted until the vibration levels were reduced and that consideration should be given to the use of a damped vibration absorber for that purpose.

2.3 On receipt of the ARL report, a meeting was convened in June 1975 with representatives of ARL, Department of Transport (Shipbuilding Division), Naval Operations and Naval Ship Design in attendance and subsequent to the meeting, further vibration trials were arranged. These were carried out onboard HMAS TARAKAN during June and July 1975.

2.4 During these tests the two-node and three-node hull vertical bending modes and frequencies were measured for various values of ship's loading. The ARL designed, low frequency, mechanical vibrator was used to excite the ship for these tests. These measurements revealed that the binnacle was attached to a panel of the deckhead of the bridge, which distorted significantly in the hull three node bending mode. This problem was cured by stiffening the panel.

2.5 In addition to this work a small vibration absorber of mass 1360 Kgs (3000 lbs) (mass ratio 1/180) was fitted to the ship and its effectiveness tested at sea. Detailed analysis of the results showed that this absorber had effectively increased the damping of the hull by 1.4 to 1.8%.

2.6 ARL produced Structural Tech Memo 233 as the result of these tests and recommended the design of a vibration absorber of a mass ratio of the order of 1/40 or 1/50 ie. an absorber weight of 10 to 15 tonnes positioned in the forward trimming tanks. ie. as far forward as possible. The reason for this position was that the required mass of the absorber varies as the square of the distance from the node, the further from the node, the smaller the mass required.

2.7 The effectiveness of the absorber depends on its mass ratio, the ratio of the absorber mass to the equivalent mass of the ship. The latter has 2 components, the mass distribution of the ship and the added virtual mass of water (in this ship, because of its high ratio of beam/draft this is of the order of 3 times the mass of the ship) - see Figure 2. It is because of the large effect of virtual mass of water that an increase in hull displacement of 45% only produces a change of 12% in the natural frequency of the two node mode.

2.8 The recommended design for the vibration absorber was for it to consist of a pivoted framework of steel beams holding cast steel weights which could be moved fore and aft within the framework if required. This was supported by air springs above and below the framework with the motion damped by shock absorbers. For details of absorber design refer to PART 2.

3. Installation

3.1 Careful structural design was necessary to fit the absorber into the recommended position because of the restricted size of the space. The existing structure had a centreline watertight bulkhead (which extended only the length of No 1 WB Tank) and 2 transverse web frames (Frames 68 and 70). These were removed and replaced by a fabricated keel and frames. (See Figure 3A and B). The original longitudinal stiffening of the hull was not altered. The absorber itself was hung from a portable deck plate cut from the tank deck plating which had corresponding deck beams added. All deck plate stiffening was stopped on a carling plate running around the perimeter of the portable deck plate. (See Figure 3C). The frames were radiused up the existing continuous longitudinal bulkheads and bracketed onto the deck beams. (See Figure 3D). The seats for the air springs were fabricated onto the shell and deck plating and were of such a height as to give the required clearances for the springs with the absorber in its lower or upper position and also to give the required additional air volume - this helping the linearity of the spring stiffness. Endstops, to prevent the absorber from hitting ship structure and to support the absorber in its lower or upper position, were arranged:

a. across the longitudinal bulkheads (and pillared to each longitudinal); and

b. across the deck girders.

Cheek plates for the bearings, which were of the 'rubber in shear' type, were located at the forward edge of the portable plate. Seatings for the shock absorbers were arranged on the shell and deck plating and absorber framework so that the maximum/minimum lengths of the shock absorbers were not exceeded. (See Figure 3E). A maximum of 12 shock absorbers was initially provided allowing the damping to be varied by selection of which absorbers were actually fitted.

3.2 The detailed design of the absorber proceeded with the equipment selected chosen from that which was available 'off the shelf'. The air springs and shock absorbers were those normally used on truck suspension systems and the rubber in shear bearings were from first maintenance period after re-entering service, structural damage was discovered and on investigation, it was concluded that this was the result of grounding on a small area, which had caused cracking in 2 welds, the crippling of an unflanged portion of the centre-line keel (inside the air spring seat) and the springing of the deck plate. The force of the grounding had been sufficient to cause a 25 mm deflection in 12 mm plating, stiffened by 127 x 76 x 10 mm angles spaced at .55 m centres longitudinally and 1.12 m centres transversely.

5.2 To prevent this from occurring again, additional stiffening was added (viz) frames were added at the odd numbered frame stations, frame 72 was plated in, the keel was flanged in way of the air spring seat, and the radius of the flanges up the longitudinal bulkheads was increased.

5.3 Prior to the fitting of the absorber, in September 1977 strain gauges were positioned on various structural members to monitor the stresses in those members.

6. Summary of Trials of Prototype

6.1 Once the absorber was fitted, the ship was used for 2 weeks of trials. Initially a vibrator (nicknamed the earthquake machine) was used to excite the ship whilst at rest - thus ensuring a known input was being used. After this initial work, the ship was steamed into the prevailing sea conditions which then acted as the input causing the vibration. The trials consisted of the following tests:

a. Measurement of the ships natural frequency for varying displacements;

b. measurement of the absorber's natural frequency for varying air spring pressures.

agricultural equipment. (See Figures 4A, B and C).

3.3 The air springs require LP air for their inflation. Since no air supply was available onboard, a system was designed using HP air bottles as a supply and arranged to provide air to each of the upper and lower groups of springs and also to allow isolation of each spring hence preventing leakage from one spring affecting the others. (See Figure 5.)

4. Tuned Tank Absorber

4.1 Whilst this detailed design was proceeding, attention was drawn to the use of tuned tank vibration absorbers on tankers which were being investigated by the National Maritime Institute (UK) (then NPL). After contact with NMI, the proposal was for 2 tall narrow water tanks one each side of the bow which would have stood 5.2 m above the tank deck (level with the top of the jackstaff). (See Figure 6). On investigation, it was discovered that more mass would be required (since only approximately 30% of the mass of liquid is effective as opposed to 80% for the solid system) and the proposed design did not allow for the maximum amplitudes anticipated (76 mm as opposed to 127 mm). ARL did however conduct some experiments on this system and discovered that, because the water surface at the top of the tank disintegrated at accelerations greater than 1g, the theory was no longer valid. For these reasons Navy did not proceed with this alternative.

5. Structural Modifications

5.1 During its refit March-June 1977, HMAS BALIKPAPAN's structure was modified to allow the fitting of the vibration absorber but owing to long lead times on various equipments, the ship re-entered service without the absorber fitted. On inspection during its

c. Measurement of the effect of the absorber on vibration using differing air spring pressures on varying ship displacements.

d. Measurement of the effect of the absorber on vibration of changes in absorber damping with the absorber correctly tuned to the selected ship displacement.

6.2 At the end of the trials, because the prevailing weather conditions had not been capable of causing large amplitudes of vibration, recorders and accelerometers, which could be switched on by the crew during rough weather, were left onboard to obtain rough weather performance data.

7. Summary of Results

7.1 Measurement of strains recorded during the trials and subsequently in rough weather showed that the maximum strain in rough weather was 360×10^{-6} (stress of 74.4 MPa - 4.8 tons/in^2) whereas during the trials, by deliberately ballasting the bow out of the water and running into the sea, a strain of 712×10^{-6} (stress of 147.2 MPa - 9.5 tons/in^2) was recorded. This lends more evidence to the intuitive practice of the COs of these vessels of keeping the bow ballasted down whenever possible. For this reason and for trimming ability, the centreline void immediately aft of the absorber compartment will in future be piped as a water ballast tank although its use as such will depend on the condition of the ship at the time.

7.2 Analysis of the rough weather recordings of accelerations showed the absorber to be working satisfactorily, well within the limits of its design ie. maximum recorded amplitudes of $\pm$ 75 mm whereas the allowable limit is $\pm$ 127 mm, but indicated that better performance could be achieved with increased damping. For this reason the number of shock absorbers was increased from 12 to 16 and their position moved further from the hinge.

7.3 During an inspection in June 1978, after 10 months of continuous service, it was discovered that the bolts retaining the bearing casings had been deformed and consequently larger diameter fitted bolts were refitted. Subsequent calculations showed the original bolts should have been adequate but since normal black bolts had been used it is possible the bolt was of smaller diameter than the hole hence allowing relative movement between the two.

7.4 From the experience gained during the fitting of the prototype, the detailed design was updated and the changes discussed above included. This final design has been used in the raising of an A and A for the fitting of these absorbers to all RAN LCHs.

7.5 From the results of the trials and subsequent tests carried out in rough water, a graph of optimum spring pressure against ship displacement was prepared which allows the ship's crew to adjust the absorber to optimum performance for any displacement.

7.6 The results showed that the absorber is working as it should and is reducing the levels of vibration experienced by the ship by a factor of between 6 and 8. In weather conditions that became so rough that the ship was forced to alter course and seek shelter for other reasons, the absorber only reached 55% of its maximum amplitude.

7.7 Because of the successful performance of the absorber, approval has been given to the fitting of one to HMAS BRUNEI at an estimated cost of $50,000 and it has been recommended that these absorbers be fitted to all RAN LCHs to alleviate the problems caused by the excessive vibration presently experienced in these craft.

8. ## Conclusions

8.1 Whilst it is normal for any all welded steel hull girder to have low structural damping, the problem was compounded in this case by the poor detailed design of the transition area from longitudinal stiffening to transverse framing. In this case, all longitudinals except the centre keelson end on the forward bulkhead of the engine room ie. the front of the superstructure and the only continuous longitudinal strength members are the centre keel, the shell plating and the narrow side deck plating which is transversely stiffened and actually decreases in thickness (from 10 mm to 6 mm) in this transition area. (See Figure 7). This effectively means the shallow 'flexible' hull girder forward is joined to a deep 'stiff' (because of the superstructure) structure aft by a weak connection which acts as a hinge. With the flat bottom and bow door, when running into seas, the slamming causes the bow to be forced up as well as the ship being decelerated. Because of the height of the superstructure, it continues forward and this, together with bow being thrown upwards, causes the vessel to vibrate in its 2 node mode. Since this is very lightly damped, this motion continues for a long time, often until the next wave is encountered.

8.2 The addition of a dynamic vibration absorber relies on this impulsive vertical force at the bow for its excitation - it does nothing to relieve that initial slam - it then causes the motion to be quickly damped and hence gives the crew a short interval of rest prior to the next slam. The tests showed that the absorber has effectively reduced the levels of vibration by a factor of at least 6. The rate of decay of vibration is significantly increased and the ship is free from vibration for long intervals between wave impulses. (See Figure 8).

8.3 The absorber design has proved to be reliable and easy to tune. Its effectiveness in reducing vibration levels is fairly dramatic and it is greatly appreciated by the crew. No major problems were experienced during installation and the operating forces and pressures appear to be well within the limitation of the equipment used.

PART 2 ABSORBER DESIGN

9. General

9.1 Vibration absorbers are not new and are used in a wide range of applications eg. DC9 engine, helicopter rotor heads, electricity transmission cables, Post Office Tower in Sydney; however only one application of an absorber to a ship hull is known (1). Undoubtedly greater use would have been made of these devices if the problems of spring design, reliability and safety could have been more easily overcome. With today's materials these problems are no longer serious and no doubt much greater use of the vibration absorber will be made in the future.

9.2 There are basically two types of vibration absorber, one is called the undamped absorber and the other is the damped absorber. The former is only suitable for vibration reduction, when the vibration is caused by a constant frequency excitation and must be tuned very accurately to obtain the optimum performance. The latter, since it adds damping to the total system, is useful over the full frequency range, although in general the damped absorber is used to damp the resonance of only one mode of vibration of the structure. (The transmission line damper is one example where an attempt is made to damp more than one mode). The damped absorber is not so sensitive to small variations in tuning.

10. Selection of Absorber Type

10.1 In the analysis of any vibration problem and in the choice of a method of vibration reduction the vibration engineer needs to establish the characteristics of the vibration source and the structure. In broad terms the approach of the engineer to a vibration problem will be as follows:-

1. Measure the amplitude and frequency of the vibration of the structure and determine the frequency range or ranges which are causing the most trouble.

2. Identify the sources of these vibrations eg. engine out-of-balance, propeller blade frequency excitation, wave action, vortex shedding etc.

3. If the vibration is occurring at a distinct frequency it will be necessary to determine if a resonance of the structure is being excited. This can be done by varying the frequency of excitation (by varying the operating speed of the system or by forced excitation with a shaker and recording the rate of change of amplitude with frequency).

4. If the vibration is causing a resonant excitation then there are several possible solutions:

   a. Reduce the excitation force eg. by balancing or isolating the unit.

   b. Change the system operating speed sufficiently to avoid the resonant frequency.

   c. Increase or decrease the resonant frequency of the structure by varying stiffness or mass.

   d. Add damping to the structure to minimise the resonant amplitude.

5. It is usual to measure or calculate the natural modes of vibration of the structure so that the effects of added mass or stiffness can be assessed. The damping in each mode is another parameter which should be measured.

10.2 These are the basic guide lines which the vibration engineer uses to assess the problem. In some instances only a small change in speed is necessary to reduce the vibration amplitude, in others correct balancing of equipment can reduce the input forces. Sometimes the structure is stiffened, at other times it may be made less stiff or have more mass added. If the vibration is not a resonant problem often the only solutions are increased mass or stiffness eg. vibration isolated engine bedplates are made very heavy to minimise forced vibration. Damping is only of use in resonant vibration problems and in general will not significantly reduce a forced vibration response.

10.3 The landing craft suffered vibration in primarily the fundamental hull vertical bending mode, which was very lightly damped, and the excitation was caused by impulsive wave action. The source of vibration was therefore not constant and it was not considered practical to reduce this force by re-designing the shape of the bow. Neither was it practical to stiffen the ship adequately to reduce the vibration amplitude. Consequently the only practical way of reducing the vibration levels was considered to be by increasing the damping in the ship.

10.4 It is not possible in an all-welded hull to obtain high values of damping, nor is it easy to increase the damping in the hull directly. The choice of a damped vibration absorber was therefore the only alternative available. When the ship is excited by wave action, the absorber, which has a natural frequency close to that of the ship, is excited to much larger amplitudes than the ship. This large relative motion between the ship and the absorber provides a convenient method of dissipating energy by means of damper units between the absorber and the ship. In this way the effective damping in the fundamental mode is increased. The damper is also effective in every other mode of vibration but to a very much lesser extent.

11. Basic Design

11.1 In order to provide design data for an absorber of the type used in the landing craft it is necessary to know the mode shape and frequency of vibration of the ship in the fundamental vertical mode. In this instance the ship was excited by the ARL low frequency shaker and the mode shape measured by means of a moveable accelerometer. Knowing the mode shape of vibration and the mass distribution of the ship, including the entrained water, it is possible to calculate the effective mass of the ship at the point of installation of the absorber. For the LCH this is of the order of 400 tonnes at frame 72.

11.2 Having obtained this figure the main factors which need to be selected and optimised are:-

a. The ratio of absorber mass to the equivalent mass of the ship.

b. The maximum amplitude of the absorber.

c. The choice of absorber natural frequency.

d. The choice of absorber damping.

11.3 The choice of mass ratio is largely dictated by space and other practical engineering problems. In general, the larger the absorber mass the more effective the system will be, but problems of installation, spring and damper design, ease of adjustment and failsafe considerations all influence this choice. Once the mass ratio has been selected the maximum effectiveness of the system is fixed, and the choice of natural frequency and absorber damping are such as to obtain this maximum effectiveness of the absorber. These choices in turn decide the maximum amplitude of the absorber relative to the ship. The absolute amplitude of the absorber is of course dependent on the maximum value of excitation. In this instance the mass ratio was chosen to be approximately 40 which requires an absorber with an effective mass of 10 tonnes.

11.4 Equation 2.3 of ref 2 states that the improvement in ship response, as a result of using a damped absorber, is given by the ratio

$$X = 2\,\gamma_2 \sqrt{1 + \frac{2M_2}{M_1}} \qquad 1.$$

Here $\gamma_2 = 0.006$, $\frac{M_2}{M_1} = 40$ giving $X \simeq 0.1$ ie. a reduction in response to approximately one tenth the value without the absorber. This compares the steady state responses of the ship alone and of the ship absorber system, to a constant input force at the system natural frequency. This is a measure of its steady state response improvement, which is considered to be a good indication of its improvement in transient response. From equation 1 it is clear that the improvement in response is approximately proportional to the square root of the mass ratio.

11.5 Once the choice of mass ratio has been made the optimum tuning anddamping of the absorber are given by the equations

$$g_r = \frac{1}{1 + M_1/M_2} \qquad 2.$$

$$\gamma_1 = \sqrt{\frac{3M_1/M_2}{8(1 + M_1/M_2)^3}} \qquad 3.$$

Also the ratio of absorber amplitude to ship amplitude can be approximated by

$$X_s = \frac{1}{2\gamma_1} \qquad 4.$$

Equations 1-4 are all that are required to optimise the absorber design. It is apparent that we need only know the damping in the ship and the equivalent mass at the location of the absorber, in order to determine the main characteristics of the absorber.

Equation 4 is a particularly simple formula to determine the absorber amplitude. Most standard texts on absorber theory, because they are based on a different co-ordinate system, provide design equations which are considerably more difficult to use than the 4 equations given above. It is considered that these are sufficient for most absorber designs.

11.6 Using $M_2/M_1 = 40$ equations 2-4 give the optimum absorber frequency as 97.5% of the ship frequency, the required absorber damping as approximately 10% of critical and the ratio of absorber to ship amplitude as 5. The maximum expected ship amplitude in rough seas was determined from sea trials on HMAS LABUAN as $\pm$ 12 mm which would indicate a maximum absorber amplitude of $\pm$ 60 mm. In practice the design allowed for an amplitude of $\pm$ 127 mm.

12. Detailed Design

12.1 The ideal absorber is one in which the mass moves linearly on a linear spring; however for ease of construction a mass on a beam, pivoted about one end, was chosen (Figure 4A). This constrains the mass to move in the arc of a circle and the moment of inertia of the system about the pivot becomes important. It can be easily shown that such a system is dynamically equivalent to one mass at the pivot and another at the centre of percussion. The only effective mass therefore is the latter one, which is of magnitude $M_1 = m\left(\frac{R^2}{R^2 + k^2}\right)$ and acts at a radius $\frac{R^2 + k^2}{R}$

where m = mass of total absorber

k = radius of gyration about the centre of gravity

R = distance of centre of gravity from the pivot

Good design will maintain this effective mass as large a fraction of the total mass as possible. The natural frequency of this system is given by

$$f_n = \frac{1}{2\pi}\sqrt{\frac{KR_s^2}{I_T}} \qquad 5.$$

where I_T = total mass moment of inertia of system about the pivot

K = spring stiffness at radius R_s from the pivot.

12.2 The proposed design is illustrated in Figure 4 Air Springs have been used in preference to steel for the following reasons:

a. Corrosion problems are reduced.

b. Air springs can give greater deflections.

c. In an emergency, the springs can be deflated, the absorber will be de-tuned and can easily be locked.

d. Variation of air pressure will change the natural frequency of the absorber.

e. Air springs are non-linear especially at large deflections and this feature can be used for amplitude limiting.

All other things being equal, a large volume air spring has a more linear spring rate than one with a small volume; however increasing the volume decreases the spring rate. The choice of a particular unit depends on deflection, mean pressure, maximum pressure, volume, mean load and spring rate.

12.3 As shown in Figure 4A, B and C six springs are used, each spring being isolated from the others. Failure of one or more of the springs will not be catastrophic, since the maximum amplitudes of vibration occur only when the natural frequency of the absorber is tuned closely to that of the ship. Any large variations from the optimum, will result in smaller vibration amplitudes of the absorber, which will then behave in the same way as any other resiliently mounted equipment on the ship, and its response will be adequately controlled by the dampers fitted.

12.4 The choice of appropriate damping units is straight-forward since the damping requirements are not arduous. Commercial-vehicle shock absorbers with a total stroke of approximately 254 mm (10 ins) have been chosen, and provision is made for fitting up to 16 of these as required.

12.5 The pivot bearings are of the rubber-in-shear type and their design is again straight-forward. A source of compressed gas and a self centring device are used to provide fine frequency tuning and automatic levelling of the absorber. Overload stops are also fitted.

13. Tuning

13.1 An absorber can be constructed and adjusted separately from the ship, but this may not always be practicable. Variations in ship's loading can cause changes in ship natural frequency which affect the optimum tuning ratio. Also variations in effective mass ratio can be caused not only by change in loading, but also by changes in mode shape. For these and other reasons it is desirable to tune the absorber while it is attached to the ship's structure.

13.2 If the absorber is fitted to the ship and the ship is excited by a simple-harmonic force, the response of the ship q_2 and of the absorber relative to the ship q_1, can be measured. If the ratio of these measurements q_1/q_2 is plotted against frequency the result will be similar to that shown in Figure 9. The frequency corresponding to the peak amplitude in this figure is the uncoupled natural frequency of the absorber, (ie. the natural frequency of the absorber not coupled to the ship) and the magnitude of the peak response (q_1/q_2) max is a measure of the absorber damping,

$$\text{ie} \quad \gamma_1 = \frac{1}{2 \frac{q_1}{q_2}_{MAX}} \qquad 6.$$

In this way the main design parameters of the absorber, ie. its frequency and its damping, can be determined after its installation.

13.3 The natural frequency of the ship, as a single-degree-of-freedom system, can now be determined by using the absorber as a tuned vibrator, as described in Reference 2. The absorber damping is reduced to zero and an exciter, attached to the absorber mass, is used to excite the ship. The natural frequency of the ship is that frequency at which the phase difference between the motion of the ship and absorber is 90°. A plot of the phase difference against frequency will be similar to figure 10A and the ship's natural frequency can be easily determined.

13.4 The magnitude of the damping in the ship can be estimated, either from the slope of this curve at resonance, or from a vector plot, such as that shown in Figure 10B. In this figure the amplitude ratio of the ship to the absorber is plotted as a vector at the corresponding phase angle θ and the damping is obtained from the diameter of the best fitting circle to the experimental points, as indicated.

13.5 We now know the natural frequencies and dampings of the ship and absorber, as uncoupled systems. The natural frequencies of the coupled system can be obtained from a graph of ship, or absorber, amplitude versus frequency. This will be of the form shown in Figure 11. Optimum absorber damping is that value at which the peaks in the ship's response are of equal height.

14. Performance Tests in HMAS BALIKPAPAN

14.1 The vibration absorber is 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.

14.2 The ship was excited by the ARL low frequency vibrator and the ship amplitude measured over a range of excitation frequencies. A typical result of such a test is given in Figure 12 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.

14.3 In Figure 13 the variation of ship natural frequency with loading is presented for a range of loadings. Over this range the variation is close to linear.

14.4 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.

14.5 In Figure 14 the absorber natural frequency is plotted as a function of air spring pressure. In the range of pressures investigated the variation is essentially linear.

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

14.7 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.

14.8 The ship was again vibrated in calm water using the ARL mechanical vibrator and the response measured, both with the absorber in operation and locked. In Figure 15 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.

14.9 From the data presented in Figures 14 and 15 it is possible to obtain a graph of optimum absorber tuning versus ship's displacement. This is presented in Figure 16. Also shown in this figure are the actual pressures used for tests on 5 loading conditions.

15. Sea Trials of Absorber

15.1 The vibration tests conducted in calm water were adequate to check the absorber design and provide a basis for optimum absorber tuning. This data was used to adjust the absorber for a light ship ballasted condition which was known to produce very severe vibration, even in calm seas.

15.2 The ship and absorber responses were measured by accelerometers located at frame 68 and recorded on a magnetic tape recorder. The ship was steered directly into the waves in order to maximise the vibration input and records of the vibration were made with the absorber locked and with the absorber free and correctly tuned.

15.3 The vibration records obtained were analysed using the Random Decrement technique (3) to obtain the impulse response of the ship-absorber system in the two-node mode of vibration. Typical records of these responses are illustrated in Figures 17 and 18. A comparison of these figures indicates how much more rapidly the response of the ship decays when the absorber is operating.

15.4 Further sea trials were conducted on a voyage from Brisbane to Sydney in sea state 4. This was the first opportunity to obtain significant rough sea data. The results of this test (4) indicate that the absorber is effective at large amplitudes of motion. The random decrement signature of this test is shown in Figure 19 and again it may be seen that the rate of decay of vibration is high, even though in this example the absorber is not precisely tuned as is evidenced by the beats occurring in the motion.

15.5 During this test the maximum recorded amplitude of the absorber was approximately $\pm$ 75 mm. A time history of the ship and absorber amplitudes is reproduced in Figure 20 (NB. only the response in the frequency range, close to the two-node mode of vibration is shown). In Figure 21 time histories of ship vibration with and without the absorber operating are compared it is clear that the two-node vibration is effectively reduced.

15.6 Since the absorber was installed, instrumentation has been maintained on board the ship to record ship and absorber vibration in rough sea conditions. A considerable amount of data has now been obtained and after approximately one year of operation, the system is still working well.

Acknowledgements

The authors would like to express their gratitude to all others concerned with this project for the assistance received during this work. In particular, the help received from LTCDR S. Leflay, CPOMTH Stapely of HMAS MORETON and LT M. Proctor (CO), CPOMTP Nation of HMAS BALIKPAPAN is gratefully acknowledged.

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1. TODD, F.H. — Ship Hull Vibration Edward Arnold (Publishers) Ltd. 1961.

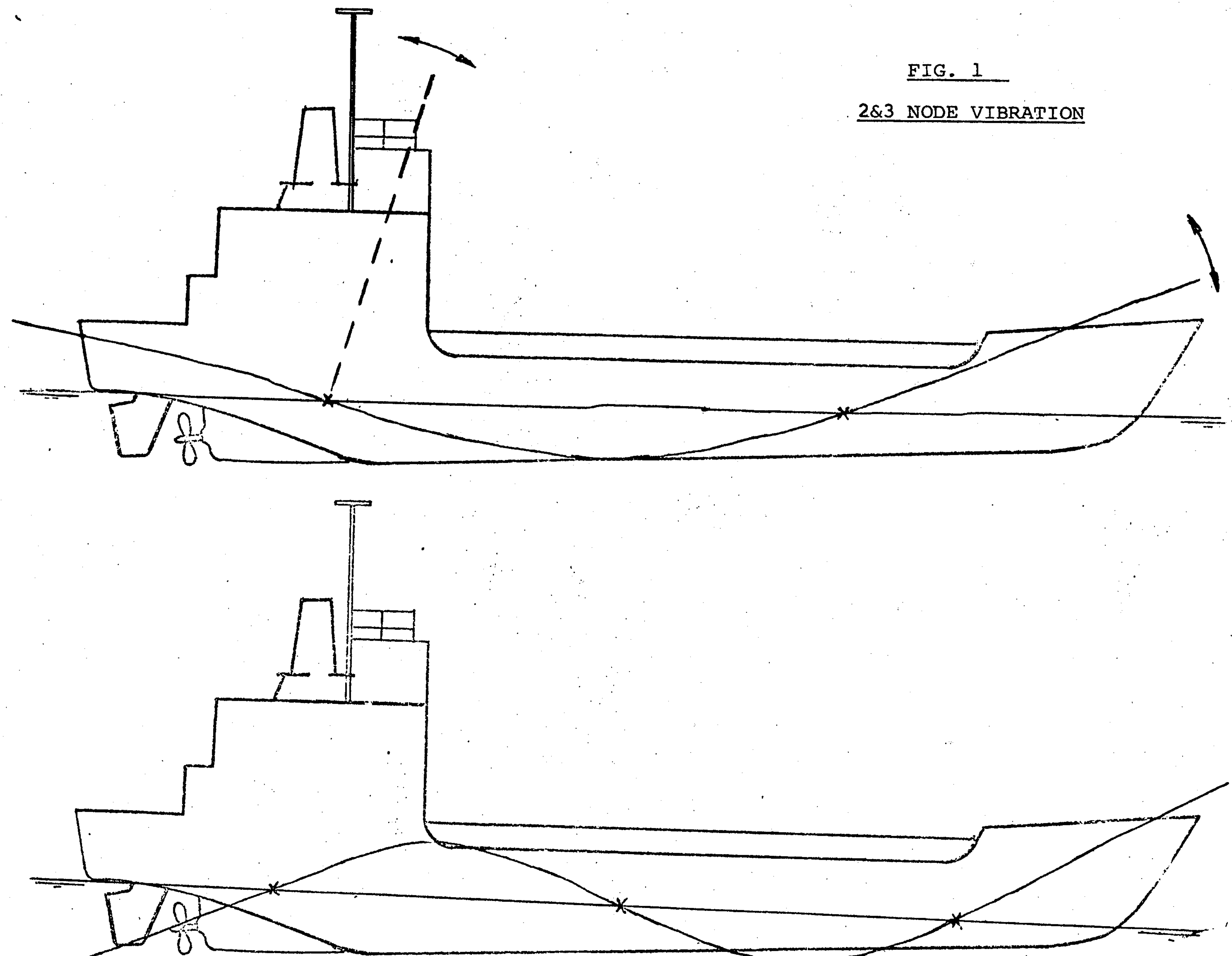
2. LONG, G. & FARRELL P.A. — The design and installation of a damped vibration absorber ARL Structures Notes 440 1977.

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

4. FARRELL, P.A. & LONG, G. — An evaluation of a damped vibration absorber in rough seas. ARL Struct. T.M. to be published.

FIG. 1

2&3 NODE VIBRATION



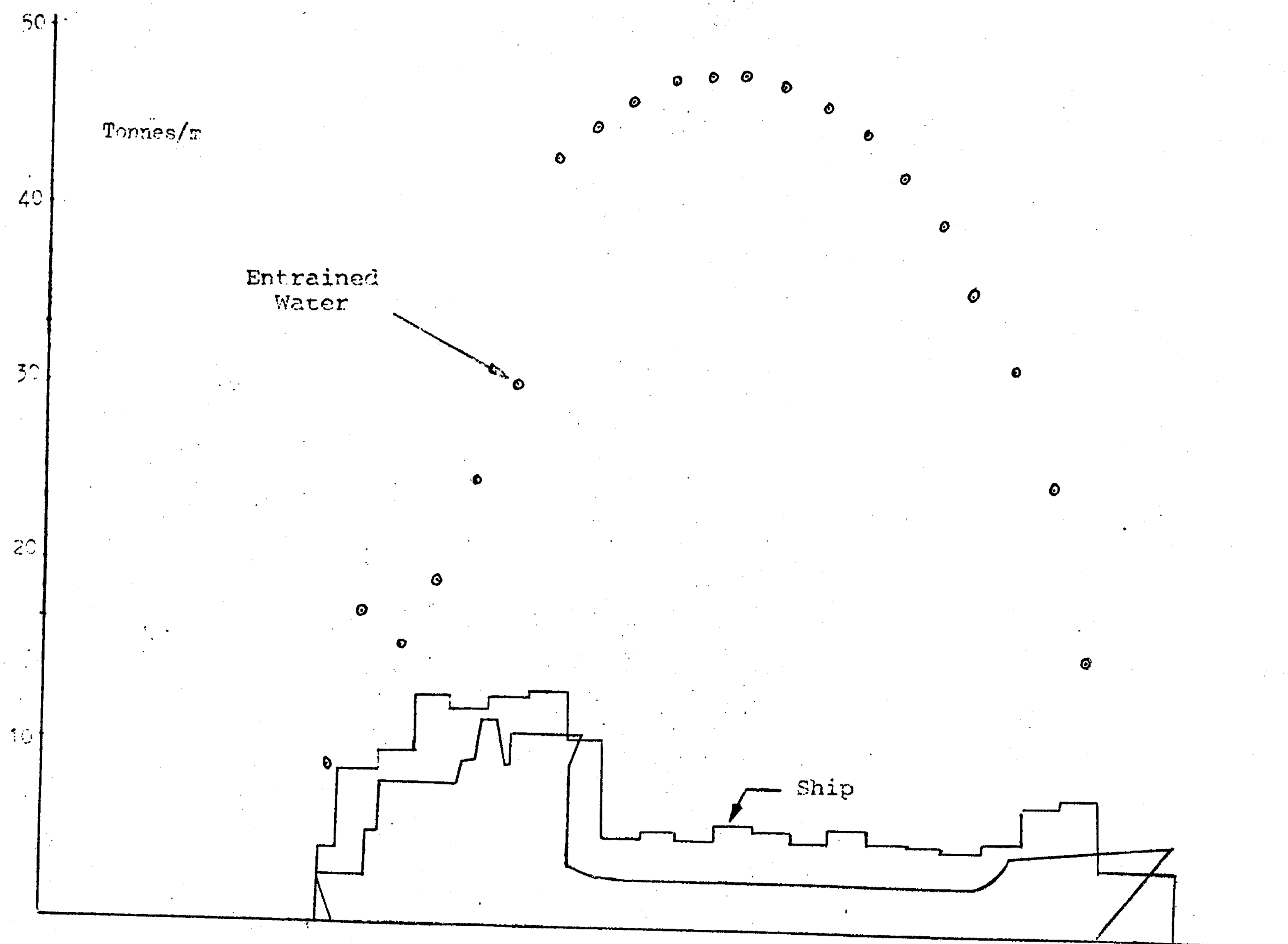
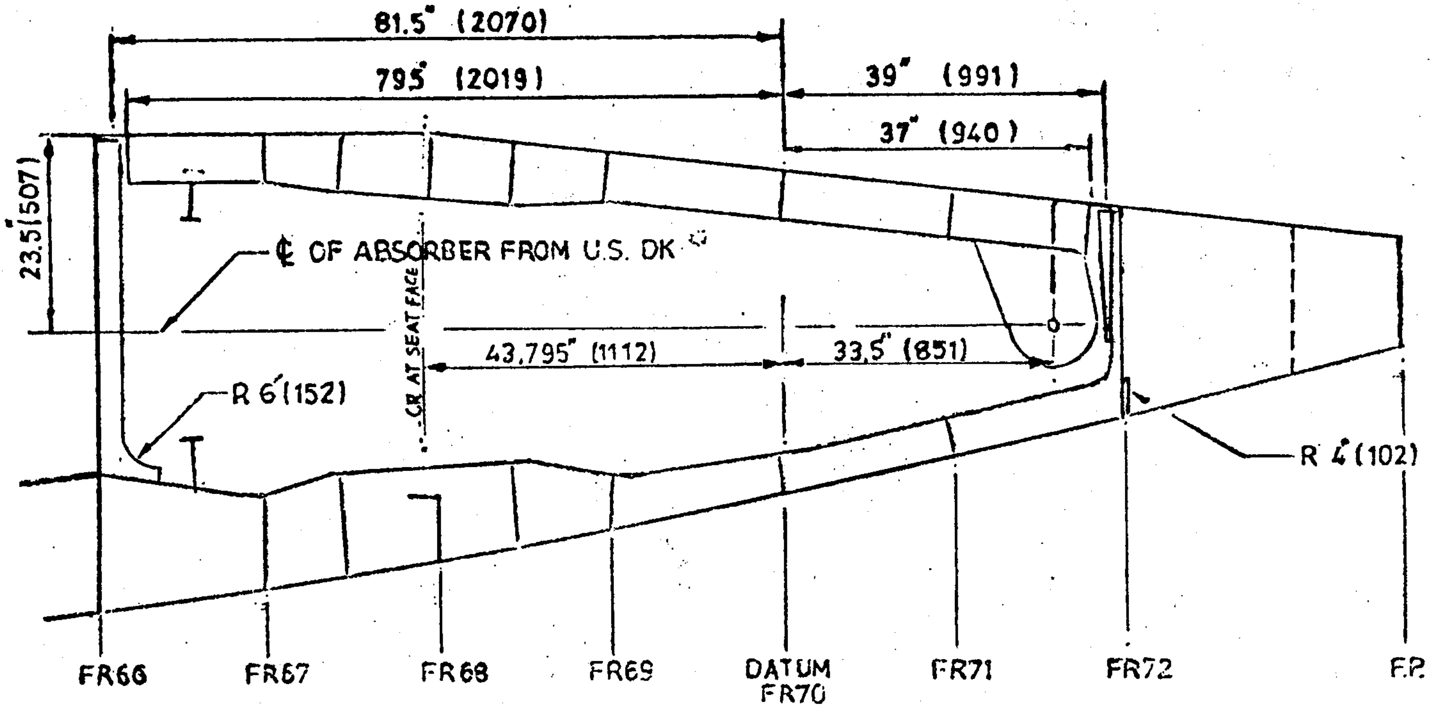


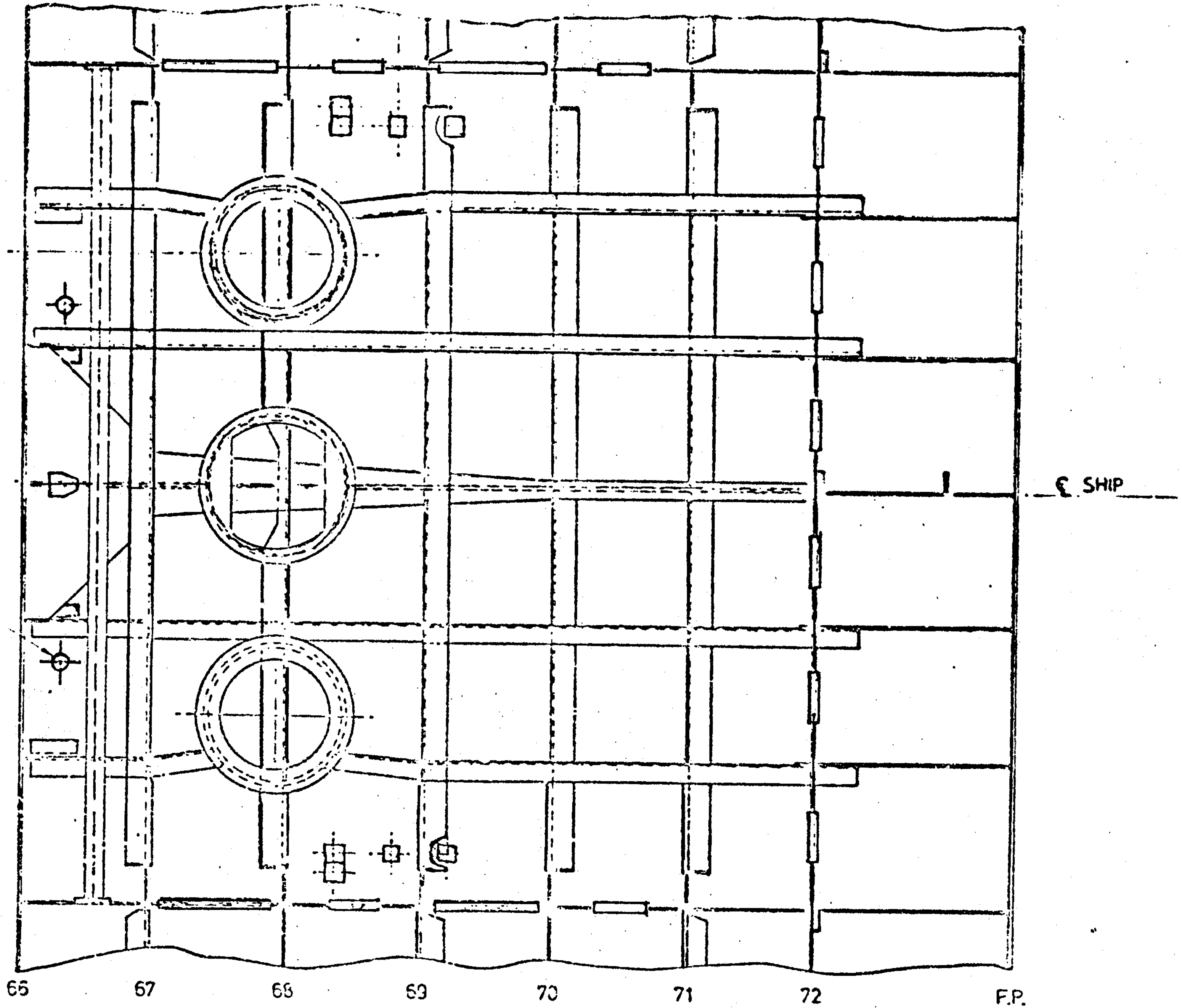
FIGURE 2

MASS DISTRIBUTION OF SHIP AND ENTRAINED WATER



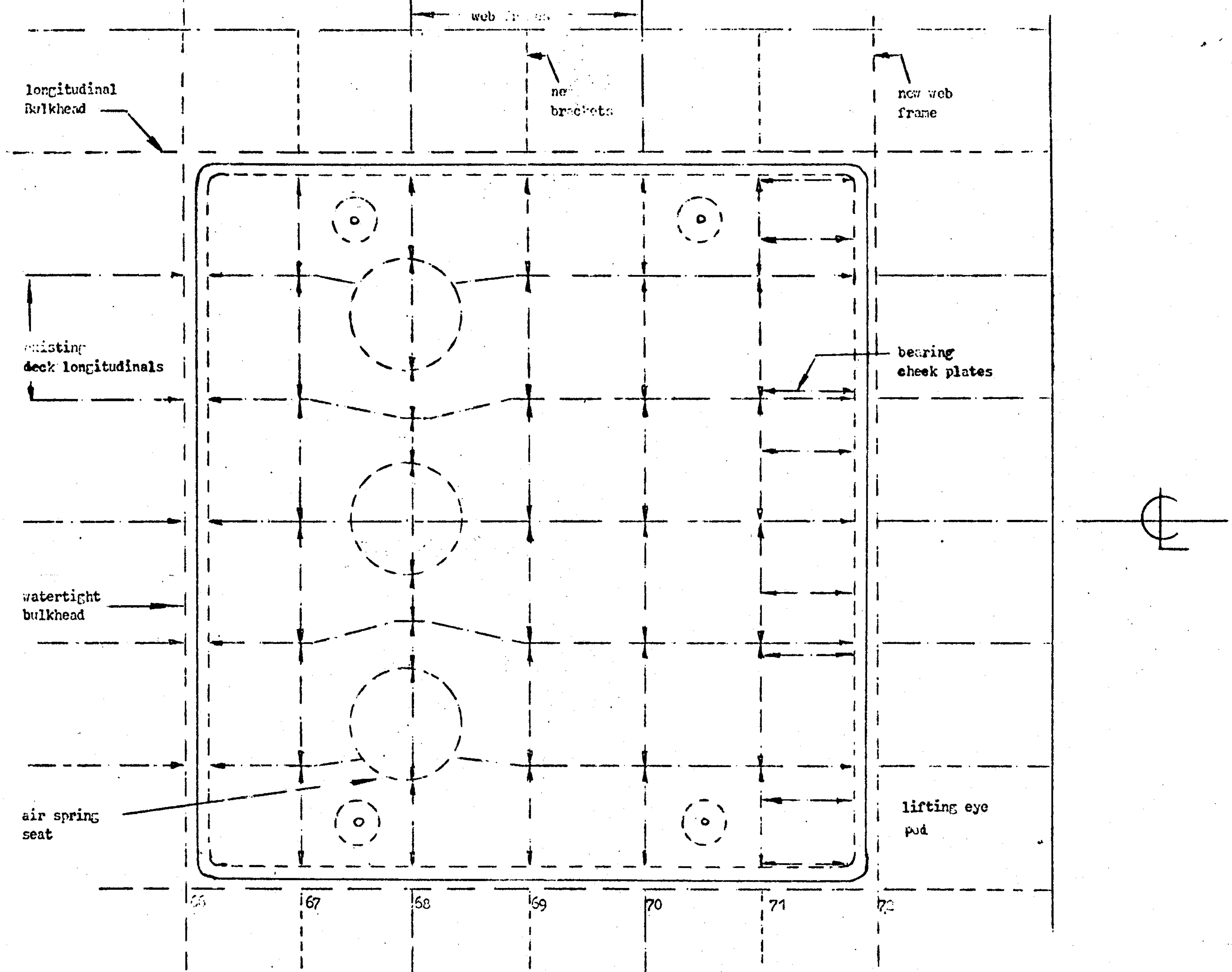
ELEVATION THRU CENTRELINE OF SHIP

FIG. 3A



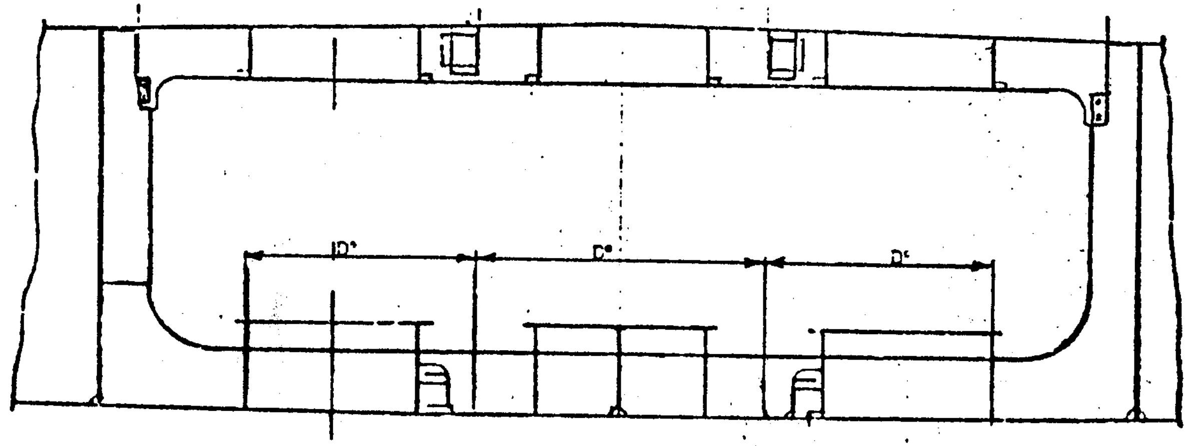
PLAN OF HULL STRUCTURE

FIG. 3B

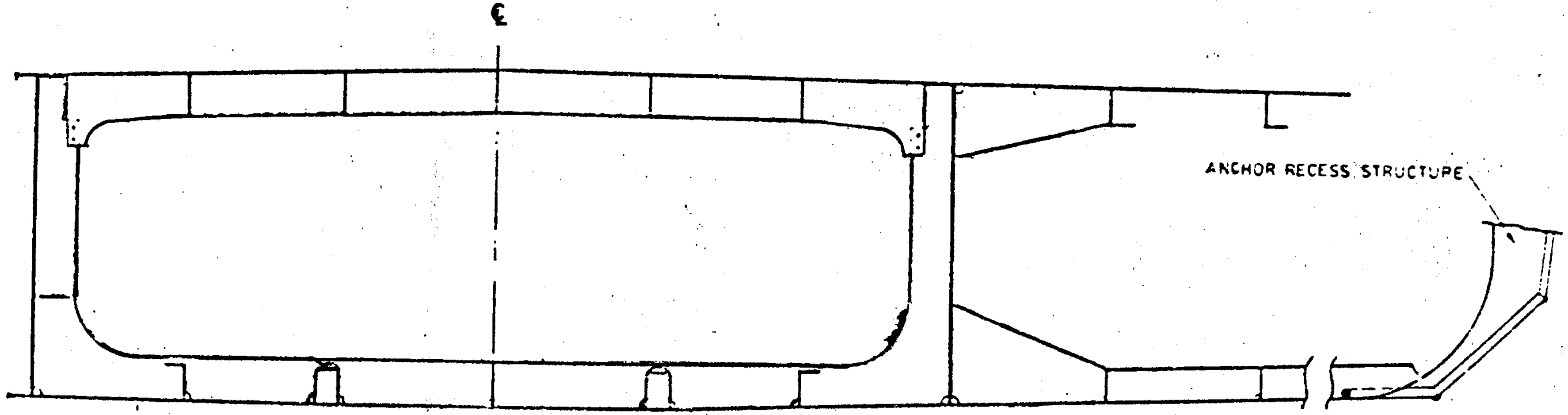


PLAN VIEW OF PORTABLE PLATE STIFFENING

FIG. 3C

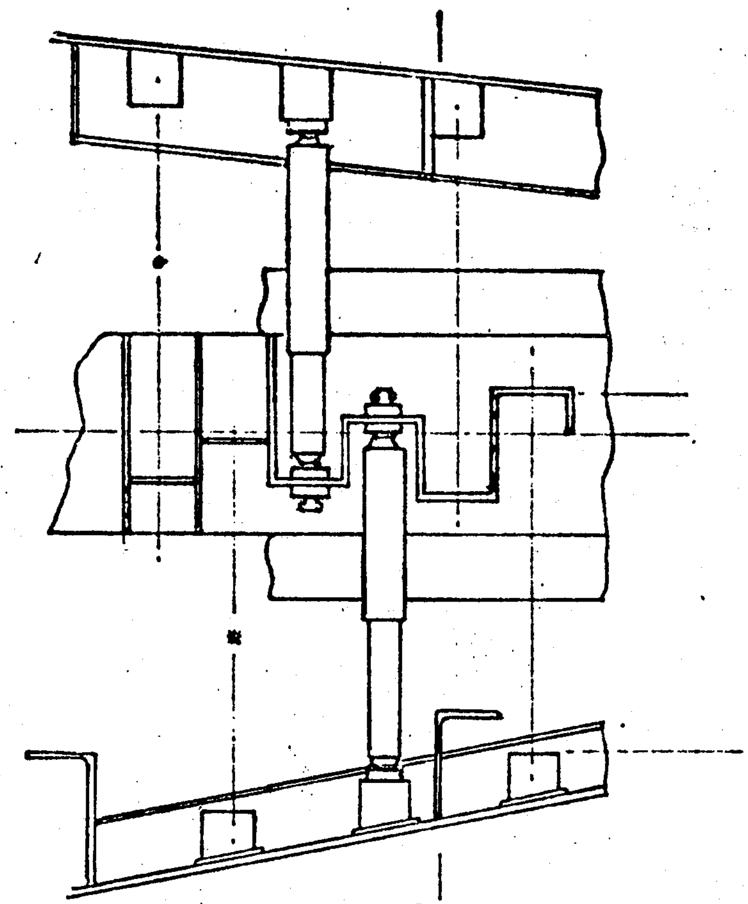


FRAME 68
LOOKING FWD.



FRAME 69
LOOKING FWD.

SECTION AT FR. 68 & 69
FIG 3D



SHOCK ABSORBER ASSEMBLY
VIEW LOOKING TO PORT

FIG. 3E

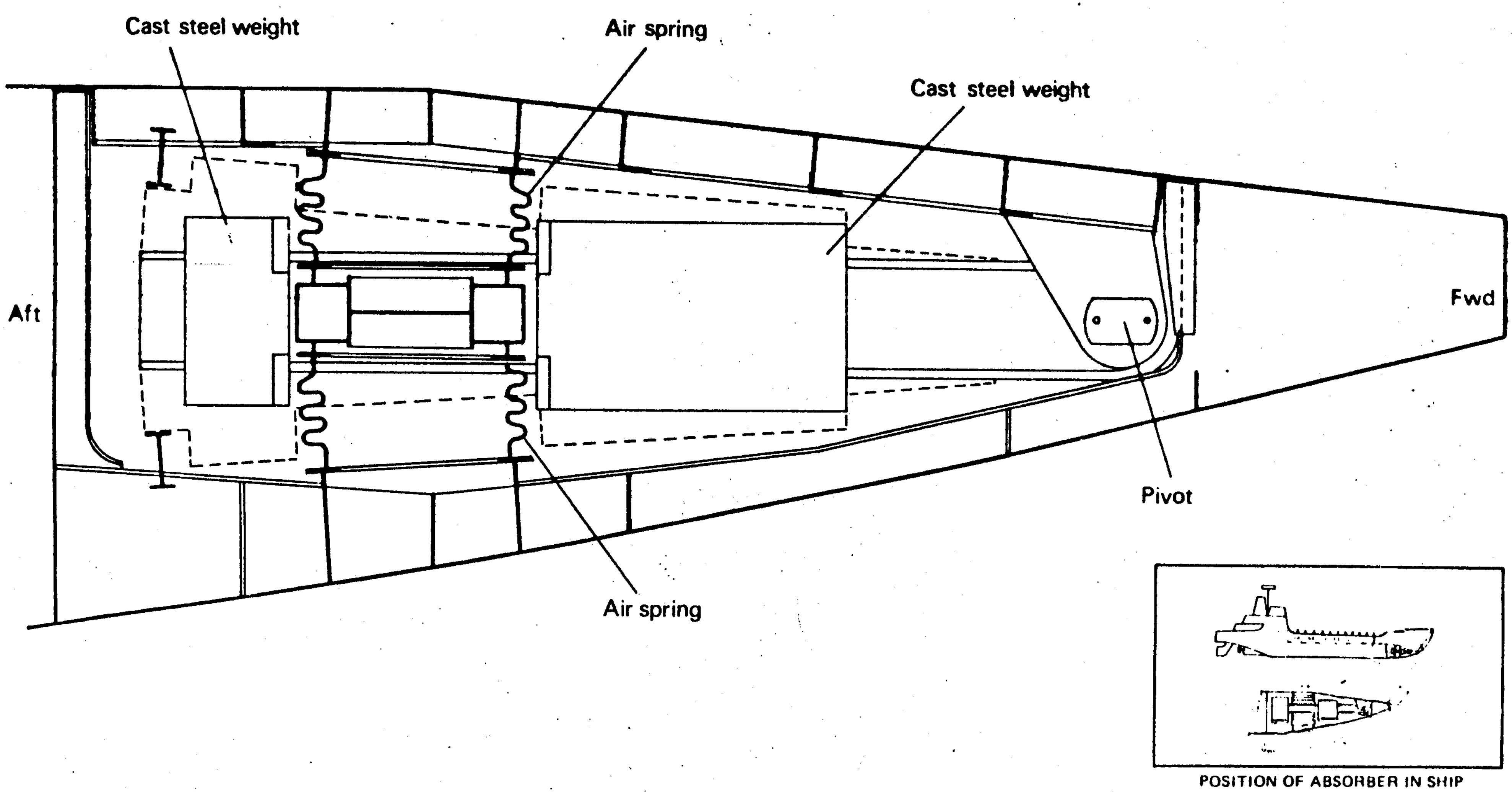


FIG. 4A ELEVATION THROUGH SHIP CENTRELINE, SHOWING ABSORBER LOCATED IN BOW

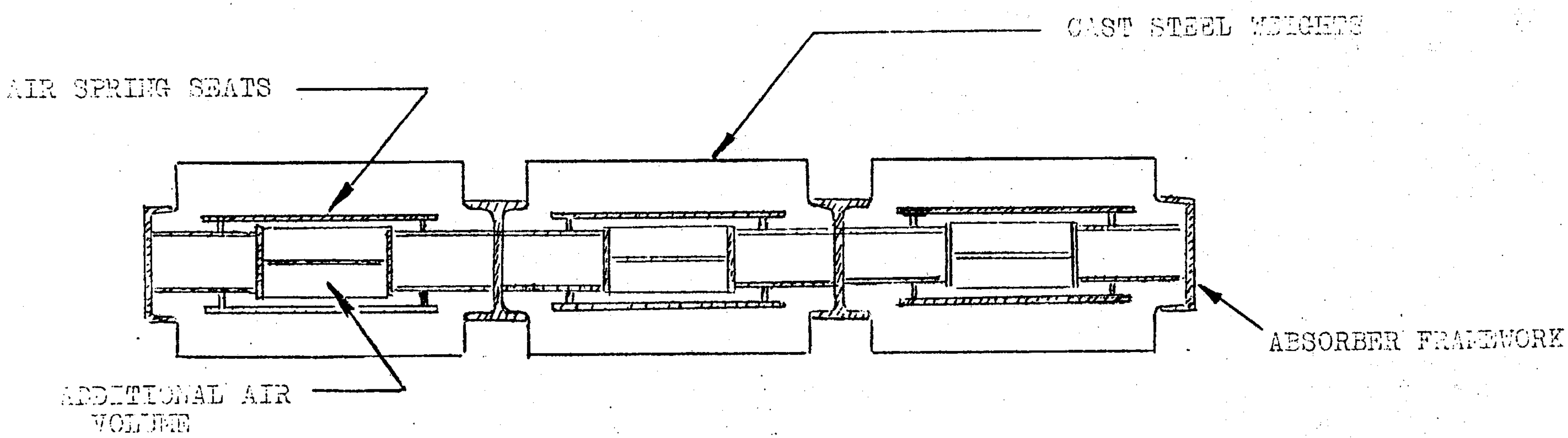


FIG. 4c

CROSS SECTION THROUGH ABSORBER IN WAY OF AIR SPRINGS

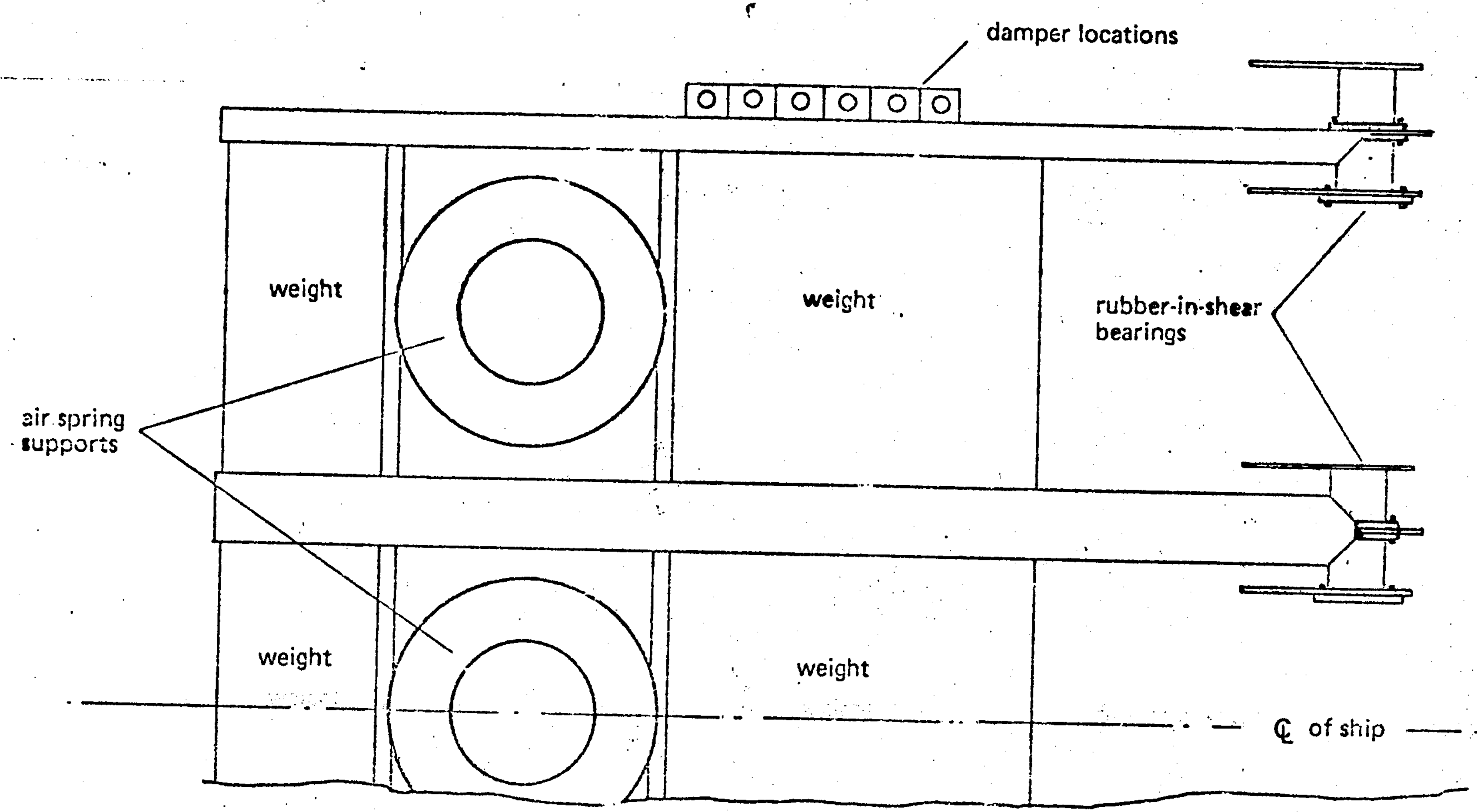
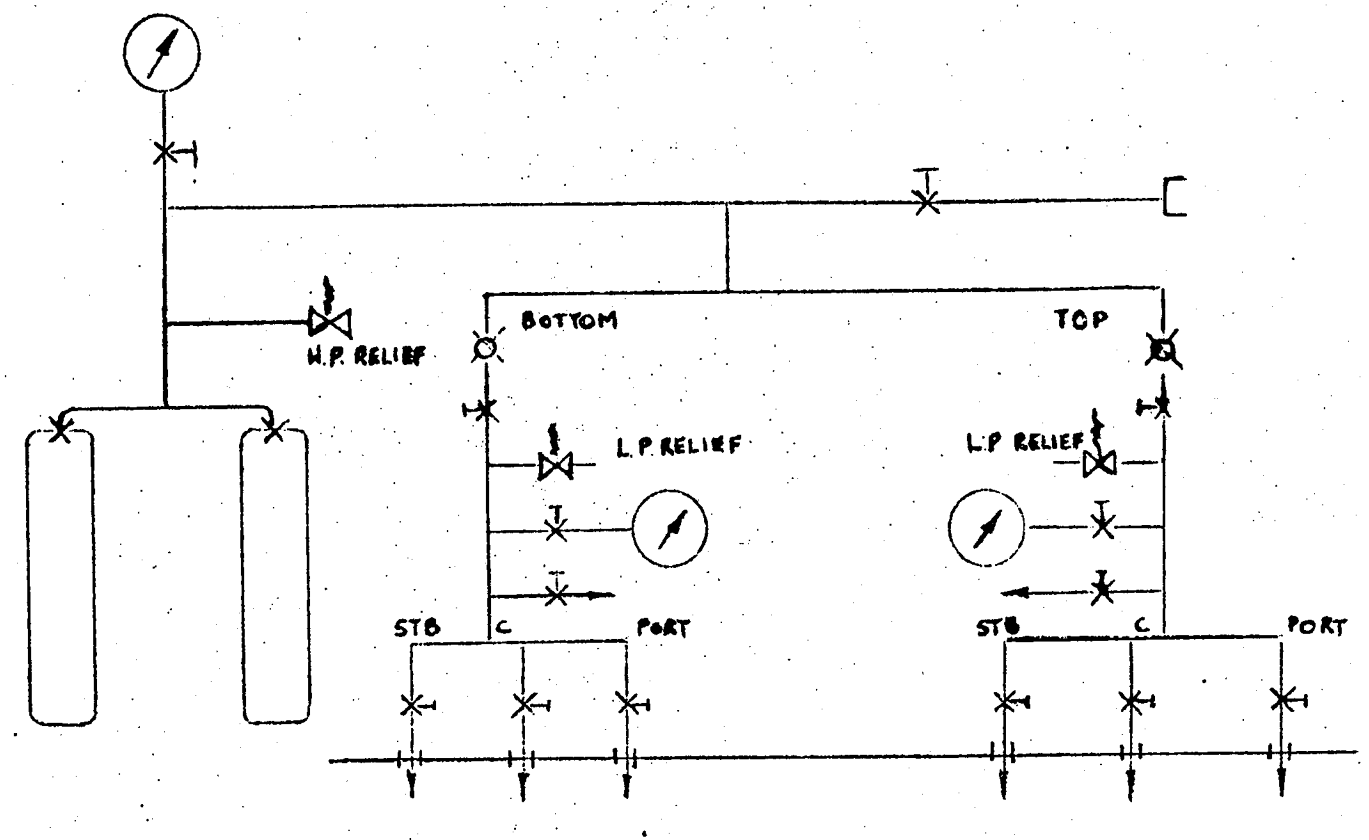


Fig.4B PLAN OF ABSORBER



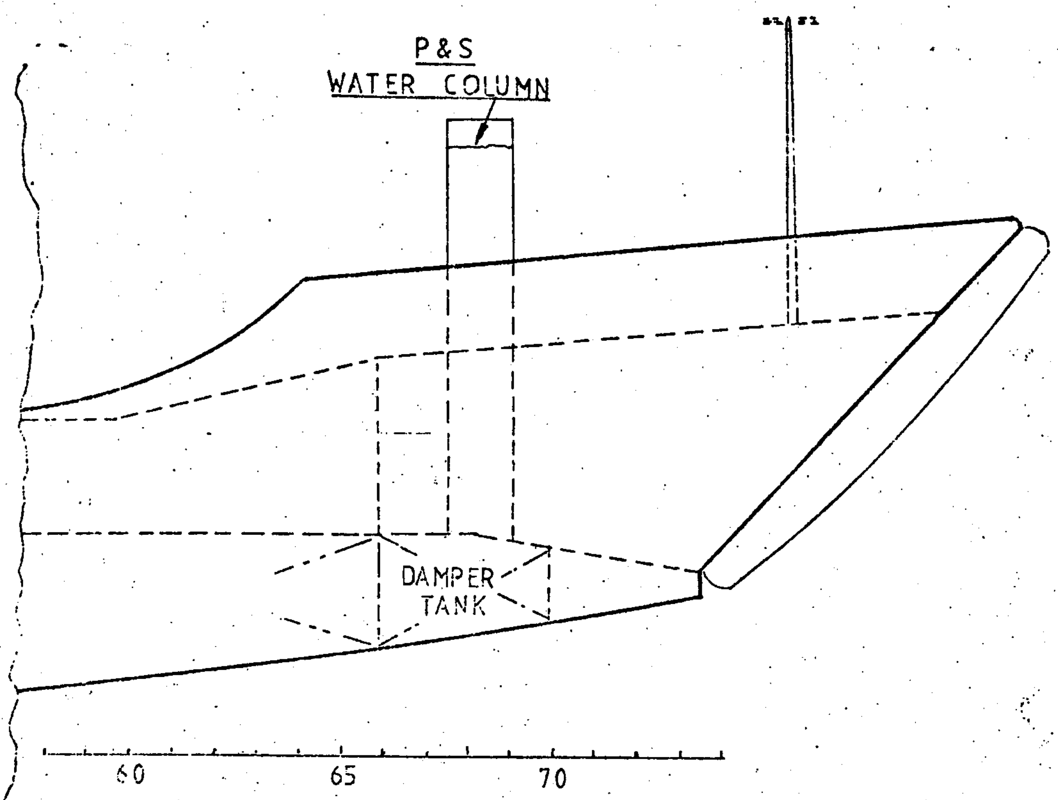
REGULATOR - should be set to 100 psig.

PRESSURE RELIEF VALVES — HP SET TO 2500 psig. LP SET TO 200 psig.

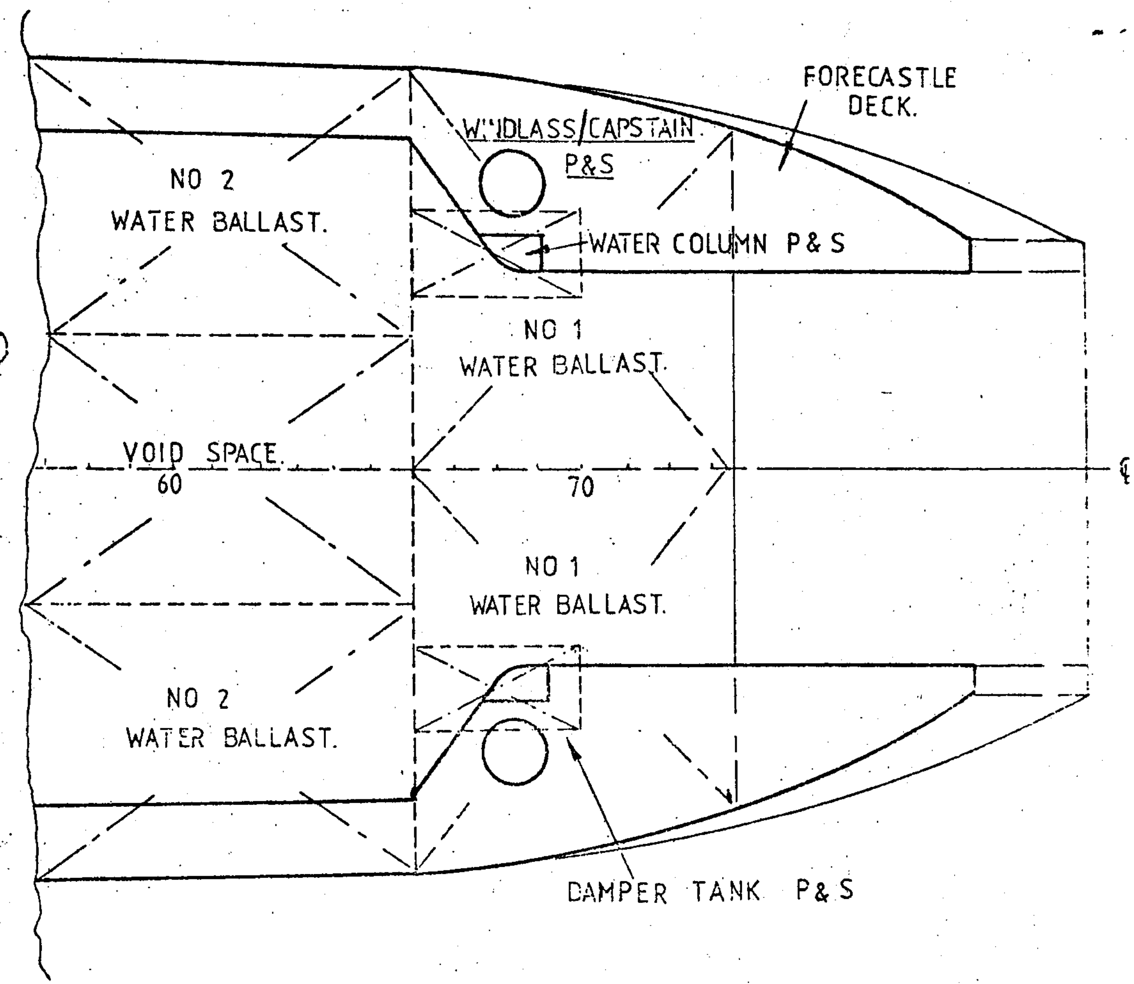
PRESSURE GAUGES

FIG. 5

DIAGRAMMATIC ARRANGEMENT OF HP/LP AIR SYSTEM



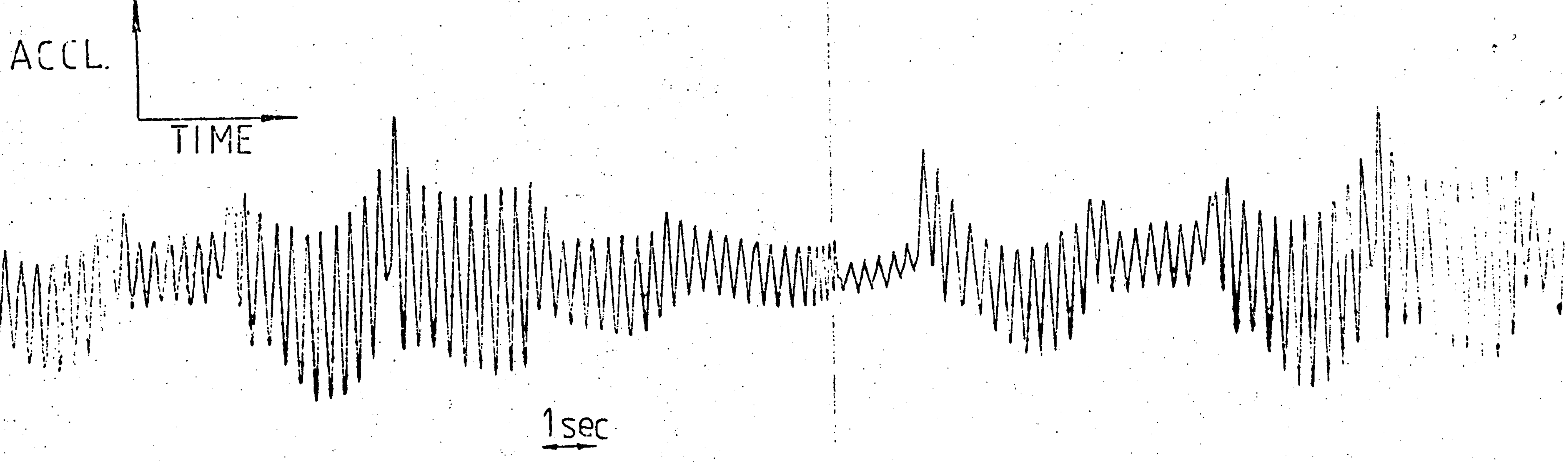
SIDE ELEVATION.



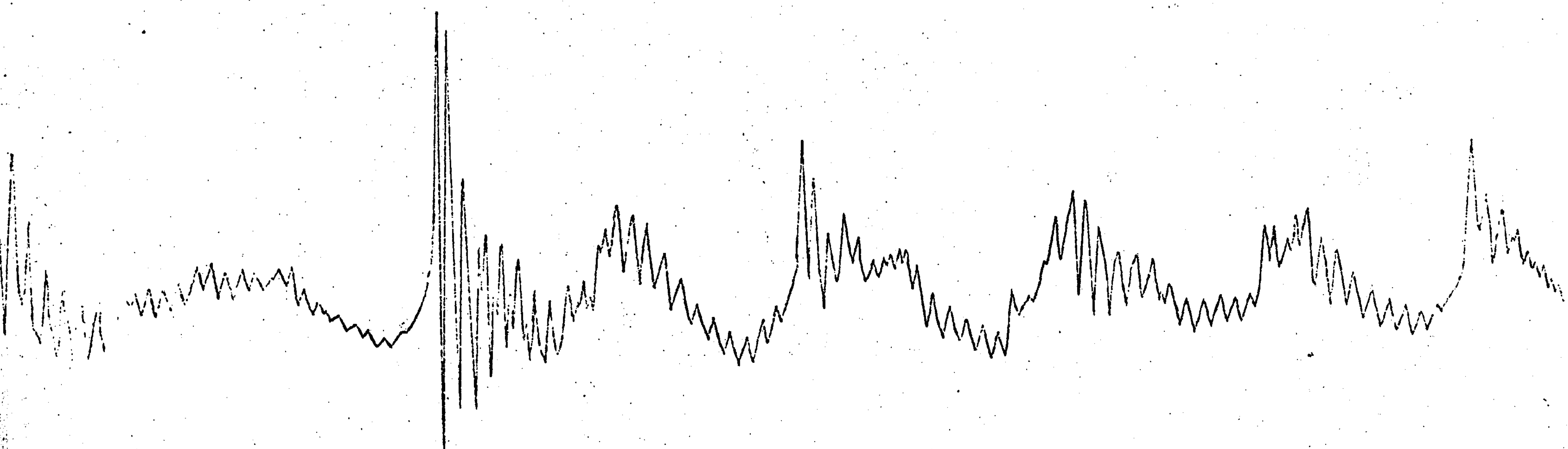
PLAN VIEW

FIG 6.

PROPOSED TUNED TANK ABSORBER.

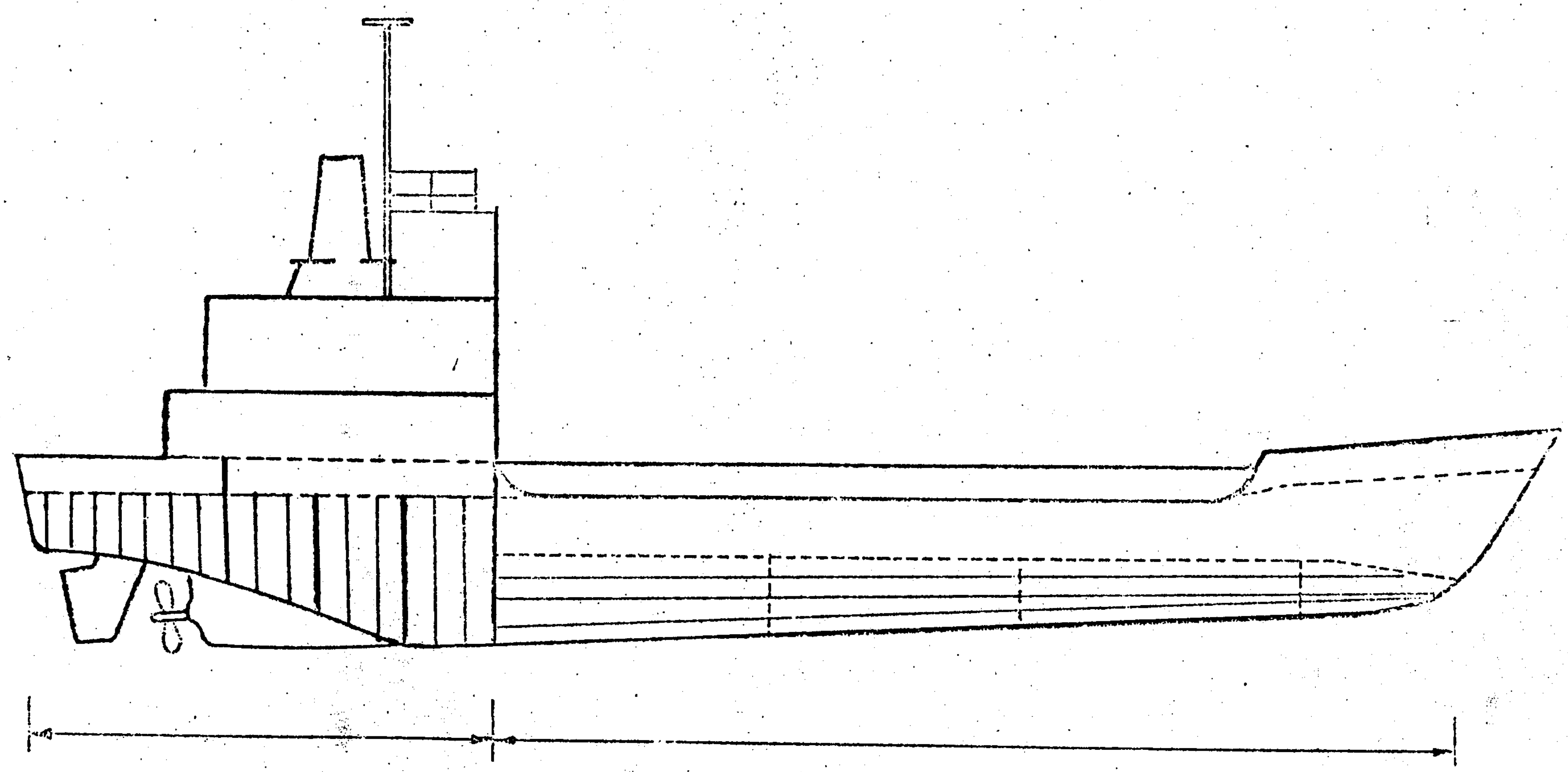


TYPICAL VIBRATION RECORDS FROM SEA TRIALS - ABSORBER LOCKED



ABSORBER FREE

FIG. 8



main transverses
200 x 10 mm with 50x8 mm flat bar
at 2667 mm spacing.

minor transverses
90 x 8 flat bar at 533 mm spacing.

bottom longitudinals
127 x 76 x 10 mm angle at 550 mm spacing.

side longitudinals
100 x 10 flat bar at 525 mm spacing.

FIG. 7.

ARRANGEMENT OF MAIN STRUCTURE.

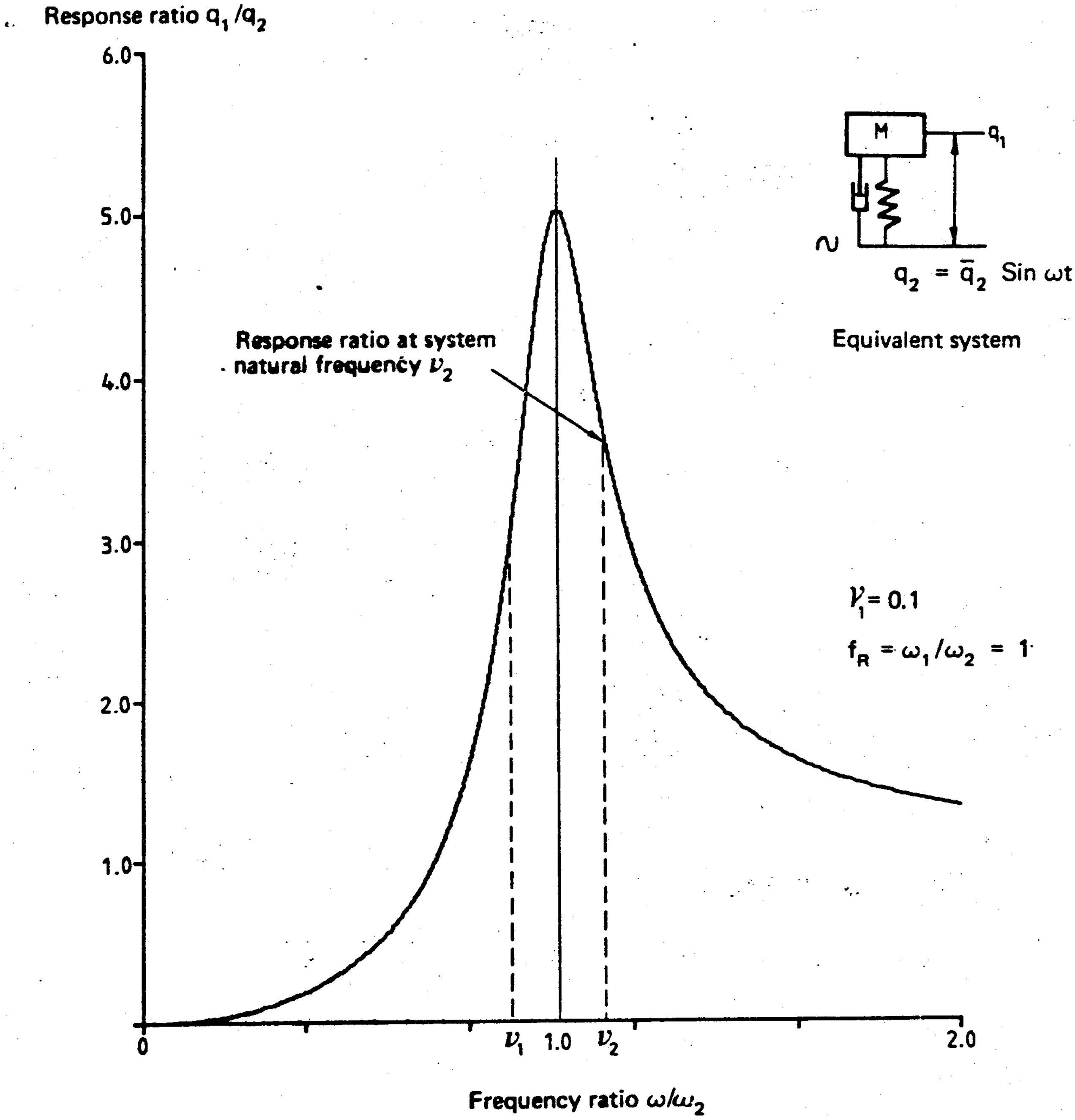


FIG. 9 EFFECT OF EXCITATION FREQUENCY ON AMPLITUDE RATIO OF MASSES

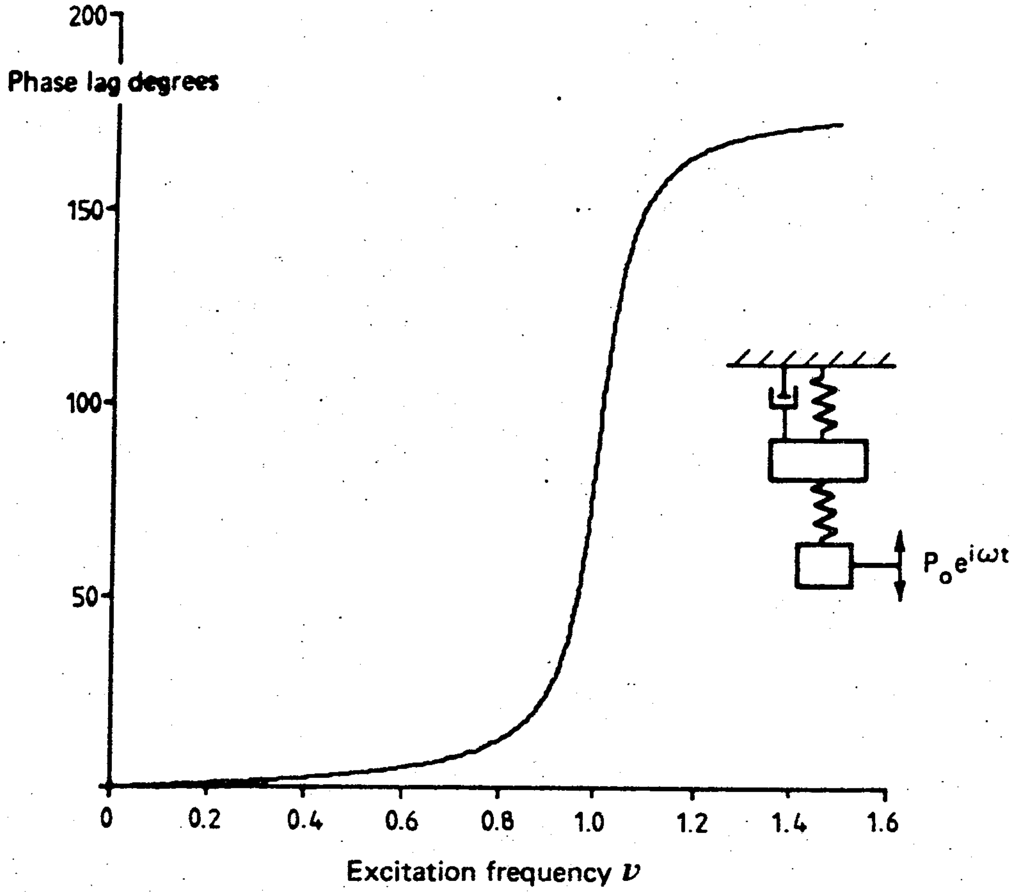


FIG.10 A PHASE DIFFERENCE BETWEEN ABSORBER AND SHIP WHEN ABSORBER IS SUBJECT TO A HARMONIC FORCE AND ABSORBER DAMPING IS SMALL

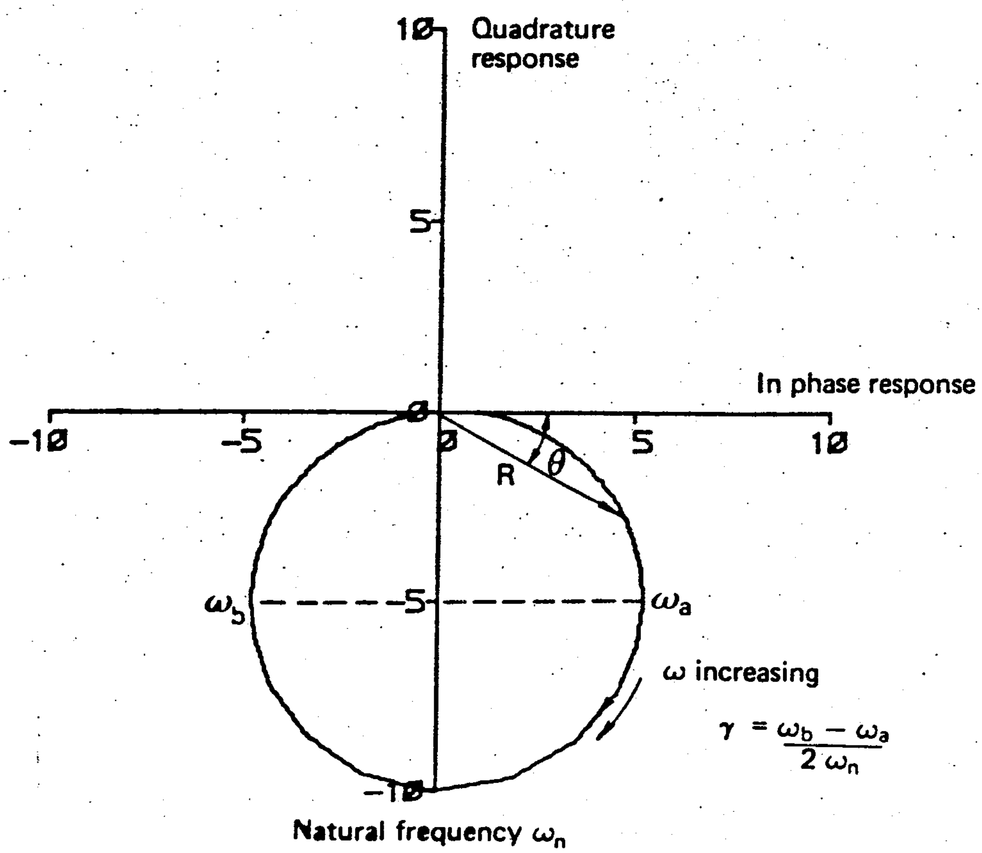


FIG.10B VECTOR PLOT OF SHIP RESPONSE

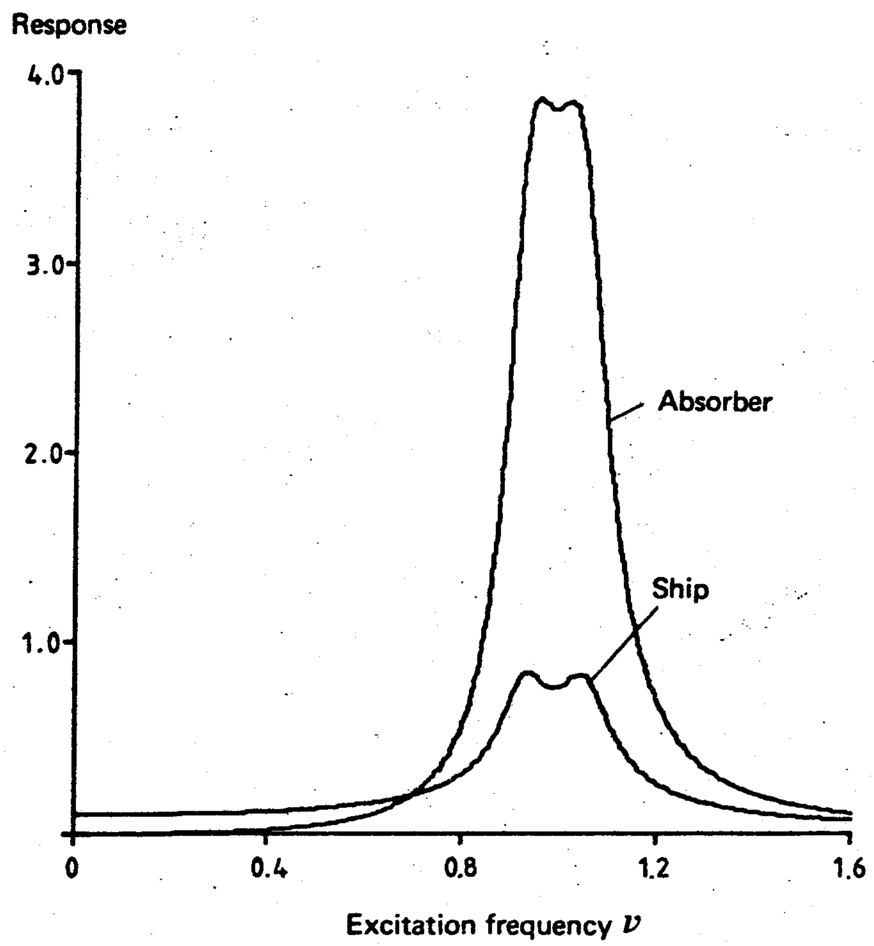


FIG. 11 EFFECT OF EXCITATION FREQUENCY ON RESPONSE OF SHIP AND ABSORBER

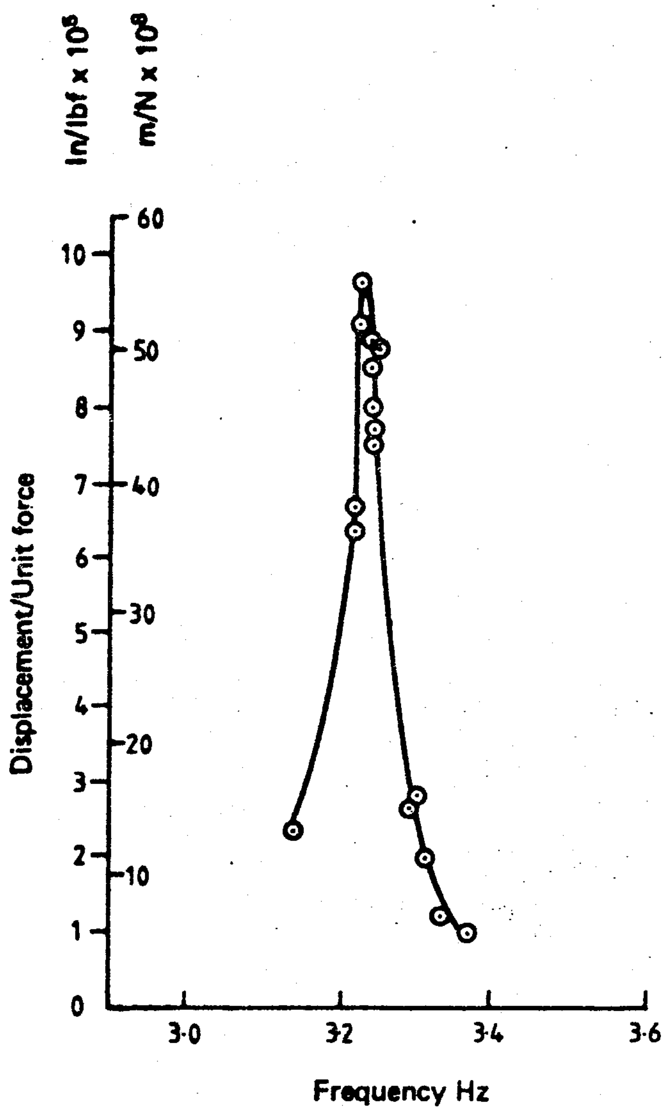


FIG. 12 VARIATION OF SHIP RESPONSE WITH FREQUENCY

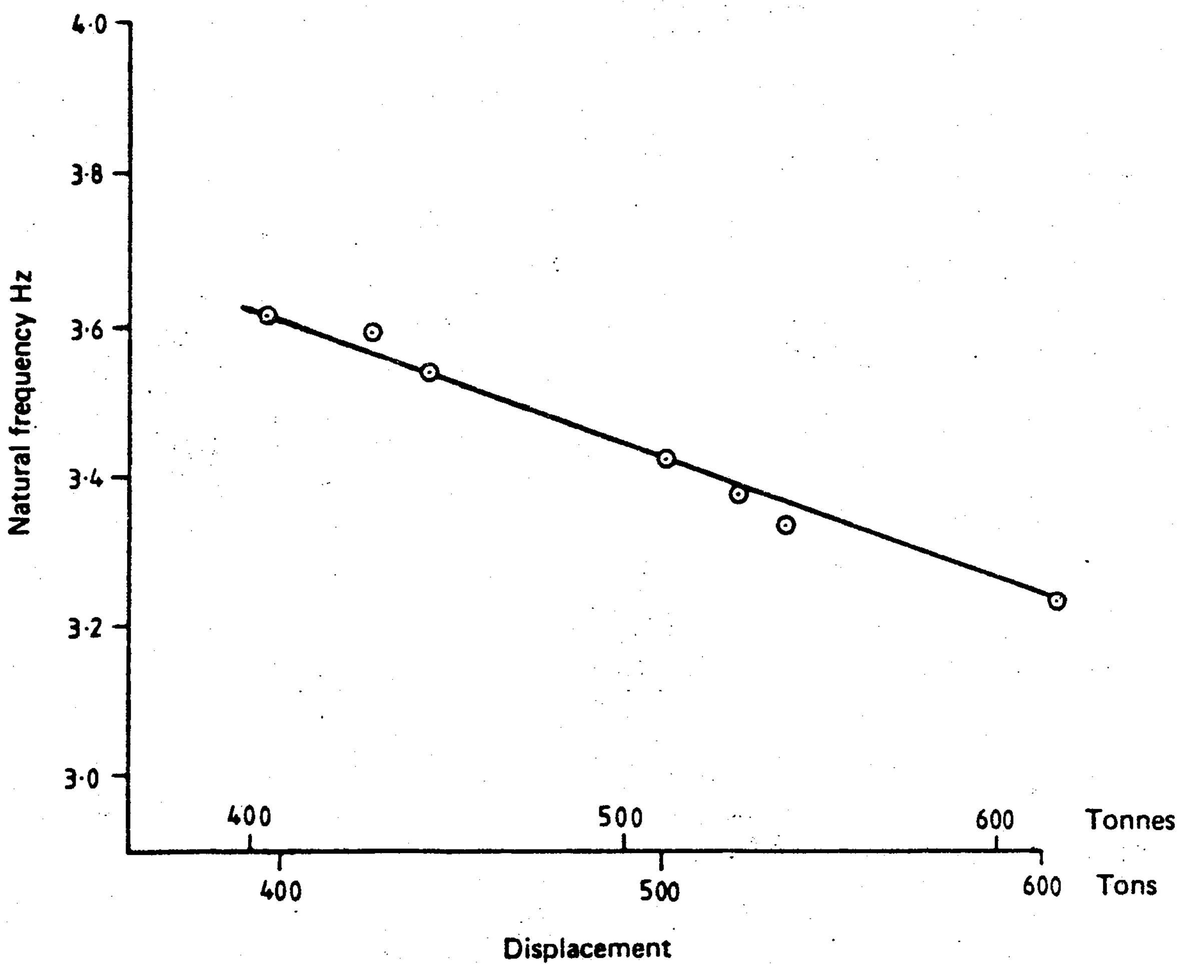


FIG. 13 VARIATION OF SHIP NATURAL FREQUENCY WITH DISPLACEMENT

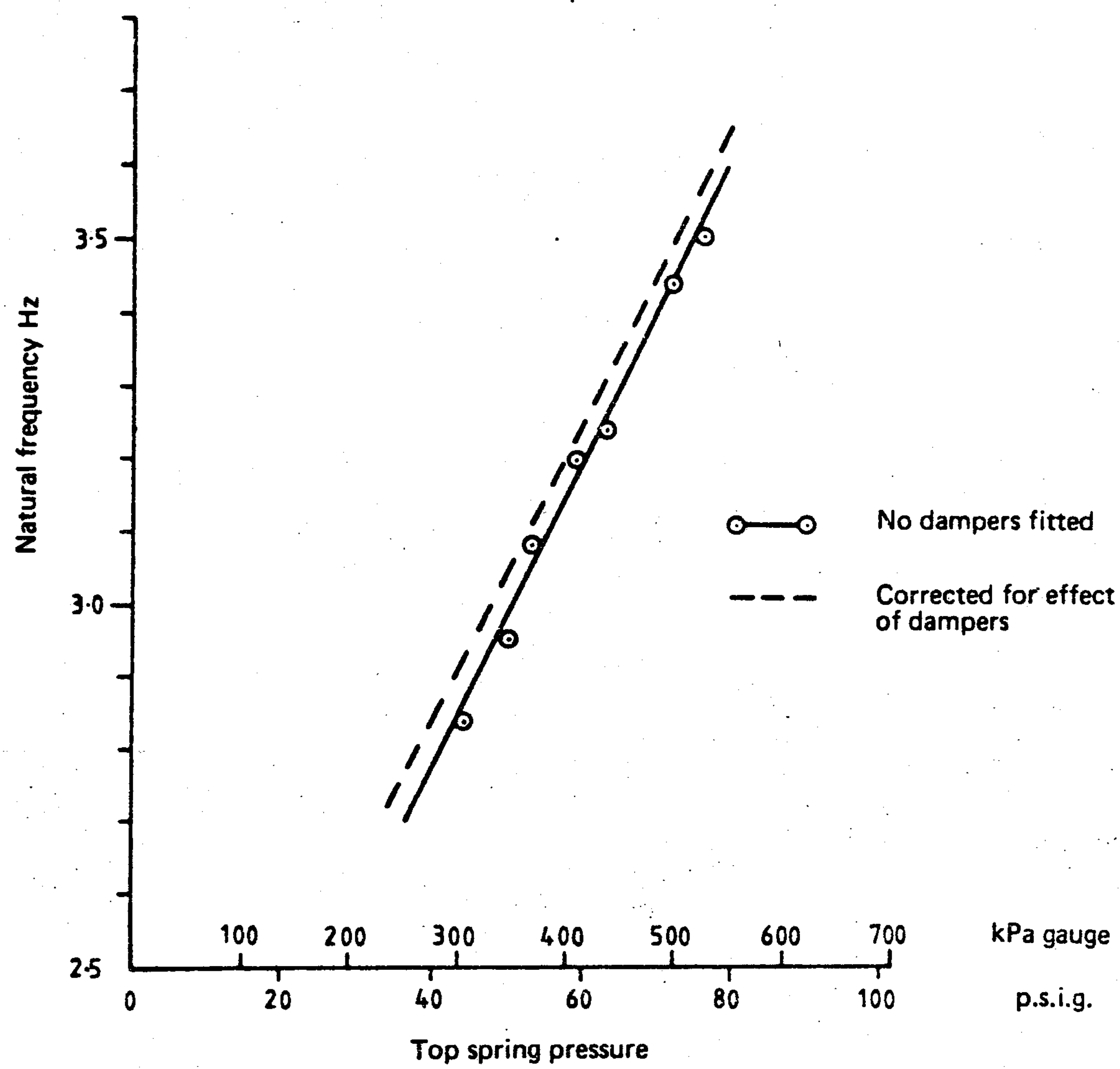


FIG. 14 VARIATION OF ABSORBER NATURAL FREQUENCY WITH PRESSURE

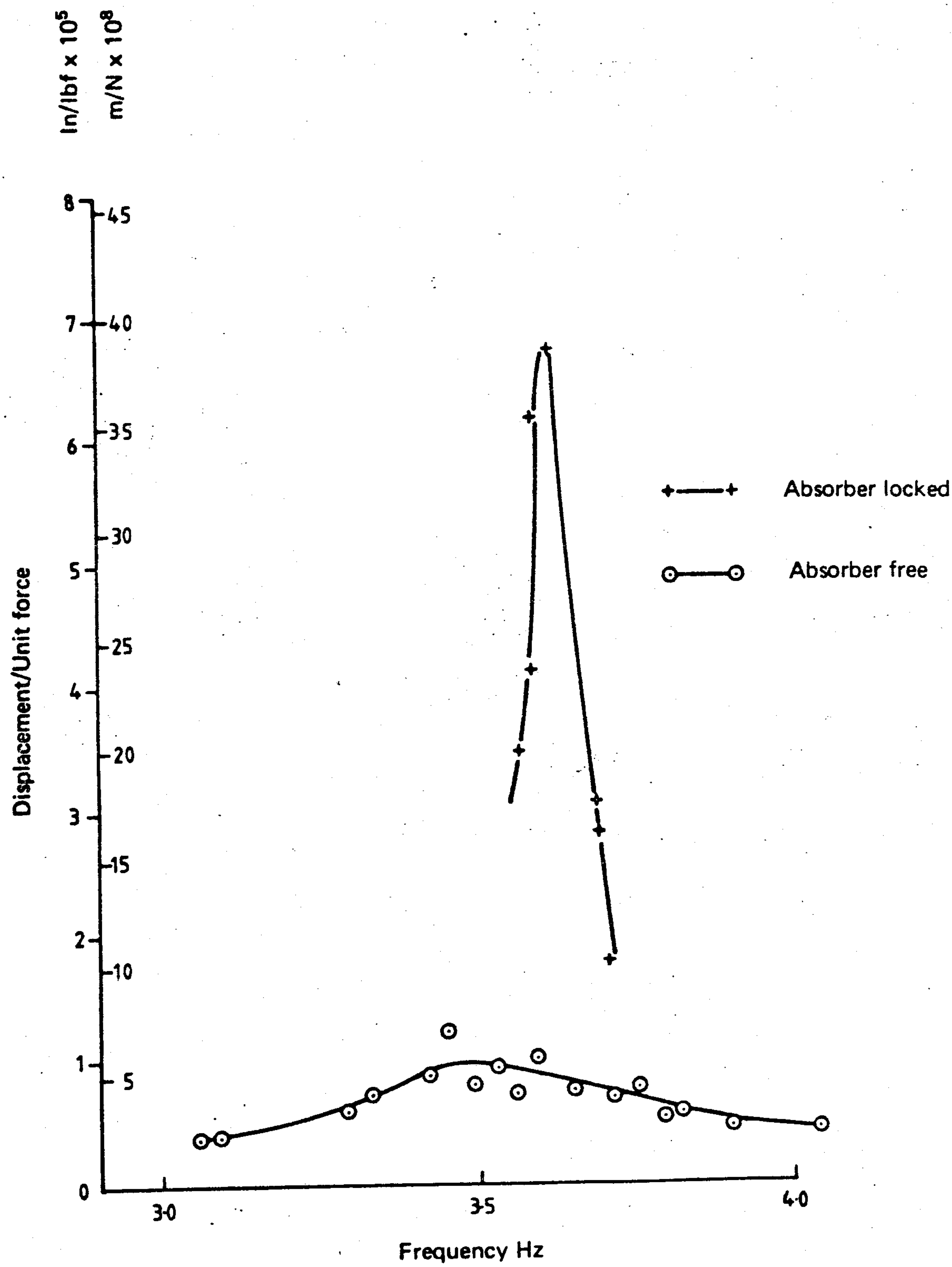


FIG. 15 EFFECT OF ABSORBER ON SHIP RESPONSE

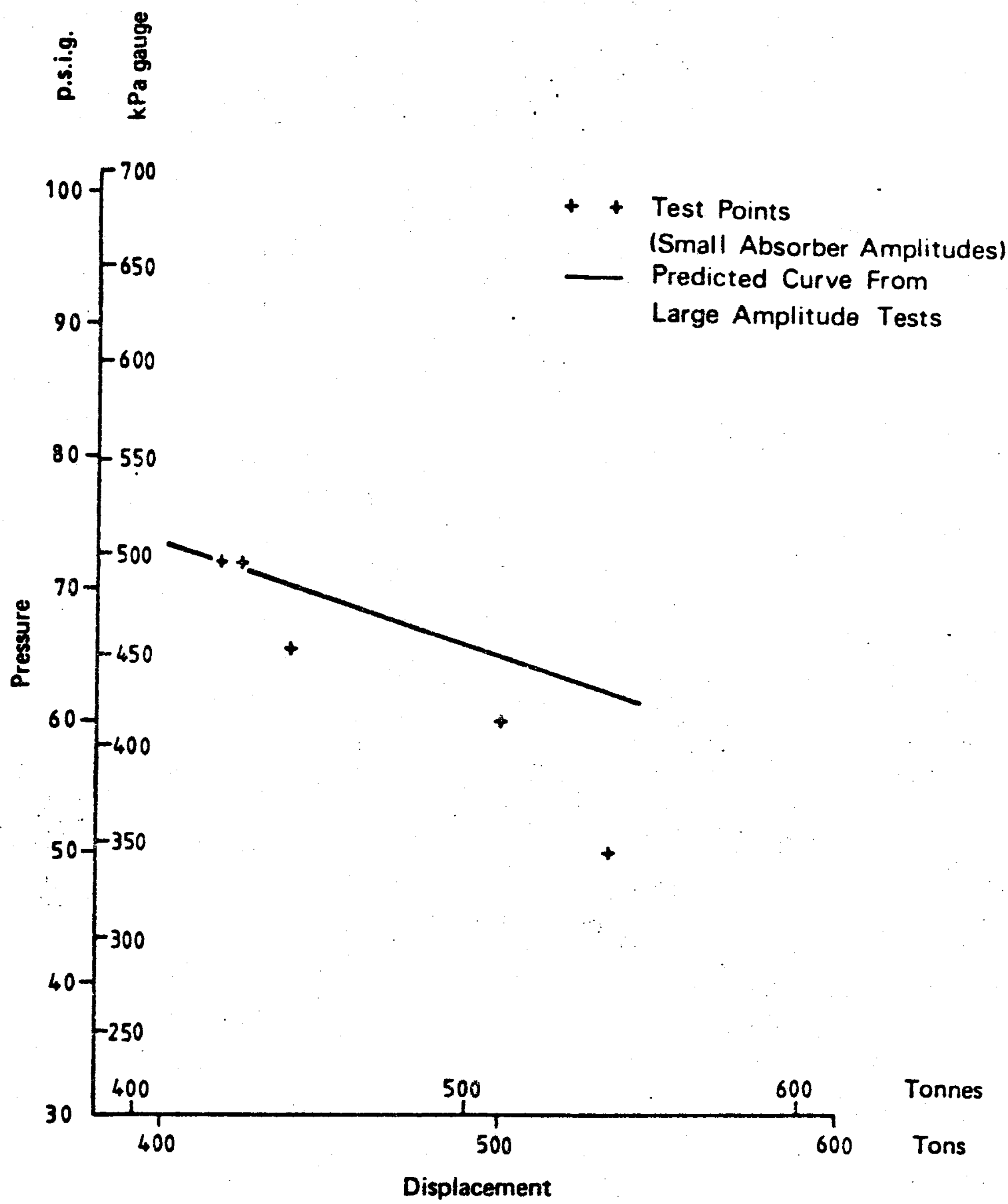


FIG. 16 RECOMMENDED AIR SPRING PRESSURE VERSUS SHIP DISPLACEMENT

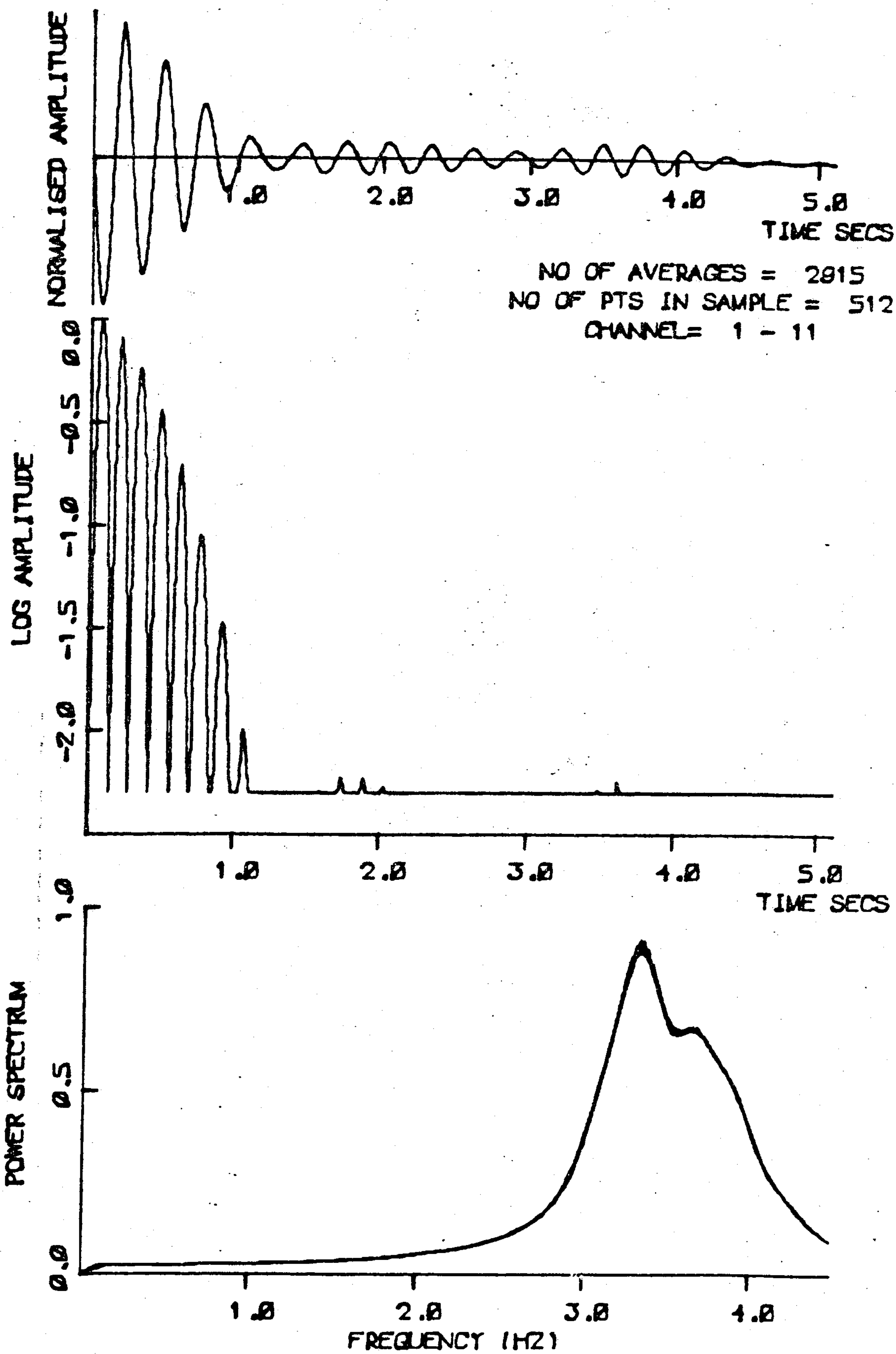


FIG. 17 RANDOM DECREMENT ANALYSIS (absorber correctly tuned)

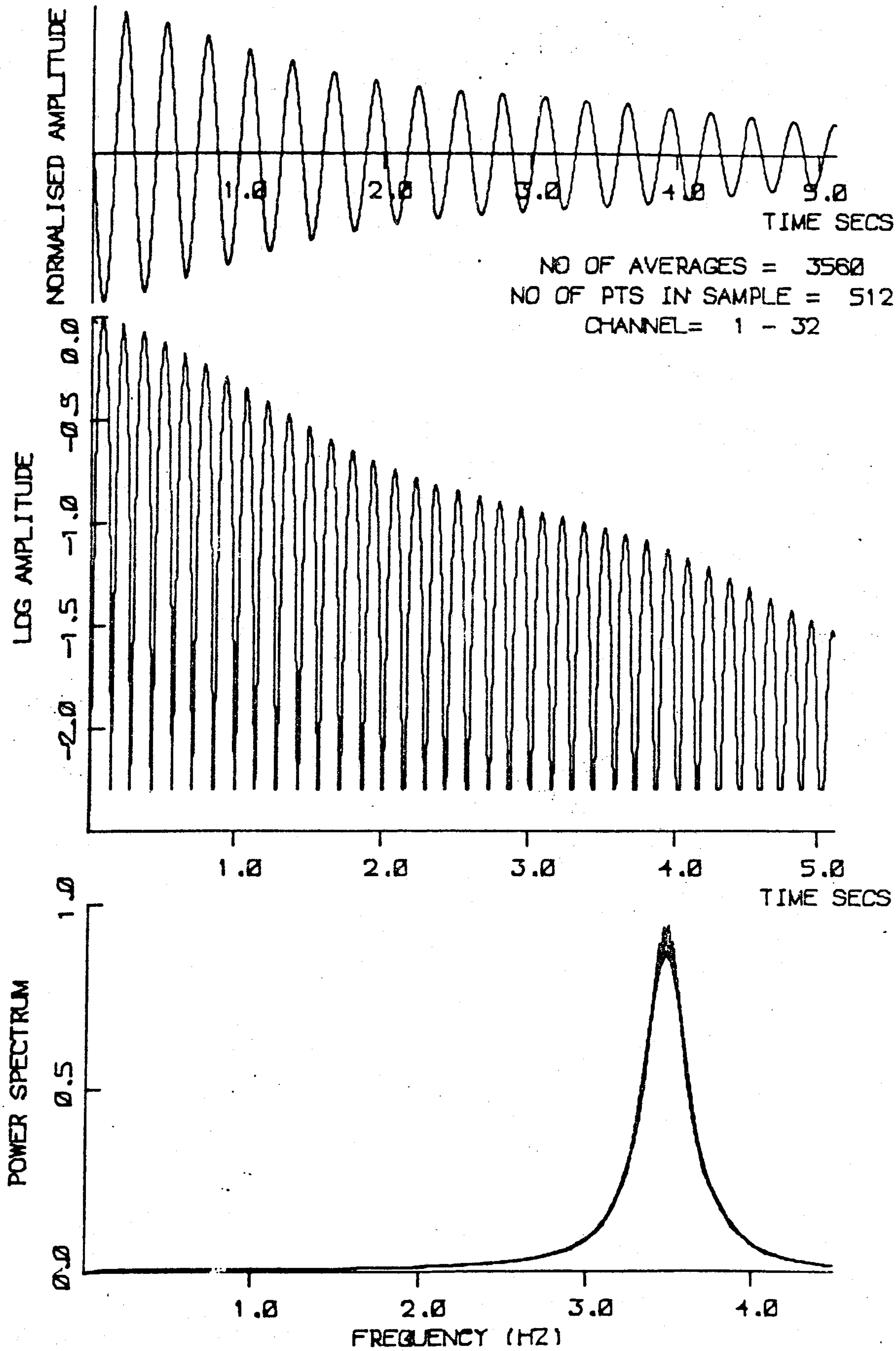


FIG. 18 RANDOM DECREMENT ANALYSIS (absorber locked)

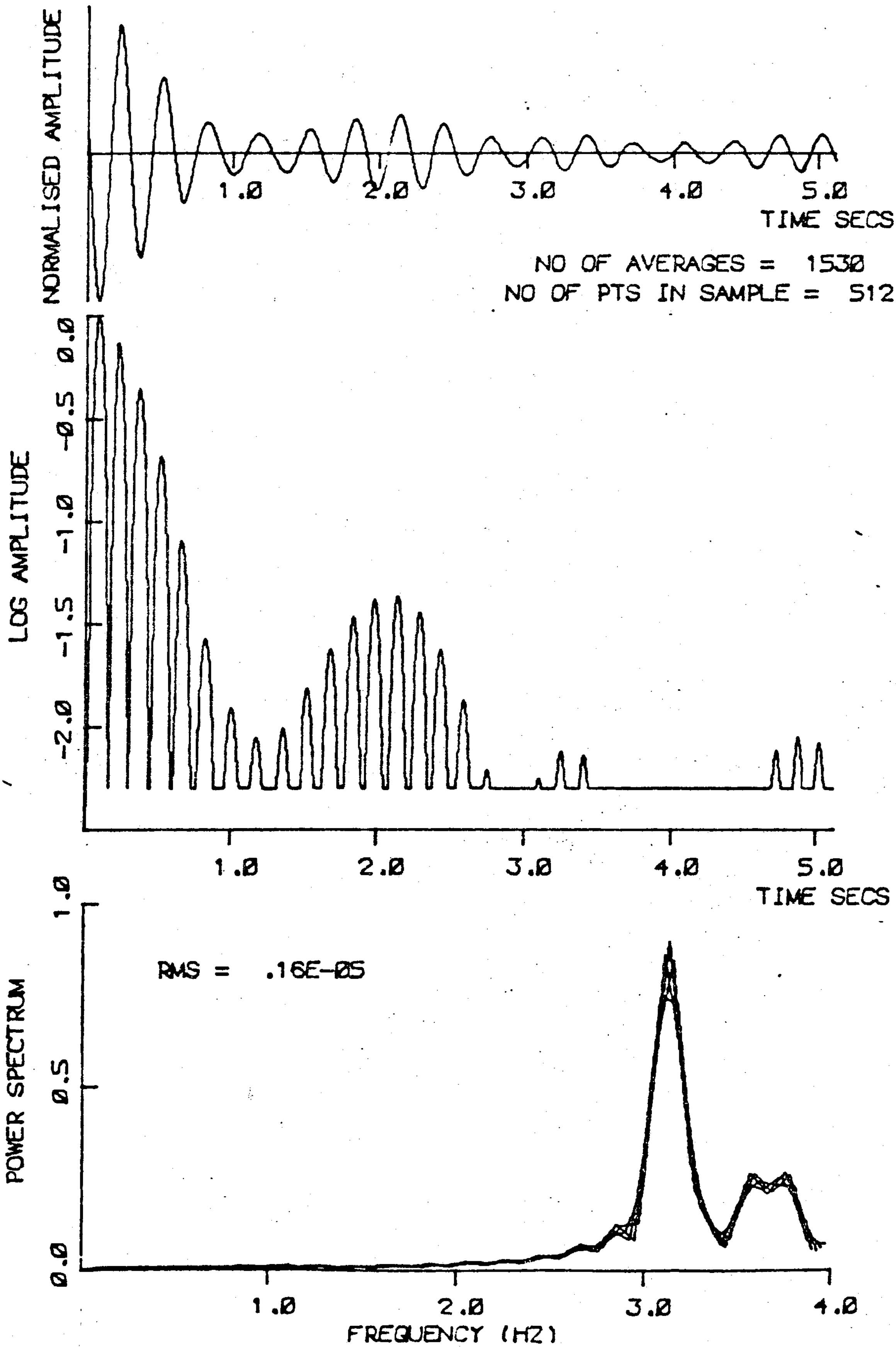
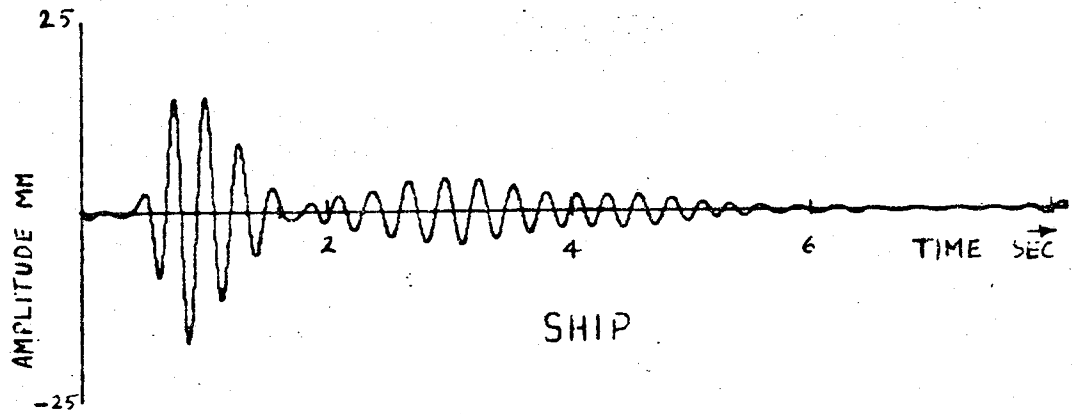


FIG. 19 RANDOM DECREMENT ANALYSIS (SEA STATE 4)



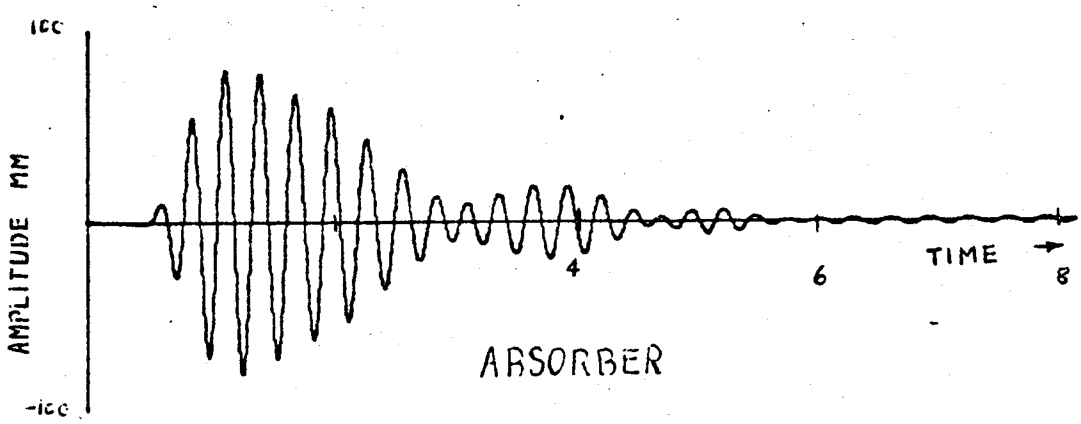


FIG.20 MAXIMUM RECORDED AMPLITUDES
OF SHIP AND ABSORBER

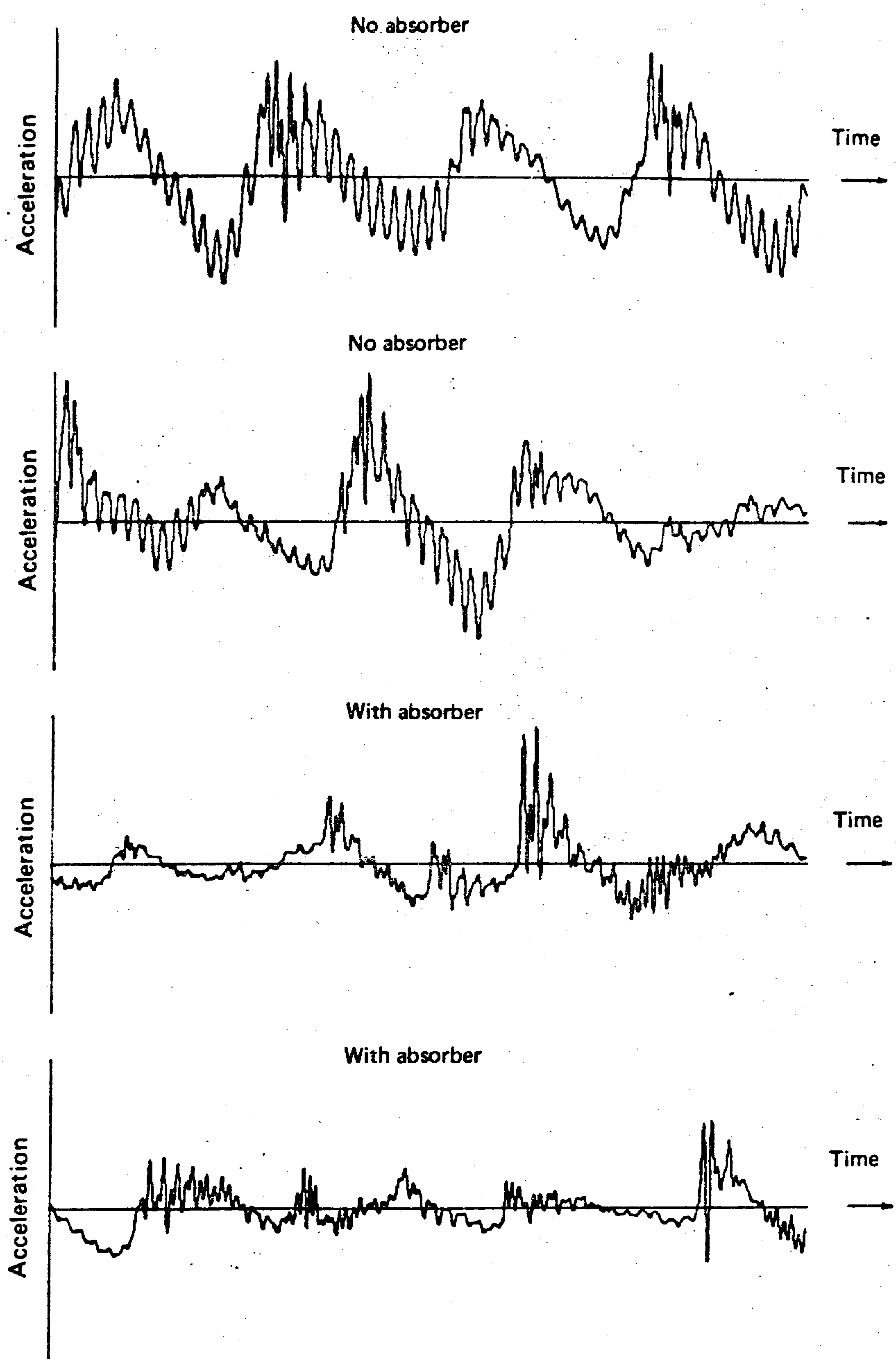


FIG. 21 COMPARISON OF SHIP RESPONSE WITH AND WITHOUT ABSORBER