Hydraulics Research
Wallingford

QUAYSHIP: A computer model of a ship against a quay in the presence of waves

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Report SR 232
February 1990

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This report describes work funded by the Department of the Environment under Research Contract PECD 7/6/111 for which the DoE nominated officer was Dr R P Thorogood. It is published on behalf of the Department of the Environment but any opinions expressed in this report are not necessarily those of the funding Department. The work was carried out in the Maritime Engineering Department of Hydraulics Research, Wallingford, under the management of Dr S W Huntington.

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ABSTRACT

A computer model QUAYSHIP has been developed to describe the linear response to waves of a vessel against a quay. This model complements UNDERKEEL which was developed to describe responses of vessels to waves in the open sea. Both these models are essential building blocks in the development of a computer model of a moored ship in waves which, in turn, is needed to satisfy the requirement for a realistic first estimate of ship responses in feasibility studies of proposed harbour developments.

QUAYSHIP has been applied to the case of a large tanker against a solid vertically faced quay. Hydrodynamic coefficients in sway and heave have been shown to correlate well with results obtained using a 3D source method. The advantage of QUAYSHIP, here, is that much less computation is needed and difficulties representing small clearances with the 3D source method are avoided.

Hydrodynamic coefficients for all the degrees of freedom of vessel movement have been found to be affected by a "manometer" resonance in which flow moves vertically, in the clearance between the vessel and the quay, and horizontally under the keel. Surprisingly, these flows appear to be cancelled to a significant degree by flows due to wave diffraction, leaving vessel responses to waves largely unaffected by the resonance.

Comparisons of the calculated hydrodynamic coefficients with those measured in a physical model show that the effect of the manometer resonance is much less pronounced in the physical model. A simpler model than QUAYSHIP was developed for two dimensional flow, in planes at right angles to the quay, in order to study the effect of friction on the resonance. The friction expected in physical models appears sufficient to explain the differences between potential theory and the physical model. At full scale it appears that some effect of friction remains on the hydrodynamic coefficients. However, the simplified two dimensional model indicates that vessel movements due to waves are unaffected by friction in both the physical model and at full scale : a result that would be consistent with the finding that actual vessel responses appear to be largely unaffected by the manometer resonance.

Recommendations are made for the further research needed into the roll responses calculated in UNDERKEEL and QUAYSHIP, and the added inertia coefficients for berthing vessels. The latter area being of concern in jetty design due to the size of berthing impacts experienced with large modern vessels. The research reported here also indicates that some simplification in modelling linear vessel responses may be possible. This could greatly assist in the subsequent computation of non-linear wave forces acting on a ship moored to a quay. It is recommended, therefore, that simplifications in modelling linear responses be sought.

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FIGURES (CONT'D)

1 INTRODUCTION

A suite of computer models is presently under development at Hydraulics Research (HR) to satisfy the requirement for a realistic first estimate of ship response to wave action. These models, coupled with the Boussinesq model describing waves in harbours, will enable more accurate feasibility studies of proposed harbour developments to be carried out prior to detailed studies using physical models. The ship models are vital in feasibility studies due to the extreme variability of ship response to wave parameters: direction and period being particularly important. This variability invalidates judgements of berth downtime based on wave height alone.

The computer model UNDERKEEL has been developed to a stage where it will describe the linear response of a free ship to waves (Ref 1). UNDERKEEL has been proved against a physical model of a tanker underway in random waves and it is used in project work for initial optimisation of dredged depths of navigation channels that are exposed to waves (Refs 2, 3, 4, 5).

As it stands, UNDERKEEL can also be used to describe the linear response of a ship moored at an open berth, ie a piled jetty at some distance from any reflecting boundaries. This is because waves pass through vertical piles (assuming typical spacing) without significant reflection thereby simulating the conditions that apply to a vessel in the open sea. But the situation of a vessel moored to a quay is different since both the incident waves and waves created by movement of the ship will reflect from the quay to some degree. With a solid vertical quay constructed of masonry blockwork, or of steel sheet piling, this reflection will be almost total making the water flow around the vessel noticeably different from the case of an open berth.

It was thought originally that the computer model UNDERKEEL could be adapted to the case of a ship moored to a quay without a great deal of additional work but this has not proved to be the case. In developing UNDERKEEL for the open sea situation use was made of the fact that surge, pitch and heave (see Fig 1) create flows around the vessel that are symmetrical about the longitudinal axis of the vessel, and sway, yaw and roll create antisymmetric flows. This feature simplifies the solution of the resulting equations. With a reflecting boundary nearby the motion of the vessel in a particular oscillating mode (surge, sway etc) leads to flows on each side of the ship which, in contrast to the open sea situation, cannot be classified as being either completely symmetric or completely antisymmetric. This means that a single oscillating mode results in a force and a turning moment on the ship with components in all three directions. Although the extra coupling was expected, solving the resulting equations has proved to be a much greater task than anticipated and a new model has been developed to satisfy the requirement. This model has been named QUAYSHIP in recognition of the fact that flows in the side clearance between the ship and the quay will play an important part in controlling vessel response just as flows under the keel become important with a small underkeel clearance.

In parallel with this work, the many non-linear wave forces that can act on a vessel are being computed as functions of wave frequency making use of equations derived in an earlier report (Ref 6). This is being done for the case of a vessel moored at an open berth where the linear velocity potentials for vessel motion and wave diffraction from UNDERKEEL are used in the formulation of the non-linear wave forces. The same equations given in Reference 6 can be used to compute the non-linear wave forces acting on a ship against a quay. Only in this case, the linear velocity potentials used in formulating the non-linear wave forces must come from QUAYSHIP. Once this stage has been reached, the mathematical models will be capable of describing the motions of vessels moored at open berths, and to berths with a quay face, although only for the case of linear moorings where a frequency space description of vessel motion is valid. Further work is then needed to represent non-linear wave forces in a random sea as functions of time. This is to allow their incorporation into SHIPMOOR (Refs 7 and 8) : a time domain computer model which is needed to represent important ship motion effects like subharmonic sway responses due to fenders being stiffer than mooring lines. It is necessary that the ultimate model of a moored ship be a time domain model as only then can the non-linear characteristics of conventional mooring systems be described.

From the above discussion it can be appreciated that a vital building block in computing moored ship motions against a quay is the linear response to wave action and that forms the subject matter of this report.

Theoretical aspects are discussed in Section 2 and an application of QUAYSHIP to a large tanker moored to a solid vertical quay is described in Section 3. The effect of friction on the flows induced under the keel of the vessel are investigated in Section 4. Conclusions and recommendations for further research follow in Sections 5 and 6.

2 THEORETICAL CONSIDERATIONS

The method used to calculate wave forces follows the same pattern as before (Ref 1) ie for each mode of vessel motion the flow beneath the hull is simplified and the flow round the sides of the vessel is represented by suitable distributed sources where source strengths are determined by the boundary conditions on the vessel's surface. However, the potential (Green's function) representing flow from a source must now also satisfy the boundary conditions at the quay face. In addition, to calculate the force on the vessel arising from diffraction, the potential of the total incident wave system for the vessel must be defined.

A simple situation for which these potentials can be easily derived is that of a ship moored at a perfectly reflecting straight vertical wall in an otherwise open situation (Fig 2). This can be taken as representative of a berth inside a harbour. In this case the velocity component normal to the wall must be zero and to satisfy this condition the required Green's function is equal to the sum of the open sea potentials of the source and its image with respect to the wall.

We denote the potential of the incident wave by the real part of,

$$\phi_o = \frac{iag}{\omega} \frac{\cosh K (z+c)}{\cosh Kd} e^{iK (\lambda x + \mu y)} e^{-i\omega t}$$

where a, ω are amplitude and radian frequency of the wave, β is the angle of propagation ($2\pi > \beta > \pi$, Fig 2) with

$$\lambda, \mu = \cos\beta, \sin\beta$$

and K satisfies the usual dispersion relation for surface waves,

$$\omega^2 = Kg \tanh Kd$$

Here d is the water depth and c is the height of the origin of coordinates above the bed.

The boundary condition at the quay wall requires

$$\frac{\partial}{\partial y}(\phi_o + \phi_r) = 0 \quad \text{at } y = -e$$

This is satisfied if the potential of the reflected wave (ϕ_r) is given by,

$$\phi_r = R \frac{iag}{\omega} \frac{\cosh K(z+c)}{\cosh Kd} e^{iK(\lambda x - \mu y)} e^{-i\omega t}$$

where

$$R = e^{-2iK\mu e}$$

Then the total wave system incident on the ship is given by

$$\phi_o + \phi_r$$

As explained in the Introduction, motion of the vessel in a given oscillatory mode will couple with all the other modes. One consequence is that the 6x6 matrices representing inertia and damping each have 36 non zero components instead of having just 18 components as described in Reference 1 for the ship in an open sea situation. This results in a more complicated method of solution for the amplitudes of vessel motion compared with the open sea case.

2.1 Manometer resonance

One of the most startling differences with the open sea situation is that added inertias can become negative for a ship against a quay (Ref 9). This will be demonstrated in the subsequent section where results are presented for a tanker moored near a solid vertical wall. It must be remembered that added inertia is just a way of describing the component of water flow around a vessel that is in phase with vessel acceleration. In an open sea situation an extra mass of water has to be accelerated with the vessel so the added inertia is always positive. But for a ship against a quay, the water trapped in the clearances between the ship and the quay, and between the ship and the seabed, can go into resonance with the result the added inertia changes its sign at the resonant frequency. In effect, the water trapped in the clearances acts like a spring. This type of motion can be described with reference to Figure 3.

We consider oscillations of the water in the clearances b and δ with velocities w and v, respectively, producing an increased elevation ζ in the clearance with the quay (b). The vessel's beam and draft are taken to be B and D, respectively.

Continuity of the flow requires,

$$wb = v\delta$$

Momentum equations in the clearances show that the additional pressure P_1 over and above atmospheric pressure, produced at the surface in the clearance between the ship and the quay is given by,

$$P_1/\rho = -\dot{w}D - \dot{v}B$$

We can relate this pressure to the increased elevation ζ through

$$P_1/_\rho = g\zeta$$

and express vertical velocity in the form

$$w = \dot{\zeta}$$

Eliminating v, w and P_1 we obtain the following equation for the vertical movement of the water surface in the clearance between the vessel and the quay,

$$\ddot{\zeta}\,[D + b\,{}^B/_\delta] + g\,\zeta = 0$$

So the natural "manometer" frequency is given by,

$$\omega^2 = {}^g/_{[D + b\,{}^B/_\delta]} \qquad (1)$$

It can be appreciated that small vessel movements at this resonant frequency, in heave or sway for example, will produce large flows in the clearances. Thus, exactly on resonance the added inertia will vanish and significant waves will be produced in the open sea side of the vessel, indicating large damping of the vessel motion. In practice, though, friction effects on the flows in the clearances will limit the size of these effects (Ref 10) and this aspect of behaviour is considered further in Section 4 of this report.

3 RESULTS FOR A VESSEL NEAR A SOLID VERTICAL QUAY

Here, we describe the application of QUAYSHIP to the case of a tanker aligned parallel to, but at various distances from, the quay face. The vessel details are given in Table 1.

This particular situation has been chosen because it relates to what appears to be the only comprehensive set of data in the literature for a ship near a solid vertical quay (Ref 9). In this reference, van Oortmerssen presents both physical model and theoretical results for some of the hydrodynamic coefficients of the vessel, ie added inertia and damping coefficients in sway and heave. The physical model scale was 1 to 82.5 and his theory made use of sources distributed over the submerged surface of the hull with strengths that satisfied the relevant boundary conditions assuming linear potential flow (3D source method).

Cross sections of the vessel at 21 equally spaced stations along its length starting at the stern, are presented in Figure 4 : stern sections appear on the left and bow sections appear on the right. These vessel characteristics were represented in QUAYSHIP and hydrodynamic coefficients for all six degrees of freedom of vessel movement (see Fig 1 for definition of movements) were calculated along with vessel responses. To allow for comparison with van Oortmerssen's results 5 different clearances, between the straight sided part of the vessel and quay, were considered (distance b in Fig 3). These ranged from 8.25m up to 41.25m.

3.1 Hydrodynamic coefficients

We consider first the results for sway and heave hydrodynamic coefficients. Comparisons with van Oortmerssen's experimental and theoretical results are presented in Figures 5 to 9. These hydrodynamic coefficients have been made non-dimensional in the following way

Sway added inertia $$A_{22} = \frac{A'_{22}}{M}$$

and damping $$B_{22} = \frac{B'_{22}}{M\sqrt{g/L}}$$

Heave added inertia $$A_{33} = \frac{A'_{33}}{M}$$

and damping $$B_{33} = \frac{B'_{33}}{M\sqrt{g/L}}$$

In these definitions, M is the displacement (mass) of the vessel, L is the length of the vessel while A'_{22} and A'_{33} are the added inertias in sway and heave, and B'_{22} and B'_{33} are the damping coefficients for sway and heave. They are plotted as functions of a non-dimensional wave frequency $\omega\sqrt{L/g}$.

The first point to notice is the correlation obtained between QUAYSHIP (solid line) and van Oortmerssen's theoretical results (closely spaced dashed line) in Figure 6. It is possible to make this comparison only for a separation distance of 16.5m from the quay as these are the only theoretical results given in Reference 9. The advantage of QUAYSHIP over the 3D source method used by van Oortmerssen is that much less computation is needed and difficulties representing small clearances with the 3D source method are avoided.

It is clear that all the results display the manometer resonance described in Section 2.1. The added inertias vanish on resonance and the shift of this resonance to smaller frequencies, seen in Figures 5 to 9 as the separation distance (b) with the quay increases, is explained by equation (1). We see this resonance causes large damping with negative added inertias for frequencies of movement just above the resonant frequency and very large positive inertias for lower frequencies. We expect this resonance to disappear in the limit of a large clearance with the quay and such a trend is visible in Figures 5 to 9 where the large positive inertias, seen with a quay separation distance of 8.25m, become less pronounced with a separation distance of 41.25m. It is remarkable, though, how important the effect of the resonance remains even when the separation distance is of the order of the beam of the vessel. In practical situations, where a vessel is moored against a vertical quay, the separation distance will be of the order of a few metres at most, ie the order of the limit of compression of the fenders. In such situations the manometer resonance will be even more pronounced. For the ship being studied here, a clearance with the quay of 2m results in resonance occurring at a period of some 13 seconds (equation (1)) which is well within the range of possible wave periods. However, the important parameter for a moored ship is the final vessel response and the complexity of this response means that one particular factor, like the manometer resonance, is not necessarily dominant (see Section 3.2).

The other obvious point emerging from Figs 5 to 9 is the apparent overestimation, by both QUAYSHIP and van Oortmerssen's 3D source method, of the resonance effects present in the physical model results. This is most pronounced at the smaller quay clearance of

8.25m (Fig 5). But the difference between results from QUAYSHIP and results from van Oortmerssen's experiments appear minimal at the largest quay clearance of 41.25m (Fig 9). It will be shown in the next section that these differences between theory and experiment can be explained by friction effects on the flow under the keel. Resonance enhances these flows with a small quay clearance eg 8.25m, making the flows more sensitive to any resistance due to friction. As resonance effects decrease with an increasing quay clearance eg 41.25m, friction can be expected to become less important so a theory without friction effects can be expected to apply.

For completeness, we present non-dimensional hydrodynamic coefficients from QUAYSHIP for all six degrees of freedom in Figures 10 to 15. Surge added inertia A'_{11} and damping B'_{11} are divided by the same factors as the sway and heave coefficients ie

$$A_{11} = \frac{A'_{11}}{M}$$

$$B_{11} = \frac{B'_{11}}{M\sqrt{g/L}}$$

The added inertia and damping coefficients for roll (A'_{44}, B'_{44}) pitch (A'_{55}, B'_{55}) and yaw (A'_{66}, B'_{66}) have been made non-dimensional in the following way,

$$A_{ii} = \frac{A'_{ii}}{M L^2}$$

$$A_{ii} = \frac{B'_{ii}}{M L^2\sqrt{g/L}} \qquad i = 4, 5, 6$$

Unfortunately, there are no results presented in Ref 9 for surge, roll, pitch and yaw hydrodynamic coefficients for the vessel near a solid vertical quay. But it is clear that these additional vessel movements exhibit the "manometer" resonance already identified in sway and heave. There also appears to be some "structure" at higher frequencies than the manometer resonance in some of the coefficients, eg A_{66} in Figure 15. This correlates with a half wavelength resonance across the gap between the side of the vessel and the side of the quay. Such effects appear negligible in sway and heave and may well prove to be unimportant in practice.

3.2 Vessel responses

We present responses for all six degrees of freedom in Figures 16 to 20 for wave directions (see angle β in Fig 2) of 270° (beam seas), 285°, 300° 315° and 330° respectively. The responses have been made non-dimensional for surge, sway and heave by dividing the amplitude of movement by the amplitude of the incident wave. This explains, for example, why the heave amplitude tends to double the incident wave amplitude for low frequencies ie the vessel follows the vertical movement of the water surface formed by wave reflection from the quay face (wave antinode). The angular movements of roll, pitch and yaw have been expressed in degrees per metre of incident wave height.

Perhaps the most startling result is the smoothness of the response curves at frequencies corresponding to the manometer resonance. Considering sway in a beam sea (Fig 16(b)) and comparing it with the behaviour of the sway added inertia and damping coefficients (Fig 11) we see little sign of an effect on resonance. For example, the response in Figure 16(b) with a quay clearance of 8.25m is relatively smooth from low frequencies right up to a value of 3 for the non-dimensional frequency. Figure 11 shows large changes in the hydrodynamic coefficients over this frequency range for a quay clearance of 8.25m due to the manometer resonance which occurs at a frequency of about 1.7. This sort of result indicates that flows induced under the keel by wave diffraction must be counteracting the manometer resonant flows induced by movement of the vessel. Such cancelling effects also appear operative in surge (Fig 16(a)), heave (Fig 16(c)) roll (Fig 16(d)) pitch (Fig 16(e)) and yaw (Fig 16(f)).

There also appears to be an anomolously large heave response in Figure 16(c) at a frequency of about 2 for a quay clearance of 41.25m. This appears to be driven, at least in part, by coupling of a very large roll response at this quay clearance (see Fig 16(d)) into heave. The roll response is very high in this case because the centre line of the vessel happens to lie close to a node which produces a large roll couple close to the resonant roll period of the vessel. In practice viscous damping will lead to a much reduced roll response which in turn means the heave response predicted by QUAYSHIP is exaggerated. This problem can only be overcome by representing viscous damping in the computer model.

Returning to sway responses in a beam sea ie Figure 16(b), we see a maximum in the response occurs at a non-dimensional frequency that varies from about 3.5 for a quay clearance of 8.25m to about 2 for a quay clearance of 41.25m. These frequencies are also consistent with a node (place of largest horizontal water particle movement) forming along the centre line of the ship due to wave reflection from the quay face. One might expect to find the largest sway response at such frequencies as the vessel will experience the largest sway force in that situation, leaving aside forces due to wave diffraction around the vessel.

The other feature to notice in the sway responses (Fig 16(b)) is the behaviour in the limit as wave frequency tends to zero. The response appears to increase with the separation distance from the quay. This feature can be explained using a simple two dimensional model which can also be used to study the effect of friction on the manometer resonance. These aspects are considered in the next section.

4 FRICTION EFFECTS ON THE MANOMETER RESONANCE

QUAYSHIP has been shown to be effective in Section 3 through comparison with theoretical and experimental results obtained by van Oortmerssen (Ref 9). The comparison with physical model experiments has also shown that potential theory appears to exaggerate a manometer resonance that effects the vessel's hydrodynamic coefficients. Here we use a simplified model to investigate the effect of friction on the flows caused by that resonance.

The method of solution employed in QUAYSHIP allows for flow normal to the quay in the space between the side of the vessel and the quay (side clearance). In practice, however, the distance of moored vessels from the quay is often small implying that flow normal to quay can be neglected. In this case the flow parallel to the quay satisfies a two dimensional surface wave equation and is readily soluble once the boundary conditions at the ends of the vessel are defined. However, for the purposes of investigating the effect of friction we assume the vessel is infinitely long so that flow along the quay is neglected. The problem then becomes two dimensional with flow only occurring in planes perpendicular to the quay. For an infinitely long ship with a rectangular (block) cross section the flow will be vertical in the side clearance and horizontal in the underkeel clearance (see Fig 3). Given these conditions it is possible to derive a linear relation between the potential and the fluid velocity on the open sea side of the underkeel clearance space. In this case, the potential on the open sea side of the block ship can be derived using a similar treatment to that adopted in UNDERKEEL for a vessel in the open sea.

4.1 Application to a block ship

Here, we apply the simplified two dimensional model described above to the situation portrayed in Fig 3 with the following parameters

$$\frac{\delta}{D} = 0.067, \ \frac{B}{D} = 2.5, \ \frac{b}{D} = 0.287$$

The resulting added inertia and damping coefficients in sway are shown in Figures 21 and 22 along with the sway response to waves at normal incidence (beam sea) calculated in the absence of vessel heave. As the vessel length is infinite, parameters are expressed relative to the vessel's beam. Thus, sway added inertia (A'_{22}) and damping (B'_{22}) have been made non-dimensional using,

$$A_{22} = \frac{A'_{22}}{\rho BD}$$

$$B_{22} = \frac{B'_{22}}{\rho BD \sqrt{g/B}}$$

and the wave frequency is made non-dimensional using,

$$\omega \sqrt{\frac{B}{g}}$$

In Figure 21 there are two sets of results. One is derived using linear potential theory, which also forms the basis of QUAYSHIP and the 3D source method employed by van Oortmerssen. In this case damping is due solely to waves radiating energy away from the vessel on the open sea side of the block ship. It can be verified that the manometer resonance occurs at a frequency consistent with equation (1). The second set of results in Figure 21 is derived allowing for friction effects one expects to find in a 1 to 100 scale physical model.

A corresponding set of results is presented in Figure 22 allowing for the friction effects one expects to find in prototype, ie for a real ship.

Figure 21 shows the sort of difference found between linear potential theory and the physical model results of van Oortmerssen where sway hydrodynamic coefficients on resonance in the experiments were about half of those calculated (see Fig 6(a)). It can also be seen, by comparing the hydrodynamic coefficients in Figure 22 with the linear potential theory results in Figure 21, that friction effects at full scale, ie for real ships, have a smaller effect on the manometer resonance.

Two friction effects have been represented. One is the head loss in the underkeel flow (v) due to viscous shear (τ) on the boundary surfaces. As in steady flow, this energy loss is assumed to depend on whether the flow is laminar or turbulent. The controlling factor is the Reynold's number (R_e)

For laminar flow : $\tau/_{\rho v^2} = 6/_{R_e}$, when $R_e < 1000$

For turbulent flow : $\tau/_{\rho v^2} = 0.0336/_{R_e^{1/4}}$,

when $R_e > 1000$

Here, ρ is the density of water and the Reynolds number is defined by,

$$R_e = \frac{|v|\delta}{\nu}$$

where ν is the kinematic coefficient of viscosity. The pressure loss over the underkeel clearance is then given by

$$\Delta p/_{\rho} = \frac{2B}{\delta}\, \tau/_{\rho}$$

The second friction effect arises from eddy losses in the flow as it separates at entry to, and exit from, the underkeel clearance during the oscillation. As in steady flow, the sum of entry and exit losses are taken to be proportional to the square of velocity,

$$\Delta p/_{\rho} = \xi\, v^2/_2$$

where ξ is a coefficient obtained from experiment (ξ = 1.44, Ref 10). In fully turbulent flow the head losses will be approximately proportional to the square of the flow velocity under the keel. In these cases a linearised friction coefficient proportional to velocity is defined to allow solution of the equations. This is why the results are sensitive to the amplitude of the incident wave (Figs 21 and 22).

The behaviour of the damping coefficient B_{22} near resonance is shown in greater detail in Figure 23. We again see that the linear potential theory result is roughly halved in magnitude due to the friction effects expected in a 1 to 100 scale physical model. It is also apparent that the damping in the physical model is not very amplitude dependent. This is because the flow expected under the keel is in the laminar range where energy losses are more nearly proportional to velocity, rather than the square of velocity. In full scale, the flow is expected to be turbulent making damping more amplitude dependent, although the friction effect is noticeably smaller than in the model.

It might be thought strange that including the effect of friction actually reduces the damping coefficient B_{22} (Fig 23). However, the effective damping factor can be considered to be $B_{22}/|1 + A_{22}|$ after allowing for added inertia, ie dividing the damping coefficient by the displacement mass plus the added mass. This damping factor is shown in Figure 24 for linear potential theory and full scale conditions on the left-hand side and for a 1 to 100 scale physical model on the right hand side. It can be seen the effective damping factor is indeed increased when friction is included and that the largest increase occurs in the 1 to 100 scale physical model (note the logarithmic vertical scale).

Finally, we return to the sway responses shown in Figures 21 and 22. The smooth behaviour of the sway response ζ_2 over the frequency range encompassing the rapid ranges in the hydrodynamic coefficients (due to the manometer resonance) is again apparent, as noted in Section 3.2 for the three dimensional model QUAYSHIP It is also apparent that in the 2D model at least, the addition of friction has little effect on sway responses. This is reassuring for physical model work using models of moored ships. However, there is little doubt that added inertia effects, so important in controlling the berthing impacts of large vessels, would not be well represented in physical models. In the case of berthing, mathematical simulations incorporating the friction effects expected at full scale, are needed for accurate results.

To verify the magnitude of the sway response it is possible to use elementary methods to solve the linear potential equations for the case of long waves ($\omega = 0$). This leads to an estimate for the sway amplitude ζ_2 which is related to the side clearance b : an effect noted earlier in Fig 16(b) with SIDEKEEL.

$$\zeta_2 / a = \left[\frac{2b}{D + \frac{D^2\delta}{3Bb}} \right]$$

This equation yields a value of 0.58 for the block ship in question which agrees with the result at $\omega = 0$ given for potential theory in Figure 21. This response appears to be maintained at higher frequencies and the observation that friction seems to have a minimal effect on the response is again consistent with the idea that flows under the keel due to wave diffraction are tending to cancel the flows due to the manometer resonance : if the resultant flow is small the importance of friction on that flow will clearly be diminished.

5 CONCLUSIONS

1. QUAYSHIP has been developed to describe the linear response of a vessel moored against a quay and subjected to wave action, a situation typical of a berth inside a harbour.

2. Correlation between QUAYSHIP and a different method of computing responses (Ref 9) has been obtained for hydrodynamic coefficients in sway and heave for the case of a tanker at a distance of 16.5m from a vertically faced quay (Fig 6). The advantage of QUAYSHIP is that simplifying assumptions about the flow under the keel enable results to be obtained with much less computation and difficulties representing small clearances are avoided.

3. In comparisons of results obtained using linear potential theory (QUAYSHIP and the 3D source method used in Ref 9) with physical model results it is clear that the effects of a manometer resonance on vessel hydrodynamic coefficients are much less pronounced in the physical model (Fig 6). On resonance large flows occur under the keel of the vessel and energy losses due to friction effects appear important.

4. A two dimensional model (infinitely long ship) has been developed to study the effects of friction on the flow under the keel. Head losses have been represented due both to viscous shear on the boundary surfaces and to eddying as the flow separates at entry to, and exit from, the underkeel clearance. This work indicates that friction effects in physical models will be much greater than for real ships but that the turbulent flow conditions expected at full scale mean that real ship hydrodynamic coefficients will also be affected by friction on the flow under the keel.

5. In spite of finding significant friction effects on hydrodynamic coefficients, the two dimensional model result for the sway response to wave

action appears independent of friction. This suggests that flows under the keel due to wave diffraction are tending to cancel flows associated with the manometer resonance. If the resultant flow is small it can be appreciated that friction effects will become less important.

6. The above result is encouraging for physical models of harbours that use models of moored ships to judge berth downtime. It suggests that horizontal vessel movements should be well represented in the physical model even though hydrodynamic coefficients are likely to be affected by unrealistic friction effects on flows under the keel.

7. The conclusion for vessels berthing in the absence of waves is different. Here the coefficients of added mass and damping in sway are important factors controlling berthing impacts and the importance of friction in controlling added mass and damping shows that they would be poorly represented in a physical model due to scaling problems. In this case a mathematical simulation model that made use of the hydrodynamic coefficients, calculated allowing for full scale friction effects, is needed to investigate berthing vessels.

8. Horizontal vessel responses obtained with the fully three dimensional QUAYSHIP also imply that flows under the keel due to wave diffraction tend to cancel flows due to the manometer resonance. This suggests that conclusions drawn about friction from the two dimensional model should apply in three dimensions as well.

9. It is anticipated that differences observed in Reference 1 between UNDERKEEL and physical model results for vessel roll in the open sea apply equally to the roll of a vessel moored against a quay. Indeed, the very large resonant roll responses shown in Figures 16(d), 17(d), 18(d) 19(d) and 20(d) are likely to be damped by the sort of friction effects described in conclusion 4 above. This in turn should reduce the anomolous heave responses observed at some wave frequencies. As these friction effects will be important at full scale, the resonant roll responses from QUAYSHIP must be considered overestimates as found with UNDERKEEL (Ref 1).

6 RECOMMENDATIONS

The work described in this report has demonstrated a clear need to develop a greater understanding of viscous damping effects. Resonant roll responses are known to be sensitive to such damping, making the linear potential theory used in UNDERKEEL and QUAYSHIP yield overestimates of roll. And, the work on vessel sway described here has shown viscous damping to be important in controlling added mass and damping which, in turn, will affect loads on jetties due to berthing ships.

It is recommended, therefore, that further research be carried out into viscous damping for the specific situation of a vessel with a small underkeel clearance. This with a view to improving the accuracy of vessel roll predictions in UNDERKEEL and QUAYSHIP as well as improving added inertia predictions for horizontal vessel movements near a solid quay. The improved estimates of added inertia and damping should be used to develop accurate vessel berthing simulators as an aid to jetty design : an area of concern at present due to the size of berthing impacts experienced with large modern vessels.

The finding that flows induced by wave diffraction tend to cancel the (resonant manometer) flows associated with vessel motion suggests that some simplification in modelling linear vessel responses may be possible. As this could assist greatly in reducing the amount of subsequent computation needed to calculate non-linear wave forces, for the case of a ship moored to a quay, it is recommended that research be carried out to seek any simplifications that may be possible.

7 REFERENCES

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TABLE

TABLE 1 : Vessel details (loaded) for results in Section 3

Length between perpendiculars	310m
Beam	47.2m
Draught	18.9m
Displacement	235,000m^3
Distance of centre of gravity forward of mid point	6.61m
Height of centre of gravity above keel	13.32m
Transverse radius of gyration	17.0m
Roll metacentric height	5.78m
Longitudinal radius of gyration	77.47m
Water depth (underkeel clearance = 20% of draught)	22.68m

FIGURES.

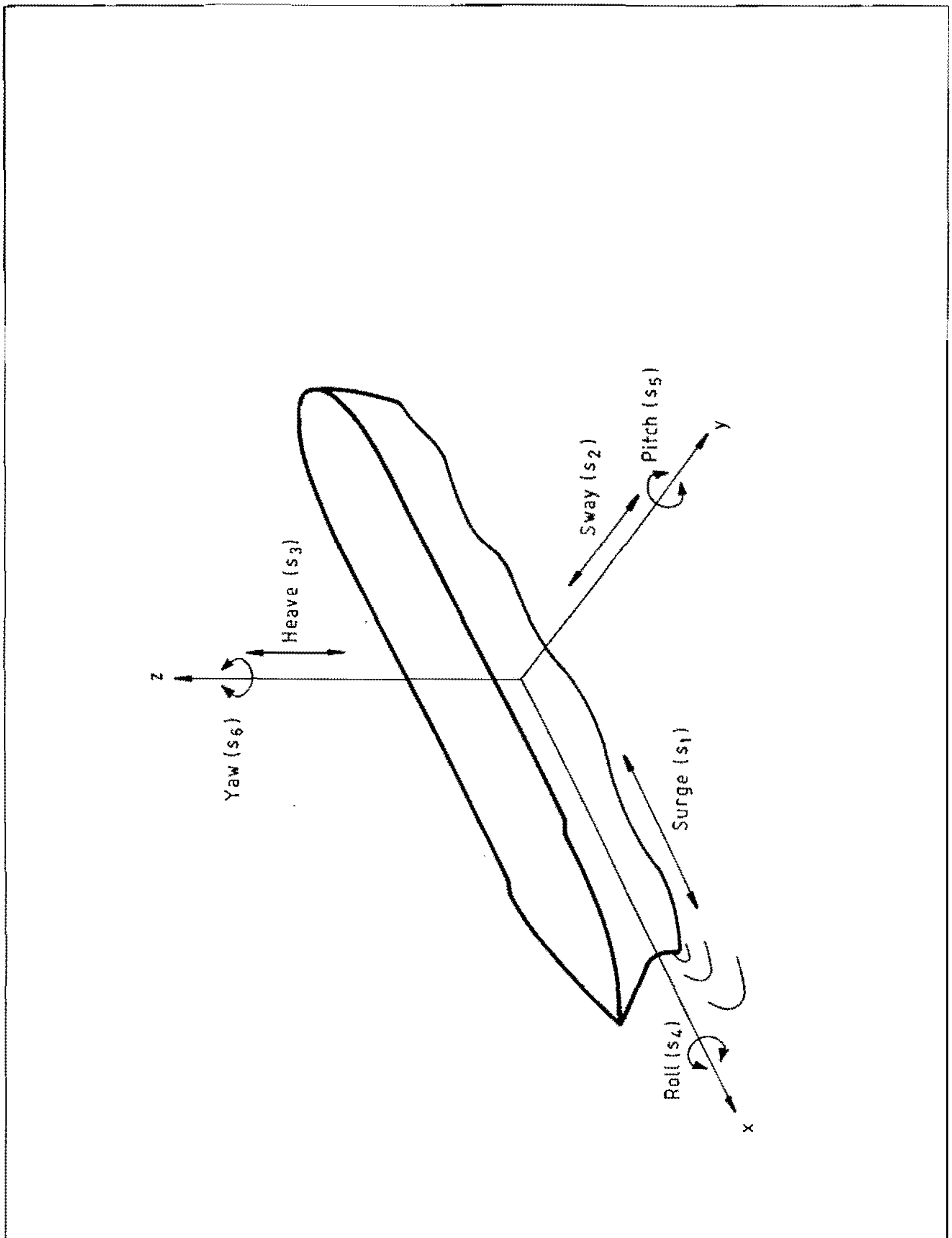


Fig. 1 The six degrees of vessel movement.

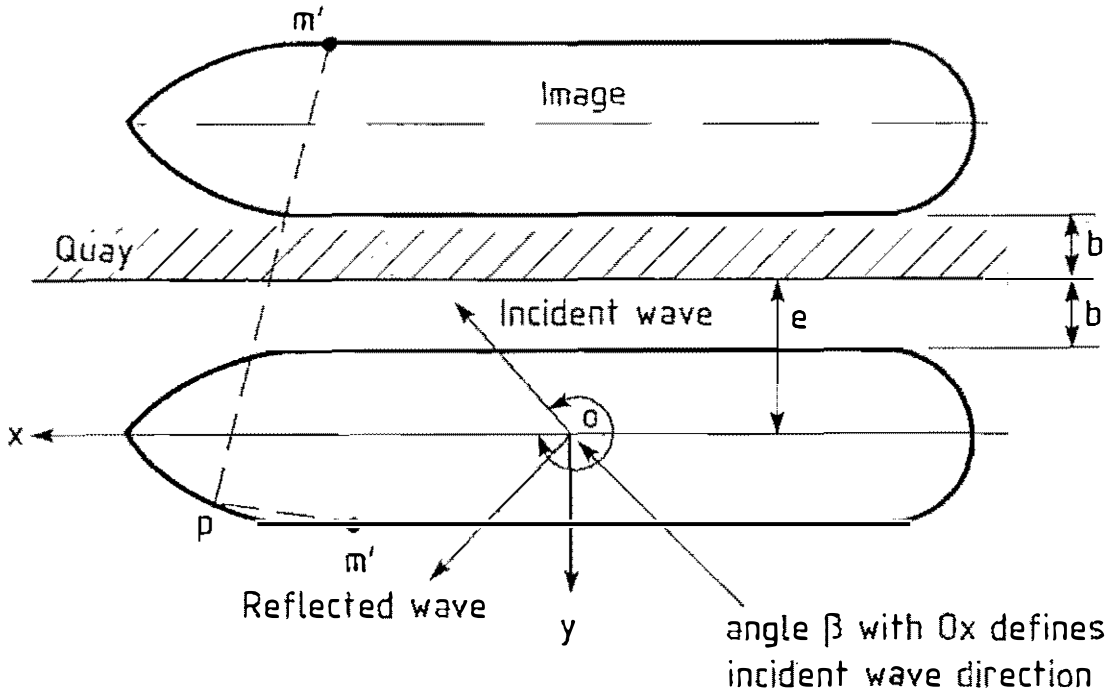


Fig 2 Definition of coordinates.

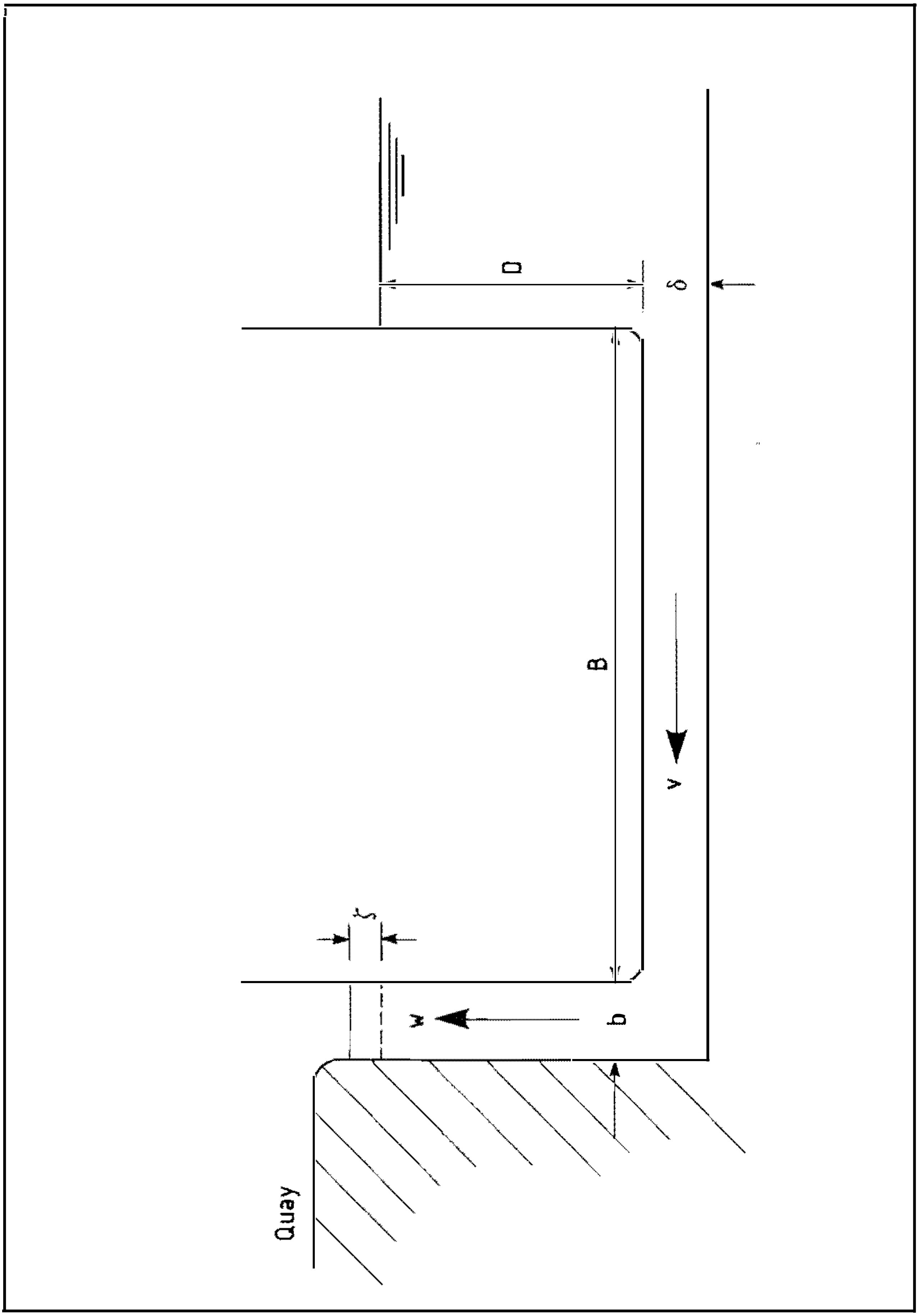


Fig 3 Manometric natural mode of motion.

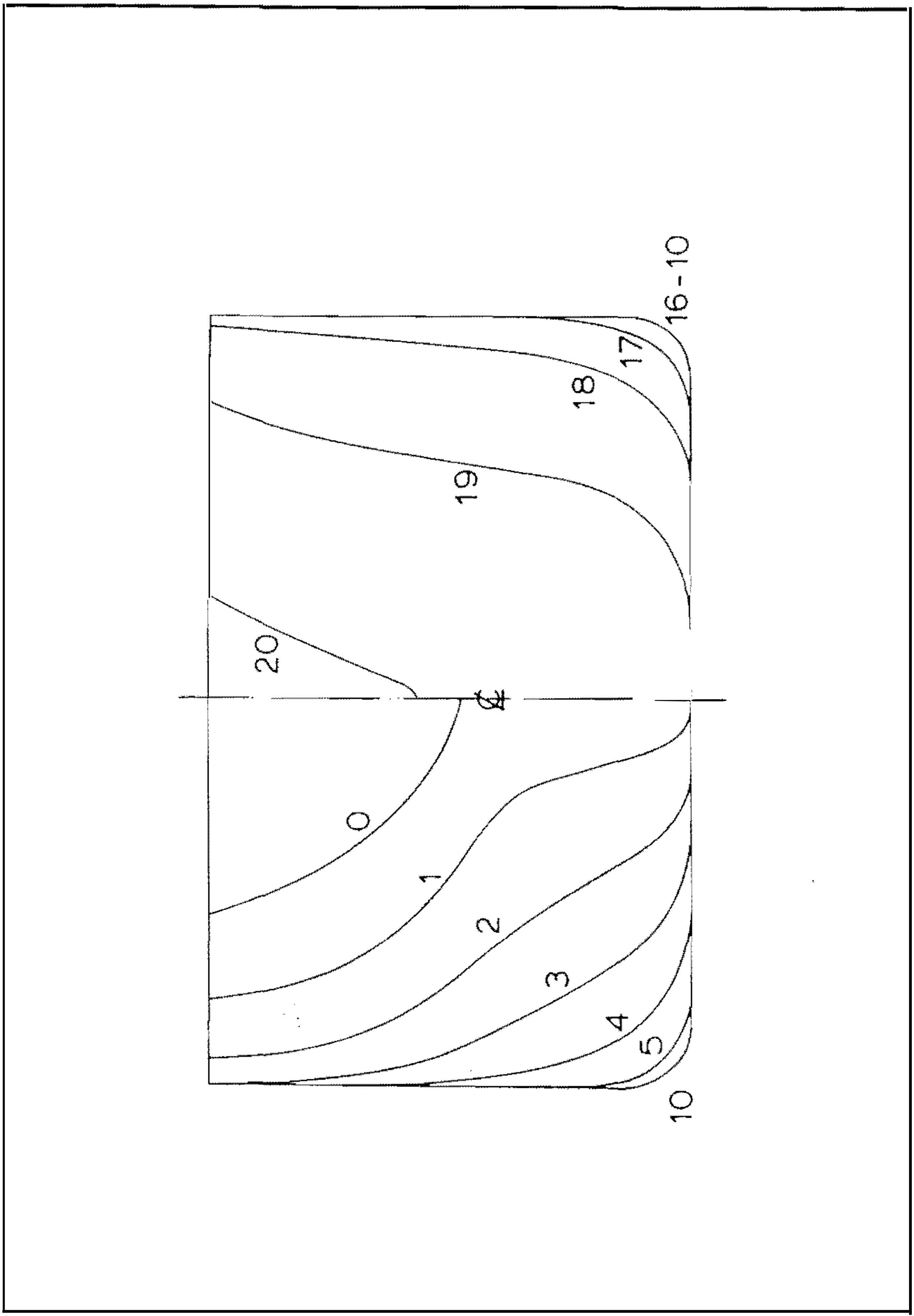


Fig 4 Vessel lines.

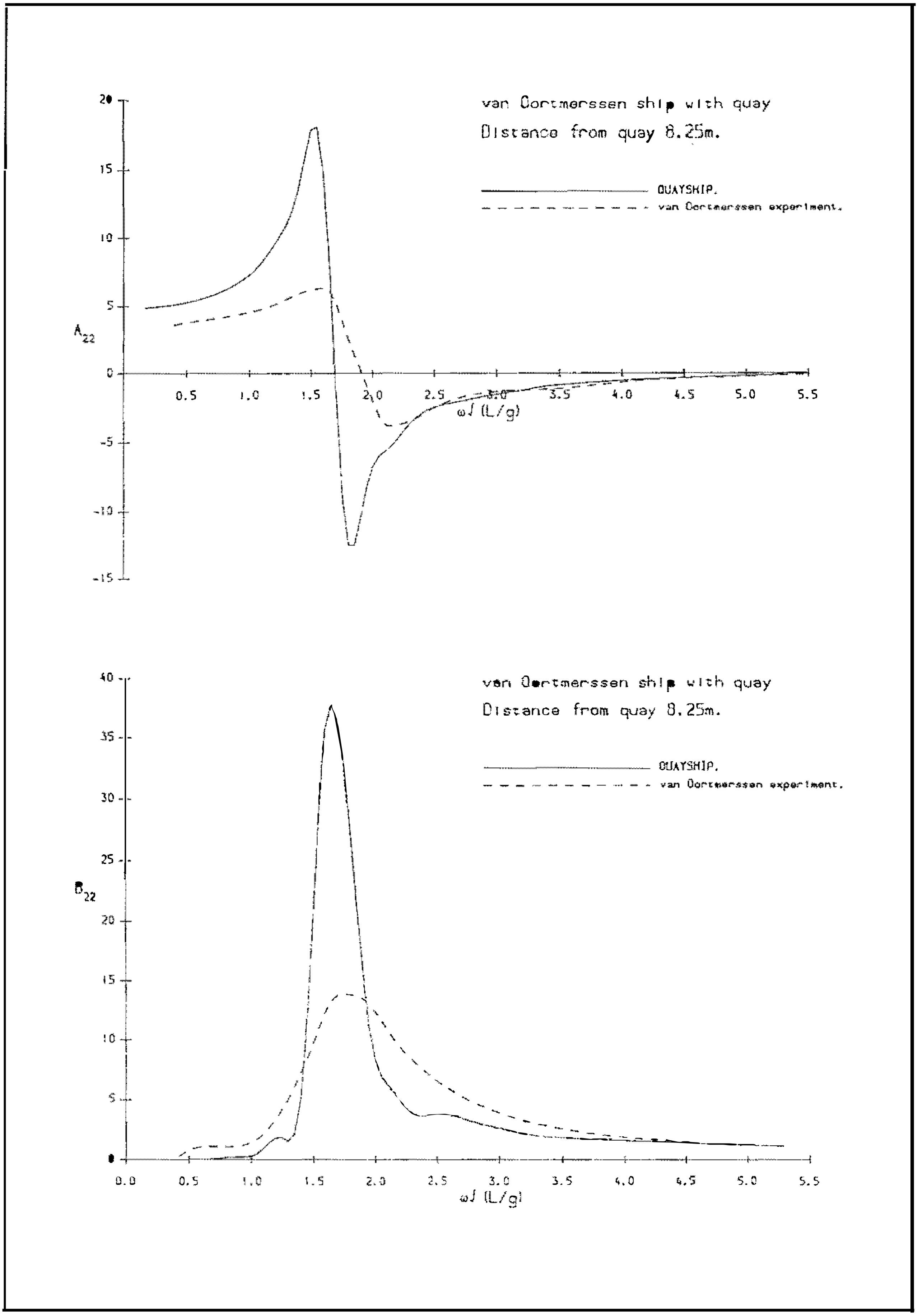


Fig 5(a) Added inertia and damping in sway-8.25m quay clearance.

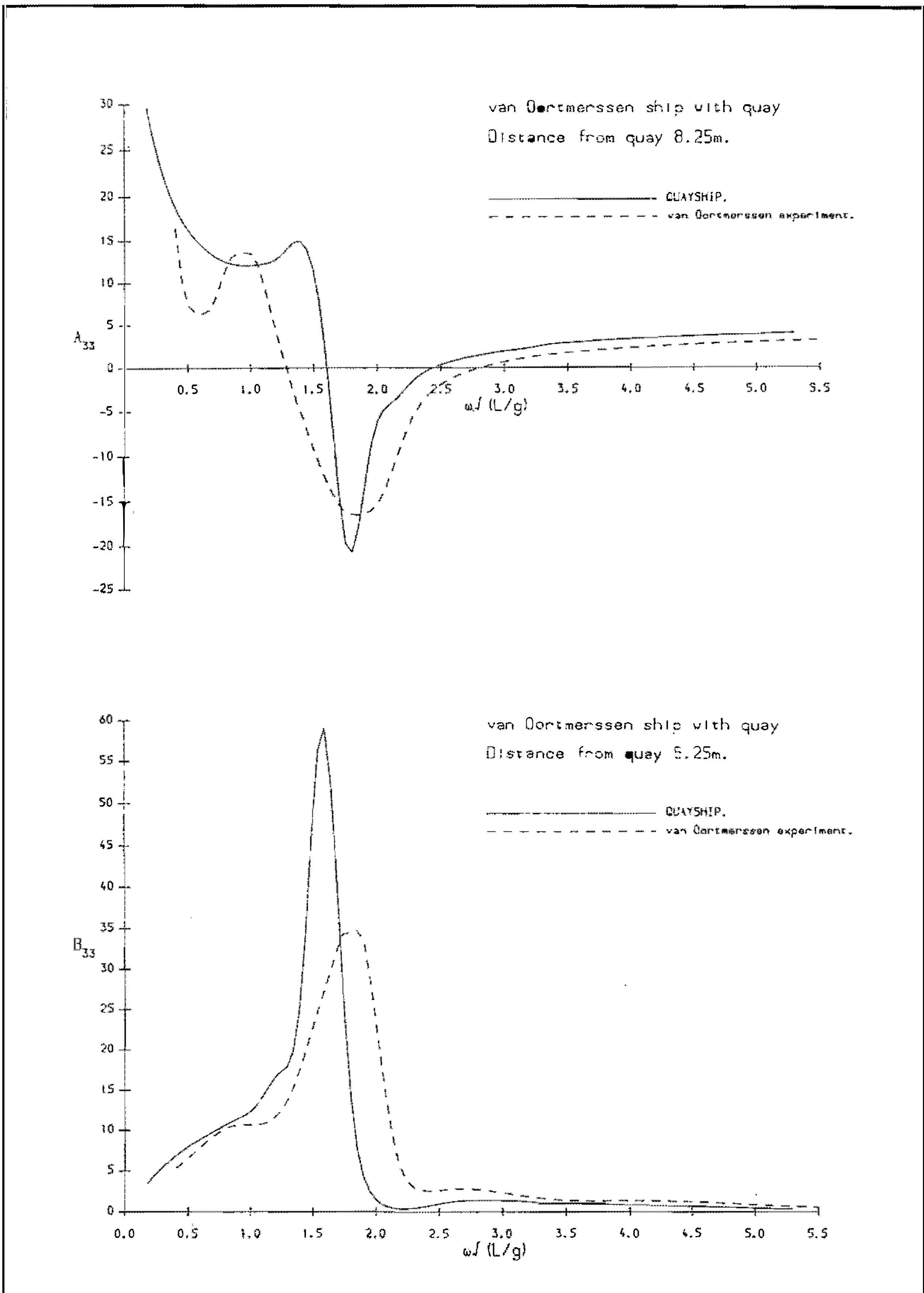


Fig 5(b) Added inertia and damping in heave-8.25m quay clearance.

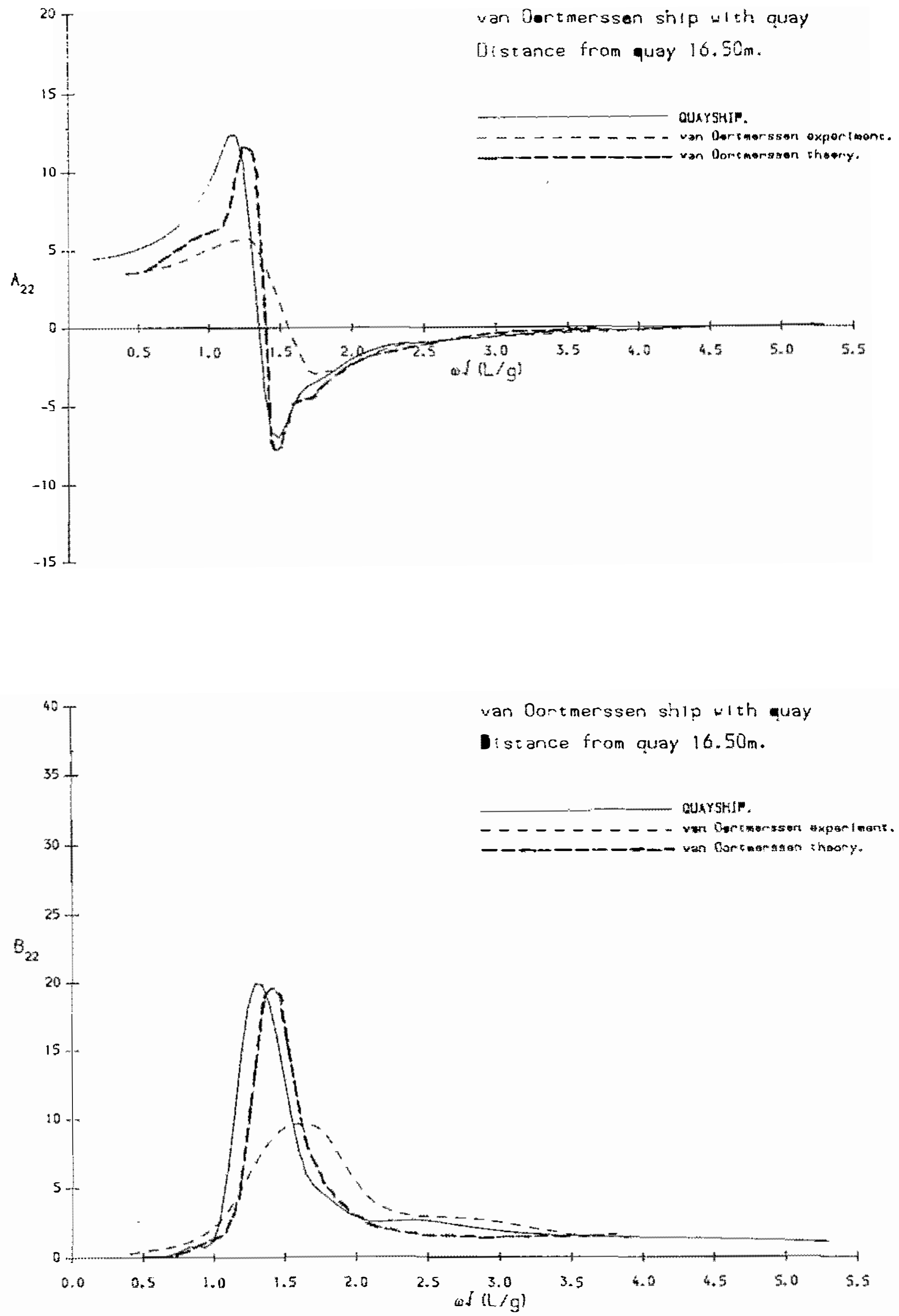


Fig 6(a) Added inertia and damping in sway-16.5m quay clearance.

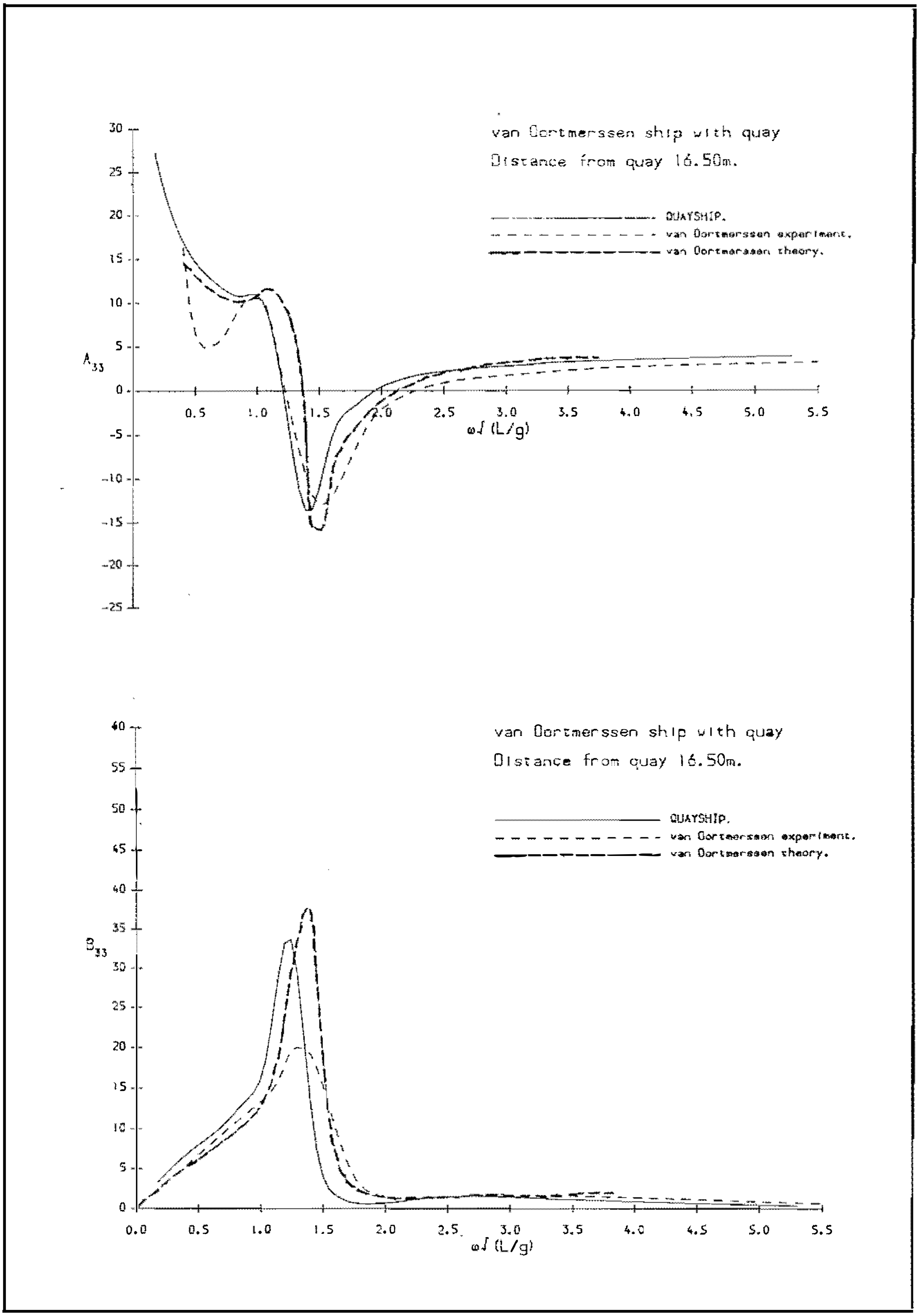


Fig 6(b) Added inertia and damping in heave–16.5m quay clearance.

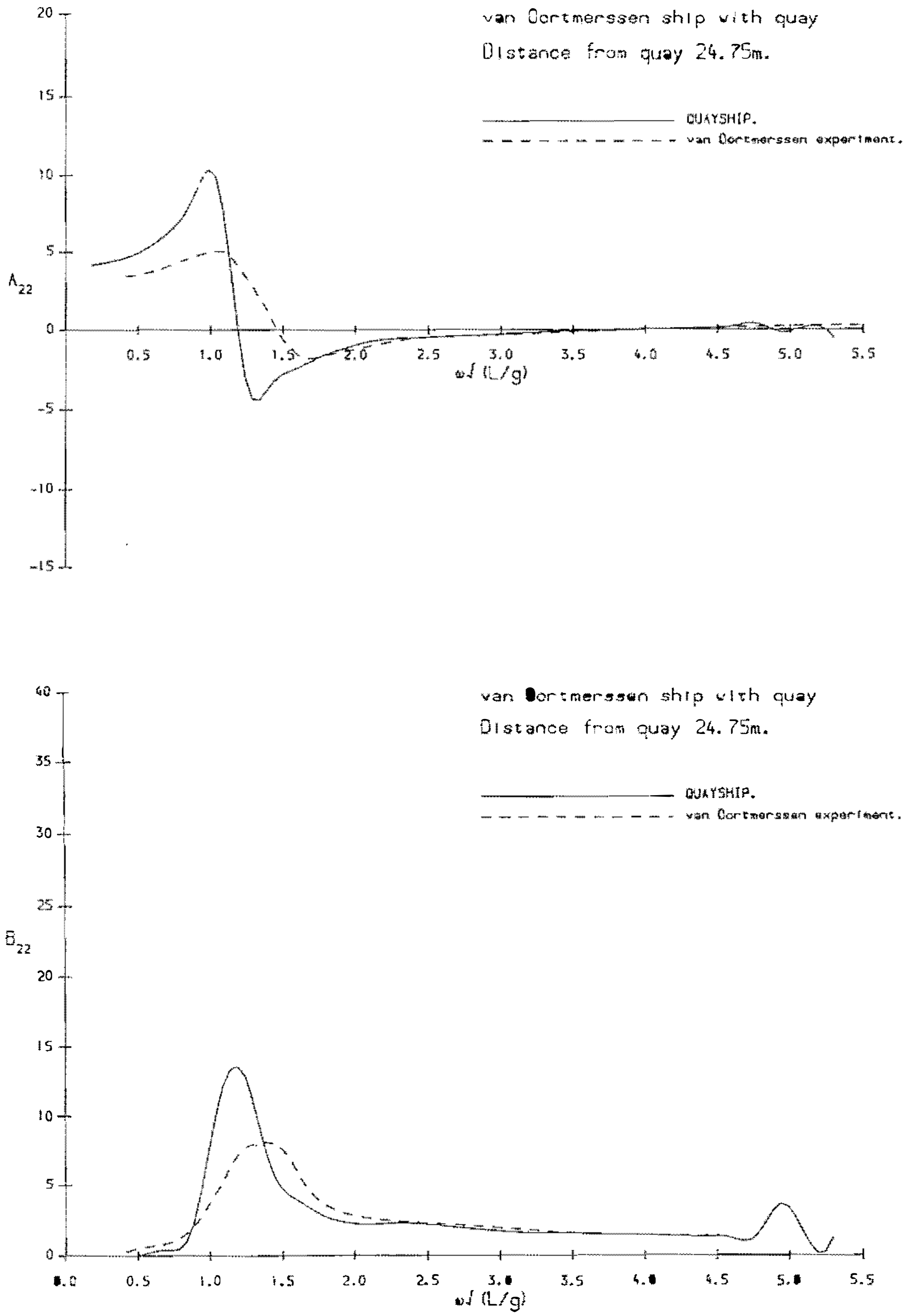


Fig 7(a) Added inertia and damping in sway-24.75m quay clearance.

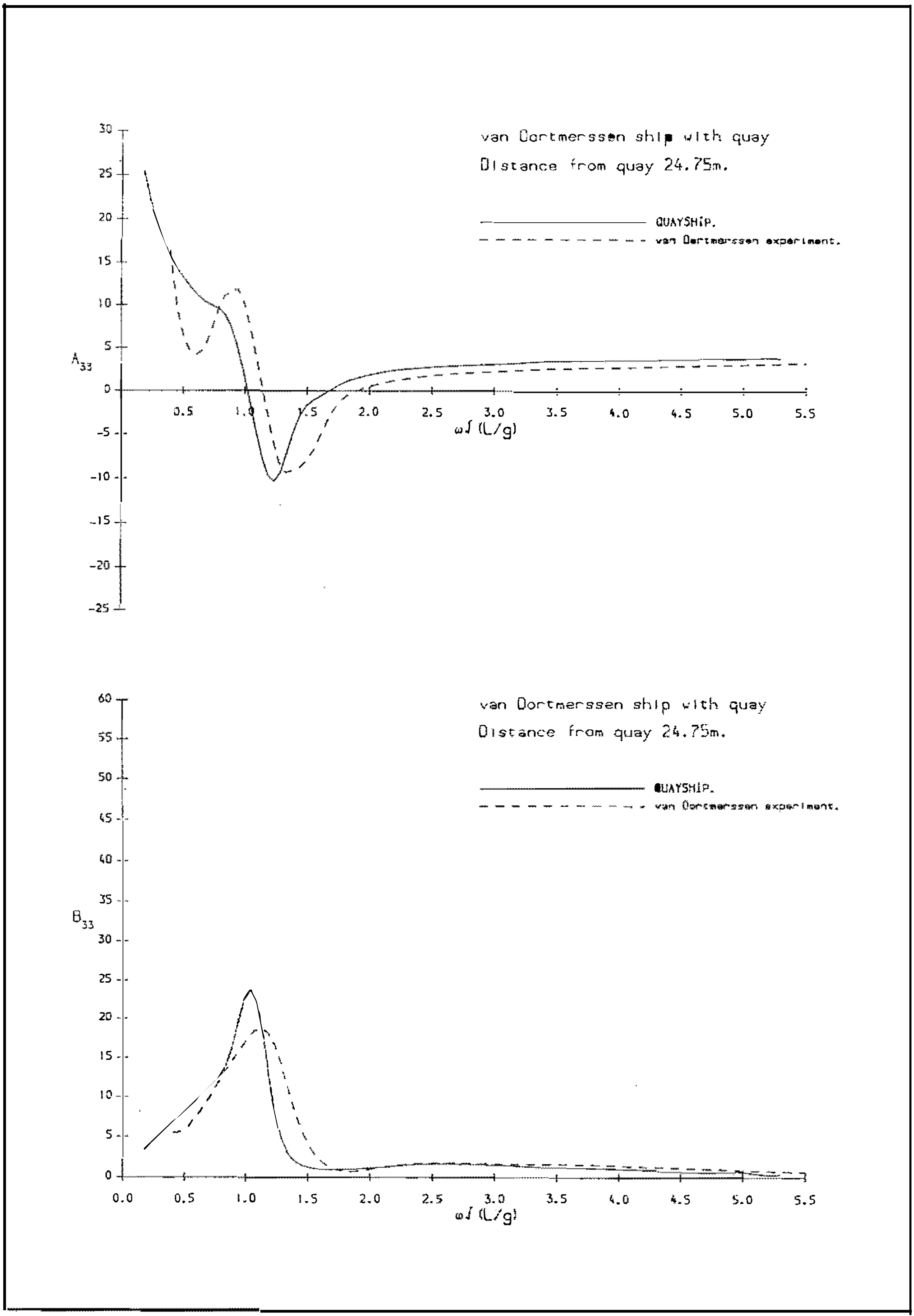


Fig 7(b) Added inertia and damping in heave–24.75m quay clearance.

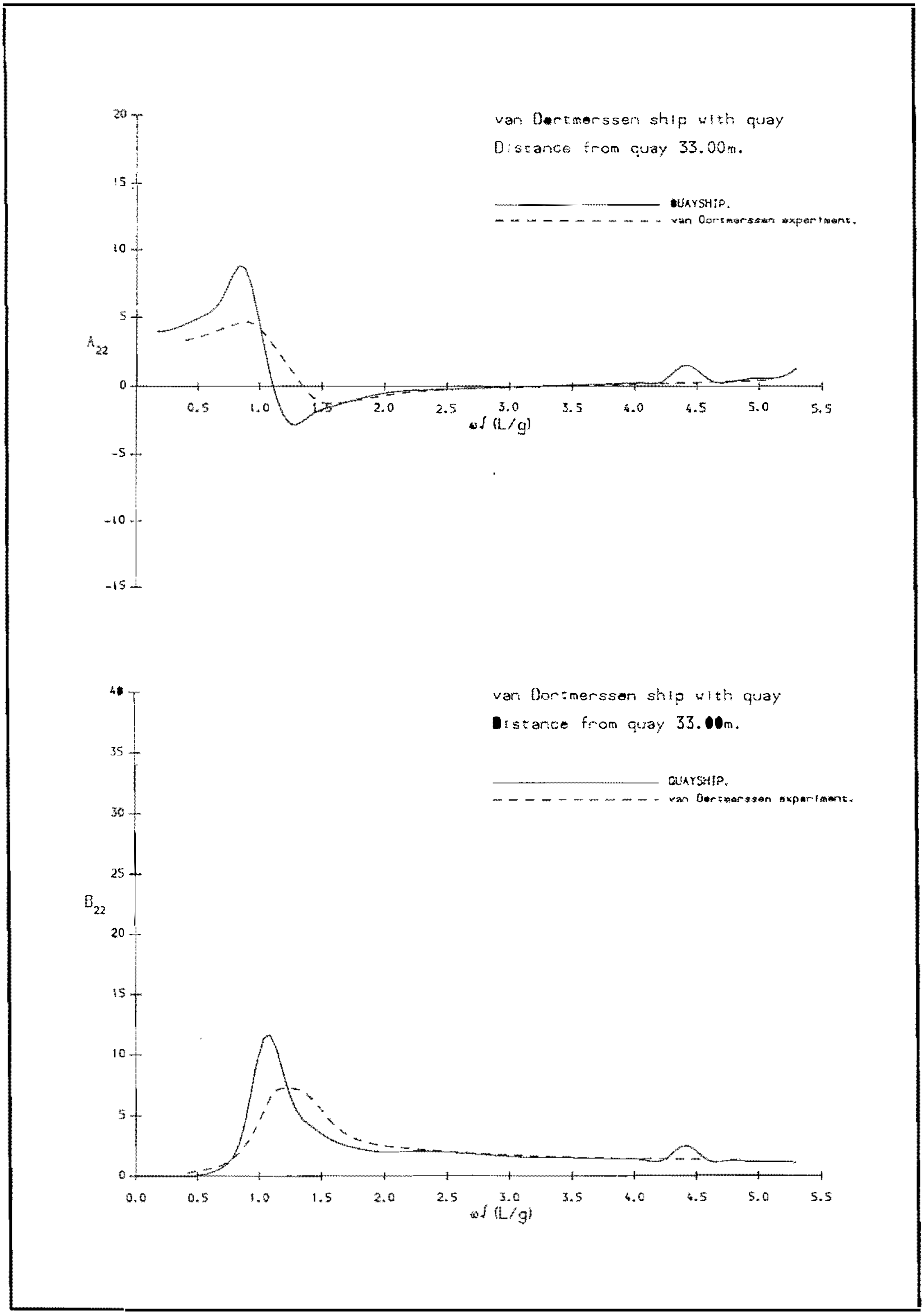


Fig 8(a) Added inertia and damping in sway-33.00m quay clearance.

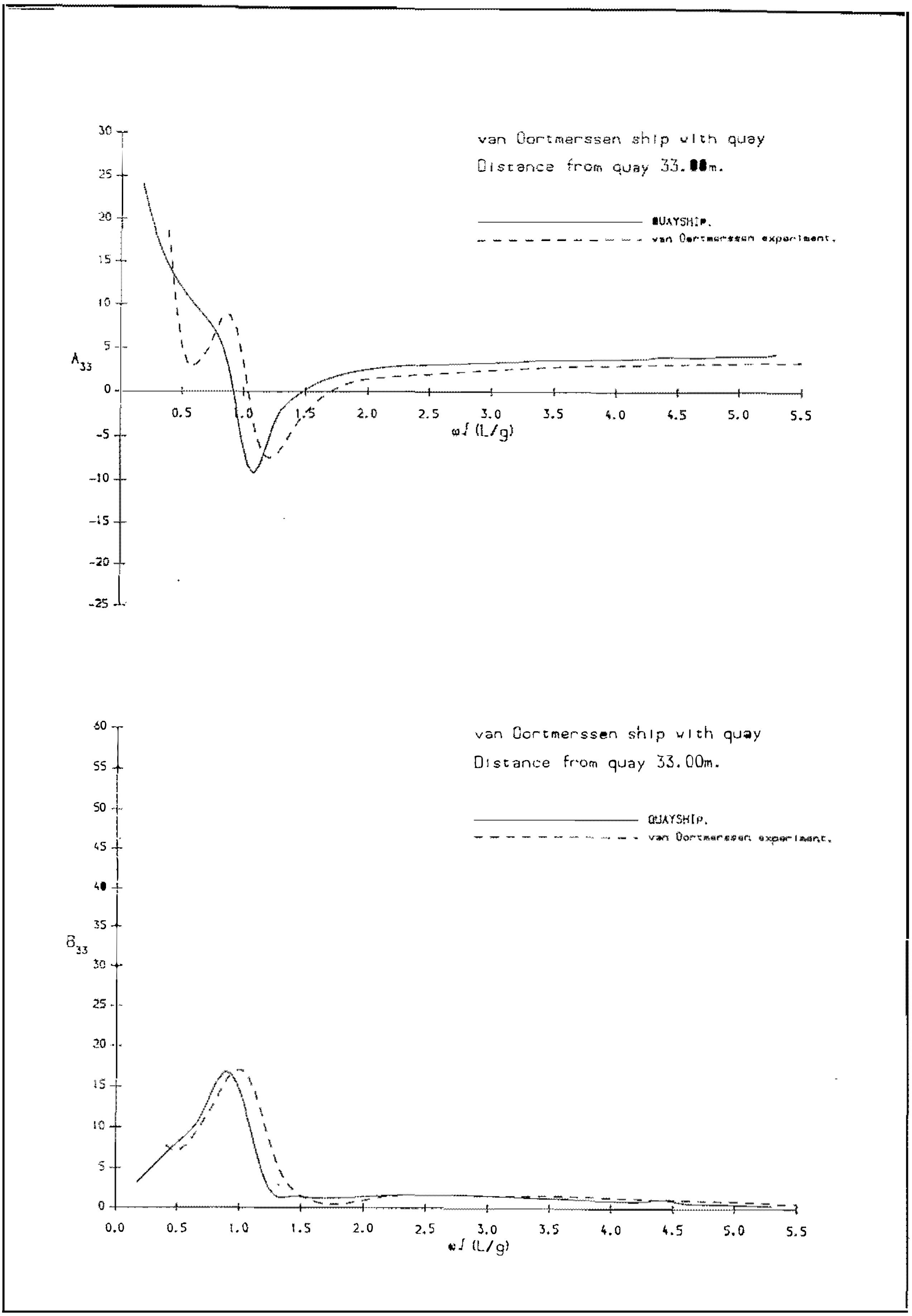


Fig 8(b) Added inertia and damping in heave-33.00m quay clearance.

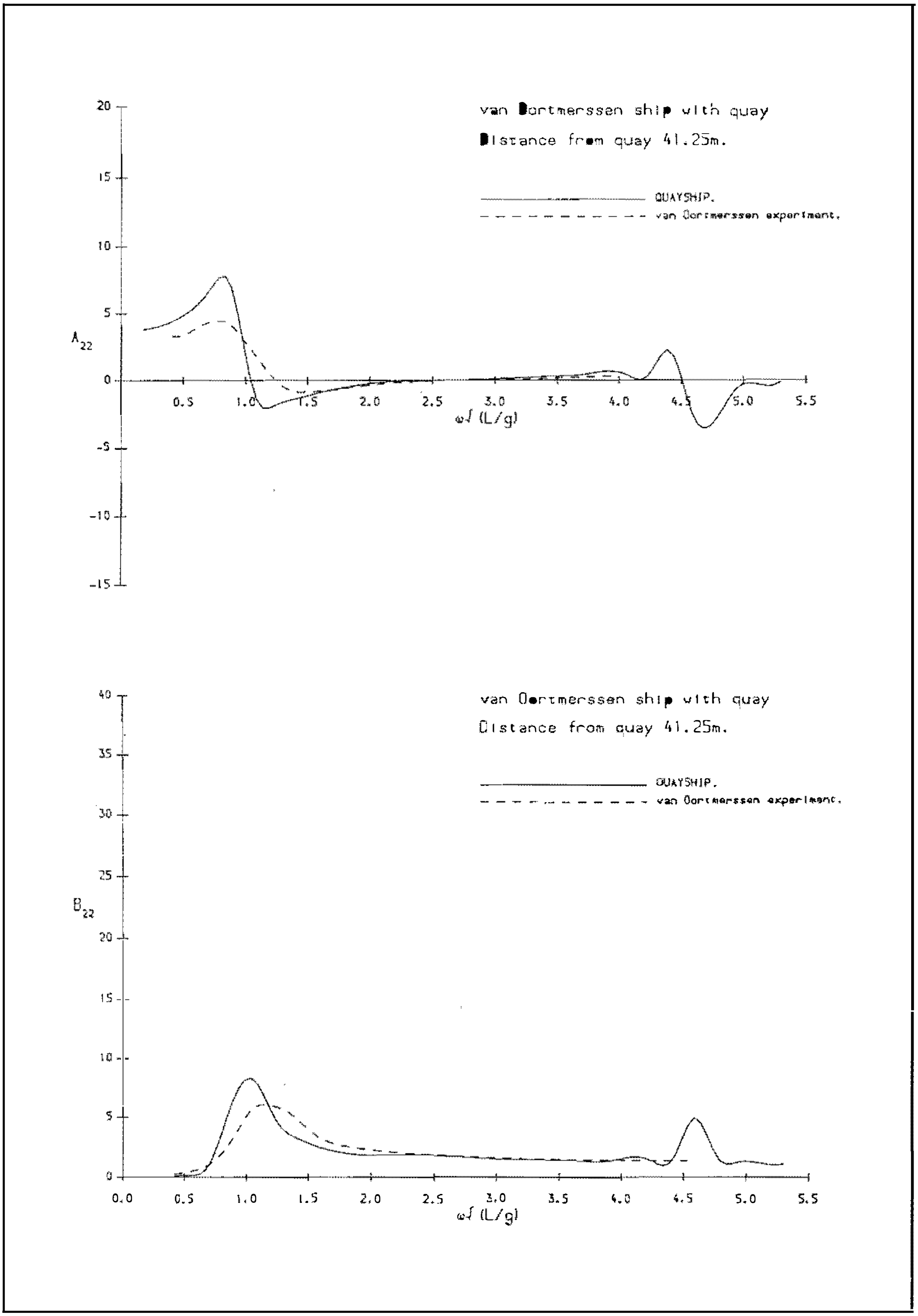


Fig 9(a) Added inertia and damping in sway-41.25m quay clearance.

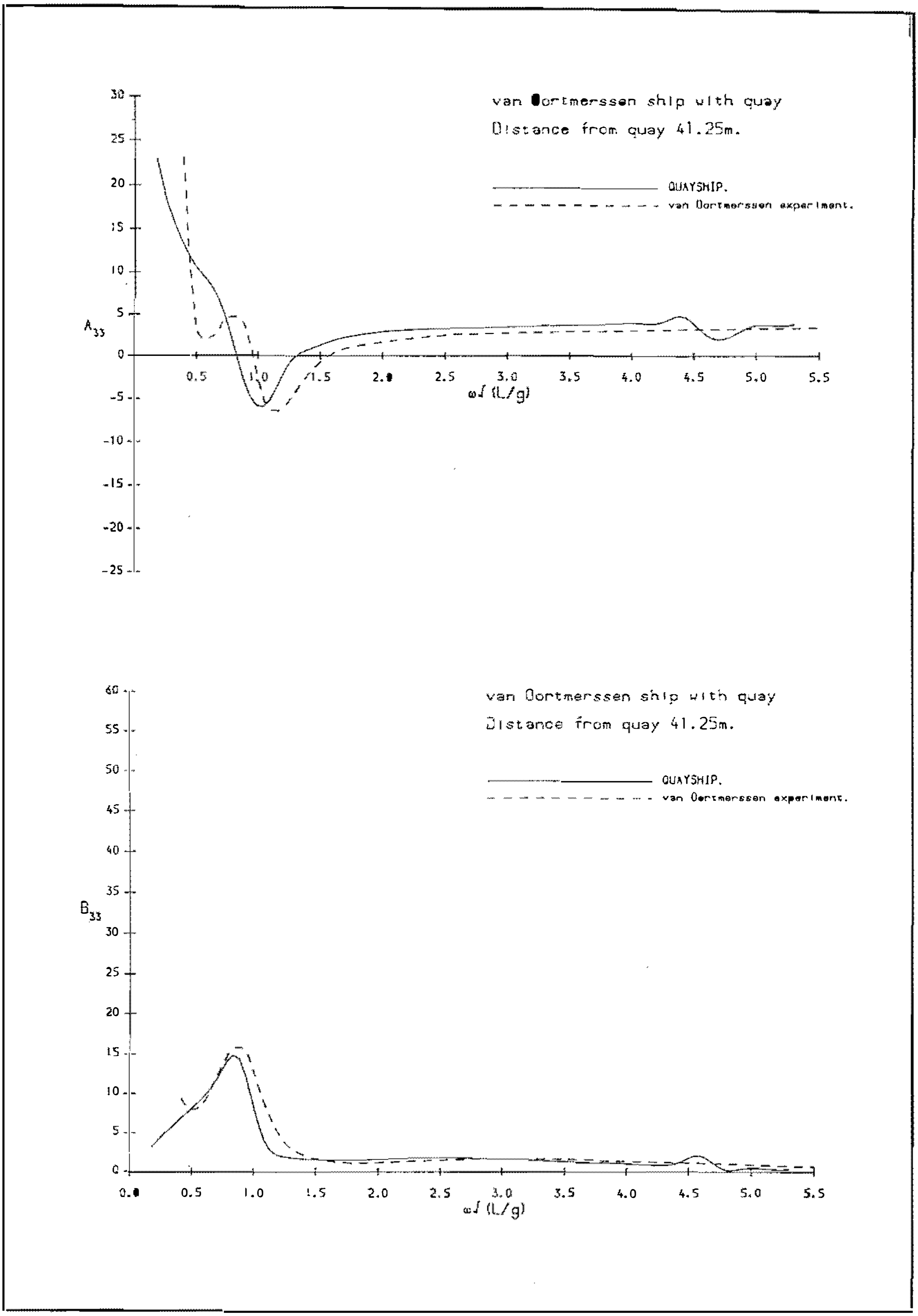


Fig 9(b) Added inertia and damping in heave-41.25m quay clearance.

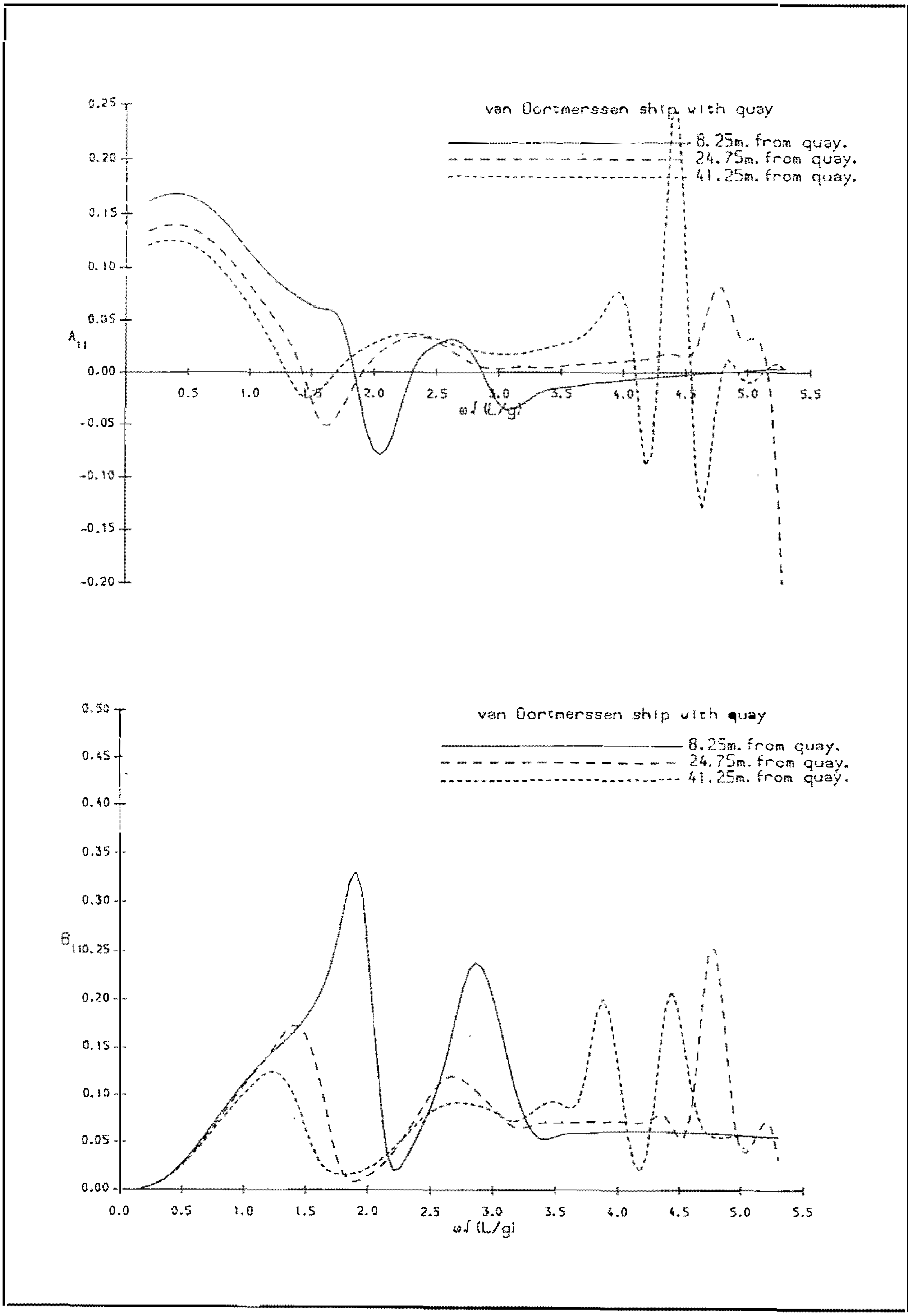


Fig 10 Added inertia and damping in surge predicted by QUAYSHIP.

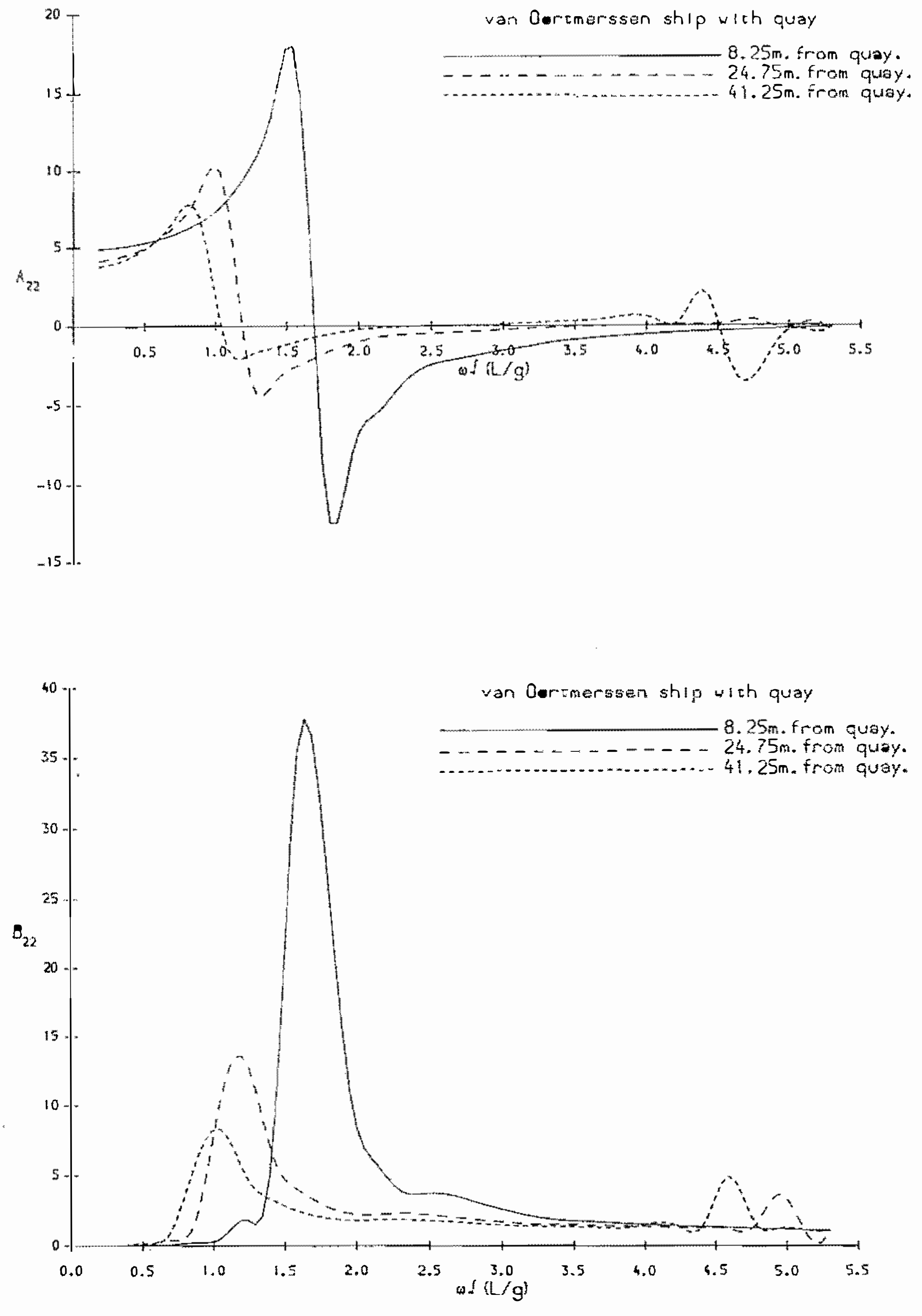


Fig 11 Added inertia and damping in sway predicted by QUAYSHIP.

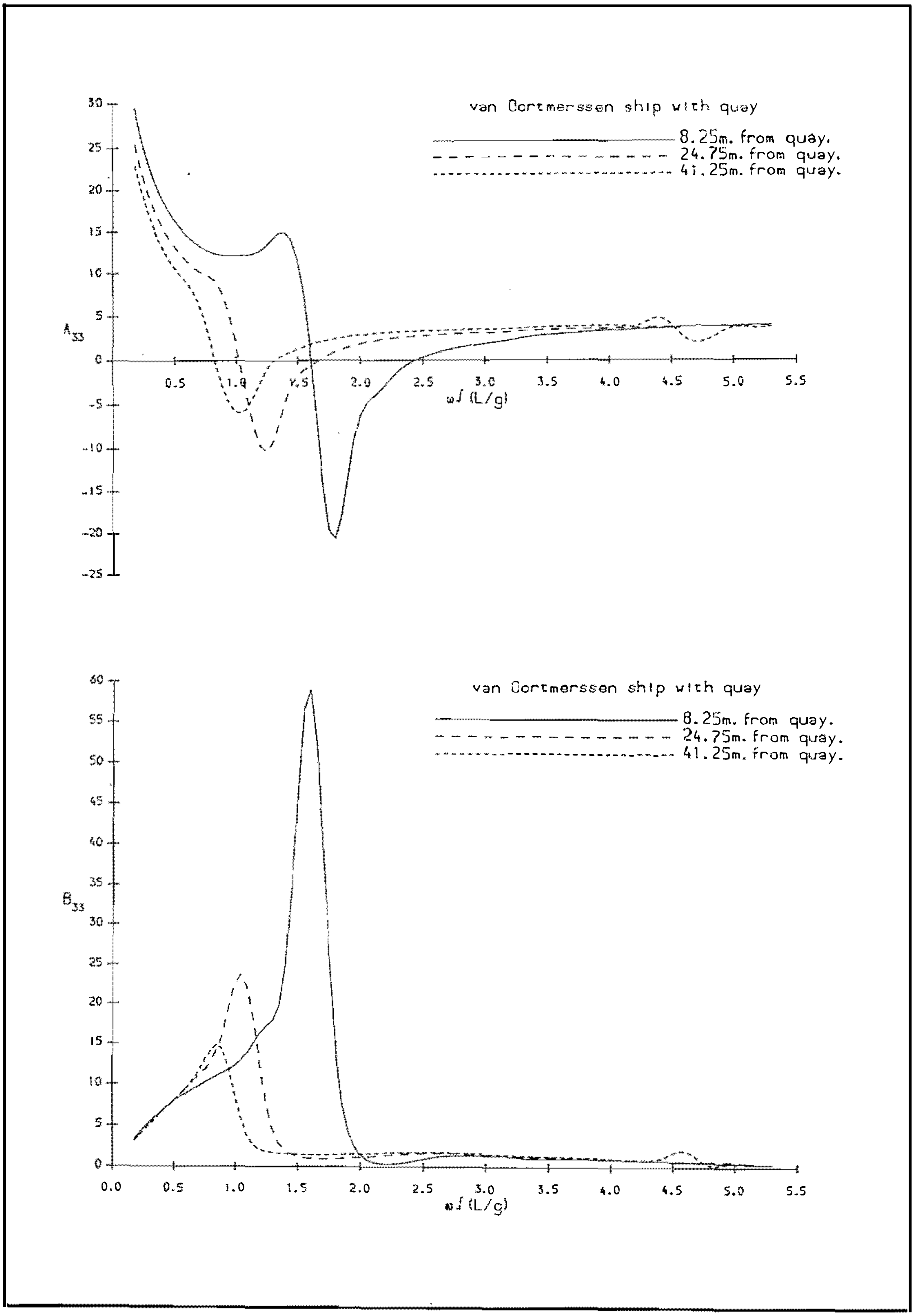


Fig 12 Added inertia and damping in heave predicted by QUAYSHIP.

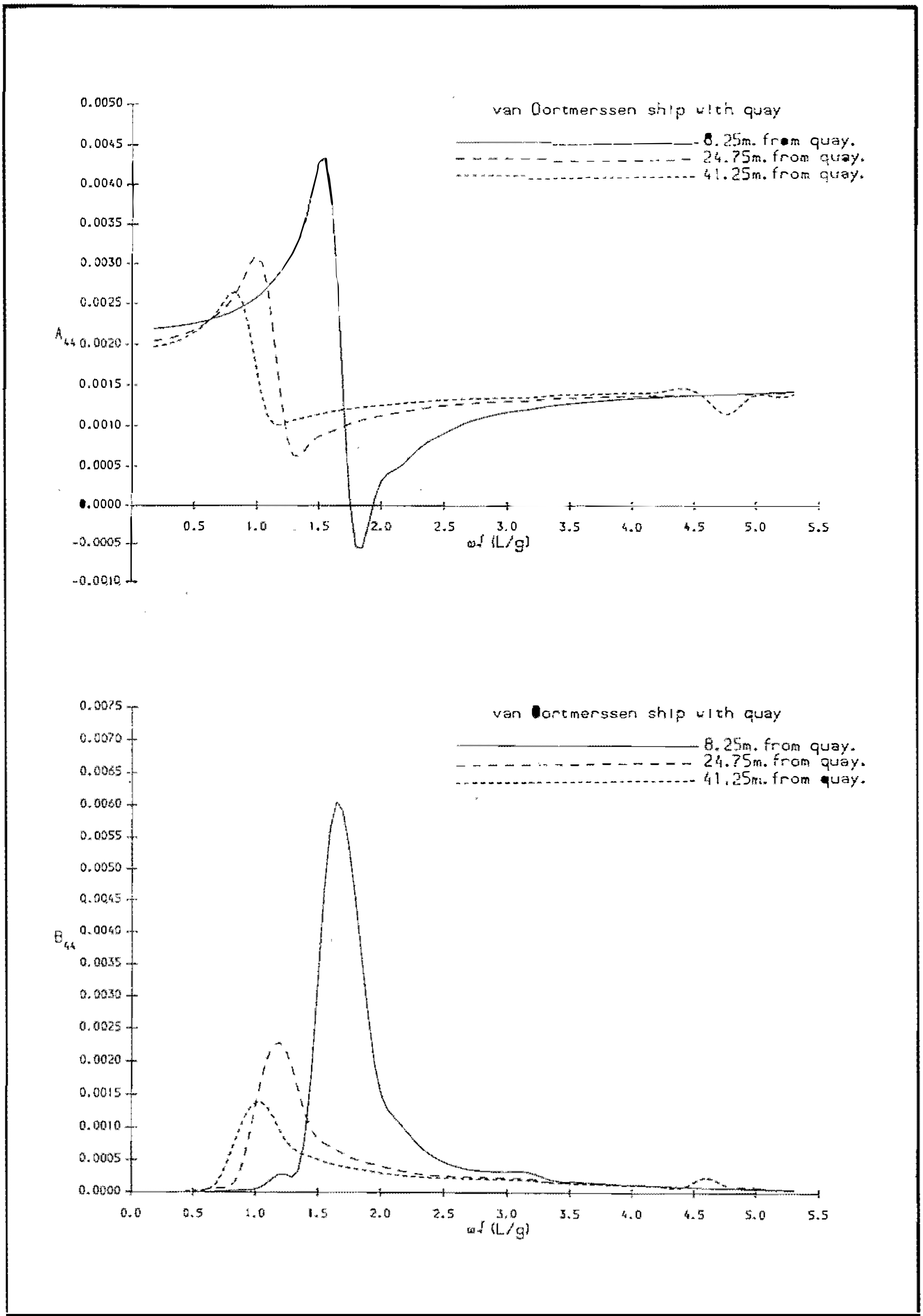


Fig 13 Added inertia and damping in roll predicted by QUAYSHIP.

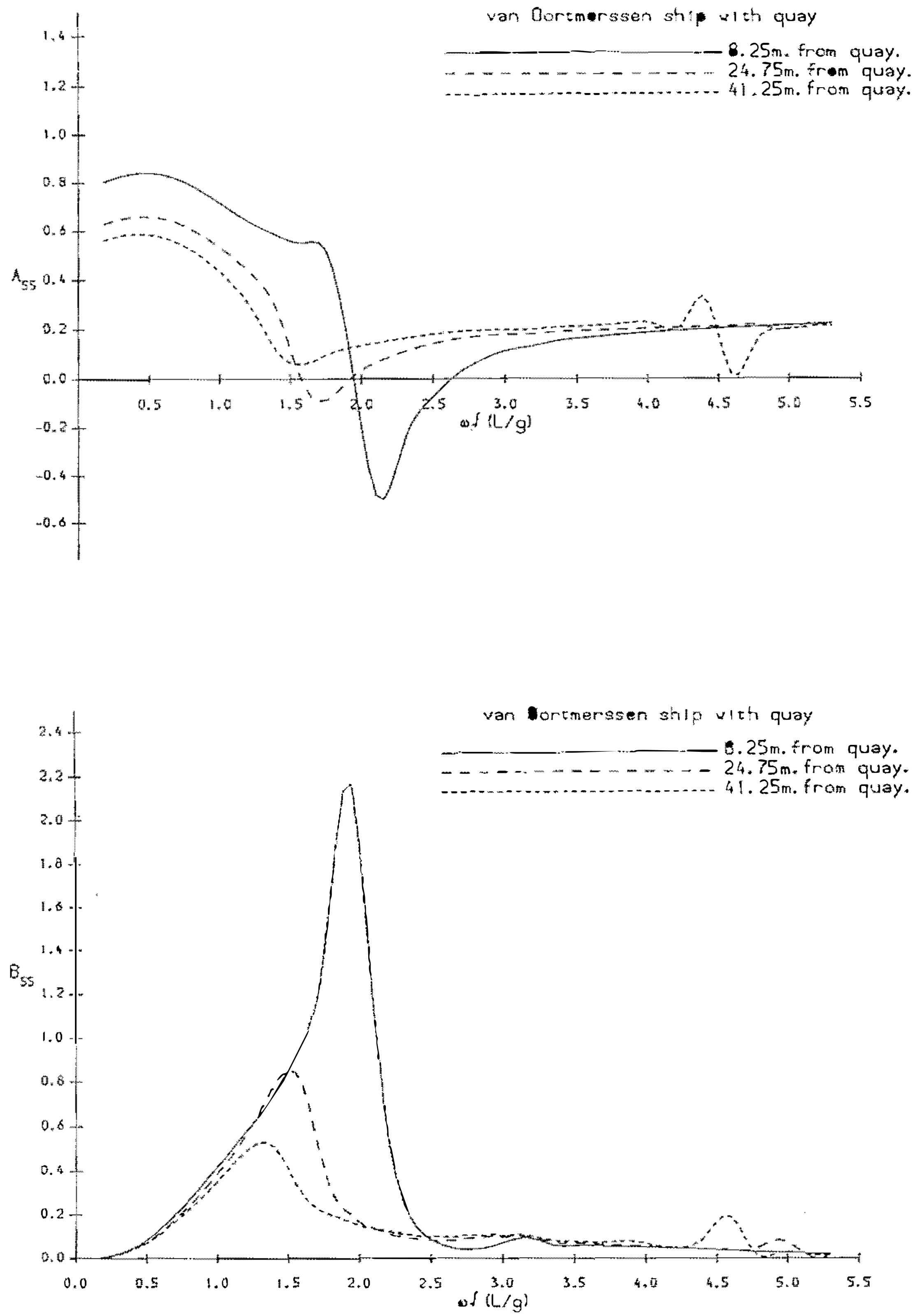


Fig 14 Added inertia and damping in pitch predicted by QUAYSHIP.

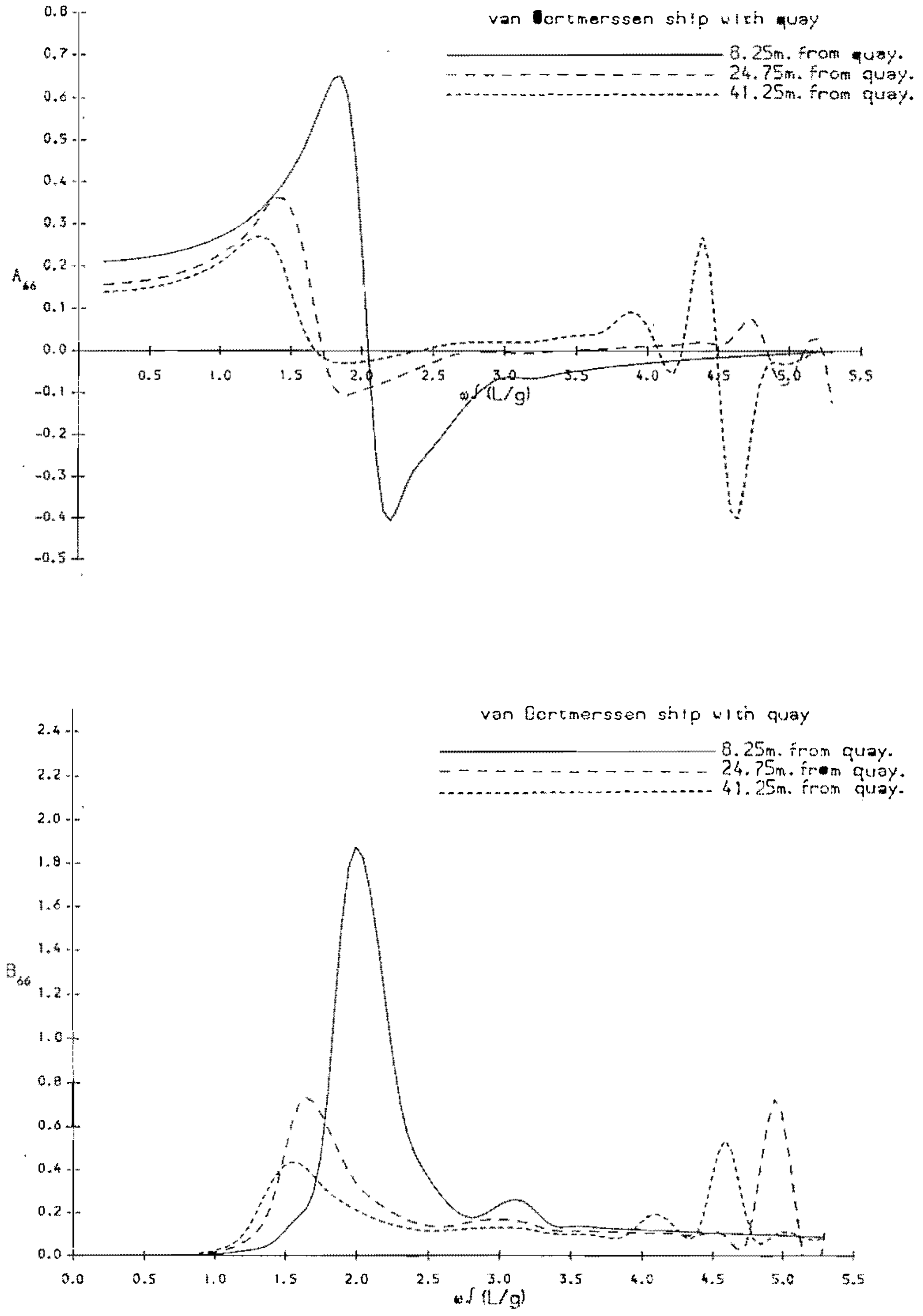


Fig 15 Added inertia and damping in Yaw predicted by QUAYSHIP.

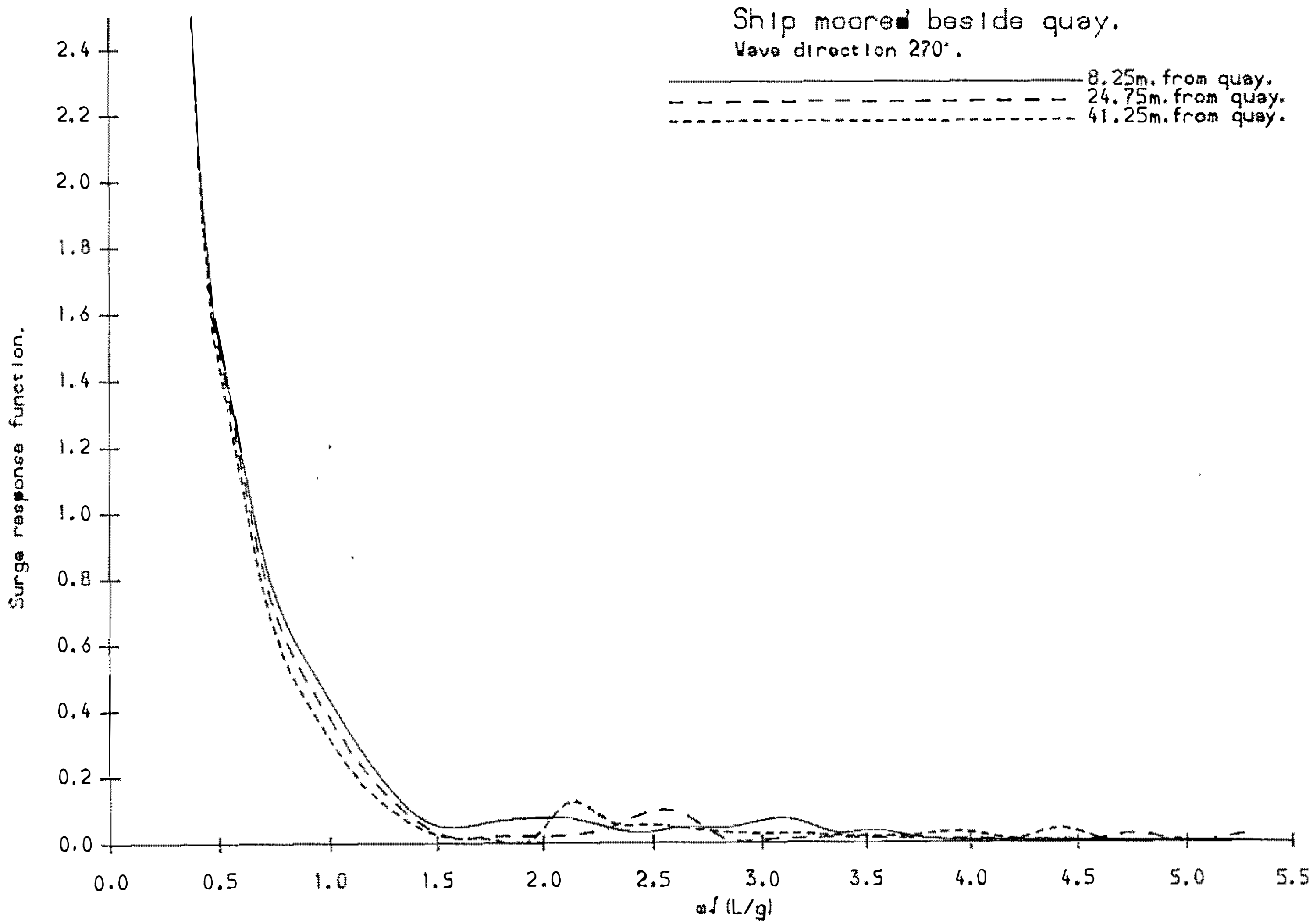
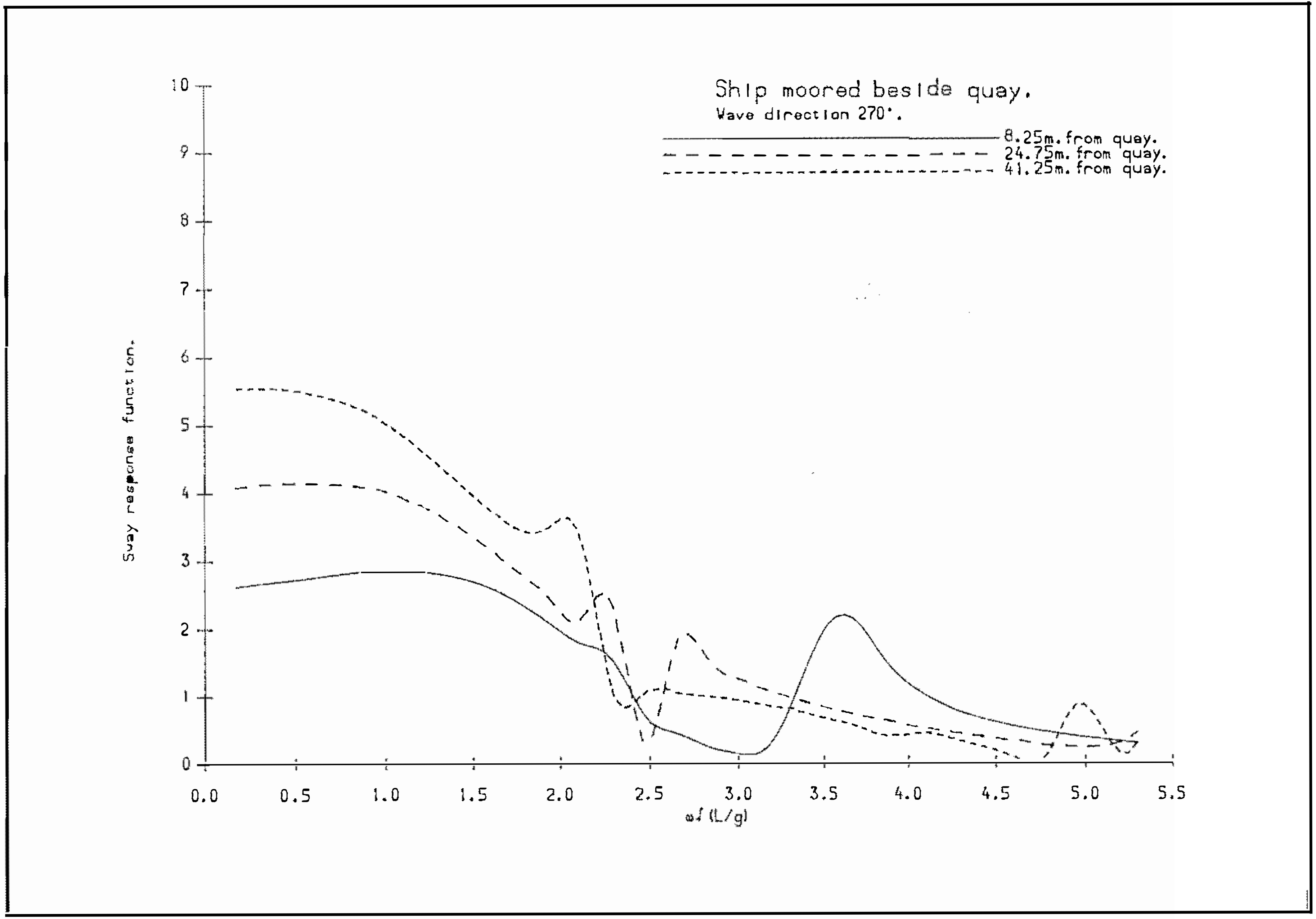


Fig 16(a) Surge-270° wave direction (beam sea).

Fig 16(b) Sway-270° wave direction (beam sea)



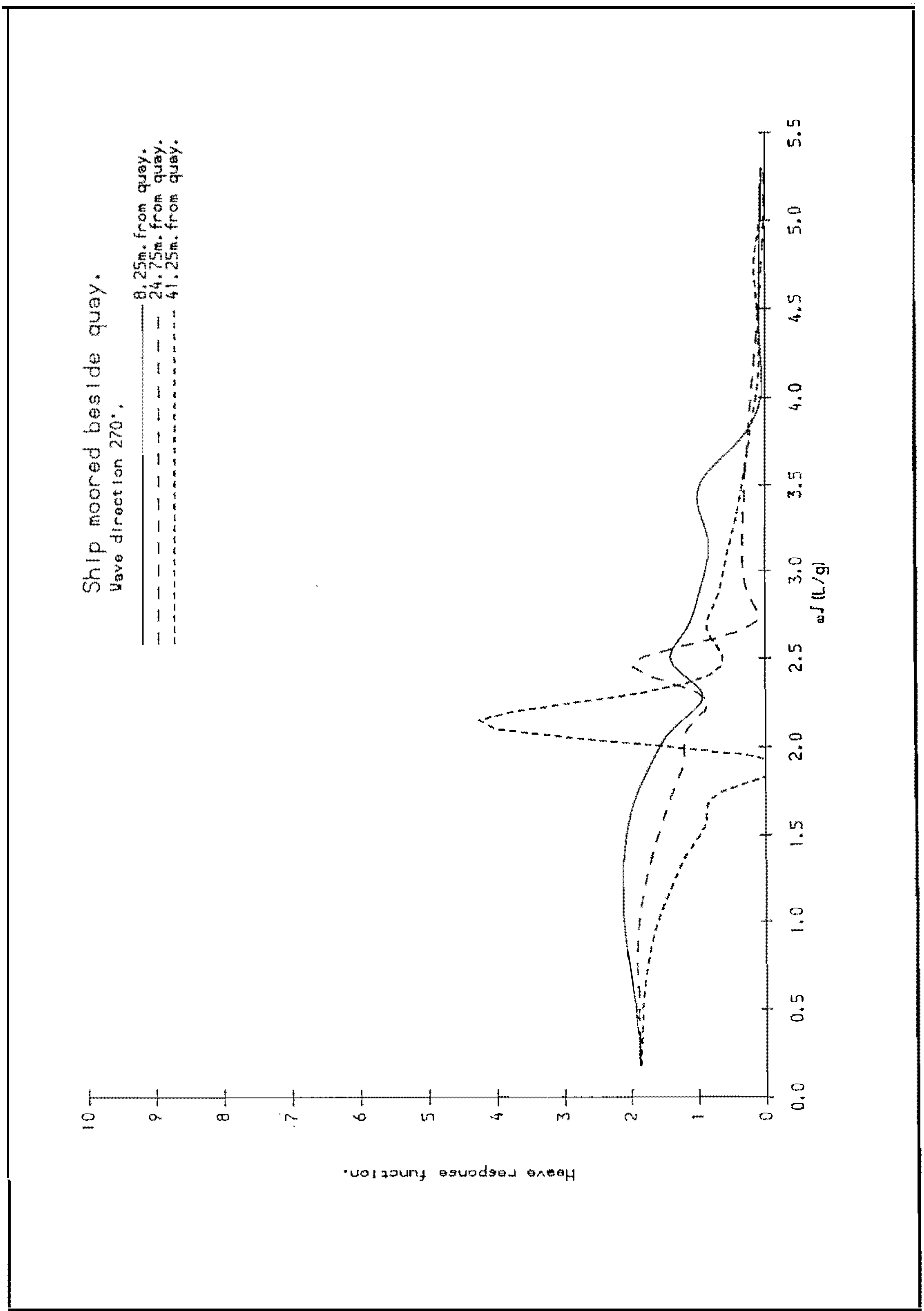


Fig 16(c) Heave-270°wave direction (beam sea)

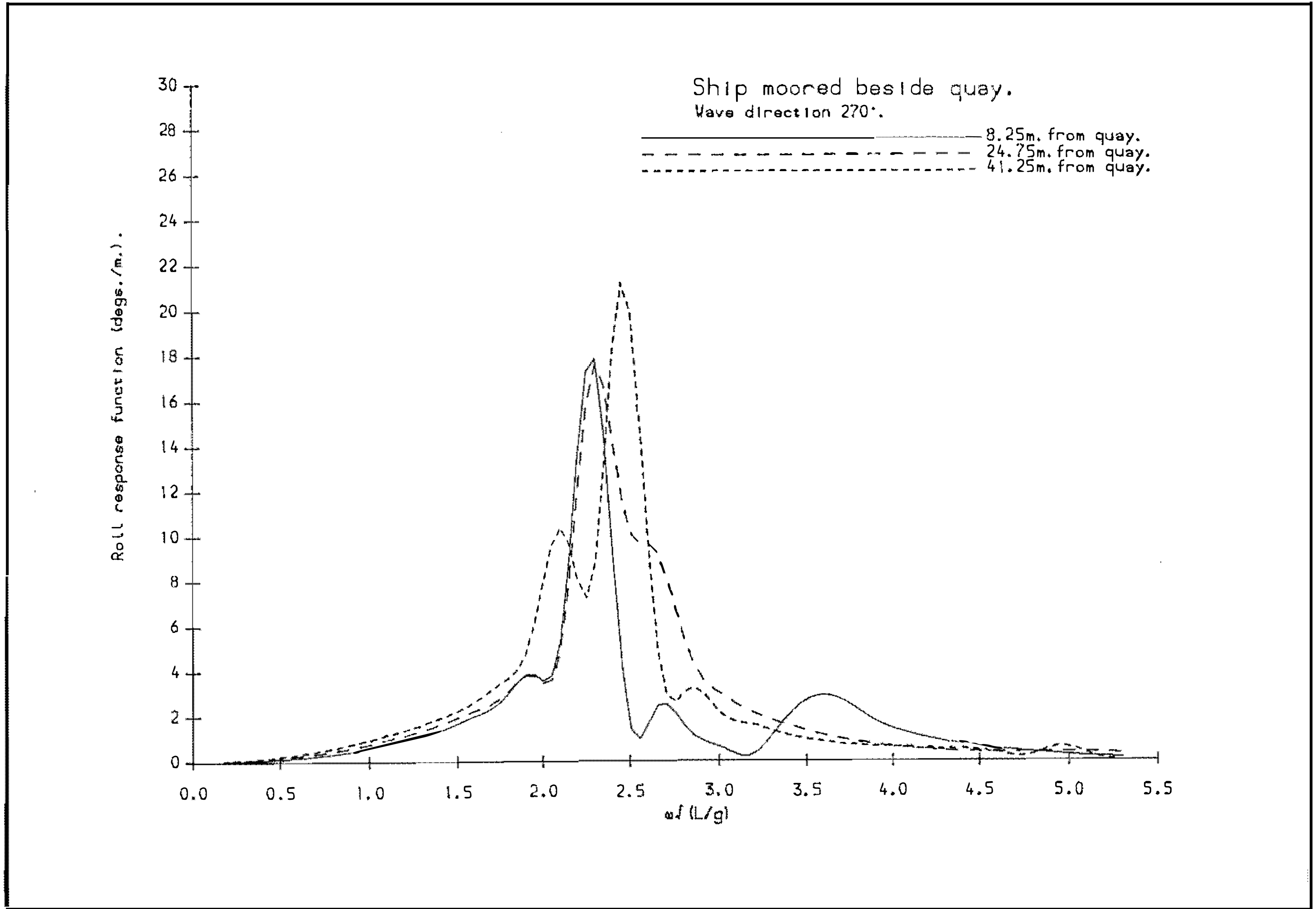


Fig 16(d) Roll–270° wave direction (beam sea).

Fig 16(e) Pitch-270° wave direction (beam sea)

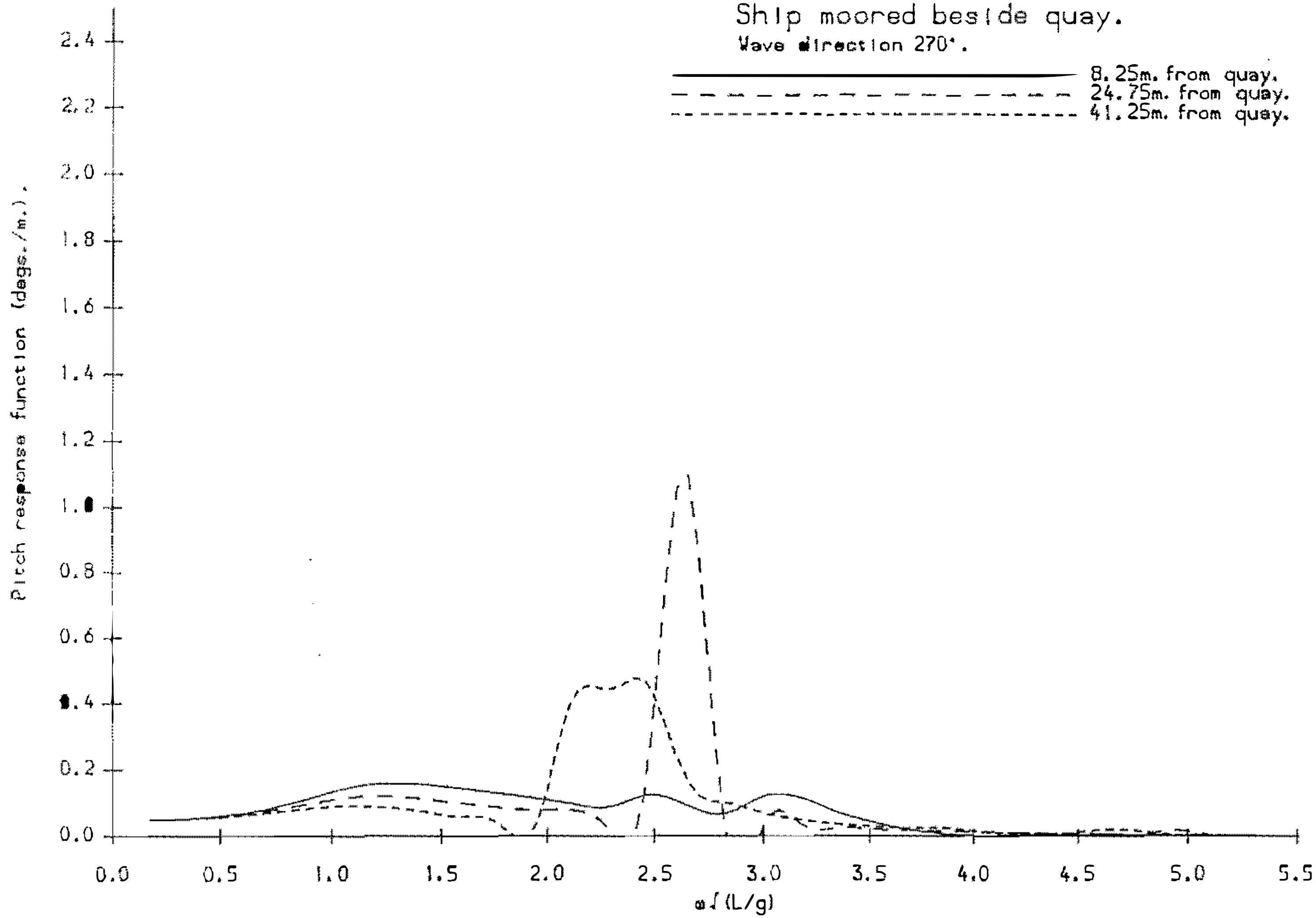
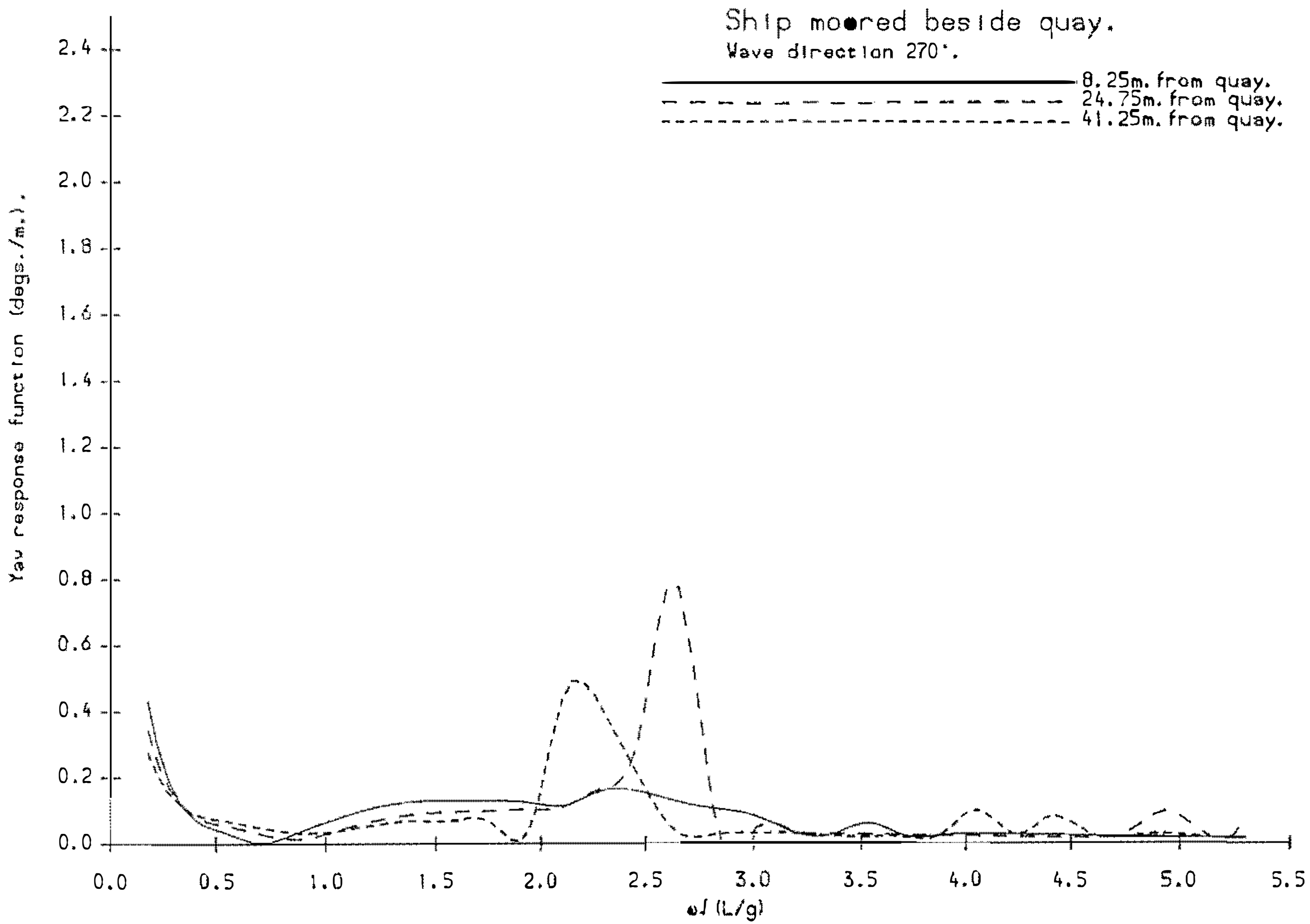


Fig 16(f) Yaw-270° wave direction (beam sea)



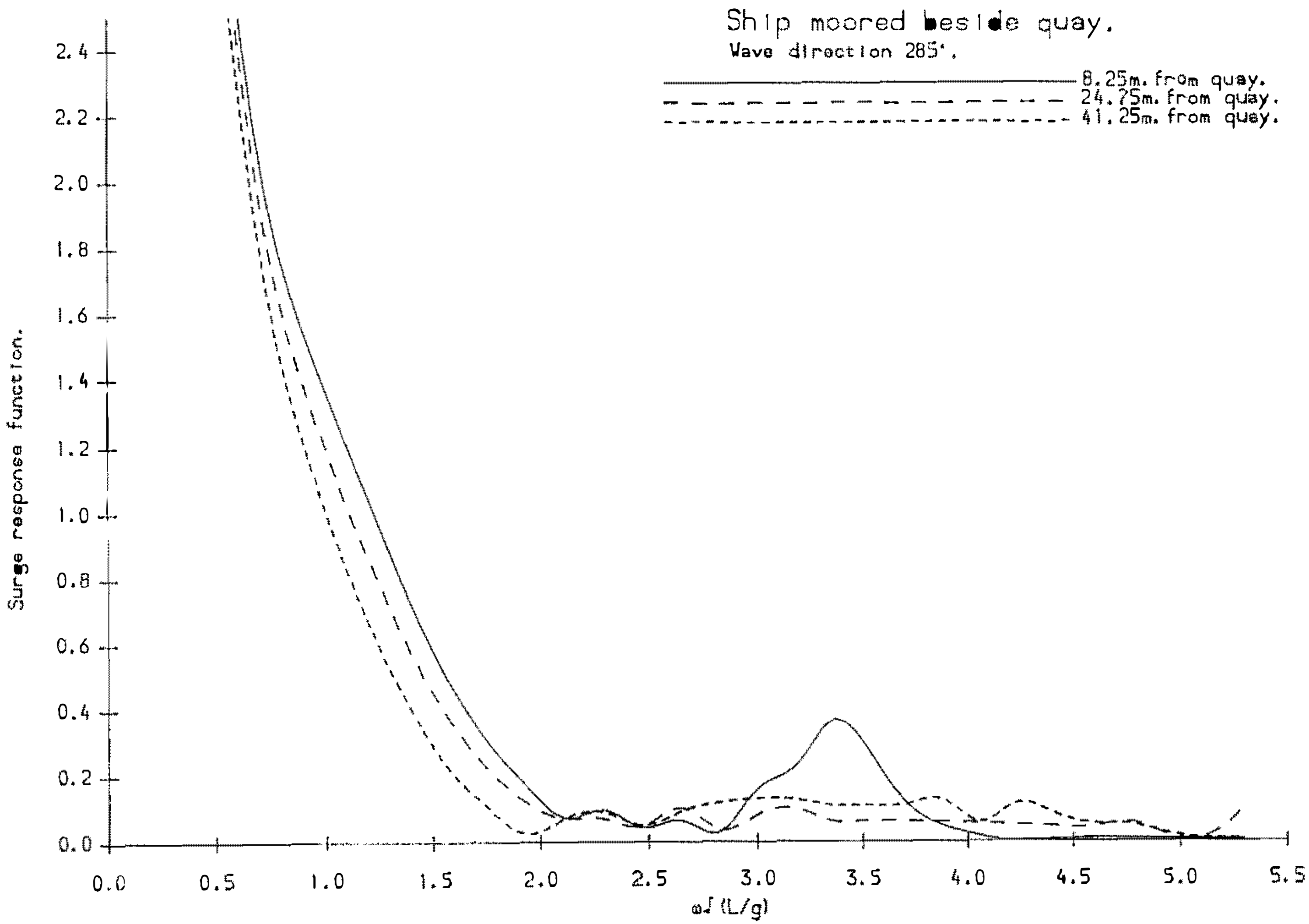


Fig 17(a) Surge-285° wave direction

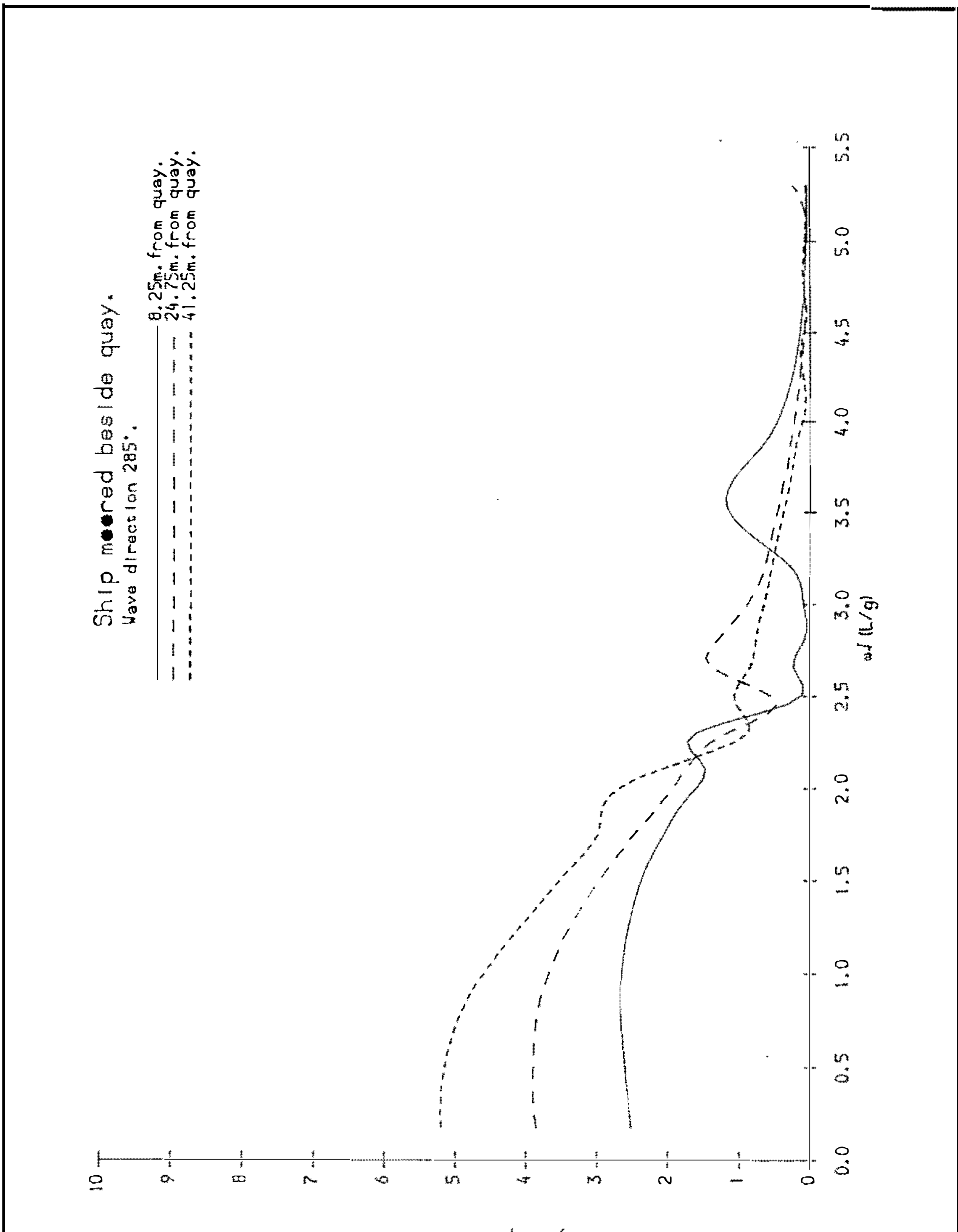


Fig 17(b) Sway-285° wave direction

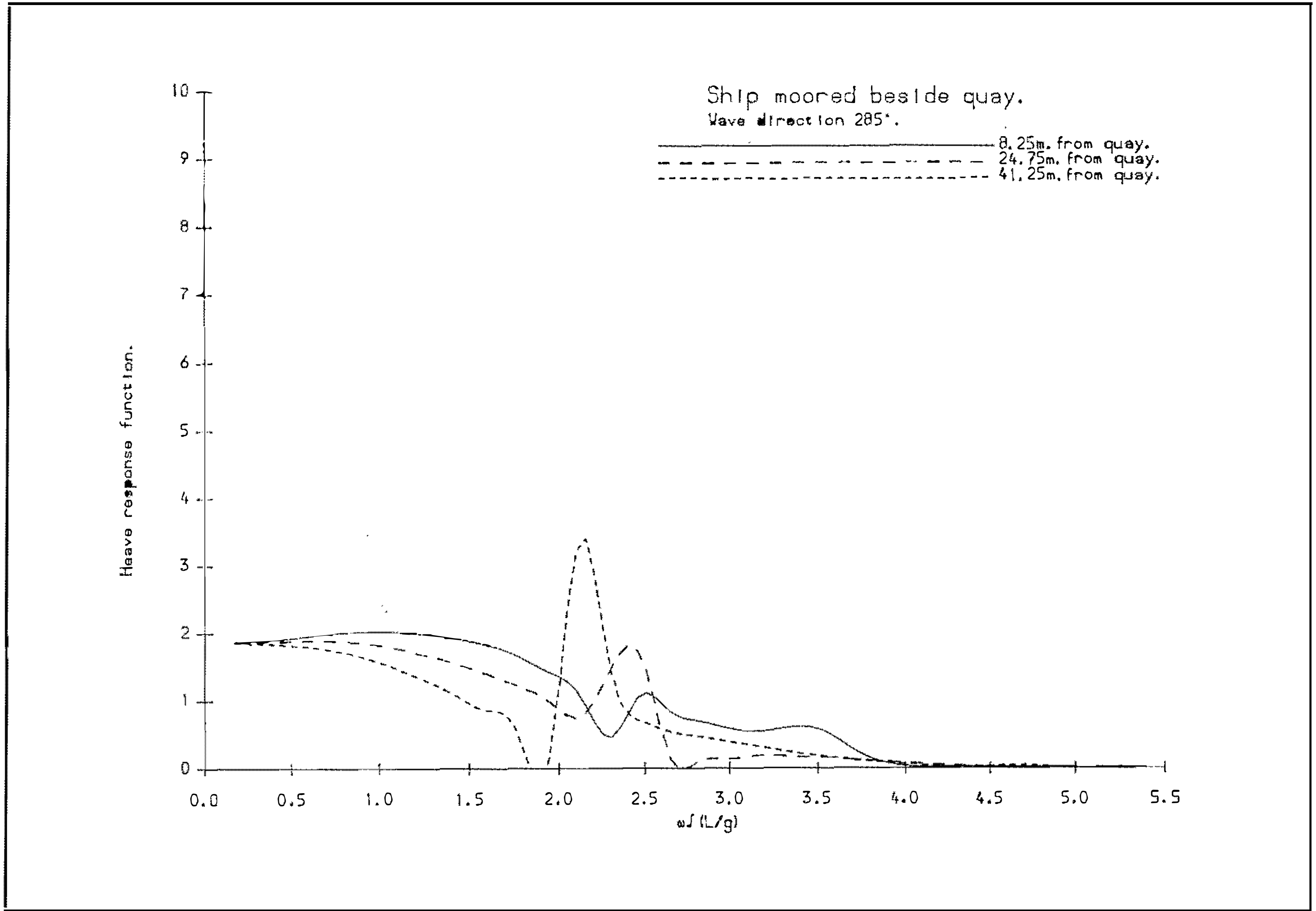


Fig 17(c) Heave-285° wave direction

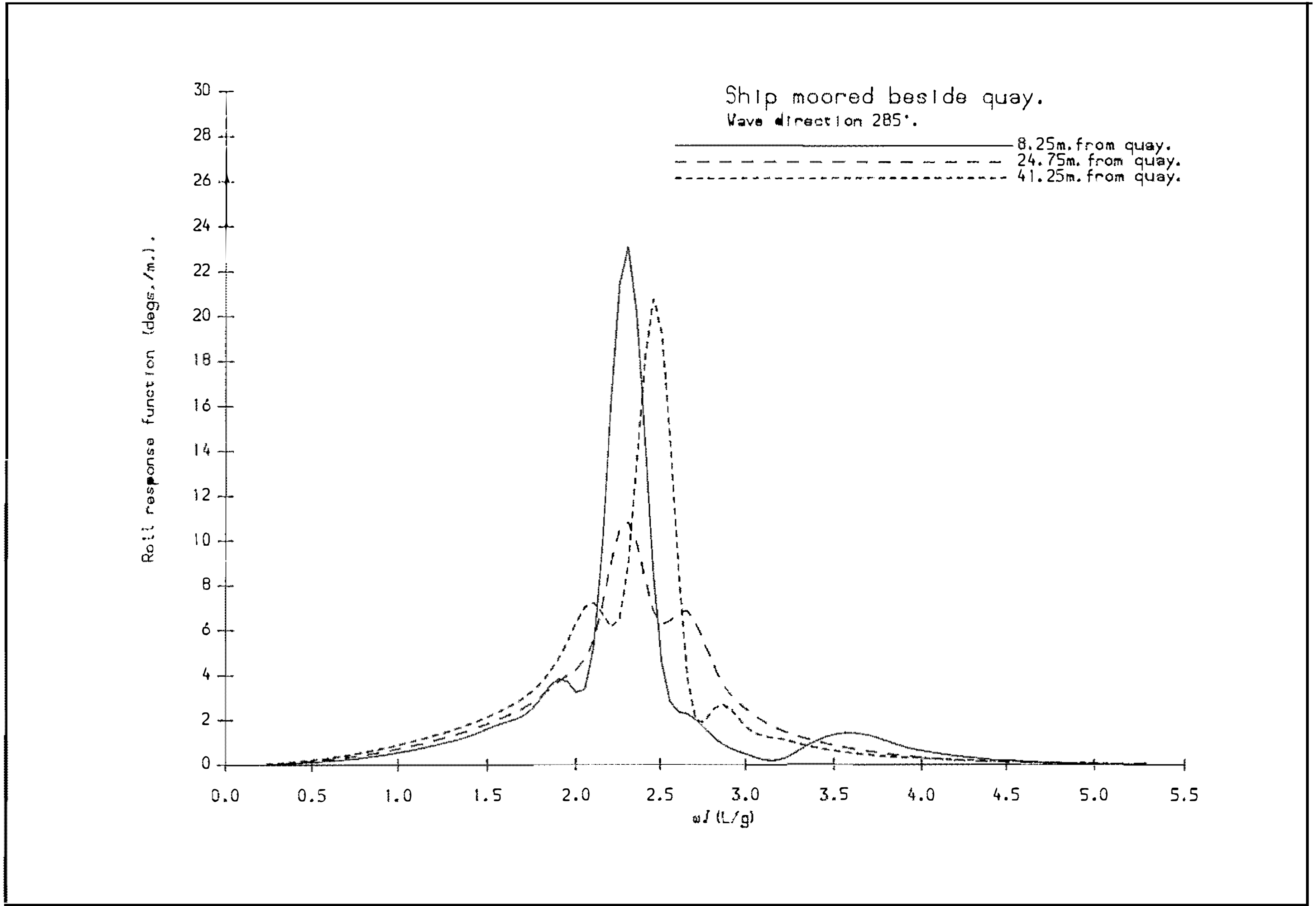


Fig 17(d) Roll-285° wave direction

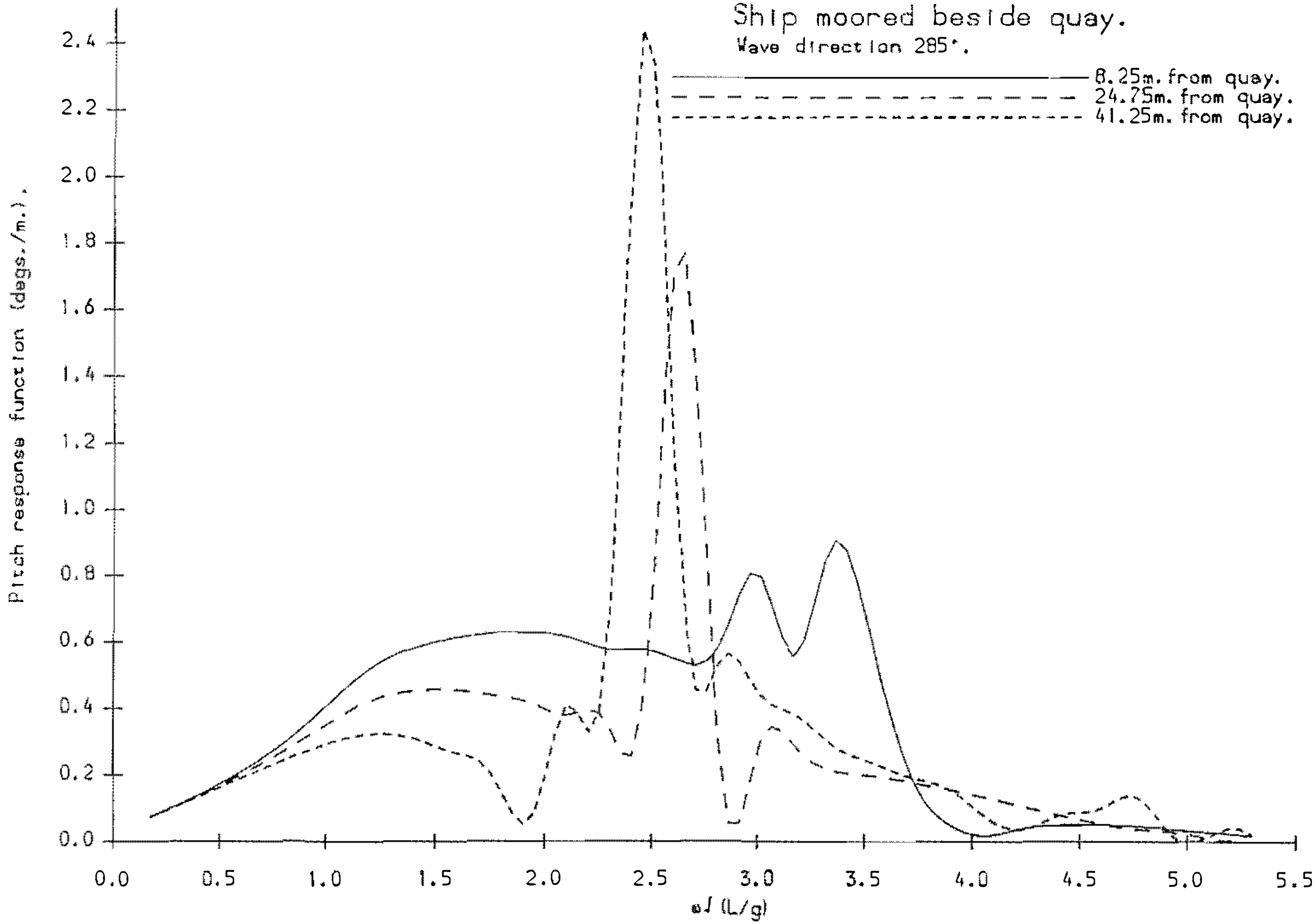


Fig 17(e) Pitch-285° wave direction

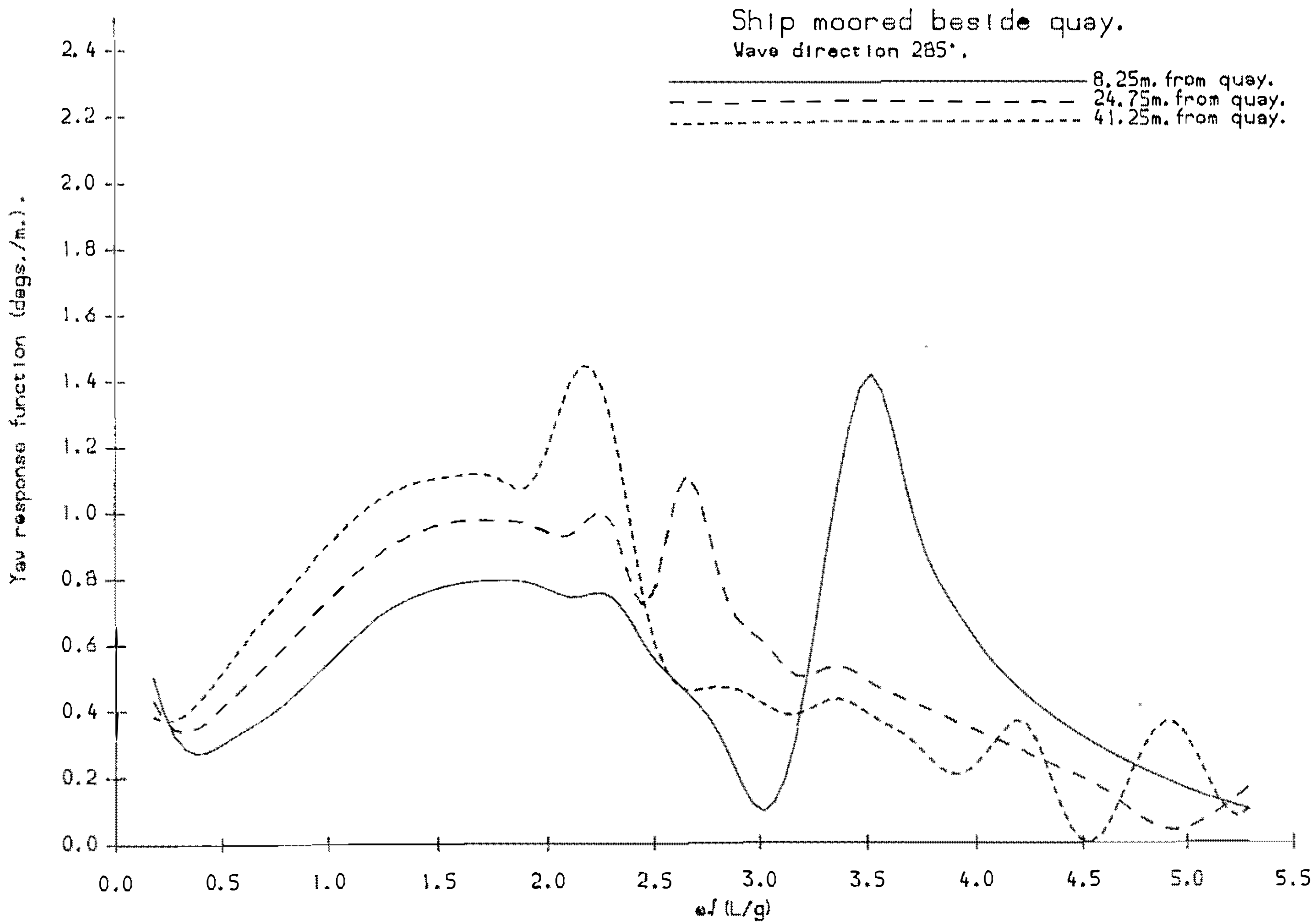


Fig 17(f) Yaw-285° wave direction

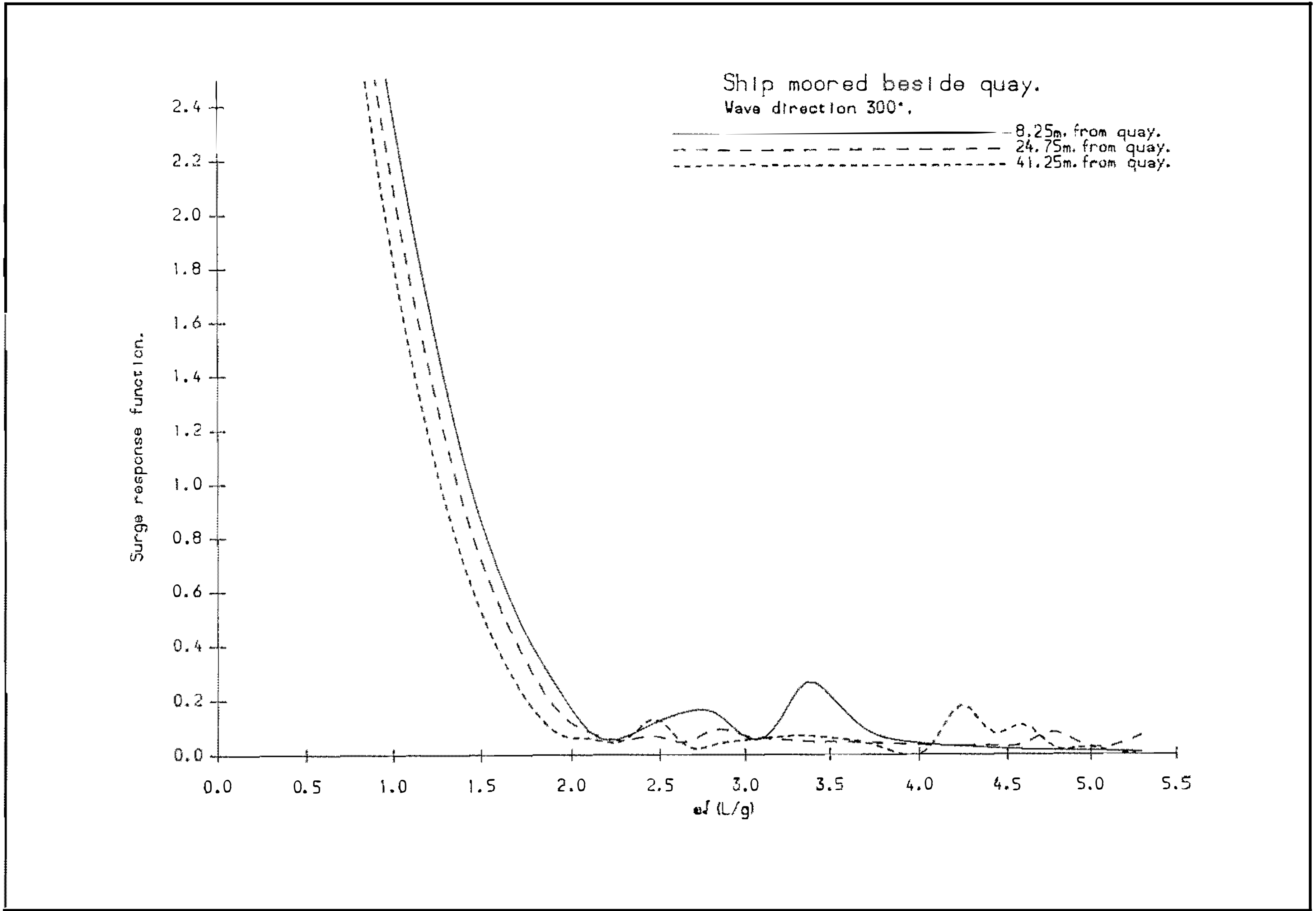


Fig 18(a) Surge-300° wave direction

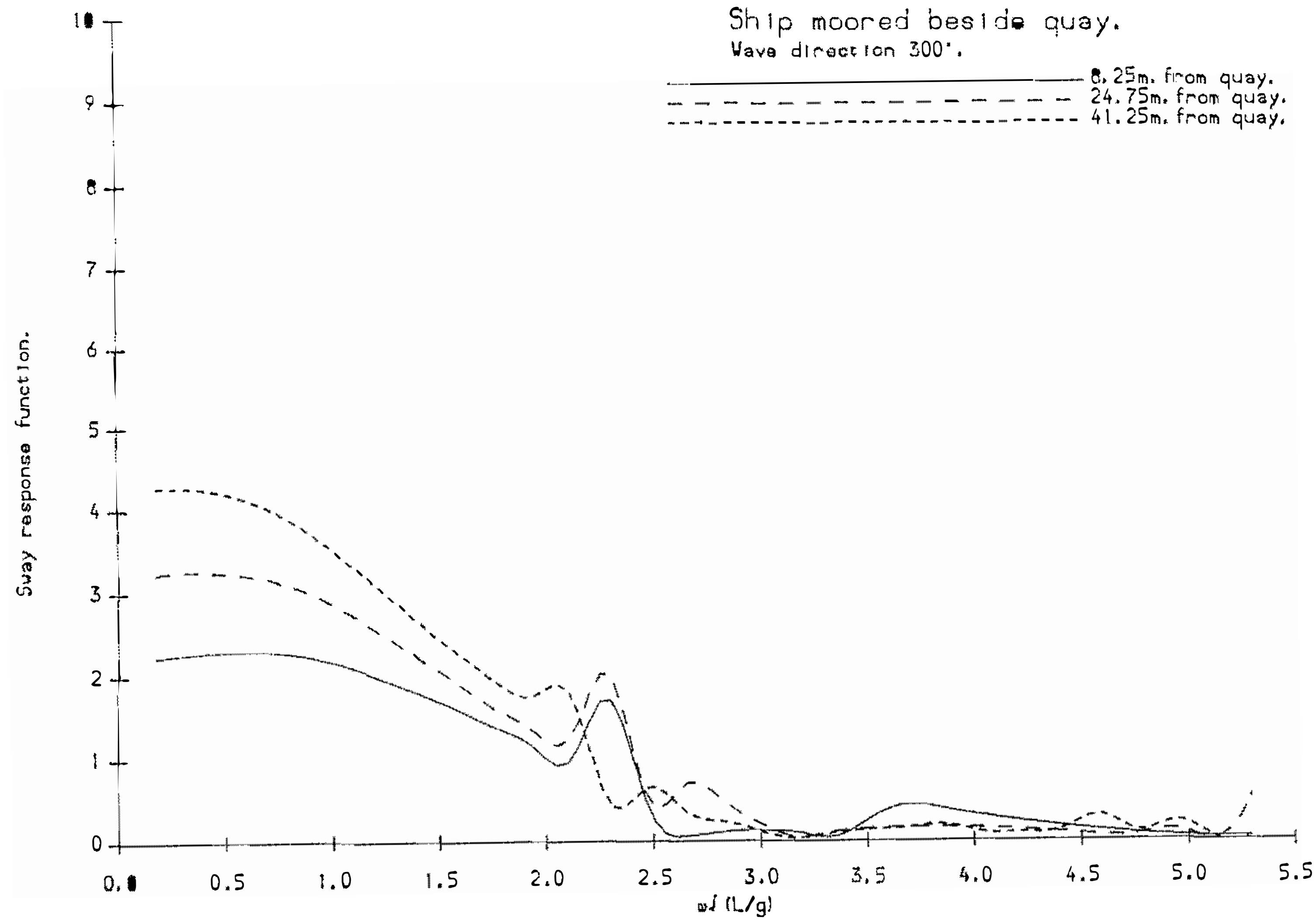


Fig 18(b) Sway-300° wave direction

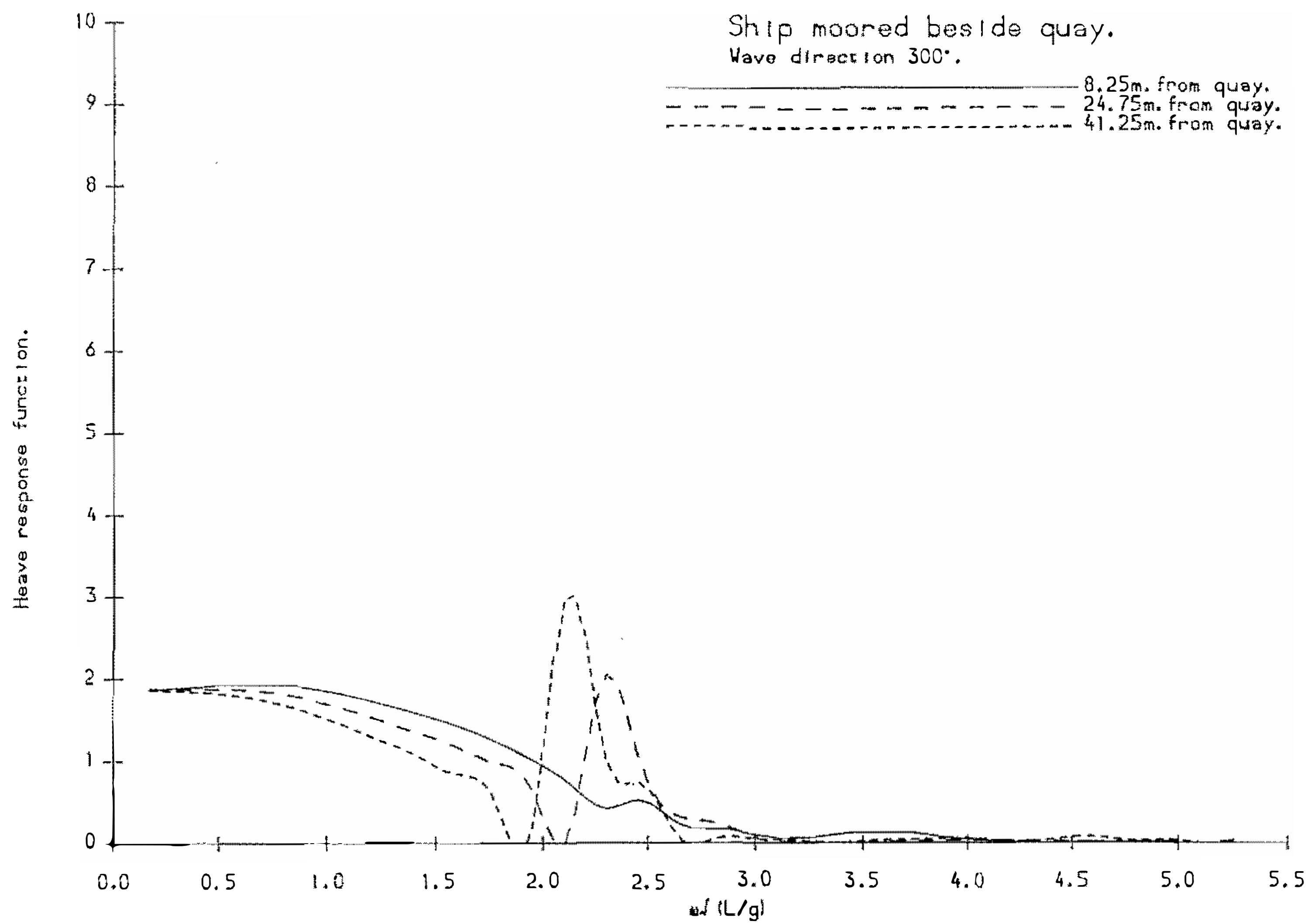
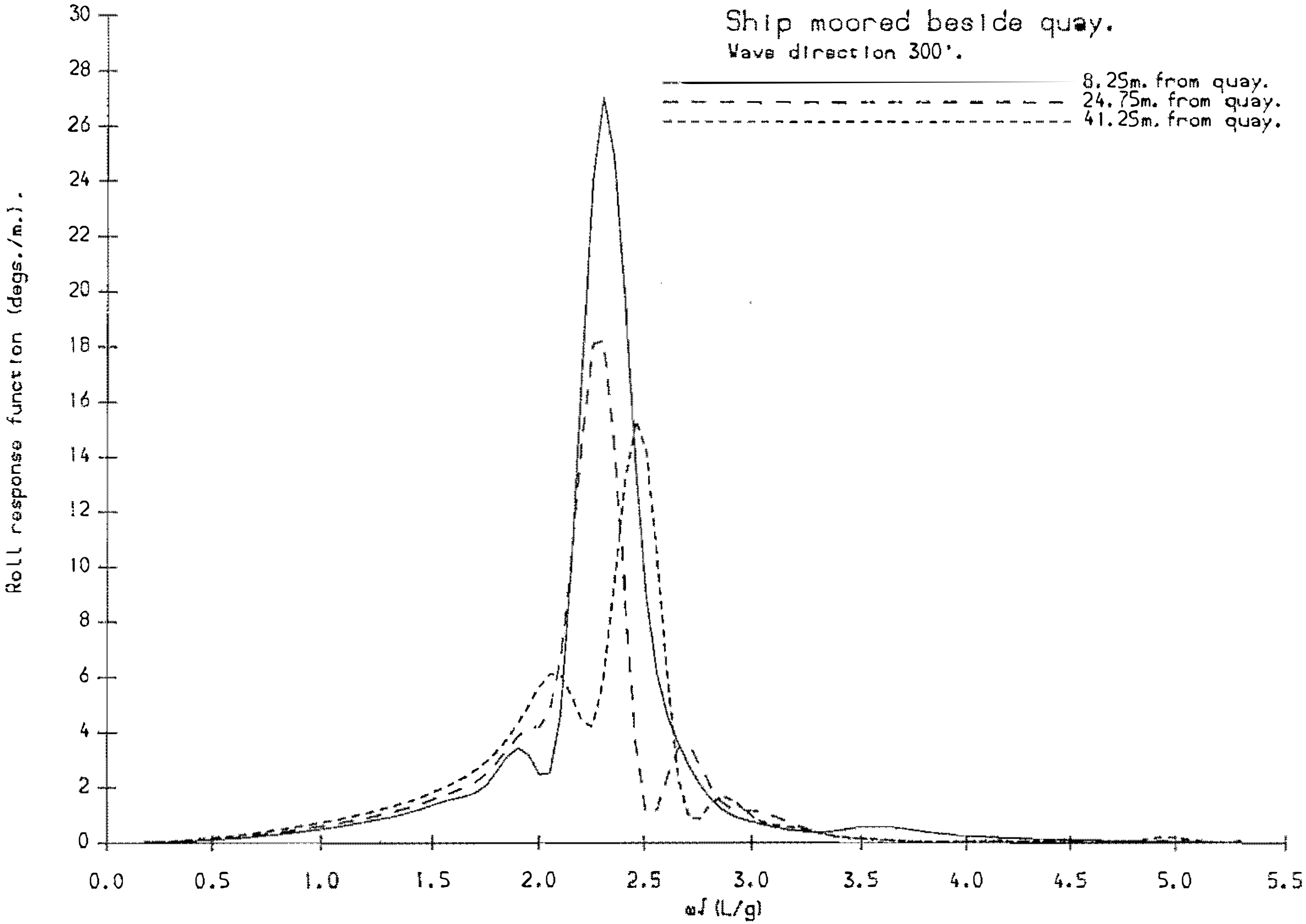


Fig 18(c) Heave-300° wave direction

Fig 18(d) Roll-300° wave direction



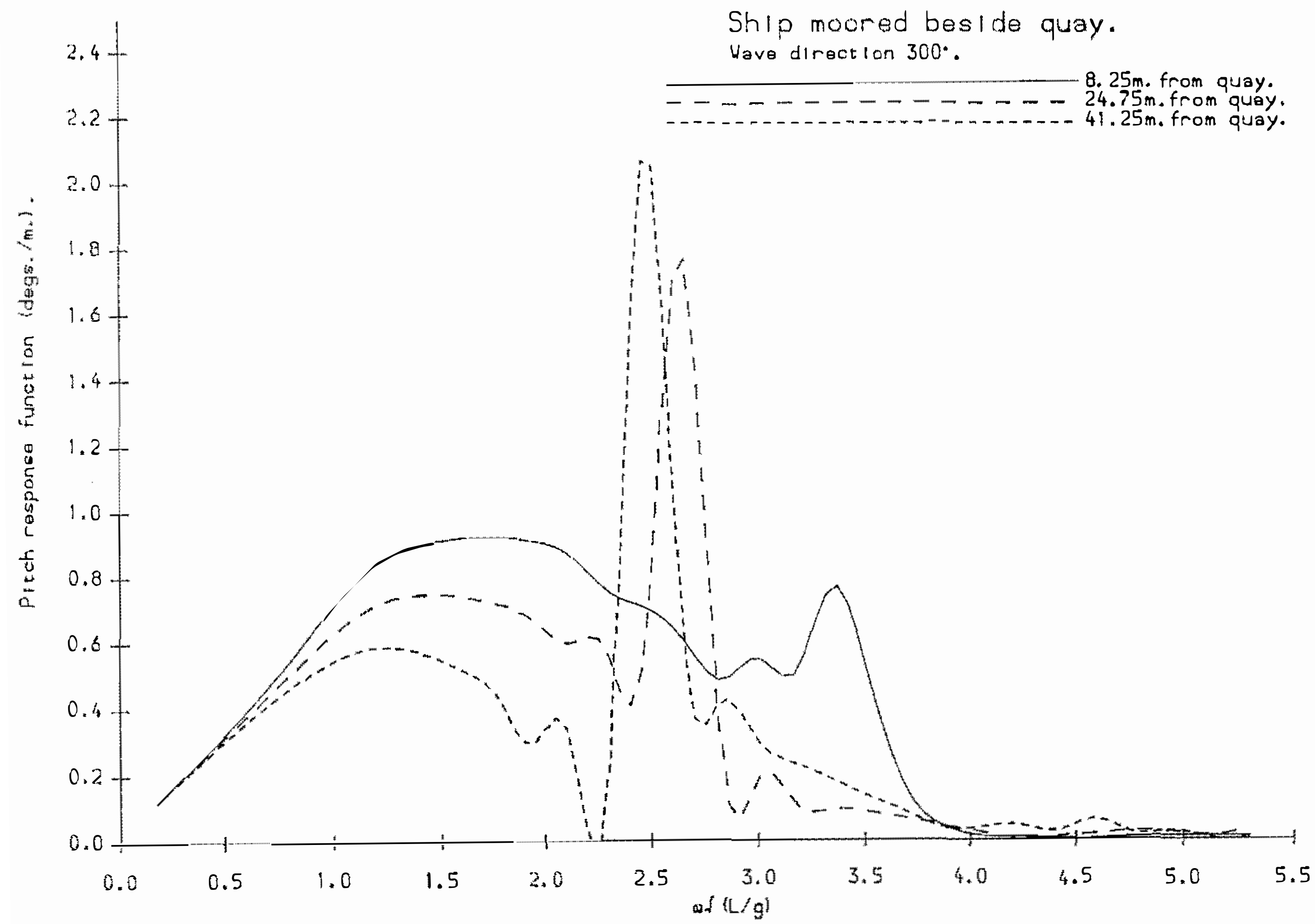


Fig 18(e) Pitch-300° wave direction

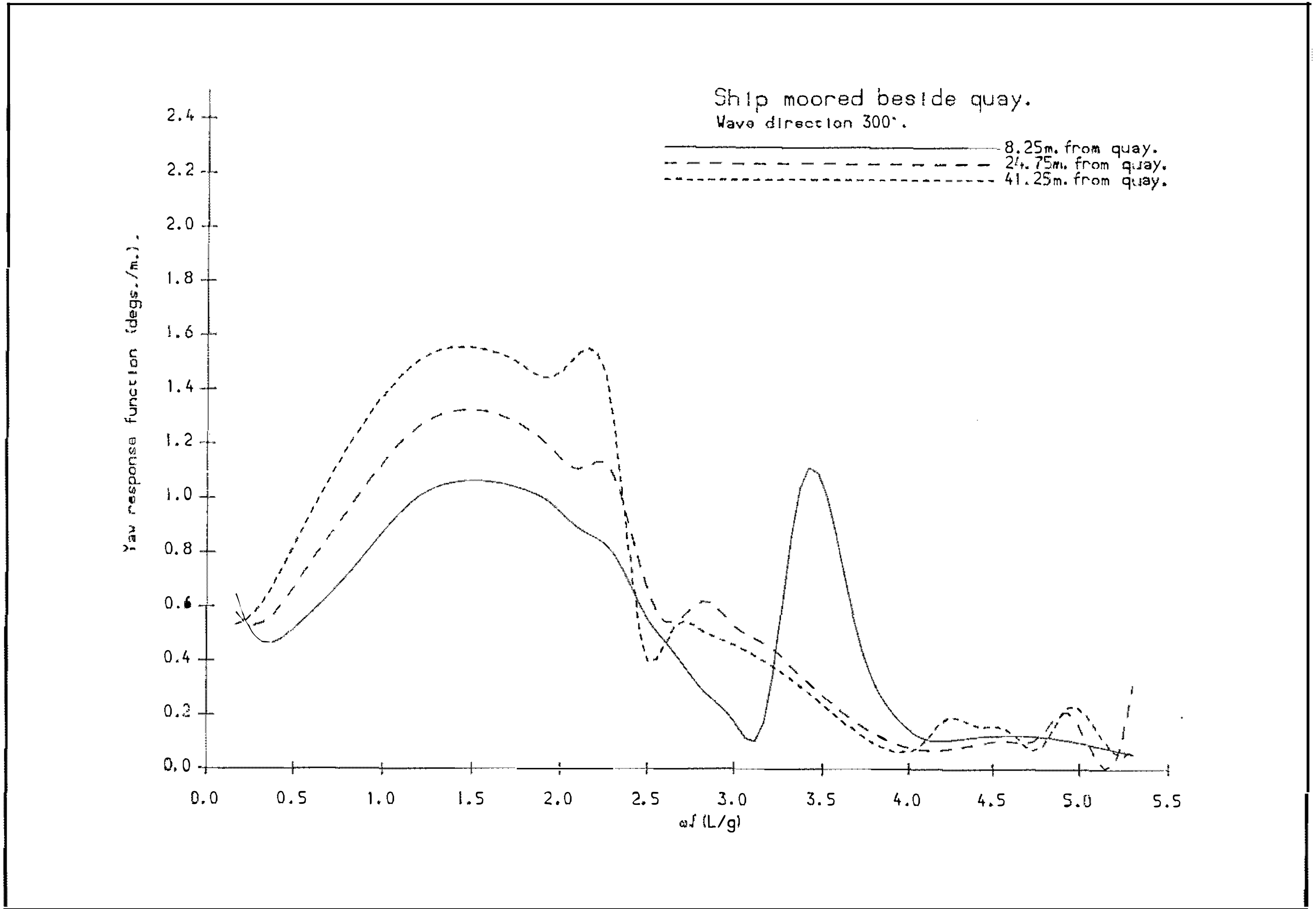
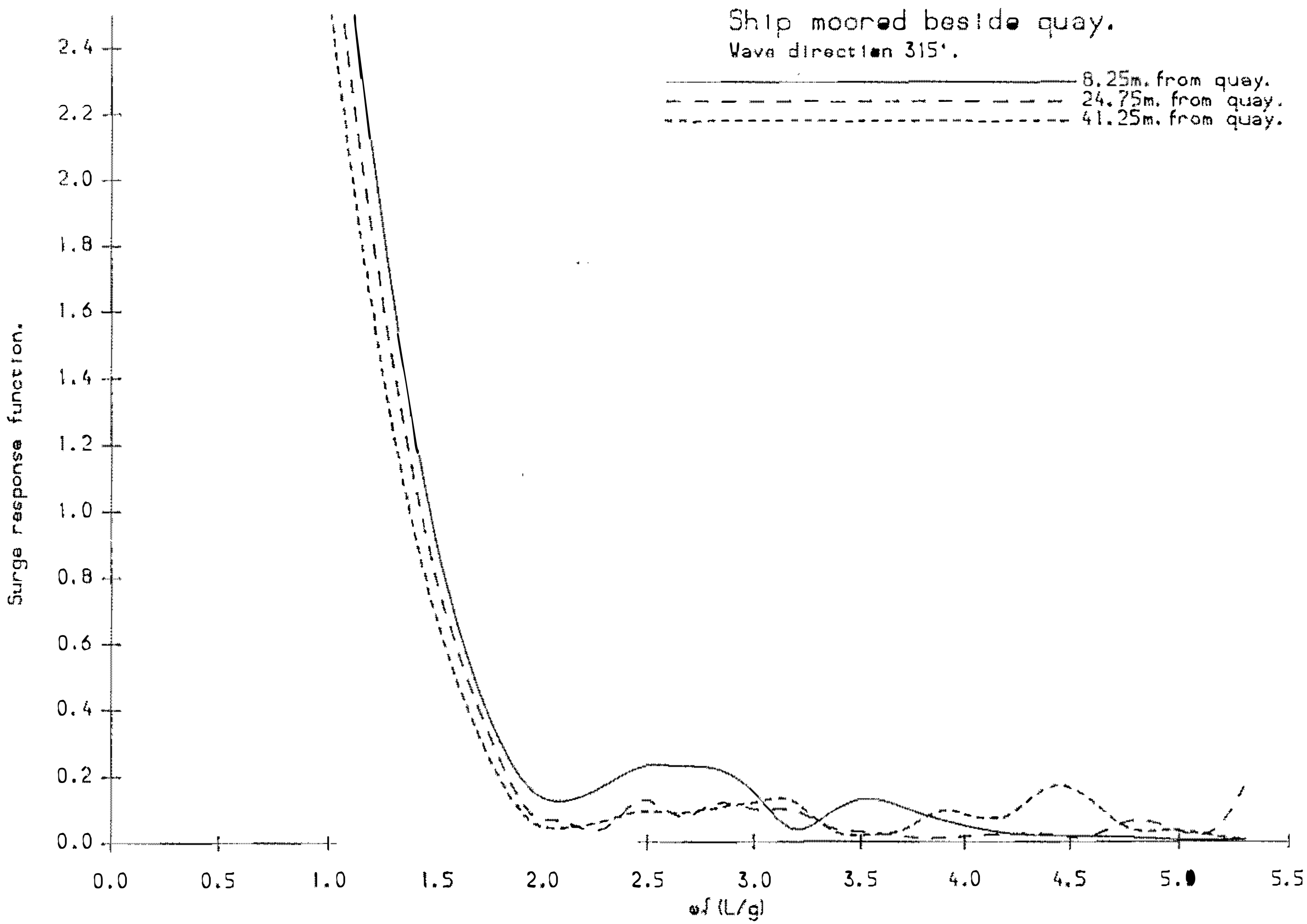


Fig 18(f) Yaw-300° wave direction

Fig 19(a) Surge-315° wave direction



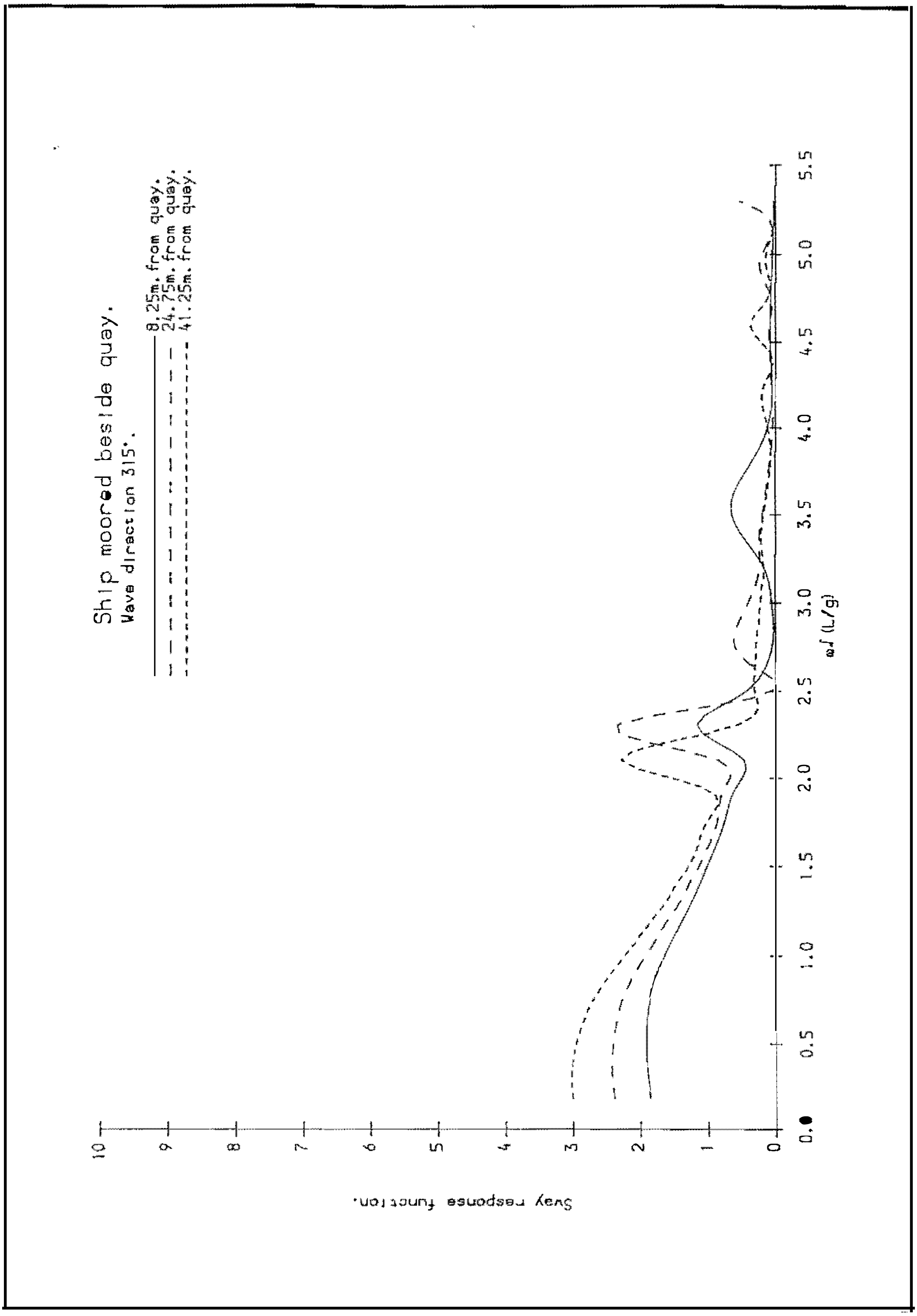


Fig 19(b) Sway-315° wave direction

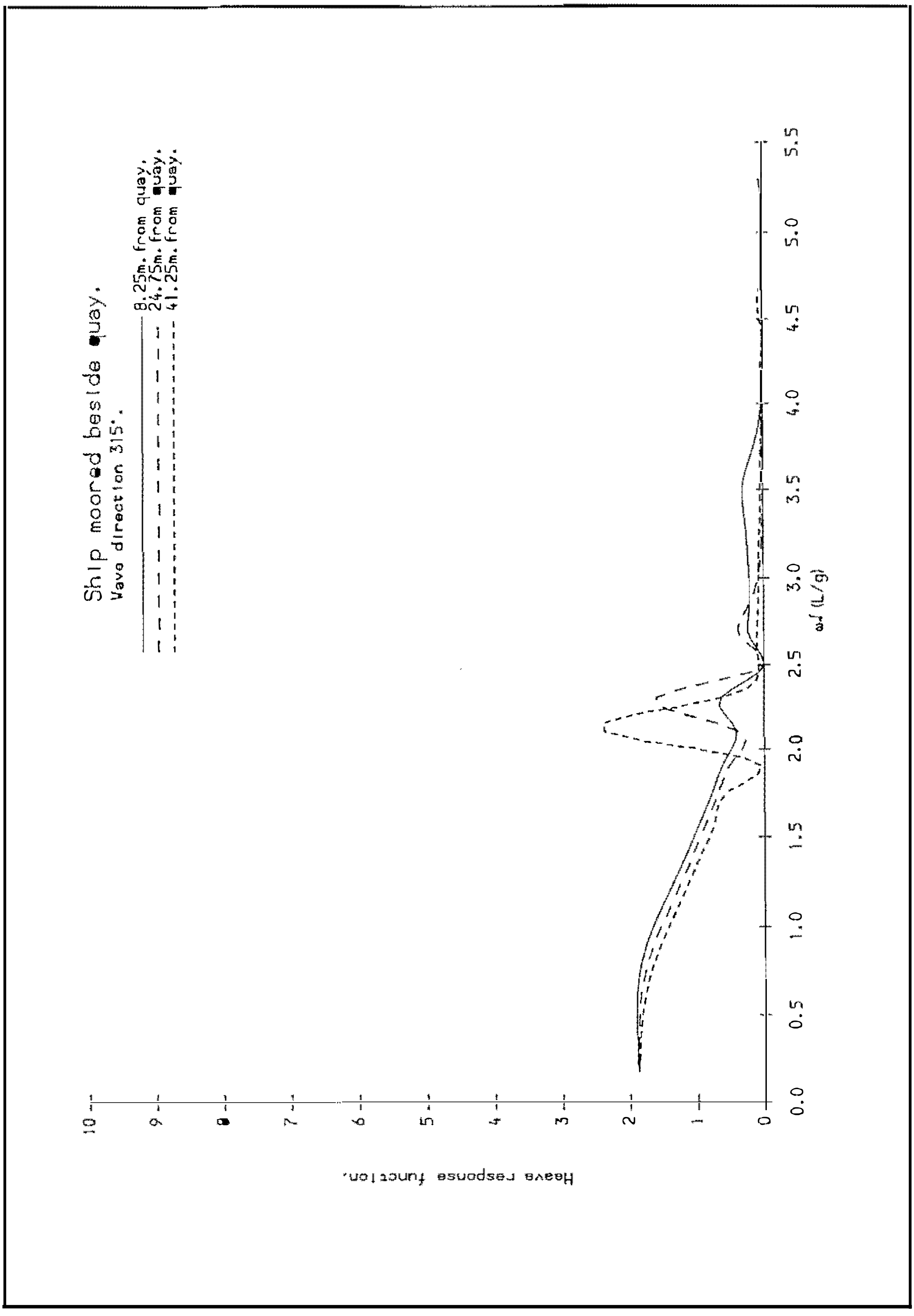
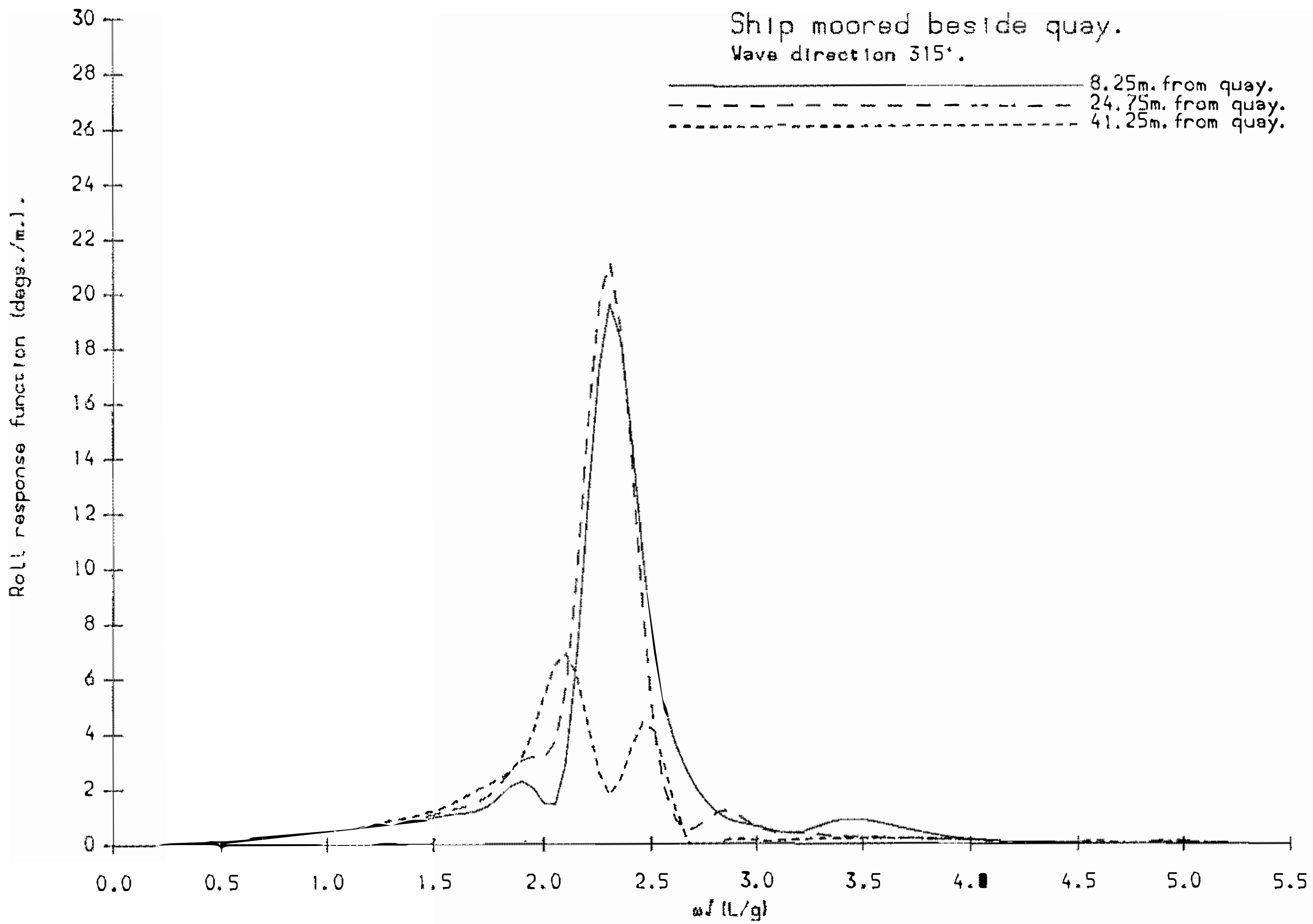


Fig 19(c) Heave-315° wave direction

Fig 19(d) Roll-315° wave direction



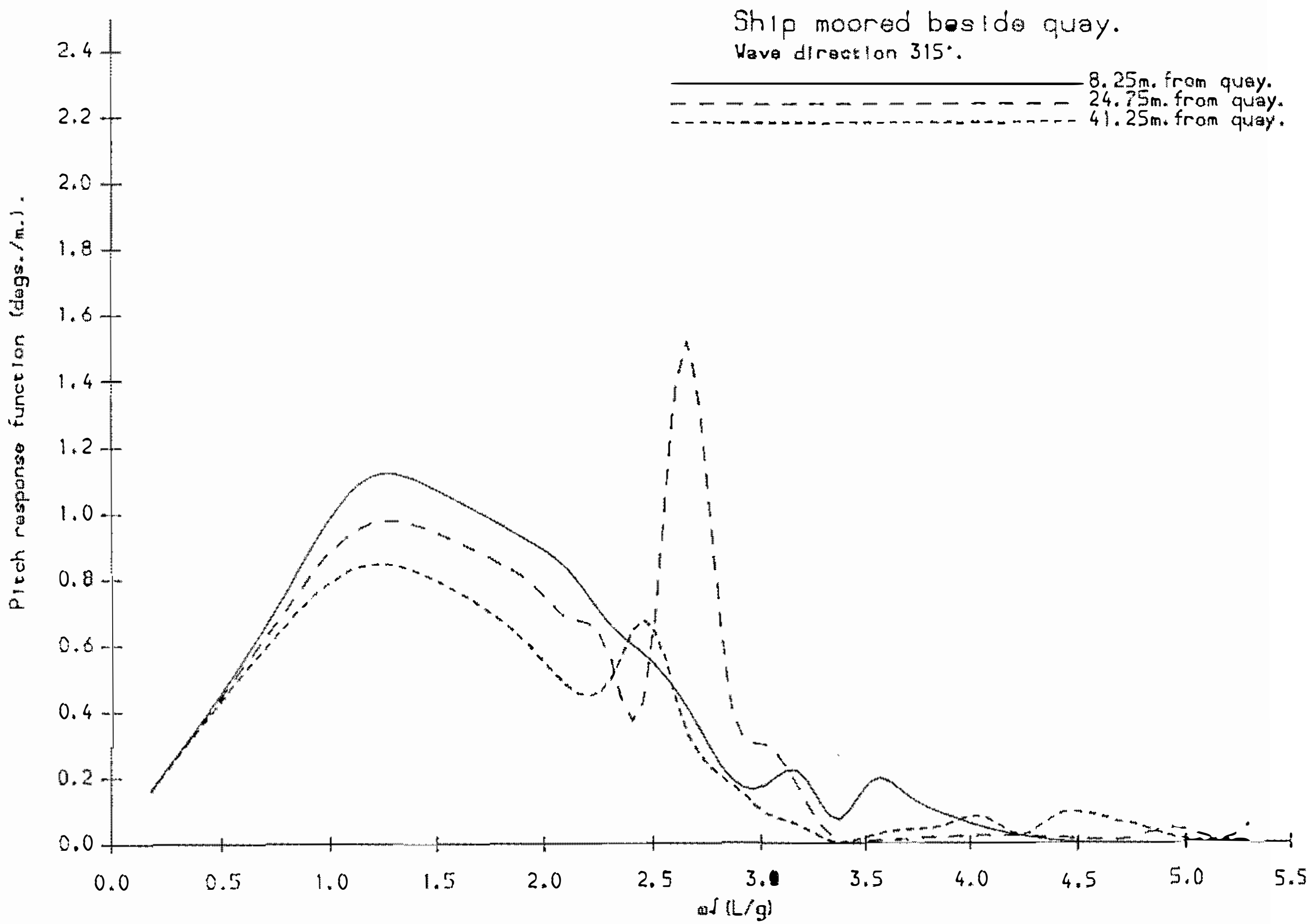


Fig 19(e) Pitch-315° wave direction

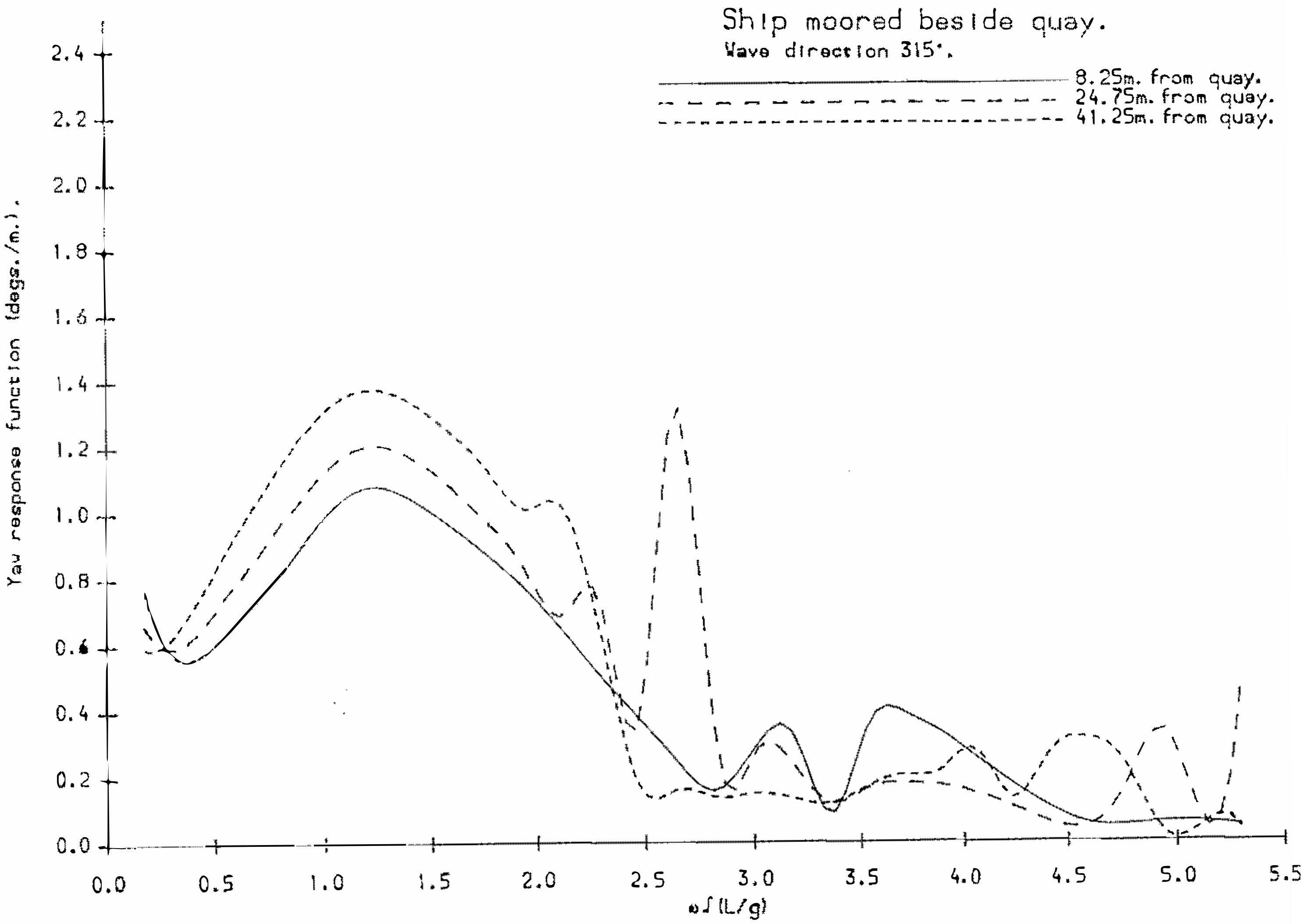


Fig 19(f) Yaw-315° wave direction

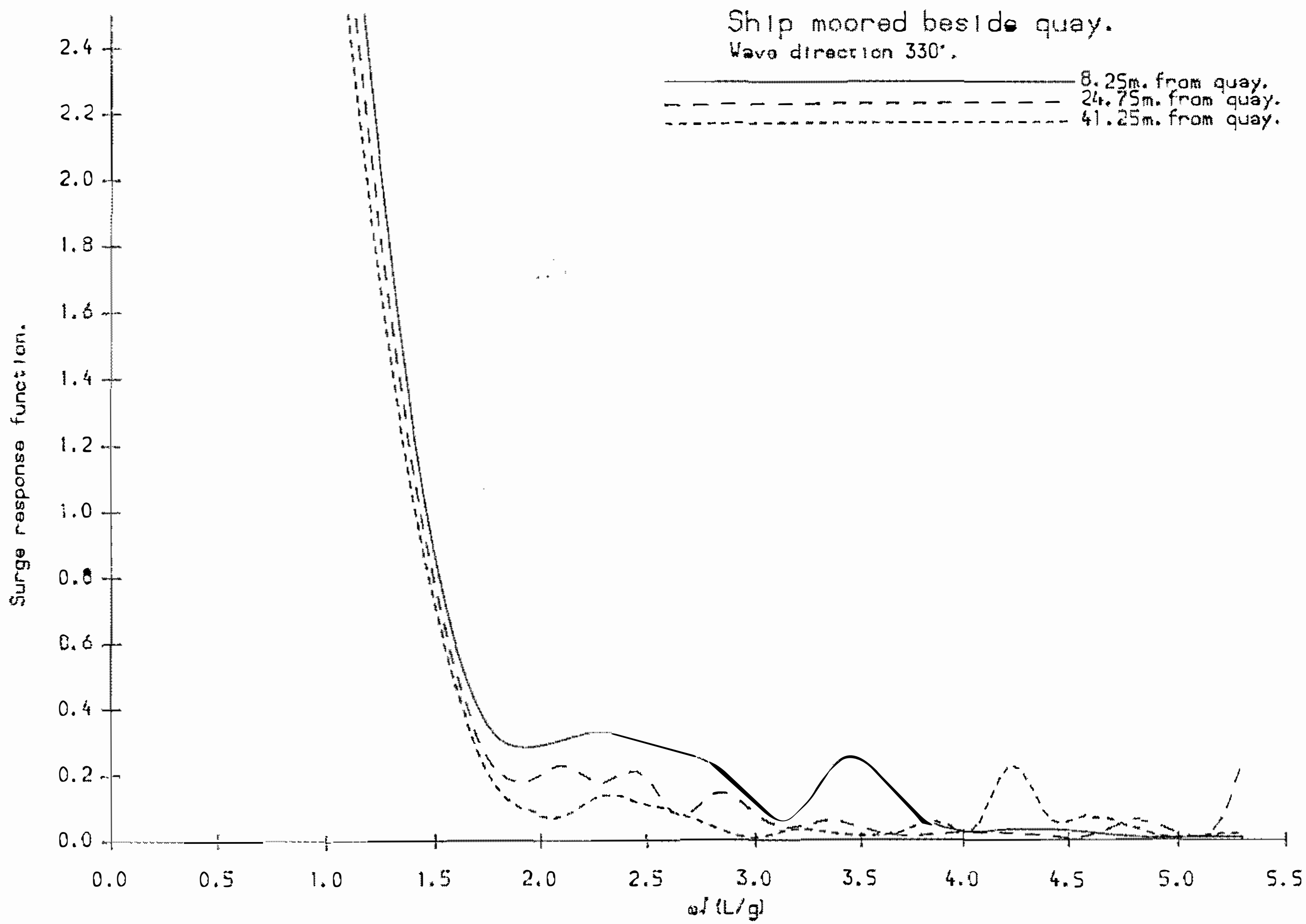


Fig 20(a) Surge-330° wave direction

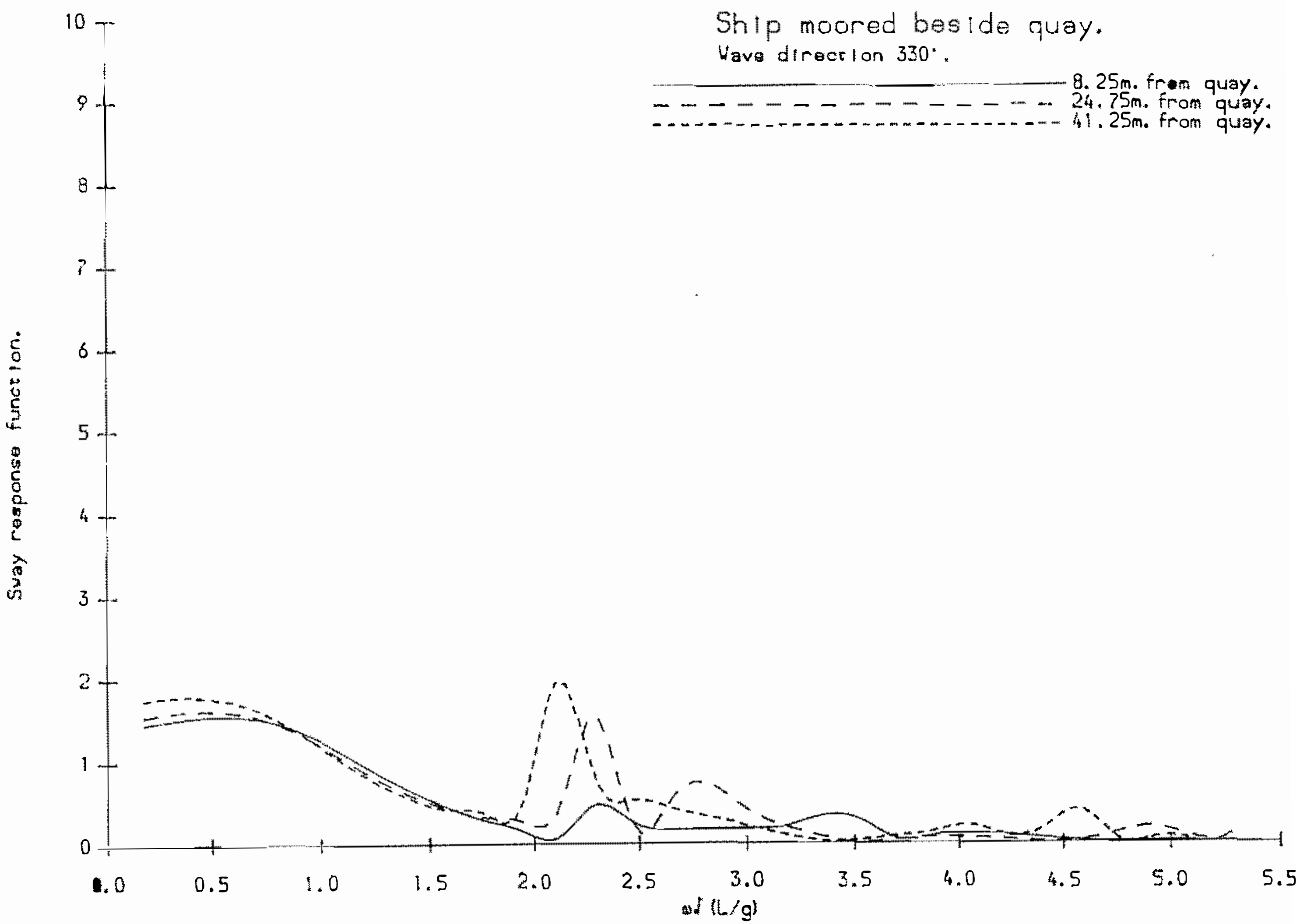


Fig 20(b) Sway-330° wave direction

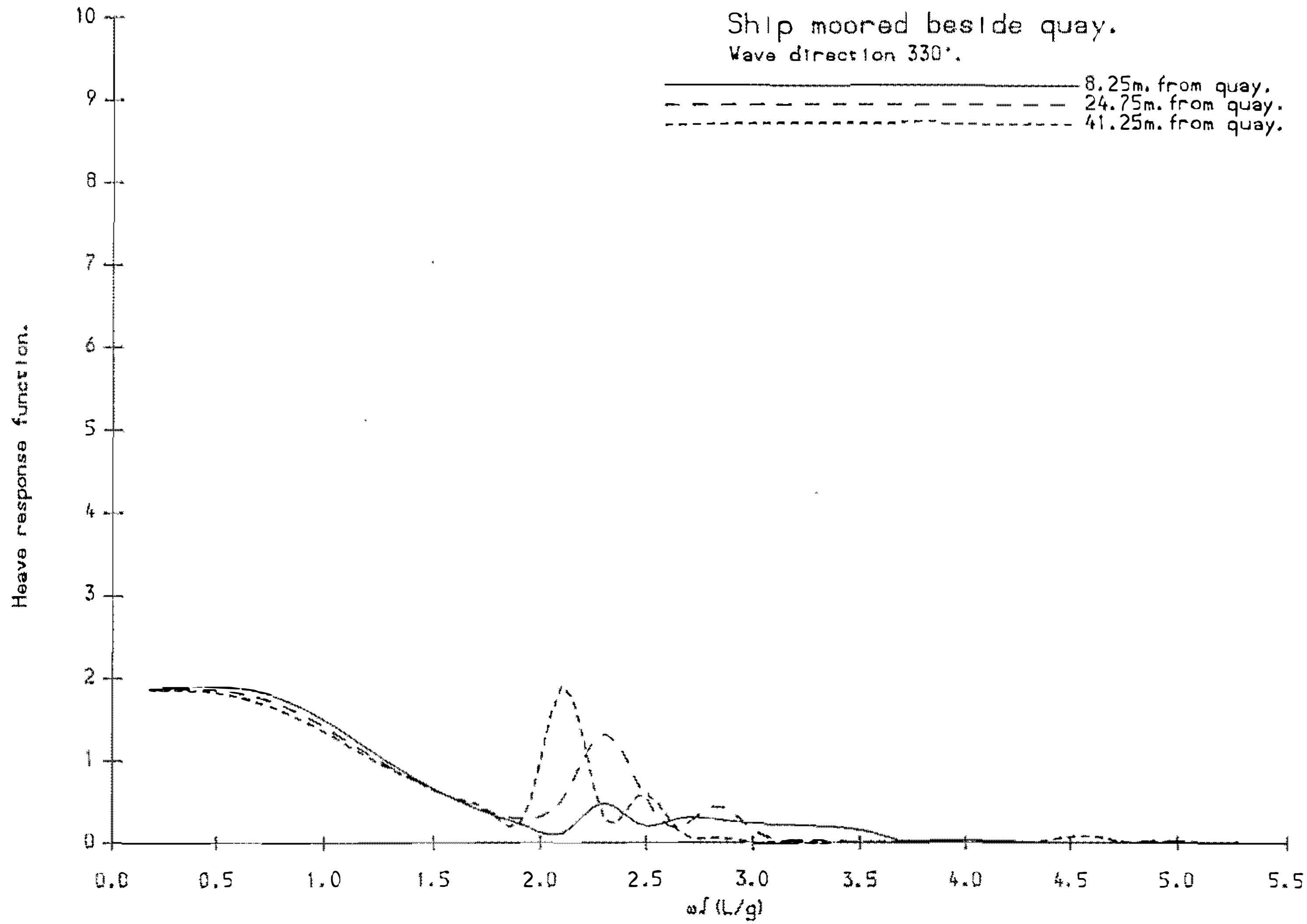
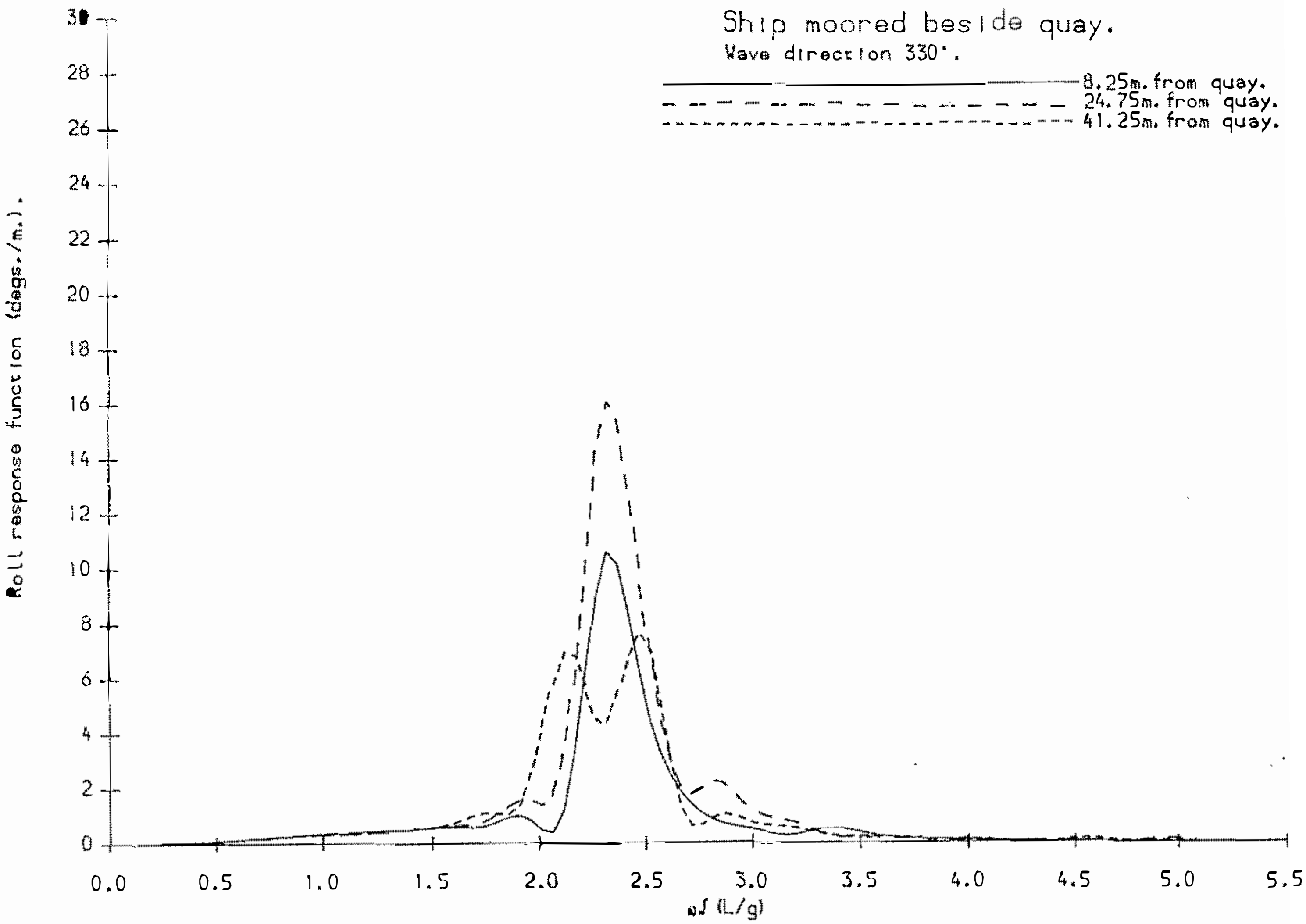


Fig 20(c) Heave-330° wave direction

Fig 20(d) Roll-330° wave direction



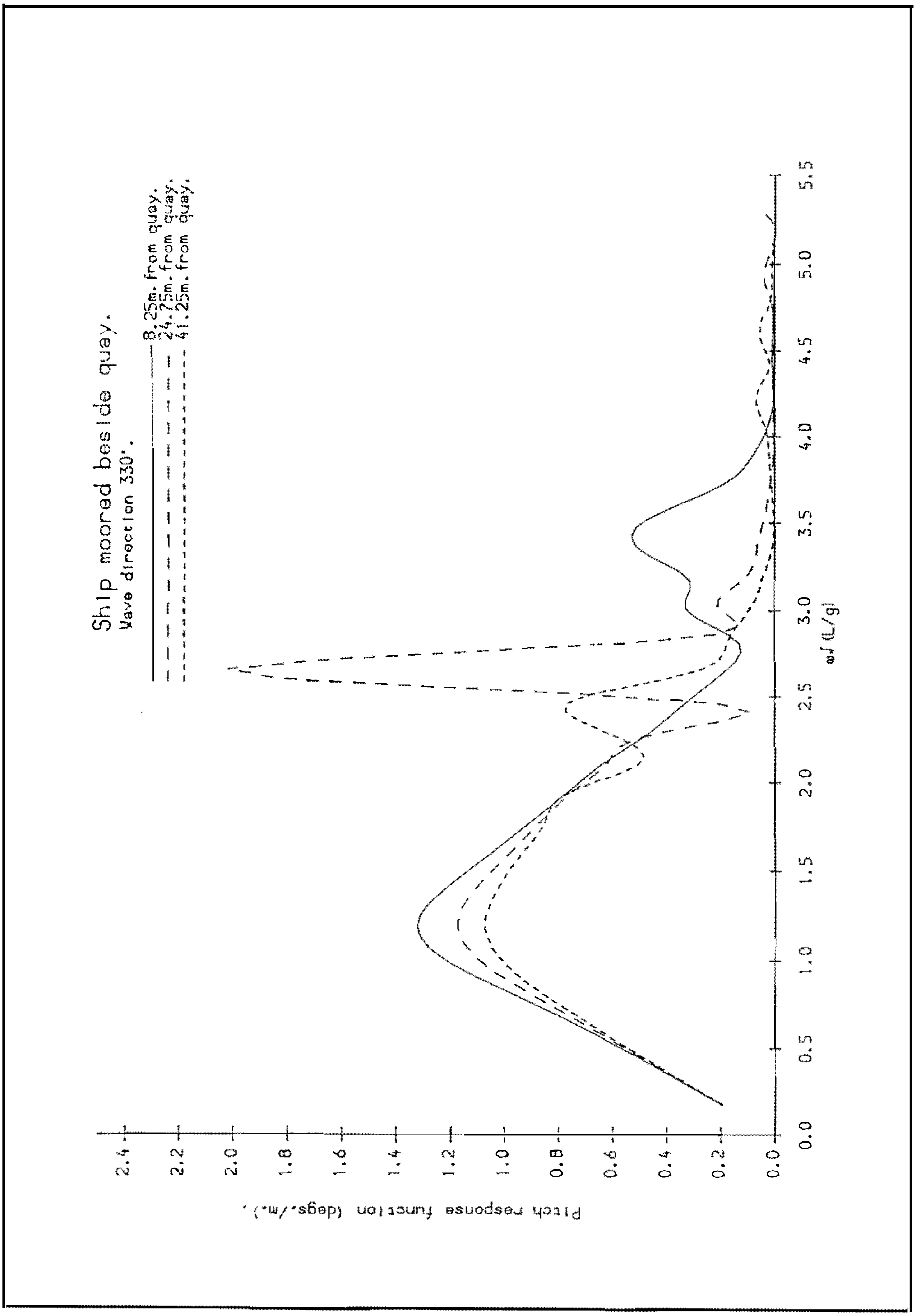
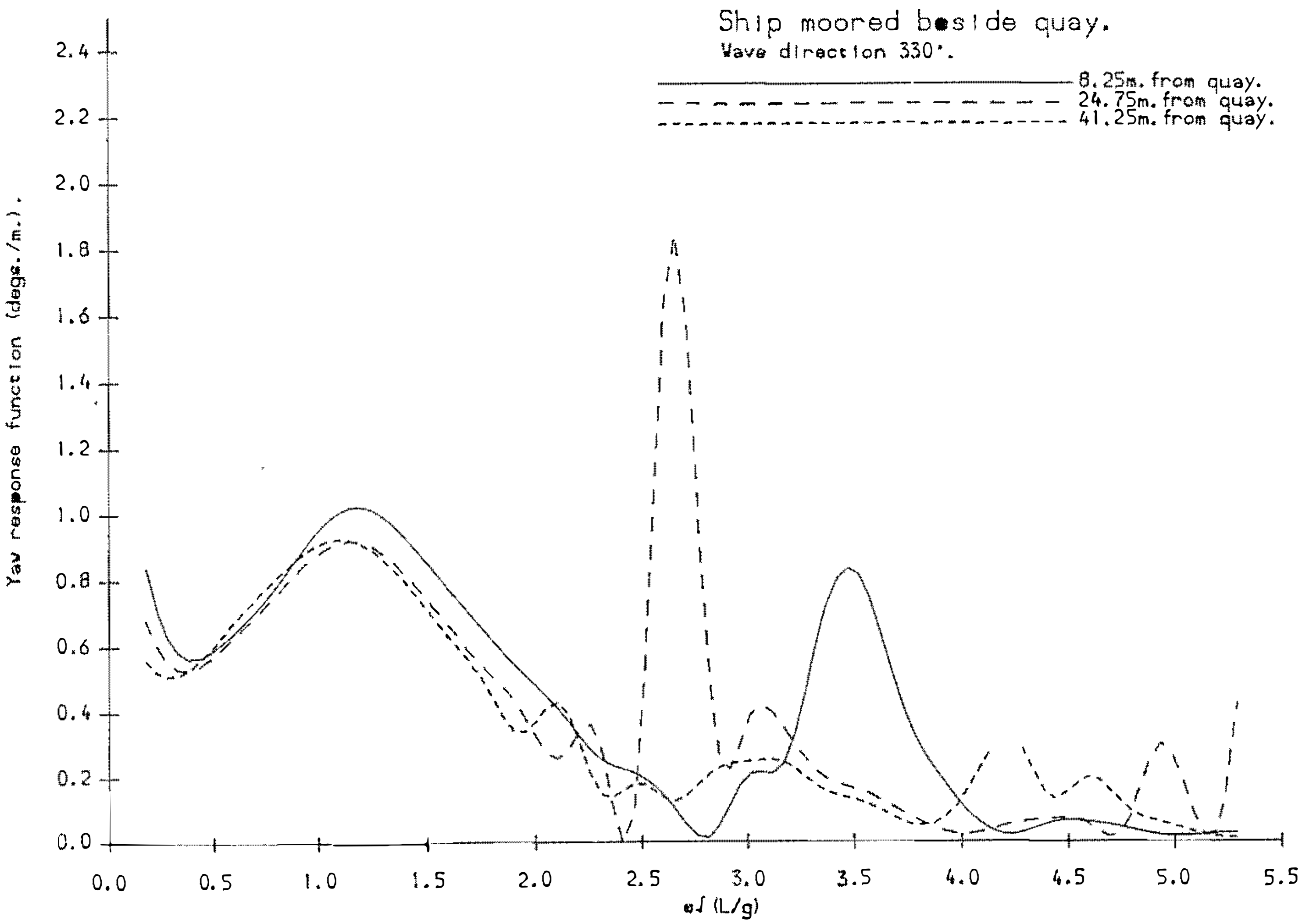


Fig 20(e) Pitch-330° wave direction

Fig 20(f) Yaw-330° wave direction



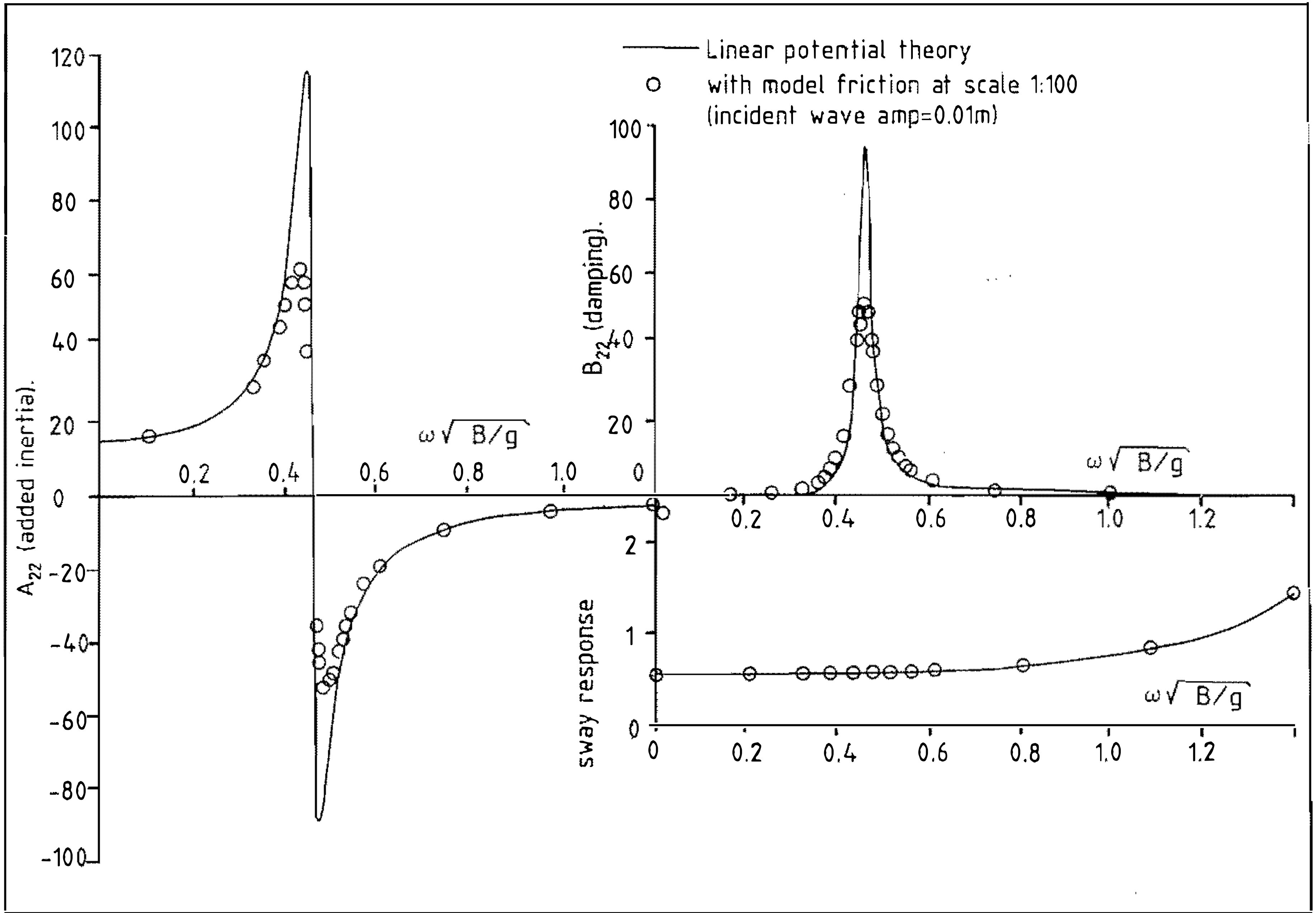


Fig 21 2D Sway:Linear potential theory and model friction.

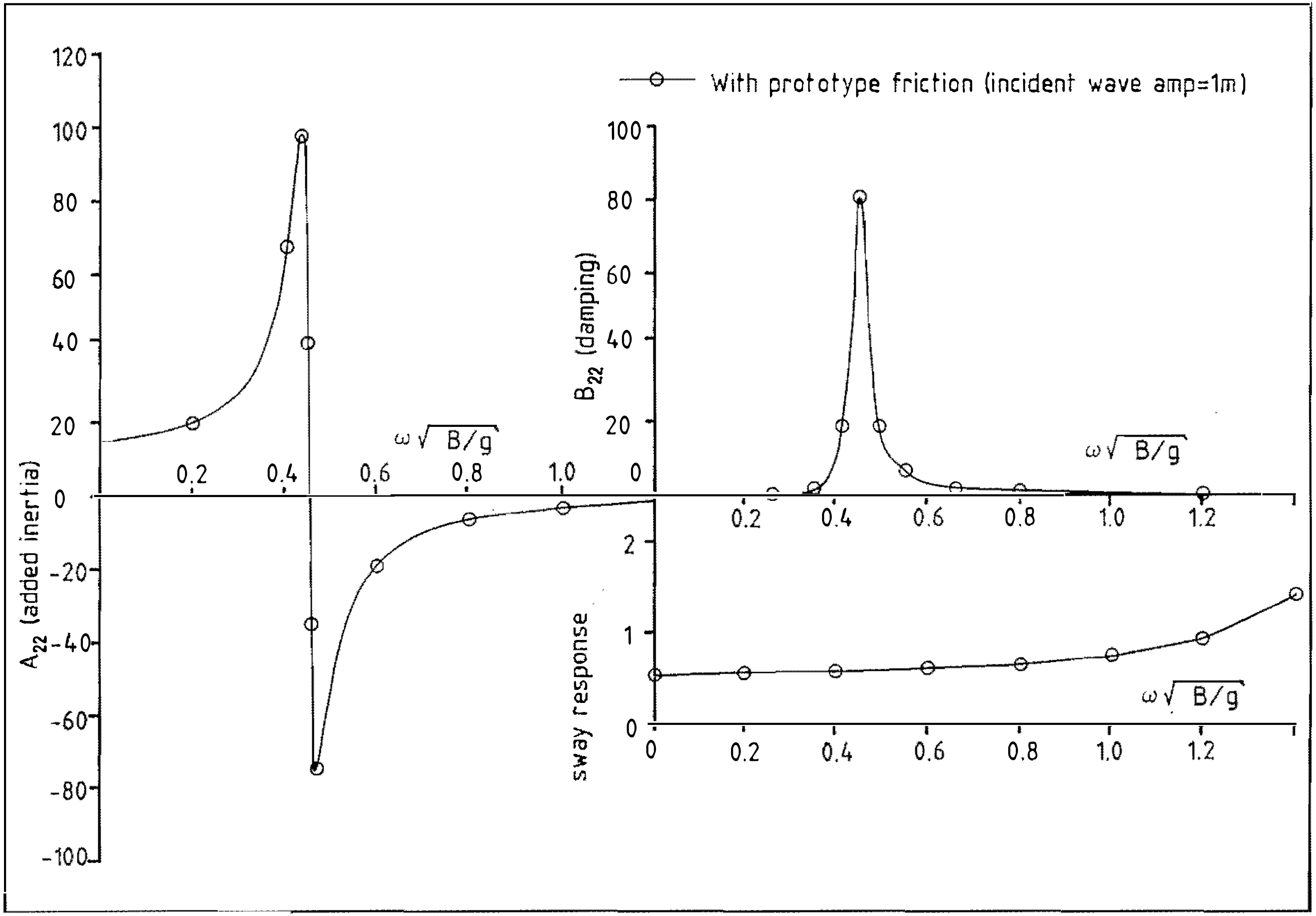


Fig 22 2D Sway:prototype friction

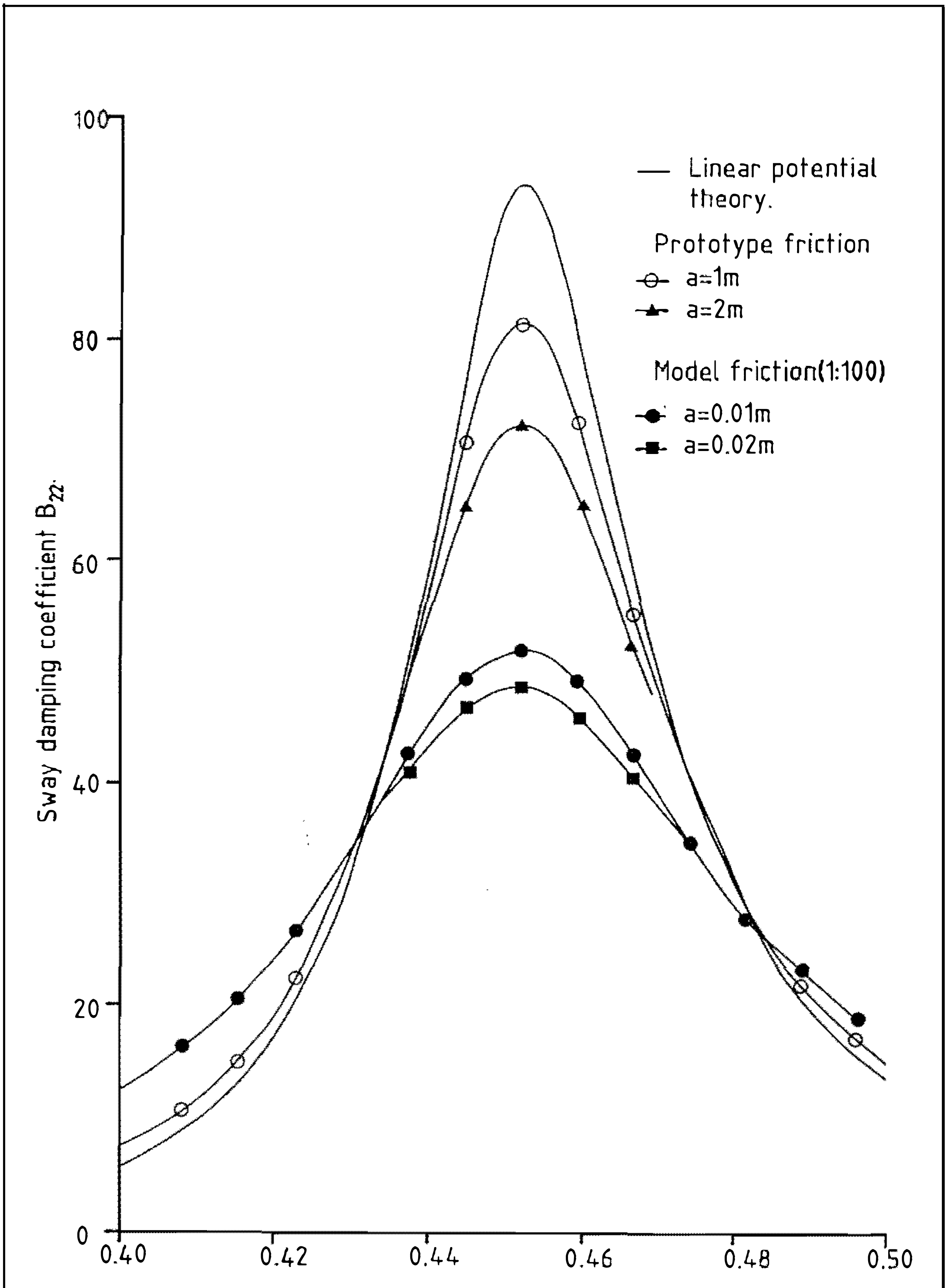


Fig 23 Variation of damping coefficient near resonance with incident wave amplitude (a).

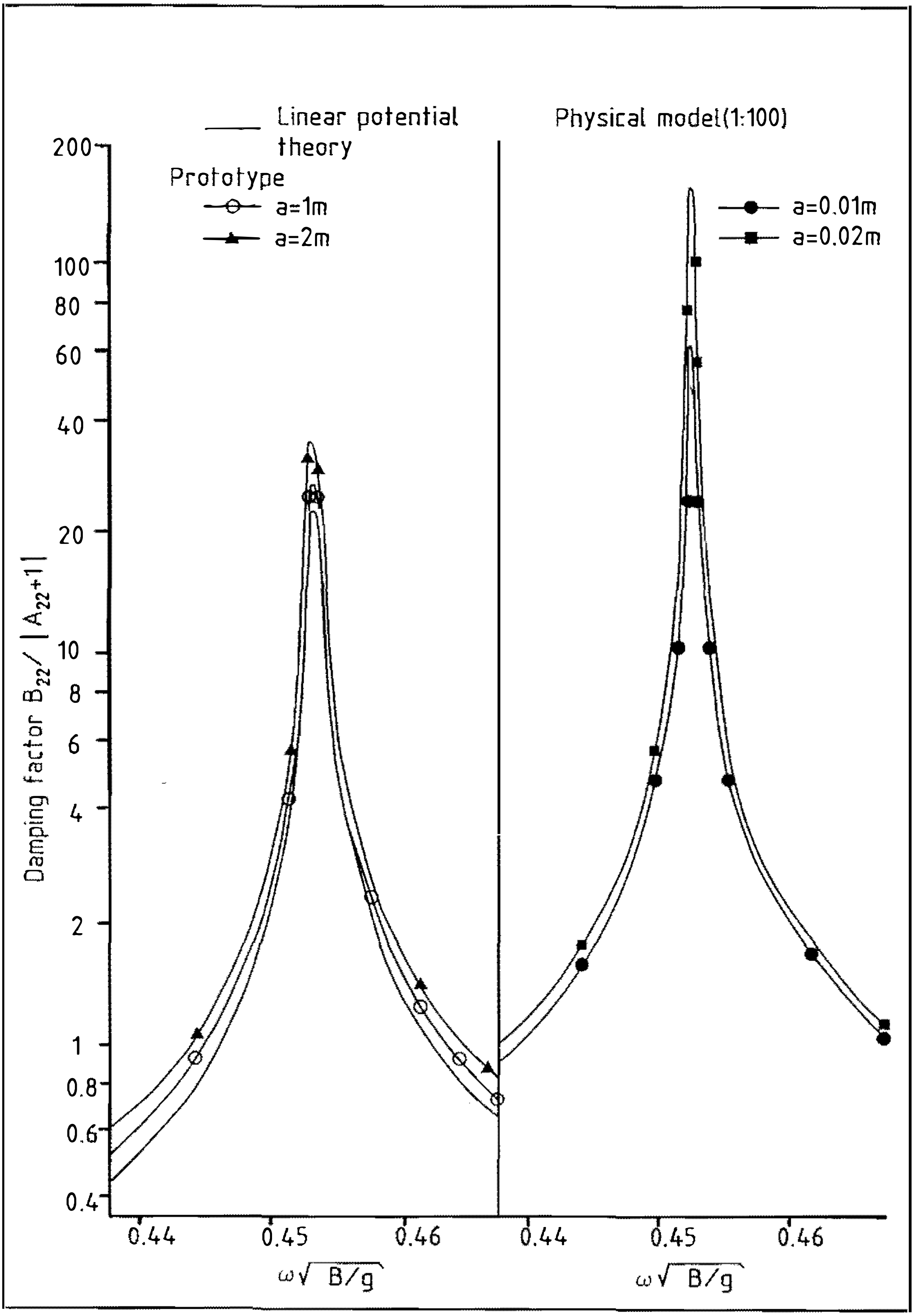


Fig 24 Variation of damping factor near resonance with incident wave amplitude (a).