FORCED OSCILLATION EXPERIMENTS

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Forced oscillation experiments

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Forced oscillation experiments with scale models are carried out to determine hydrodynamic characteristics of ships, with respect to motions in waves or steering and manoeuvring qualities. Depending on the considered motion components, in a horizontal or vertical plane, various methods are used to induce forced oscillations which are discussed briefly. Some results of forced oscillation experiments are presented as examples of this technique and compared with calculations based on numerical methods. The comparisons include, among others, the effects of ship speed and restricted water depth.

1. Introduction

Forced oscillation experiments with scale models are used in ship research for the determination of hydrodynamic forces associated with oscillatory motions of ships and other maritime constructions. The results obtained with oscillating models have been used to determine the coefficients of the equations of motion, for instance to validate theoretical models of the considered dynamic flow problems.

These have included simplified engineering solutions for the prediction of the motion of a ship advancing in irregular sea waves, as well as advanced numerical three-dimensional methods for the determination of hydrodynamic forces, based on rational theories.

In these oscillation techniques a scale model is forced to carry out harmonic oscillations of known amplitude and frequency. The required force is split up in a component in phase with the motion of the body to obtain the hydrodynamic or added mass, whereas the quadrature component is associated with damping.

The experiment may concern one particular mode of motion, for instance heaving of a ship, or more complicated coupled motions, generated by a so-called planar motion mechanism, to obtain linear and nonlinear hydrodynamic coefficients of the equations of motion in a horizontal plane for the simulation of steering and manoeuvring of ships.

As far as I know the first published forced oscillation experiment with a ship model has been carried out by Haskind & Rieman (1946). Forced heaving motions at zero forward speed were carried out in a range of frequencies and amplitudes of motion. The results showed frequency-dependent damping, vanishing at high and low frequencies, and frequency dependent hydrodynamic mass. The influence of nonlinearities appeared to be small for the considered wall-sided mathematical ship model. Later similar techniques have been used in various ship hydrodynamic laboratories as a consequence of the increased interest in the dynamics of ships.

Some results of forced oscillation experiments, carried out in the Delft Shiphydrodynamic Laboratory, are briefly summarized here to illustrate the

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possibilities of this experimental technique with respect to the analysis and validation of theoretical methods as used in ship science and engineering applications.

In addition, some recent results concerning the influence of high forward speed and restricted water depth on hydrodynamic motion parameters of a ship will be presented.

2. Vertical motions

The experimental set-up as used by Haskind & Rieman has been developed to include the determination of the hydrodynamic coefficients of the heave and pitch equations of motion at forward speed.

These may be written as two linear coupled equations with frequency-dependent coefficients:

$$\left.\begin{aligned}(a+\rho\nabla)\ddot{z}+b\dot{z}+cz+d\ddot{\theta}+e\dot{\theta}+g\theta &= \bar{F}\exp(\mathrm{i}\omega t) \quad \text{(heave)},\\ (A+I_{yy})\ddot{\theta}+B\dot{\theta}+C\theta+D\ddot{z}+E\dot{z}+Gz &= \bar{M}\exp(\mathrm{i}\omega t) \quad \text{(pitch)},\end{aligned}\right\} \tag{1}$$

where z is the heave, θ is the pitch, ω is the circular frequency, ∇ is the volume of displacement and I_{yy} is the mass moment of inertia. $\bar{F}$ and $\bar{M}$ are the complex exciting force and moment amplitudes.

The agreement between motion amplitudes and phase characteristics, as derived from model tests in regular and irregular waves and corresponding calculations, using hydrodynamic coefficients a,b,d,e,A,B,D,E obtained from forced oscillation experiments as well as measured wave forces and moments F,M in the equations of motion, is satisfactory for conventional hull forms at moderate forward speeds as shown by Gerritsma & Beukelman (1967).

Strip theory computations using two-dimensional approximations for added mass and damping of ship cross sections neglect the mutual interference of the flow between those sections. Also three-dimensional effects at the ends of a ship, which in particular are important for pitching motions, are not taken into account.

Nevertheless computed motion response functions for heave and pitch agree quite well with experiments in many cases, including length-beam ratios as small as 2.5.

A systematic model series derived from one particular hull form (Series Sixty $C_B = 0.70$), of which the length-beam ratio varied from $L/B = 4$–20, has been force oscillated in still water at forward speeds corresponding to $Fn = 0.2$ and 0.3. In addition, motion response and added resistance experiments for heave and pitch in regular waves have been carried out (Gerritsma *et al.* 1974). The results confirm in more detail the applicability of the strip method. An example of measured damping compared with a strip theory calculation is given in figure 1.

However, predictions of motions in following waves as well as predicted relative motions and added resistance at high forward speeds using strip theory calculations do not agree with experimental results in all cases. Also, strip theory calculations cannot be used for the determination of forces in waves which are very short compared with the length of the ship. Pitching amplitudes of long slender ships are overestimated and they are underestimated in the case of sailing yachts.

To study strip theory methods in more detail two-dimensional calculations for shiplike cross sections have been compared with experimental results. First of all this concerned Ursell's (Ursell 1957) solution for the vertical oscillation of a circular cylinder and its generalization for actual shiplike cross sections by Tasai (1960) and others.

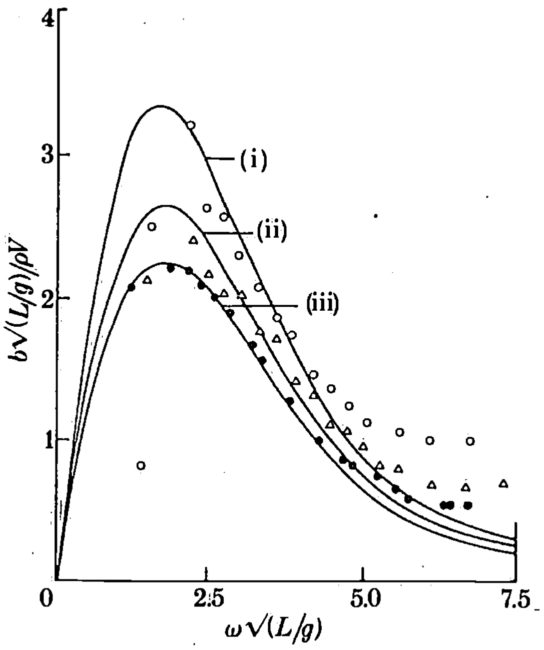


Figure 1. Heave damping. (i) $L/B = 4$; (ii) $L/B = 5.5$; (iii) $L/B = 7$. ——, Strip theory damping.

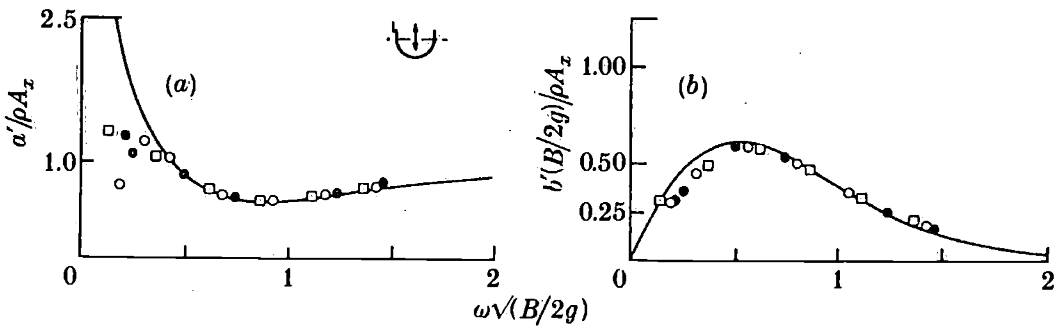


Figure 2. Added mass and damping of a heaving circular cylinder. ——, Calculation; ●, $z_a = 0.01$ m; ○, $z_a = 0.02$ m; □, $z_a = 0.03$ m.

Forced oscillation experiments with cylinders have been carried out by Vugts (1970). These tests include a half-circular cross section as well as a triangle, and other shiplike cross sections. The cylinders spanned the width of the towing tank to avoid three-dimensional effects at the ends of the cylinders, and beaches on both ends of the towing tank have been used to absorb the radiated damping waves.

In figure 2 the experimental result for a half-circular cross section is compared with computed mass and damping.

In general the agreement is satisfactory, also for the other cross sections: rectangular sections with beam-draught ratios 2, 4 and 8, two shiplike cross sections, and a triangle. The experiments confirm the applicability of Ursell's work, which has been of great value for the progress in ship motion research.

Also computations based on finite-element methods have been carried out, showing similar accuracy. These proved to be useful in cases of complicated bottom and/or wall geometry.

Based on the results of the cylinder tests it may be accepted that generally the effects of viscosity can be neglected for vertical motions with moderate motion amplitudes.

Figure 3

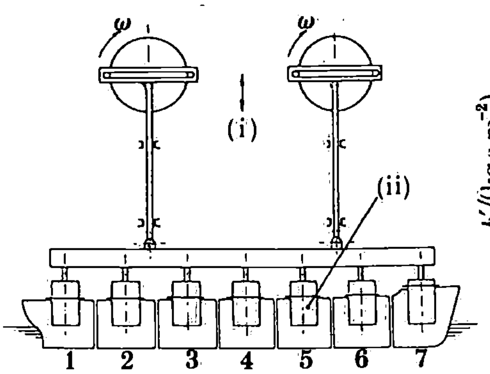


Figure 4

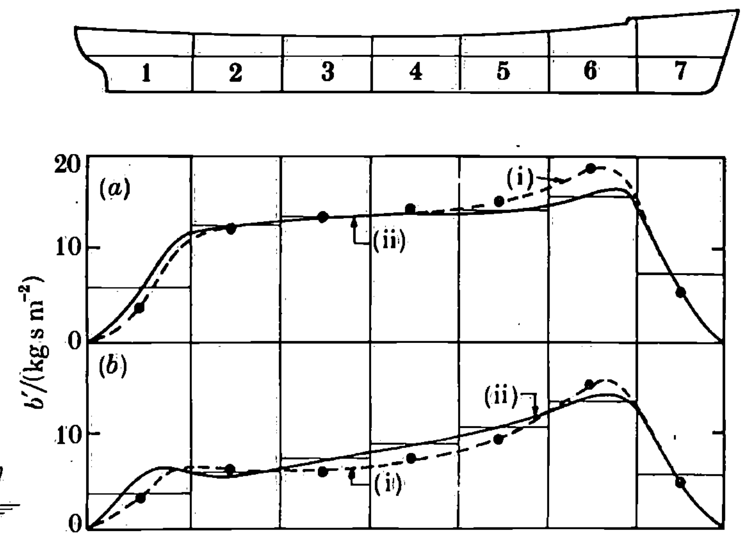


Figure 3. Arrangement of oscillation experiment with a segmented model. (i) $z = r \sin \omega t$; (ii) $F_i \sin (\omega t + \epsilon_i)$.

Figure 4. Longitudinal distribution of damping coefficient, $Fn = 0.20$, $L = 2.3$ m. (*a*) $\omega = 6$ rad s^{-1}; (*b*) $\omega = 8$ rad s^{-1}. (i) Calculation; (ii) experiment.

Forward speed effects on the distribution of the hydrodynamic forces along the length of a ship are important, because they introduce longitudinal hydrodynamic asymmetry and corresponding coupling between heave and pitch motions.

To investigate forward speed and three-dimensional effects a segmented model technique has been introduced. The experimental set-up is shown in figure 3.

Each separate segment of the ship model is connected to a stiff beam by means of a dynamometer to measure the vertical hydrodynamic force acting on the segments. The beam oscillates vertically with an harmonic motion and the forces on each section are reduced to added mass and damping components. A continuous line, as drawn in figure 4, approximates the distribution of the measured added mass and damping.

Strip theory methods predict the effect of forward speed on damping reasonably well for conventional shipforms and moderate Froude numbers, say $Fn < 0.40$.

According to strip theory the longitudinal distribution of the damping coefficient b' is given by:

$$b' = N' - V\,\mathrm{d}m'/\mathrm{d}x, \tag{2}$$

where N' is the two-dimensional damping coefficient of the cross section at x, and m' is the corresponding added mass coefficient.

In particular the linear speed dependency of the damping cross-coupling coefficients of pitch and heave and their symmetry relation have been confirmed by experiments (Gerritsma & Beukelman 1967).

In figure 4 the longitudinal distribution of the heave damping coefficient for a Series 60, $C_{\mathrm{B}} = 0.70$ hull form at $Fn = 0.20$, calculated with a strip theory method, is compared with measured distributions.

To investigate three-dimensional effects, the longitudinal distribution of the hydrodynamic forces at zero forward speed has also been calculated with the three-dimensional radiation/diffraction panel method WAMIT, developed at MIT (WAMIT 1988).

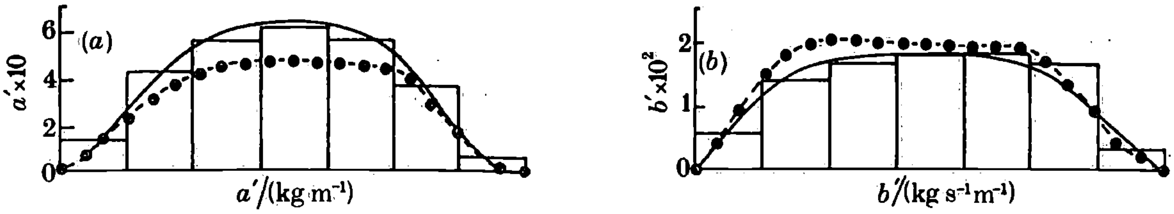


Figure 5. Comparison of calculated added mass and damping distribution at zero forward speed, $L = 2.3$ m, $\omega = 4$ rad s^{-1}. ●, Two-dimensional calculation; ——, three-dimensional calculation; ⊓, experiment.

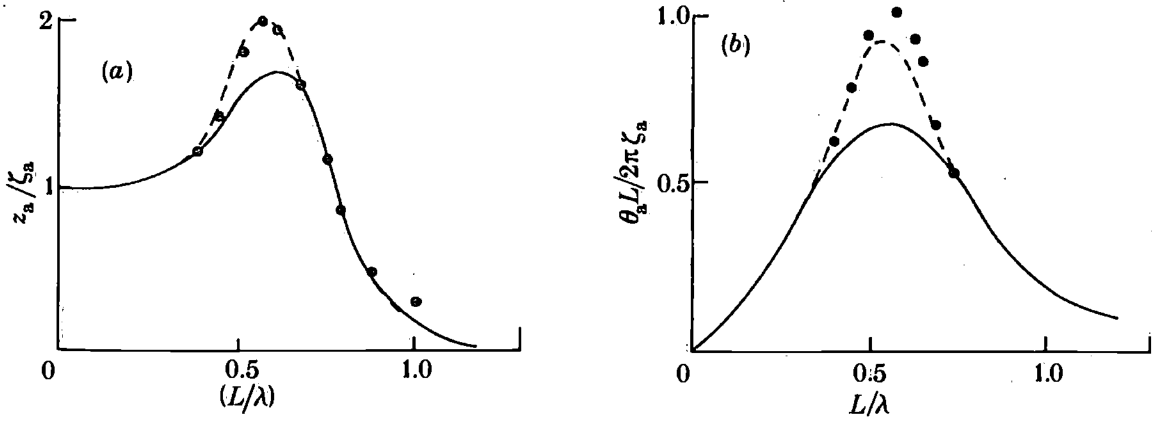


Figure 6. (*a*) Heave and (*b*) pitch amplitude of a sailing yacht. ——, Strip theory; ----, three dimensional; ●, experiment.

For a number of ship forms Adegeest (1989) determined the contribution of each segment of the hull to the total hydrodynamic mass and damping, using a suitable panel distribution to fit the considered model segments.

Such panel distributions can be generated also for conditions with trim and heel angle and in addition they may include the wave contour along the hull as well, if this is known.

For a Series Sixty $C_B = 0.70$ model at low frequencies of oscillation $\omega\sqrt{(L/g)} = 1.9$ and zero forward speed the WAMIT result agrees better with the experiment as compared with the two-dimensional strip theory prediction, see figure 5, but for higher frequencies very little difference between the two computed results exist.

A similar improvement for the low-frequency damping and added mass distribution for this particular model has been found earlier by Maruo (1978) with a revised formulation of his slender-body approximation.

The three-dimensional distribution of the damping coefficient at zero forward speed, as computed by the panel method, may be transformed to the case of forward speed by using expression (2) as a practical approximation.

For a sailing yacht hull form with $L/B = 4.5$, $B/T = 2.5$ this improved considerably the predicted heave and pitching motions in the range of maximum motion amplitudes, as shown in figure 6. The same method proved to be successful in the case of asymmetric cross sections, which have to be considered when a yacht sails with a heel angle.

An extensive experiment to measure hydrodynamic coefficients and wave forces as well as the motions and the added resistance in regular head waves, has been carried out with a $L/B = 10$ symmetric mathematical Wigley hull form (Gerritsma 1988). King & Beck (1989) developed a three-dimensional time domain seakeeping calculation method using a linear theory to determine the hydrodynamic forces on the hull due to motions, as well as the exciting forces due to the incident waves. The potential flow problem was solved in the time domain rather than in the frequency domain, but the results have been presented in frequency domain coefficients of the motion equations to compare with experiments and strip theory results. This approach covers the three dimensionality of the hull and forward speed effects to some extent.

King & Beck make an extensive comparison of this three-dimensional method with strip theory calculations and the Delft experiments, including also the wave exciting forces. Generally the agreement of both calculation methods with the experiment is good, with some slight improvements of the three-dimensional panel method compared with the strip theory method.

The heave–pitch damping cross-coupling coefficients, as predicted by this three-dimensional method, are close to the measured values. Apparently the longitudinal distribution of the damping coefficient as a result of forward speed is quite accurately predicted method for this symmetric slender model.

3. High forward speed

So far the forced oscillation experiments concerned moderate ship speeds and conventional displacement type hull forms. An important area of interest is the seakeeping behaviour at high speeds. In particular the operability of a fast ship depends to a large degree on the motion response and the vertical accelerations of the ship in a seaway. The reliable prediction of these quantities is a valuable tool for the designer of such vessels.

The applicability of strip theory methods for high speed naval ships has been investigated by Blok & Beukelman (1984). They concluded that motion amplitude and vertical acceleration predictions, based on strip theory, agree very well with measured motions for a high-speed displacement type hull form at speeds as high as $Fn = 0.57$ and 1.14, see for instance figure 7. Also added resistance and relative motions were predicted well for $Fn = 0.57$, but some differences occur at $Fn = 1.14$. The prediction of these two important aspects of ship behaviour in waves depends on the phase of the motion with respect to the wave and this in turn depends to some extent on the longitudinal distribution of damping and added mass.

The speed range considered in this case exceeds the commonly accepted limits for the applicability of strip theory methods. The hydrodynamic pressure distribution at high forward speed and zero frequency of oscillation in calm water may be characterized by pronounced trim, sinkage, dynamic lift and nonlinear ship waves.

Generally these phenomena are not included in prediction methods based on calculations.

To study these effects in more detail Keuning carried out forced oscillation experiments with a segmental model of the same hull form as used by Blok & Beukelman at the same high speeds (Keuning 1990).

The model, with $L/B = 8$, $B/T = 4$, has flat sections aft, but the relatively low rise of the centre of gravity, observed during the resistance tests in calm water, indicated only a moderate influence of dynamic lift in the considered speed range.

The oscillation experiments have been carried out with trim and sinkage, corresponding to the considered forwards speeds. The in-phase component of the hydrodynamic force contains a contribution of the restoring force.

Figure 7

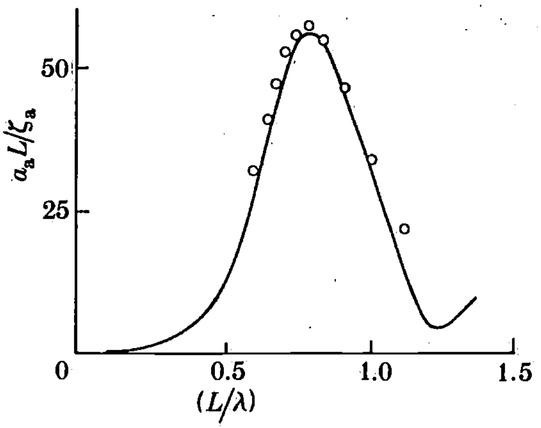


Figure 8

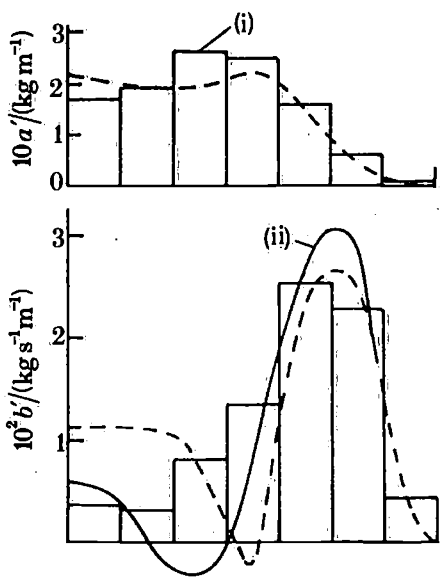


Figure 7. Vertical acceleration amplitudes at $Fn = 1.14$. ——, Calculated; o, experimental.
Figure 8. Added mass and damping distribution at high forward speed; $Fn = 1.14$, $L = 2$ m, $\omega = 15$ rad s^{-1}. ---, Strip theory; (i) experiment; (ii) calculated with experiment, dm'/dx.

For moderate speeds and conventional ship forms the linear hydrostatic restoring force at zero speed and frequency may be used as an acceptable simplification, but at high Froude numbers the pressure distribution differs considerably from this assumption. At zero forward speed the hydrostatic restoring forces of the separate segments are only slightly nonlinear with respect to vertical displacement, but high forward speed introduces significant nonlinearities.

To include the actual restoring force in the determination of added mass an additional experiment has been carried out to determine the restoring force of each section as a function of vertical displacement at $Fn = 1.14$ and zero frequency of oscillation. To this end the model was vertically displaced in small steps and the resulting restoring moment was measured as a function of the heave displacement. The total vertical force recordings in the time domain were reduced by the measured nonlinear restoring forces. The in-phase component of the corrected vertical force was used to determine the added mass. The experimental results for added mass, obtained in this way, have been compared with strip theory calculations in figure 8. The experimental longitudinal distribution of the added mass is now almost independent of the oscillation frequency and there are no negative values for some of the sectional added masses, which resulted when linear hydrostatic restoring forces were used in the analysis. The agreement between experiment and computation is improved by this procedure, but the added mass of the flat sections aft is overestimated, as shown in figure 8.

The determination of the damping distribution is not affected by this method. As shown in figure 8 the damping coefficients according to strip theory calculation are also overestimated in the aft part of the hull.

This is due to the erroneous sectional added mass prediction in this part of the hull, using hydrostatic restoring force coefficients.

A better correlation is obtained when the longitudinal damping distribution is based on the slope of the experimental added mass distribution as discussed, using equation (2) for the forward speed transformation, see figure 8.

Apparently this approximation of the forward speed effect on the damping distribution is also useful at high Froude numbers.

Three-dimensional computations, using the WAMIT computer program, have been carried out for this hull form, using the experimental values for trim, sinkage and the wave profile along the hull. The agreement with the experiment is not improved, compared with strip theory results, except for the predicted added mass of the aft sections, which is closer to the measurements (Adegeest 1989).

In view of motion response calculations the wave exciting forces for this particular hull form have been measured using a restrained model in regular waves. The strip theory predicts these forces quite well, as found earlier for much lower Froude numbers (Gerritsma & Beukelman 1967). This may be due to the relatively small diffraction part, compared with the Froude–Krylov force, resulting from the undisturbed wave pressure integrated over the hull surface. Even gross errors in the computed diffraction forces may be masked in the predicted total wave force. This does not apply to all cases. For instance for very small wavelengths, say $\lambda/L < 0.5$, which are of interest for hydroelastic behaviour of long ships, the wave exciting forces cannot be predicted by strip theory methods, as shown by Moeyes (1976). He used a 24-segment model of a tanker to determine wave forces to study springing phenomena, caused by wave excitation in the frequency range of the two-node natural frequency of vertical ship vibrations. It is expected that three-dimensional panel methods are more suited to compute the wave excitation in this case.

For the displacement-type hull-form trim, sinkage and wave profile have not been taken into account using the strip theory calculation for the determination of the sectional hydrodynamic forces and hydrostatic values for the restoring forces have been assumed. Apparently errors due to this rather strong simplifications tend to cancel each other.

For $Fn = 1.14$ the motion amplitude prediction is still satisfactory, but the added resistance is underestimated. It may be concluded that simplified calculation methods for the prediction of heave and pitch amplitude response functions still hold for high forward speeds. The exceptions concern related phenomena, such as relative motions and added resistance, which depend to some extent on the longitudinal distribution of the hydrodynamic forces. The distribution of these forces at high speeds cannot be determined without experimental data, because of the speed-dependent, strongly nonlinear restoring forces and moments.

4. Restricted water depth

Vertical motions in shallow water are important in view of the allowable keel-clearance of very large ships entering coastal waters.

As in the case of unlimited water depth, added mass, damping and their longitudinal distributions have been determined for the Series Sixty $C_B = 0.70$ ship model for water-depth/draught ratio's $h/T = 1.15$–2.4 (Beukelman 1982). The results show increasing added mass and damping coefficients with decreasing water depth, in particular when $h/T < 1.5$.

The distribution of added mass, normalized with the total added mass, is not greatly influenced by the water-depth/draught ratio, but a significant increase of damping near the bow with decreasing water depth is observed.

The results of mass and damping calculations, using strip theory and multipoles for the two-dimensional cross-section approximations, as well as two- and three-dimensional numerical panel methods, using source distributions, do not give a pronounced preference for one of these methods (Beukelman *et al.* 1983).

The use of numerical methods assuming potential flow, is not fully obvious here, because an important influence of viscosity could be expected, in particular for very small keel clearances.

However, the calculations agree rather good with the experiment, except for frequencies lower than $\omega\surd(L/g) = 1.9$, where the three-dimensional method gives a slightly better result than the other two methods.

It may be concluded that each of these numerical methods is applicable for the prediction of vertical ship motions in shallow water for engineering purposes.

The horizontal hydrodynamic forces acting on a slowly oscillating ship in shallow water are of interest for the determination and analysis of the steering and manoeuvring qualities of ships. The same Series-Sixty ship model has been used to carry out forced low-frequency sway and yaw oscillations, as well as static drift angle tests (Beukelman & Gerritsma 1983).

For a small water depth, $h/T = 1.15$, the agreement between calculation and experiment is not satisfactory, in particular for sway damping. Apparently viscosity has an important influence on the damping force distribution in the after part of the ship. Also the prediction of added mass for $h/T = 1.15$ does not agree with the model experiment.

For relatively high Froude numbers very important forward speed effects on the horizontal hydrodynamic forces may be expected and the Froude number based on water depth: $Fnh = V/\surd(gh)$ should be considered in this respect. Strong non-linearities occur when the ship speed approaches $\surd(gh)$.

Recently an experiment with a model of a RO–RO passenger ferry has been carried out in the Delft Ship Hydrodynamics Laboratory to investigate the directional stability as a function of trim and forward speed in shallow water. The tests were carried out with rudders fixed in the neutral position and propellers running at the self-propulsion point of the ship to avoid unrealistic propeller loading and a corresponding increase of the effective rudder area.

For the fixed-rudder case linearized equations of motion are used to determine the stabilty roots σ_i:

$$\left.\begin{aligned} (m-Y_{\dot{v}})'\,\dot{\beta}+Y'_{\beta}\beta+Y'_{\dot{r}}\dot{r}'+(Y_r-m)'\,r' &= 0 \quad \text{(sway)},\\ N'_{\dot{v}}\,\dot{\beta}+N'_{\beta}\beta+(I_{zz}-N_{\dot{r}})'\,\dot{r}'-N'_r r' &= 0 \quad \text{(yaw)}, \end{aligned}\right\} \tag{3}$$

with the solution:

$$\beta = \beta_i\,\mathrm{e}^{\sigma_i t}, \quad r' = r'_i\,\mathrm{e}^{\sigma_i t}.$$

Positive stability roots indicating directional instability, were found in all considered conditions, which included water depths ranging from $h/T = 1.4$–2.2 and speeds corresponding to $Fn = 0.262$. For speeds exceeding $V/\surd(gh) = 0.8$ the fixed control instability increases very rapidly and trim by the bow enhances this effect.

For $h/T = 2.2$ and two trim conditions the stability roots as a function of forward speed are depicted in figure 9. Although a certain amount of fixed control instability is acceptable in many cases, the sharp increase of the instability at certain combinations of trim, forward speed and water depth should be a matter of concern,

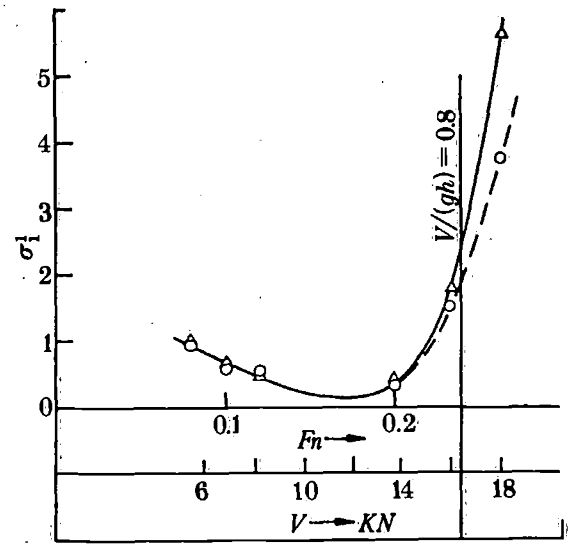


Figure 9. Stability roots of a roll-on–roll-off passenger ferry in shallow water. $h = 12.5$ m, $L = 126.1$ m, $B = 22.7$ m, $T = 5.69$ m, $h/T = 2.2$. o, Even keel; △, trim by the bow 0.75 m.

as pointed out by Bishop *et al.* (1988), in particular for high-powered ships, with a high-speed potential in shallow water.

It should be remarked that this model experiment has been carried out in a conventional towing tank and consequently the width of the tank caused a virtual depth decrease. In the considered case this would correspond to a depth decrease of 15% and a 7% decrease of the critical wave speed.

References

Haskind, M. D. & Rieman, I. S. 1946 Method for the determination of the heave and pitch characteristics of a ship (transl.). *Bull. Acad. Sci. URSS, Classe Sci. Tech.* no. 10.

Gerritsma, J. & Beukelman, W. 1967 Analysis of the modified strip theory for the calculation of ship motions and wave bending moments. *Int. Shipbuilding Prog.* **14**, 319–337.

Gerritsma, J., Beukelman, W. & Glansdorp, C. C. 1974 The effects of beam on the hydrodynamic characteristics of ship hulls. *Tenth Symposium Naval Hydrodynamics, Cambridge, Massachusetts*, pp. 3–33.

Ursell, F. 1957 On the virtual mass and damping of floating bodies at zero speed ahead. *Symposium on the behaviour of ships in a seaway, Wageningen*, pp. 374–387.

Tasai, F. 1960 Measurements of the wave height produced by the forced heaving of cylinders. *Rep. Res. Inst. appl. Mech., Kyushu University*, **8**, 279–310.

Vugts, J. H. 1970 The hydrodynamic forces and shipmotions in waves. Thesis, Delft University of Technology.

WAMIT 1988 A radiation–diffraction panel program for wave–body interaction. Dept of Ocean Engineering, MIT.

Adegeest, L. J. M. 1989 Analysis of a three dimensional method for the calculation of the distribution of hydrodynamic coefficients and ship motions at forward speed. Thesis, Delft University of Technology, Faculty of Mechanical Engineering and Maritime Technology.

Maruo, H. 1978 Prediction of hydrodynamic forces and moments on ships in heaving and pitching by taking into account of the three-dimensional effect. *15th International Towing Tank Conference, The Hague*, pp. 53–54.

Gerritsma, J. 1988 Motions, wave loads and added resistance in waves of two Wigley hull forms. *Rep. no.* 804. Delft Shiphydrodynamics Laboratory.

King, B. K., Beck, R. F. & Magee, A. R. 1989 Seakeeping calculations with forward speed using time-domain analysis. *Seventh Symposium on Naval Hydrodynamics, The Hague*, pp. 577–596.

Blok, J. J. & Beukelman, W. 1984 The high-speed displacement-ship systematic series hull form seakeeping characteristics. *SNAME*, *New York* **92**, 125–150.

Keuning, J. A. 1990 Distribution of added mass and damping along the length of a ship model moving at high forward speed. *Int. Shipbuilding Prog.* **37**, 123–150.

Moeyes, G. 1976 Measurement of exciting forces in short waves. *Rep. no.* 437. Delft Shiphydrodynamics Laboratory.

Beukelman, W. & Gerritsma, J. 1982 The distribution of hydrodynamic mass and damping of an oscillating shipform in shallow water. *Int. Shipbuilding Prog.* **29**, 297–315.

Beukelman, W., Huysmans, R. H. M. & Keuning, P. J. 1983 Calculation methods of hydrodynamic coefficients of ships in shallow water. *Rep. no.* 571A. Delft Ship hydrodynamics Laboratory.

Beukelman, W. & Gerritsma, J. 1983 The longitudinal distribution of low frequency hydrodynamic derivatives for lateral motions in shallow water. *Rep. no.* 603. Delft Ship hydrodynamics Laboratory.

Bishop, R. E. D., Price, W. G. & Temarel, P. 1988 On the dangers of trim by the bow. *Trans. R. Inst. nav. Architects* **131**, 281–303.

Discussion

R. Eatock Taylor (*University of Oxford, U.K.*). Professor Gerritsma has shown some impressive comparisons between the results of experiments and strip theory. There are several variants of the theory, and my discussion is prompted by the approach published by Newman (1977). This is considerably simpler than the 'unified' slender-body theory which came later, and there is some evidence that it provides superior predictions to those of earlier strip theories. It is in the expression for distributed vertical force that Newman's 1977 formulation differs significantly from those of Gerritsma & Beukelman (1967), Salvesen *et al.* (1970), and others.

The most marked difference between Newman's formulation and others appears to be the selection of frequency at which the two-dimensional added mass and damping coefficients are evaluated: in the former the wave frequency is used where as other theories use the wave encounter frequency. This and other differences were investigated by Andrew (1985), who also made an extensive series of measurements of the distributed wave force. The experiments were conducted at the Admiralty Research Establishment (Haslar) using a model of a destroyer, some of the equipment being borrowed from Professor Gerritsma's laboratory.

Andrew found that the phase angles of the integrated heave force and pitch moment, which were not well predicted by Salvesen *et al.*'s theory (STF) were in general predicted much more reliably by the Newman theory (N). By implication the latter should give better predictions of motion responses, particularly relative motions. Andrew also observed that N was generally better than STF in correlating with the magnitude and phase of the distributed vertical force over the afterbody region of the hull particularly at higher speeds and wave frequencies. Over the forebody region, however, N was found to overpredict the force to a greater extent than STF. Near the middle of the hull, Andrew noted that these two theories gave generally similar results but underpredicted the measured values.

It appears that Newman's 1977 formulation has not received the attention accorded to other strip theories. It may provide better predictions of motions, although estimates of local bending moments and shear forces may not be improved to the same extent. There is clearly therefore a need for fully three-dimensional analyses such as presented in the next paper. Comparison of results with those from careful experiments on segmented models, such as presented by Professor Gerritsma, will form a vital part in the validation of these latest numerical analyses. Although his paper emphasized the forced oscillation problem, his comments on this wider issue of hydrodynamic loading on ships would be of great interest.

J. GERRITSMA. The distribution of vertical wave loads, as discussed by Professor Eatock Taylor, has been measured and calculated for the considered destroyer hull form. The calculated distributions of the vertical wave force, according to a strip theory method and with the three-dimensional approximation mentioned in the paper, were both very near to the experimental result. Apparently the diffraction part of the wave force is small compared with the Froude–Kriloff part of the total wave force. Therefore an improvement of the calculation, as proposed, has little influence on the total wave force in this particular case. For less slender hull forms, small wave lengths and low frequencies improvements with three-dimensional methods can be expected. I agree that such three-dimensional solutions require careful experimental verification.

M. GREENHOW (*Brunel University, U.K.*). My comment concerns low-frequency forced oscillations. Here, to get sizeable forces, one has to drive the body with large heave amplitudes. The problem is then nonlinear since the wetted surface (or its distance from the free surface for submerged bodies) changes significantly, although the free surface stays quite flat. How then can we attribute the forces to added mass and damping type forces? Perhaps the problem is best solved as described by Beck at the preceding IUTAM Conference.

J. GERRITSMA. In general forced oscillation experiments are carried out for a range of motion amplitudes and frequencies to study nonlinearities. The accuracy of the force dynamometers limits the lowest frequencies and also reflections of the generated damping waves against the tank walls are important in this respect. For normal ship hull forms and moderate vertical motion amplitudes linearity is an acceptable approximation for damping and added mass, except for very high and very low frequencies which are less important for ship motions in waves.

D. W. ROBINSON (*Lloyd's Register, London, U.K.*). Comparisons were shown of pitch and heave transfer functions for a destroyer hull form at high forward speed and they showed excellent agreement between model experiments and strip theory. Assuming the experiments were conducted at reasonably low wave heights and observing the nonlinear nature of Professor Gerritsma's added mass and damping results, could such agreement for pitch and heave have been obtained for higher wave heights? Also, does the nonlinear nature of the addedd mass, in particular, bring into question the validity of using conventional segmented model experiments to measure moment and force transfer functions using low wave amplitudes?

J. GERRITSMA. Mr Robinson rightly remarks that the destroyer motions were calculated for moderate wave conditions. High waves could introduce nonlinearities, but these were not considered in the experiments. In the forced oscillation experiment the primary cause of nonlinearity was the strongly nonlinear restoring force at high speed. When this had been subtracted from the excitation force, the assumption of linear added mass was very acceptable, as shown in the paper.

H. Maeda (*Japan*). I remember being impressed by Professor Gerritsma's forced oscillation experiment in 1963. This test uses sinusoidal oscillations. In contrast, transient oscillations include a range of frequencies for hydrodynamic forces, and they save experiment time as one test covers the range of frequencies. This forced oscillation test was developed in Japan, mainly by Takezawa, and I have successfully applied it several times.

J. Gerritsma. Transient forced oscillation tests have been carried out in the Delft Ship Hydrodynamics Laboratory, as a substitute for harmonic oscillation. Up to now the last method is preferred because of greater accuracy in a large range of frequencies and the possibility to analyse nonlinear phenomena in more detail.

Additional references

Andrew, R. N. 1985 Distributed vertical force on a ship with forward speed. D.Phil. thesis, University of London, U.K.

Gerritsma, J. & Beukelman, W. 1987 Analysis of the modified strip theory for the calculation of ship motions and wave bending moments. *Int. Shipbuilding Prog.* **14**, 319–337.

Newman, J. N. 1977 *Marine hydrodynamics*. Cambridge, Massachusetts: MIT Press.

Salvesen, N., Tuck, E. O. & Faltinsen, O. 1970 Ship motions and sea loads. *Soc. nav. Architects mar. Engrs Trans.* **78**, 250–287.

FORCED OSCILLATION EXPERIMENTS

J.GERRITSMA

1.INTRODUCTION

Forced oscillation experiments with scale models are used in ship research for the determination of hydrodynamic forces associated with oscillatory motions of ships and other maritime constructions. The results obtained with oscillating models in still water have been used to determine the coefficients of the equations of motion, for instance to validate theoretical models of the considered dynamic flow problems.

These have included simplyfied engineering solutions for the prediction of the motion of a ship advancing in irregular sea waves, as well as advanced numerical three dimensional methods for the determination of hydrodynamic forces, based on rational theories.

In these oscillation techniques a scale model is forced to carry out harmonic oscillations of known amplitude and frequency. The required force is split up in a component in phase with the motion of the body to obtain the hydrodynamic- or added mass, whereas the quadrature component is associated with damping.

The experiment may concern one particular mode of motion, for instance heaving of a ship, but more complicated coupled motions are for instance generated by a so called planar motion mechanisme to obtain linear and non-linear hydrodynamic coefficients of the equations of motion in a horizontal plane for the simulation of steering and manoeuvring of ships.

In a few cases free running self-propelled models with an internal force- or moment-excitator have been used to investigate rolling or lateral low-frequency motions. The precession moment of gyroscopes, mounted inside the model, produced a roll excitation and the athwartship thrust of air-propellors mounted on a self-propelled model has been used to produce yawing moments and sway forces.

Full size ship rolling has been generated by oscillating the rudder or the stabilizers, but the magnitude of the exciting moment is difficult to measure in such cases.
An athwartship moving mass, designed as a roll damping device, has been used as a roll-motion excitator of which the excitation moment could be determined with acceptable accuracy.

Another category of forced oscillation experiments concerns the sloshing of a fluid with a free surface in a tank. The motion of the fluid in a tranversily arranged tank generates a rolling moment and its quadrature component produces roll damping. This property has been used to design passive anti-rolling tanks, based on systematic experiments with model tanks.

Another aspect of sloshing fluid in oscillating tanks concerns the hydrodynamic load on the construction resulting from the fluid motion, which is of interest for ships carrying liquid cargo. Model experiments with oscillating model tanks have been carried out to determine such loads.

As far as known to the Author the first forced oscillation experiment with a ship model has been carried out by Haskind and Rieman [1]. Forced heaving motions at zero forward speed were carried out in a range of frequencies and amplitudes of motion. The results showed frequency dependent damping, vanishing at high and low frequencies, and frequency dependent hydrodynamic mass. The influence of non-linearites appeared to be small for the considered wall-sided mathematical shipmodel.

The vertical harmonic motion was obtained with a Scotch yoke mechanisme, using a soft spring as a dynamometer, the measured compression of the spring being proportional to the excitation force. The phase difference between the upper and lower end of the spring provided the additional information to determine the in-phase and quadrature components of the heaving force. Later, similar techniques have been used in various ship hydrodynamic laboratories as a consequence of the increased interest in the dynamics of ships in a seaway.

Some results of forced oscillation experiments, carried out in the Delft Shiphydrodynamic Laboratory, are briefly summarized here to illustrate the possibilities of this experimental technique with respect to the analysis and validation of theoretical methods as used in ship science and engineering applications.

In addition, some recent results concerning the influence of high forward speed and restricted water depth on hydrodynamic motion parameters of a ship will be presented.

2. VERTICAL MOTIONS.

The experimental set-up as used by Haskind and Rieman has been developed to include the possibilty to determine the hydrodynamic coefficients of the coupled equations of motions of heave and pitch at forward speed.
These may be written as two linear coupled equations with frequency dependent coefficients:

$$\left.\begin{aligned}(a+\rho\nabla)\ddot{z}+b\dot{z}+cz+d\ddot{\theta}+e\dot{\theta}+g\theta&=\bar{F}e^{i\omega t}\\(A+I_{yy})\ddot{\theta}+B\dot{\theta}+C\theta+D\ddot{z}+E\dot{z}+Gz&=\bar{M}e^{i\omega t}\end{aligned}\right\}(1)$$

with: z = heave, θ = pitch, ω = circular frequency, ∇ is the volume of displacement and I_{yy} is the mass moment of inertia. $\bar{F}$ and $\bar{M}$ are the complex exciting force and moment amplitudes.

The agreement between motion amplitudes and phase characteristics, as derived from model tests in regular waves and corresponding calculations, using hydrodynamic coefficients a,b,d,e,A,B,D,E obtained from forced oscillation experiments as well as measured wave forces and moments F,M in the equations of motion, is satisfactory for conventional hull forms at moderate forward speeds as shown in [2].
Amplitude and phase characteristics, obtained from tests in regular waves with varying wave heights and tests in longcrested irregular waves, using cross-spectral analysis to obtain these characteristics, seem to indicate that the assumption of linearity is an acceptable simplification, at least for engineering applications.

Strip theory computations using two dimensional approximations for added mass and damping of ship cross sections neglect the mutual interference of the flow between those sections. Also three- dimensional effects at the ends of a ship, which in particular are important for pitching motions, are not taken into account.
Nevertheless computed motion response functions for heave and pitch agree quite well with experiments in many cases including length-beam ratios as small as 2.5.

A systematic model series derived from one particular hull form (Series Sixty CB=.70), of which the length-beam ratio varied from L/B = 4 to 20, has been force-oscillated in still water at forward speeds corresponding to Fn = 0.2 and 0.3. In addition, motion response and added resistance experiments for heave and pitch in regular waves have been carried out [3]. The results confirm in more detail the applicability of the strip method. An example of measured added mass and damping compared with strip theory calculation is given in Figure 1

However, predictions of motions in following waves as well as predicted relative motions and added resistance at high forward speeds using strip theory calculations do not agree with experimental results in all cases. Also, strip theory calculations cannot be used for the determination of forces in waves which are very short compared with the length of the ship. Pitching amplitudes of long slender ships are over-estimated and they are under-estimated in the case of sailing yachts.

To study strip-theory methods in more detail two-dimensional calculations for ship like cross sections have been compared with experimental results. First of all this concerned Ursell's solution for the vertical oscillation of a circular cylinder [4] and its generalisation for actual ship like cross sections by Tasai [5] and others. The effect of shallow water has been included by Porter [6].

Forced oscillation experiments with cylinders have been carried out by Vugts.. These tests include a half-circular cross-section as well as a triangle, and other ship-like cross sections [7]. The cylinders spanned the width of the towing tank to avoid three-dimensional effects at the ends of the cylinders, and beaches on both ends of the towing tank have been used to absorb the radiated damping waves.

In Figure 2 the experimental results for the half-circular- and the triangular cross-sections are compared with computed mass and damping.

In general the agreement is satisfactory, also for the other cross sections: rectangular sections with beam-draught ratios 2,4 and 8 and two ship-like cross sections.

The experiments confirm the applicability of Ursell's work, which has been of great value for the progress in shipmotion research.

Also computations based on finite element methods have been carried out, showing similar accuracy. These proved to be useful in cases of complicated bottom and/or wall geometry.

Based on the results of the cylinder tests it may be accepted that the effects of viscosity can be neglected for vertical motions with moderate motion amplitudes.

Forward speed effects on the distribution of the hydrodynamic forces along the length of a ship are important, because they introduce longitudinal hydrodynamic asymmetry and corresponding coupling between heave and pitch motions.

To investigate forward speed- and three-dimensional effects a segmented model technique has been introduced. The experimental set up is shown in Figure 3.
Each separate segment of the shipmodel is connected to a stiff beam by means of a dynamometer to measure the vertical hydrodynamic force acting on the segments. The beam oscillates vertically with a harmonic motion and the forces on each section are reduced to added mass and damping components. A continuous line, as drawn in Figure 4, approximates the distribution of the measured added mass and damping.

Strip theory methods predict the effect of forward speed on damping reasonable well for conventional shipforms and moderate Froude numbers, say $Fn < 0.40$.
According to strip theory the longitudinal distribution of the damping coefficient b' is given by:

$$b' = N' - V.dm'/dx \qquad (2)$$

where: N'- the two-dimensional damping coefficient of the cross section at x, and m' is the corresponding added mass.
In particular the linear speed dependency of the damping cross coupling coefficients of pitch and heave and their symmetry relation have been confirmed by experiments [2].

In Figure 4 the longitudinal distribution of the heave damping coefficient for a Series 60, CB=.70 hull form at Fn=.20, calculated with a strip theory method, is compared with measured distributions.

To investigate three-dimensional effects, the longitudinal distribution of the hydrodynamic forces at zero forward speed has also been calculated with the three-dimensional radiation/diffraction panal method WAMIT, developed at M.I.T.[8].
For a number of ship forms Adegeest [9] determined the contribution of each segment of the hull to the total hydrodynamic mass and damping, using a suitable panel distribution to fit the considered model segments.
Such panel distibutions can be generated also for conditions with trim and heel angle and in addition they may include the wave contour along the hull as well, if this is known.
For a Series Sixty Cb=0.70 model at low frequencies of oscillation and zero forward speed the WAMIT result agrees better with the experiment as compared with the two-dimensional strip theory prediction, but for all other frequencies very little difference between the two computed results exist, see Figure 5.

An similar improvement for the low frequency damping and added mass distribution for this particular model has been found earlier by Maruo with a revised formulation of a his slender body approximation [10].

The three-dimensional distribution of the damping coefficient at zero forward speed, as computed by the panel method, may be transformed to the case of forward speed using the expression (2), as a practical approximation.
For a sailing yacht hull form with L/B=4.5, B/T=2.5 this improved considerably the predicted heave and pitching motions in the range of maximum motion amplitudes, as shown in Figure 6. The same method proved to be succesful in the case of asymmetric cross-sections, which have to be considered when a yacht sails with a heel angle.

An extensive experiment to measure hydrodynamic coefficients and wave forces as well as the motions and the added resistance in regular head waves, has been carried out with a L/B=10 symmetric mathematical Wigley hull form [11]. The hull form is given by:

$$\frac{y}{B}=\left(1-\left(\frac{z}{T}\right)^2\right)\left(1-\left(\frac{2x}{L}\right)^2\right)\left(1+0.2\left(\frac{2x}{L}\right)^2\right)+\left(\frac{z}{T}\right)^2\left(1-\left(\frac{z}{T}\right)^8\right)\left(1-\left(\frac{2x}{L}\right)^2\right)^4 \qquad (3)$$

King [12] developed a three-dimensional time domain seakeeping calculation method using a lineair theory to determine the hydrodynamic forces on the hull due to motions, as well as the exciting forces due to the incident waves. The potential flow problem was solved in the time domain rather than in the frequency domain, but the results have been presented in frequency domain coefficients of the motion equations to compare with experiments and strip theory results. This approach covers the three-dimensionality of the hull and forward speed effects to some extend.

In [12] an extensive comparison has been made of this three dimensional method with strip theory computations and the Delft experiments, including also the wave exciting forces.
Generally the agreement of both calculation methods with the experiment is good, with some slight improvements of the three-dimensional panel method compared with the strip theory method.
In particular the heave-pitch damping cross coupling coefficients are better predicted by the three-dimensional method and are very close to the measured values, see Figure 7. Apparently the longitudinal distribution of the damping coefficient as a result of forward speed is predicted better by the three-dimensional method for this symmetric slender model.

3. HIGH FORWARD SPEED

So far the forced oscillation experiments concerned moderate ship speeds and conventional displacement type hull forms.
An important area of interest is the seakeeping behaviour at high speeds. In particular the operability of a fast ship depends to a large degree on the motion response and the vertical accelerations of the ship in a seaway. The reliable prediction of these quantities is a valuable tool for the designer of such vessels.

The applicabilty of strip theory methods for high speed naval ships has been investigated by Blok and Beukelman [13]. x appli-cability

They concluded that motion amplitude and vertical acceleration predictions, based on strip theory, agree very well with measured motions for a high speed displacement type hull form at speeds as high as Fn = 0.57 and 1.14. Also added resistance and relative motions were predicted well for Fn =0.57, but some differences occur at Fn = 1.14, see Figure 8. The prediction of these two important aspects of ship behaviour in waves depends on the phase of the motion with respect to the wave and this in turn depends to some extend on the longitudinal distribution of damping and added mass.

The speed range considered in this case exceeds the commonly accecpted limits for the applicability of calculation methods. The hydrodynamic pressure distribution at high forward speed and zero frequency of oscillation in calm water may be characterized by pronounced trim, sinkage, dynamic lift and non linear ship waves.
Generally these phenomena are not included in prediction methods based on calculations.

To study these effects in more detail Keuning carried out forced oscillation experiments with a segmented model of the same hull form as used by Blok and Beukelman at Fn = 0.57 and Fn = 1.14, [14].
Their model is the parent form of a high speed displacement hull form series, with L/B=8, B/T=4, Cb=0.396. Although the model has flat sections aft, the relatively low rise of the center of gravity observed during the resistance tests in calm water indicated a moderate influance of dynamic lift in the considered speed range.
The oscillation experiments have been carried out without trim and sinkage, but also with trim and sinkage corresponding to the considered forwards speeds. The former condition

appeared to be unrealistic because of the extreme bow wave and the very small immersion of the flat sections aft. Thus only the trim and sinkage condition, as measured at the considered forward speed in still water, will be considered here.

The in-phase component of the hydrodynamic force contains a contribution of the restoring force, as shown in equation (4).

For the heaving model:

$$F \cos \varepsilon = (a+\rho\nabla) z_a \omega^2 - c z_a \qquad (4)$$

where ε is the phase of the excitation force with respect to the motion.

A similar expression holds for each of the separate segments. For moderate speeds and conventional ship forms the linear hydrostatic restoring force at zero speed and frequency may be used as an acceptable simplification, but at high Froude numbers the pressure ditribution differs considerably from this assumption.

At zero forward speed the hydrostatic restoring forces of the separate segments are only slightly non-linear with respect to vertcal displacement, but high forward speed introduces significant non-linearities.

To include the actual restoring force in the determination of added mass an additional experiment has been carried out to determine the restoring force of each section as a function of vertical displacement at Fn = 1.14 and zero frequency of oscillation. To this end the model was vertically displaced in small steps and the resulting restoring moment was measured as a function of the heave displacement. The total vertical force recodings in the time domain were reduced by the measured non- x recording linear restoring forces:

$$F\,(\text{corrected}) = F\,(x, \ddot{z}, \dot{z}, z, t) - F(x, z) * z \qquad (5)$$

The in-phase component of the corrected vertical force produced the added mass.

The experimental results for added mass obtained in this way are compared with strip theory calculations in Figure 9. The experimental longitudinal distribution of the added mass is now almost independent of the oscillation frequency and there are no negative values for some of the sectional added masses, which resulted when linear hydrostatic restoring forces were used in the data reduction. The agreement between experiment and computation is improved by this procedure, but the added mass of the flat sections aft is over-estimated, as shown in Figure 9.

The determination of the damping distribution is not affected by this method. As shown in Figure 9 the damping coefficients according to strip theory calculation are also over estimated in the aft part of the ship. x over-estimated

This is due to the erronous sectional added mass prediction in x erroneous this part of the hull, using hydrostatic restoring force coefficients.

A better correlation is obtained when the longitudinal damping distibution is based on the slope of the experimental added mass distibution as discussed, using equation (2) for the forward speed transformation, see Figure 9.

Apparently this approximation of the forward speed effect on the damping distibution is also useful at high Froude numbers.

Three-dimensional computations, using the WAMIT computer program, have been carried out for this hull form, with trim and sinkage, also taking into account the wave profile along the hull. The agreement with the experiment is not improved, compared with strip theory results, except for the predicted added mass of the aft sections, which is closer to the measurements [9]. The inclusion of the wave profile along the hull as well as trim and sinkage did not improve the numerical results.

In view of motion response calculations the wave exciting forces for this particular hull form have been measured using a restrained model in regular waves. The strip theory predicts these forces quite well, as found earlier for much lower Froude numbers [2]. This may be due to the relatively small diffraction part, compared with the Froude-Krylov force, resulting from the undisturbed wave pressure integrated over the hull surface. Even gross errors in the computed diffraction forces may be masked in the predicted total wave force. This does not apply to all cases. For instance for very small wave lengths, say $/L < 0.5$, which are of interest for hydro elastic behaviour of very long ships, the wave exciting forces cannot be predicted by strip theory methods, as shown by Moeyes [15]. He used a 24 segment model of a tanker to determine wave forces to study springing phenomena, caused by wave exitation in the frequency range of the two-node natural frequency of vertical ship vibrations. It is expected that three-dimesional panel methods are more suited to compute the wave excitation in this case.

For Fn = 0.57 the predition of the pitch and heave amplitudes, vertical accelerations and added resistance, using strip theory, agrees very well with experiments in regular waves. Trim, sinkage and wave profile have not been taken into account for the determination of the sectional hydrodynamic forces and hydrostatic values for the restoring forces have been assumed. Apparently errors due to this rather strong simplifications tend to cancel each other.
For Fn = 1.14 the motion amplitude prediction is still satisfactory, but the added resistance is under-estimated.
A similar picture holds for the case of a hard chine semi planing hull, with L/B = 4.66, B/T = 3.48, CB = 0.38. In Figure 10 the comparison is shown for speeds corresponding to Fn = 0.72 and 0.91 [16]. The surprisingly good correlation may be partly due to the relatively high damping of hard chine planing hulls.

It may be concluded that symplified calculation methods for the prediction of heave and pitch amplitude response functions still hold for high forward speeds. The exceptions concern related phenomena, such as relative motions and added resistance, which depend to some extend on the longitudinal distribution of the hydrodynamic forces. The distribution of these forces at high speeds cannot be determined without experimental data, because of the speed dependent, strongly non-linear restoring forces and moments.

4. RESTRICTED WATERDEPTH

Vertical motions in shallow water are important in view of the allowable keel-clearance of very large ships entering coastal waters. For instance, the permission to enter the Eurochannel for fully loaded very large ships is based on the expected ship motions in the low frequency part of the prevailing wave spectrum. In this respect keel clearances as small as 15% of the ship's draught, or less are of interest.

As in the case of unlimited water depth, added mass, damping and their longitudinal distributions have been determined for the Series Sixty Cb.70 ship model for water-depth/draught ratio's $h/T = 1.15$ to 2.4 [17].

The results show increasing added mass and damping coefficients with decreasing waterdepth, in particular when $h/T < 1.5$.

The distribution of added mass, normalised with the total added mass, is not greatly influenced by the waterdepth / draught ratio, but a significant increase of damping near the bow with decreasing waterdepth is observed.

The results of mass and damping calculations, using strip theory and multipoles for the two-dimensional cross-section approximations, as well as two- and three-dimensional numerical panel methods, using source distributions, do not give a pronounced preference for one of these methods [18].

The use of numerical methods assuming potential flow, is not fully obvious here, because an important influence of viscosity could be expected, in particular for very small keel-clearances.

However the calculations agree rather good with the experiment, except for frequencies lower than $\omega\sqrt{L/g} = 1.9$, where the three-dimensional method gives a slightly better result than the other two methods.

It may be concluded that each of these numerical methods is applicable for the prediction of vertical ship motions in shallow water for engineering purposes.

The horizontal hydrodynamic forces acting on a slowly oscillating ship in shallow water are of interest for the determination and analysis of the steering and manoeuvring qualities of ships. The same Series-Sixty ship model has been used to carry out forced low frequency sway and yaw oscillations, as well as static drift angle tests [19].

For a small water depth $h/T = 1.15$ the agreement between calculation and experiment is not satisfactory, in particular for sway damping. Apparently viscosity has an important influence on the damping force distribution in the after part of the ship. Also the prediction of added mass for $h/T = 1.15$ does not agree with the model experiment.

For relatively high Froude numbers very important forward speed effects on the horizontal hydrodynamic forces may be expected and the Froude number based on water depth:

$Fnh = \sqrt{V/gh}$ should be considered in this respect. Strong non-linearities occur when the ship speed approaches $\sqrt{gh}$.

Recently an experiment with a model of a RoRo passenger ferry has been carried out in the Delft Ship Hydrodynamics Laboratory to investigate the directional stability as a function of trim and forward speed in shallow water.The tests were carried out with rudders fixed in the neutral position and propellers running at the self-propulsion point of ship to avoid unrealistic propeller loading and a corresponding increase of the effective rudder area.

For the fixed-rudder case linearized equations of motion are used to determine the stability roots of:

$$\left.\begin{array}{ll} (m - Y_{\dot{v}})' \dot{\beta}' + Y'_{\beta}\beta + Y'_{\dot{r}}\dot{r}' + (Y_r - m)' r' = 0 & \text{sway} \\ N'_{\dot{v}}\dot{\beta} - N'_{\beta}\beta + (J_{zz} - N_{\dot{r}})'\dot{r}' - N'_r r' = 0 & \text{yaw} \end{array}\right\} \quad (6)$$

with the solution:

$$\beta = \beta_i e^{\sigma'_i t'}, \quad r' = r'_i e^{\sigma'_i t'}$$

Positive stability roots indicating directional instability, were found in all considered conditions which included water-depths ranging from h/T=1.4 to 2.2 and speeds corresponding to Fn = 0.262. In the high speed range, trim by the bow enhances the instability.

For h/T = 2.2 and two trim conditions the stability roots as a function of forward speed are depicted in Figure 11.

Although a certain amount of fixed control instability is acceptable in many cases, the sharp increase of the instability at certain combinations of trim, forward speed and waterdepth should be a matter of concern as pointed out by Bishop and Price [20], in particular for high powered ships, with a high speed potential in shallow water.

For speeds exceeding $V/\sqrt{gh}$ =0.8 the fixed control instability increases very rapidly and trim by the bow enhances this effect.

It should be remarked that this model experiment has been carried out in a conventional towing tank and consequently the width of the tank caused a virtual depth decrease. In the considered case this would correspond to a depth decrease of 15% and a 7% decrease of the critical wave speed.

A systematic study of the directional stability of RoRo passenger ships in shallow water should include forced oscillation experiments to determine the hydrodynamic derivatives of the equations of motion, as well as real time simulator experiments ~~of~~ to analyse the steering qualities in this respect.

REFERENCES

[1] M.D.Haskind and Rieman

Method for the determination of the heave and pitch characteristics of a ship (translation).

Bulletin de l'Academie des Sciences de URSS, classe des Sciences Techniques 1946 no 10

[2] J.Gerritsma and W.Beukelman

Analysis of the modified strip theory for the calculation of ship motions and wave bending moments.

International Shipbuilding Progress 1967.

[3] J.Gerritsma, W.Beukelman, C.C.Glansdorp

The effects of beam on the hydrodynamic characteristics of ship hulls.

Tenth Symposium Naval Hydrodynamics, Cambridge, Massachusetts, 1974

[4] F.Ursell

On the virtual mass and damping of floating bodies at zero speed ahead.

Symposium on the behaviour of ships in a seaway, Wageningen, 1957

[5] F.Tasai

Measurements of the wave height produced by the forced heaving of cylinders.

Reports of Research Institute for Applied Mechanics, Kyushu University, 1960

[6] W.R.Porter

Pressure distribution, added mass and damping coefficients for cylinders oscillating in a free surface.

University of California, Institute of Engineering Research, Series 82, 1960

[7] J.H.Vugts

The hydrodynamic forces and shipmotions in waves

Thesis Delft University of Technology, 1970.

[8] WAMIT, a radiation-diffraction panel program for wave-body interaction.

Dept of Ocean Engineering MIT, 1988.

[9] L.J.M.Adegeest

Analysis of a three dimensional method for the calculation of the distribution of hydrodynamic coefficients and ship motions at forward speed.
Thesis Delft University of Technology, Faculty of Mechanical Engineering and Maritime Technology, 1989.

[10] H.Maruo
Prediction of hydrodynamic forces and moments on ships in heaving and pitching by taking into account of the three-dimensional effect.
15th International Towing Tank Conference, the Hague,1978.

[11] J.Gerritsma
Motions, wave loads and added resistance in waves of two Wigley hull forms.
Report nr 804. Delft Shiphydrodynamics Laboratory 1988.

[12] B.King
Time-domain analysis of wave-exciting forces on ships and bodies
The University of Michigan, College of Engineering, Report no 306, 1987.

[13] J.J.Blok and W.Beukelman
The high-speed displacement-ship systematic series hull forms-seakeeping characteristics.
SNAME, NewYork, 1984.

[14] J.A.Keuning
Distribution of added mass and damping along the length of a ship model moving at high forward speed.
International Shipbuilding Progress,1990.

[15] G.Moeyes

Measurement of exciting forces in short waves.

Report no 437, Delft Shiphydrodynamics Laboratory, 1976.

[16] W.Beukelman

Prediction of operability of fast semi-planing vessels in a seaway.

Report no 700, Delft Shiphydrodynamics Laboratory, 1986.

[17] W.Beukelman

The distribution of hydrodynamic mass and damping of an oscillating shipform in shallow water.

International Shipbuilding Progres 1982.

[18] W.Beukelman, R.H.M.Huysmans, P.J.Keuning

Calculation methods of hydrodynamic coefficients of ships in shallow water.

Report nr 571A Delft Ship hydrodynamics Laboratory, 1983.

[illegible] Engels als rapport no 562 A

[19] W.Beukelman, J.Gerritsma,

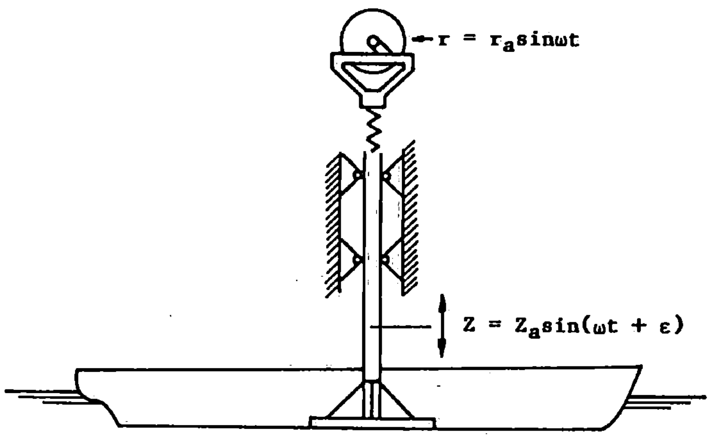
De langsscheepse verdeling van de hydrodynamische krachten bij een langzaam oscillerend schip op ondiep water.

Report nr 562 Delft Ship hydrodynamics Laboratory, 1982.

[20] R.E.D.Bishop, W.G.Price, P.Temarel,

On the dangers of trim by the bow.

Transactions Royal Institution of Naval Architects, 1988.



$r = r_a \sin\omega t$
$Z = Z_a \sin(\omega t + \varepsilon)$

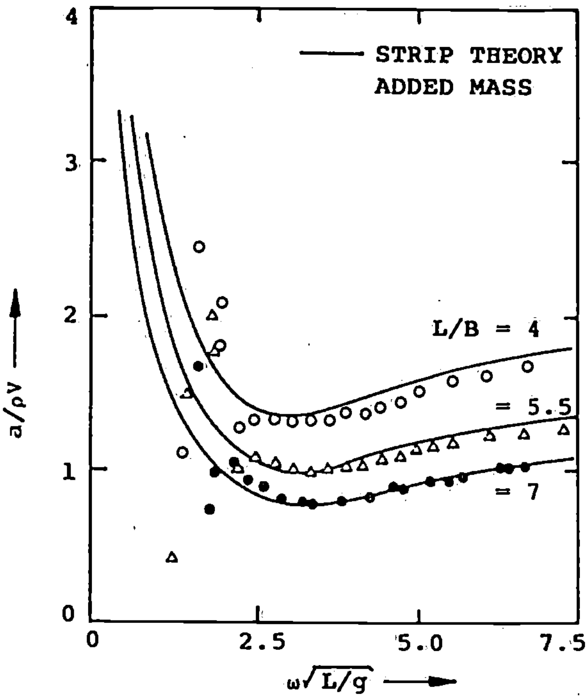


FIGURE 1A. HEAVE ADDED MASS.

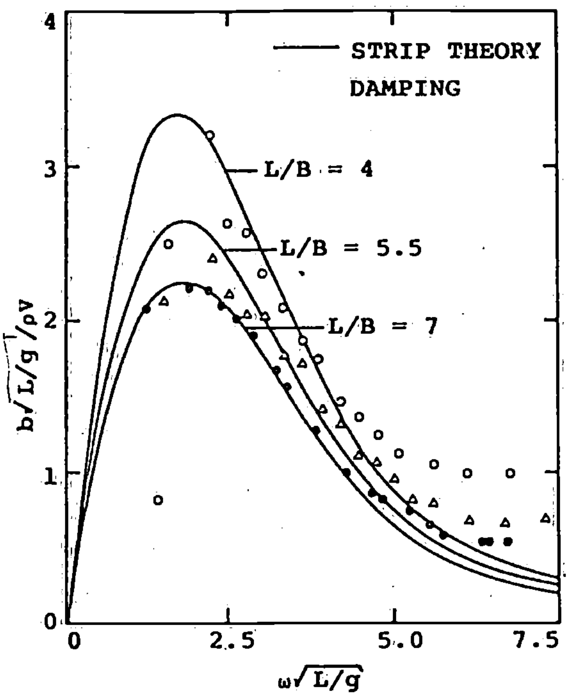


FIGURE 1B. HEAVE DAMPING.

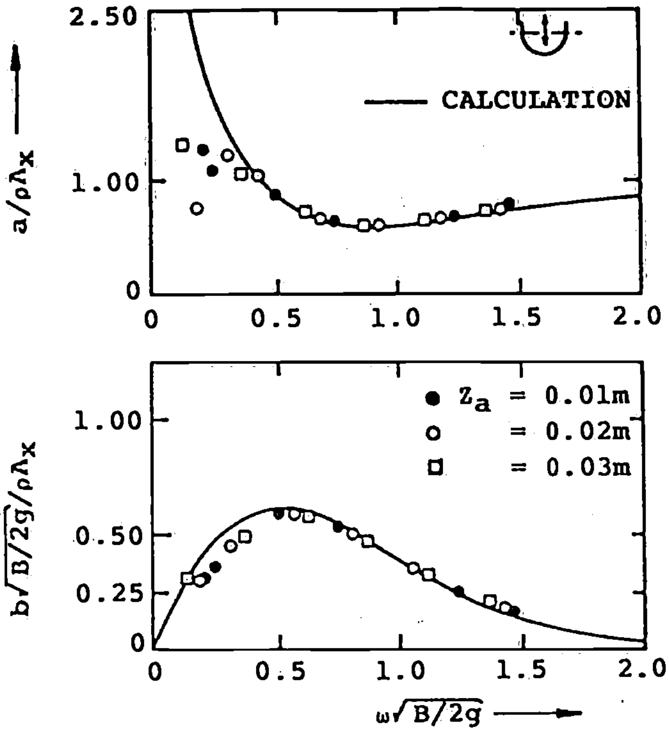


FIGURE 2A. ADDED MASS AND DAMPING OF A HEAVING CIRCULAR CYLINDER.

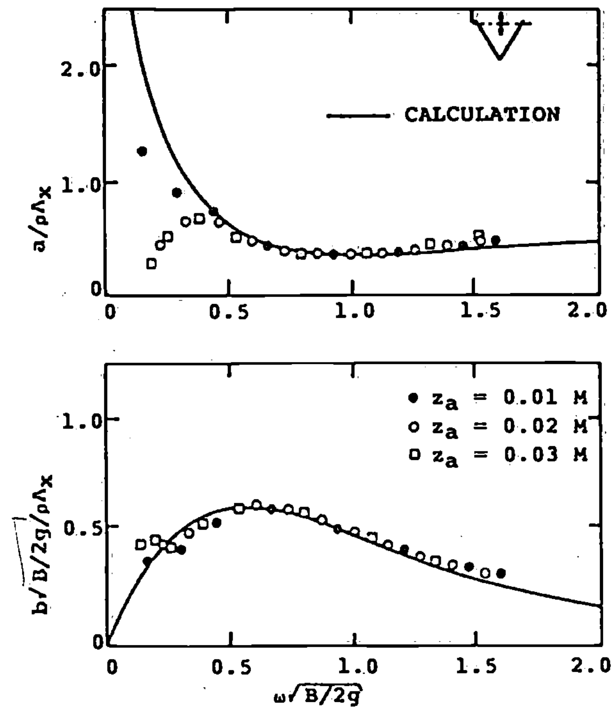


FIGURE 2B. ADDED MASS AND DAMPING OF A HEAVING TRIANGULAR CROSS SECTION.

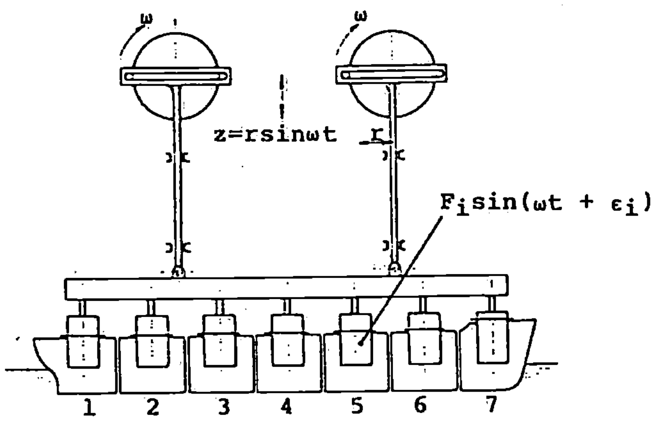


FIGURE 3. ARRANGEMENT OF OSCILLATION EXPERIMENT WITH A SEGMENTED MODEL.

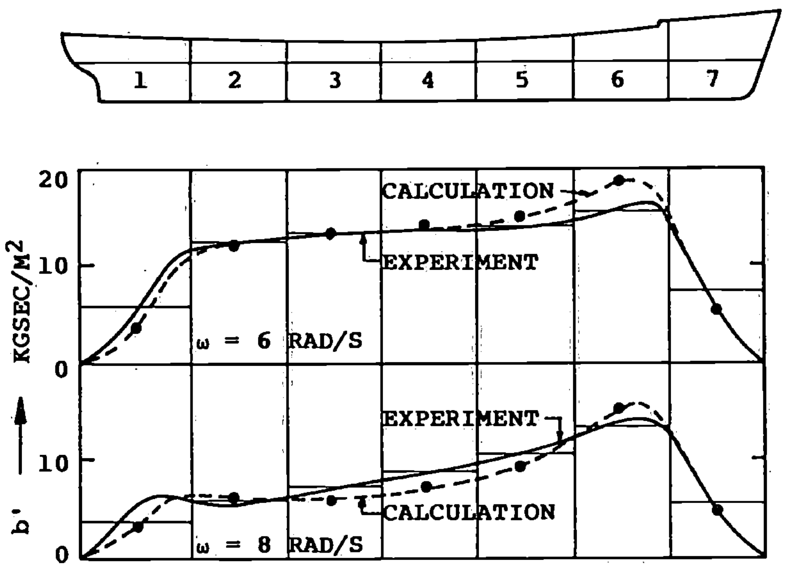


FIGURE 4. LONGITUDINAL DISTRIBUTION OF DAMPING COEFFICIENT, Fn = 0.20, L = 2.3 M.

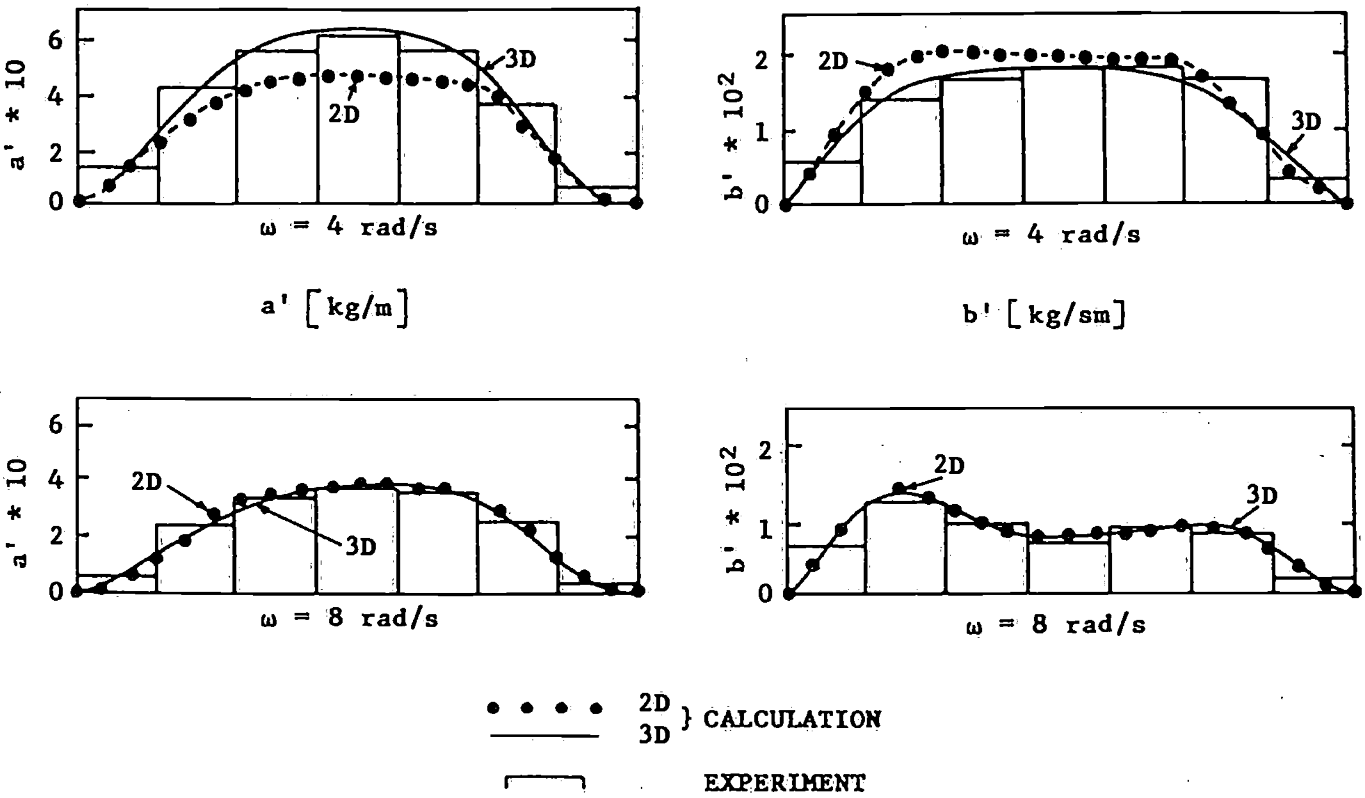


FIGURE 5 COMPARISON OF CALCULATED ADDED MASS AND DAMPING DISTRIBUTION AT ZERO FORWARD SPEED, L = 2.3 M.

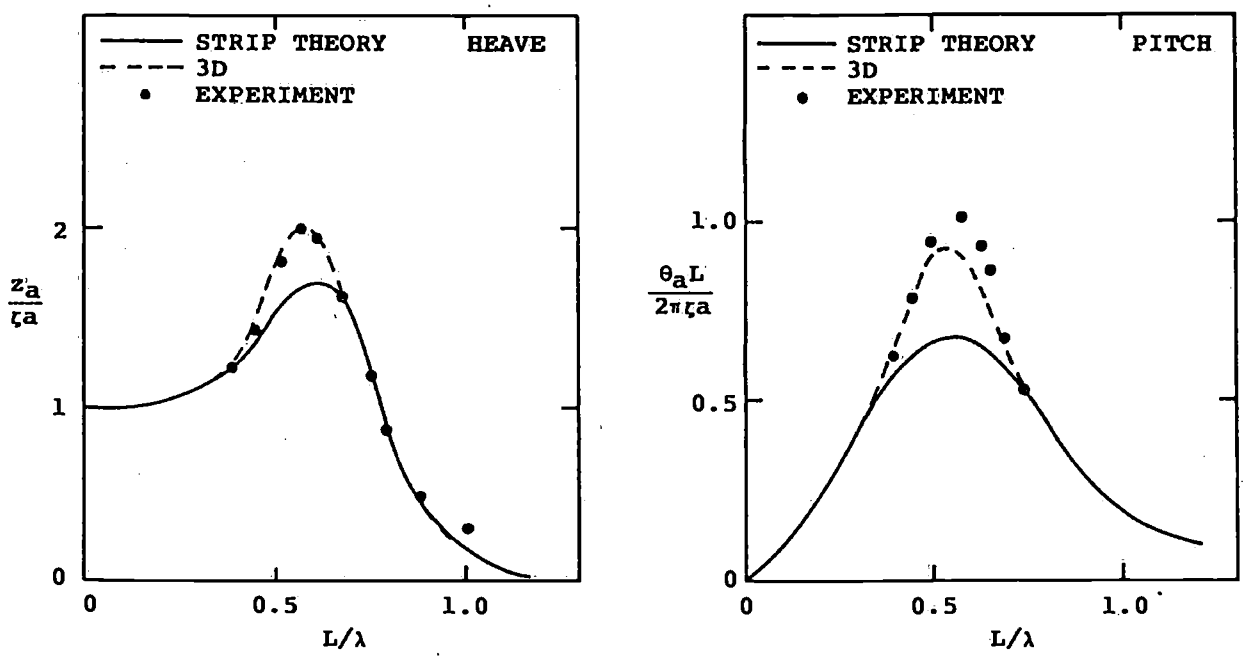


FIGURE 6. HEAVE AND PITCH AMPLITUDE OF A SAILING YACHT.

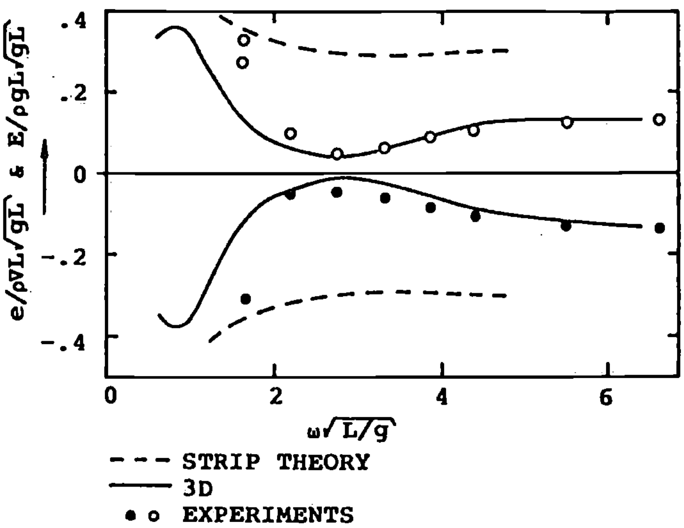
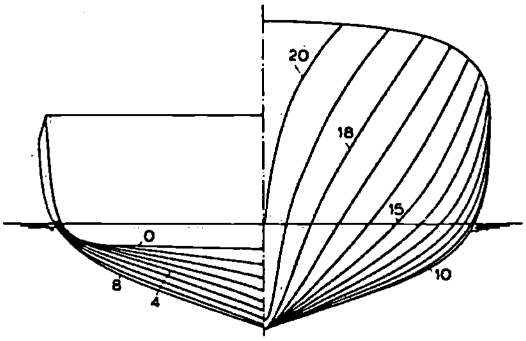


FIGURE 7. DAMPING CROSS COUPLING COEFFICIENTS
FOR HEAVE AND PITCH, Fn = 0.2.



$L/B = 8$, $B/T = 4$, $C_B = 0.4$

FIGURE 8A. BODYPLAN.

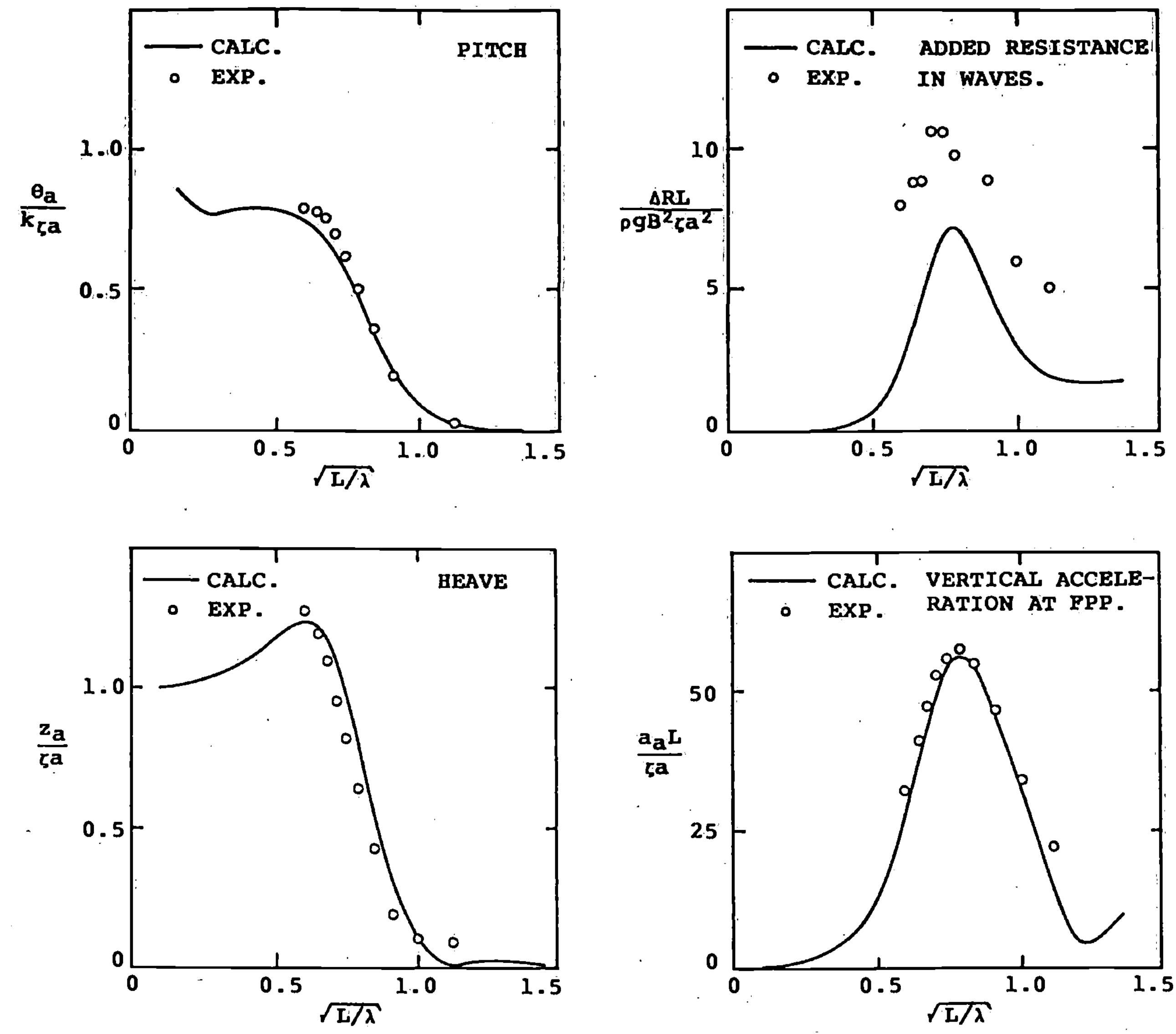


FIGURE 8B. FREQUENCY RESPONSE FUNCTIONS, Fn = 1.14.

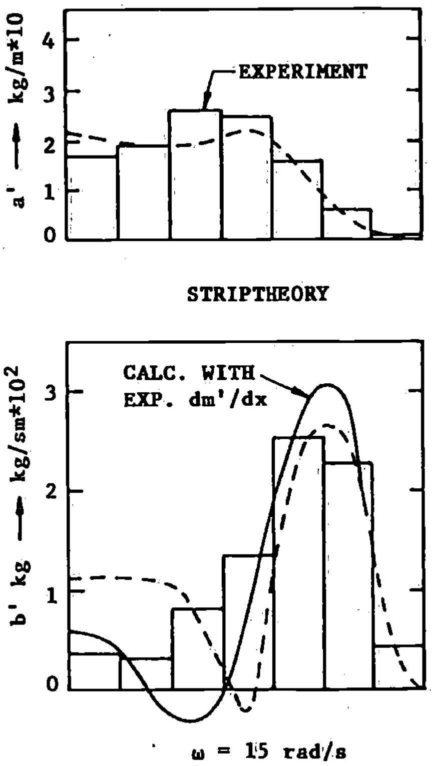


FIGURE 9. ADDED MASS AND DAMPING DISTRIBUTION AT HIGH FORWARD SPEED Fn = 1.14, L = 2 M.

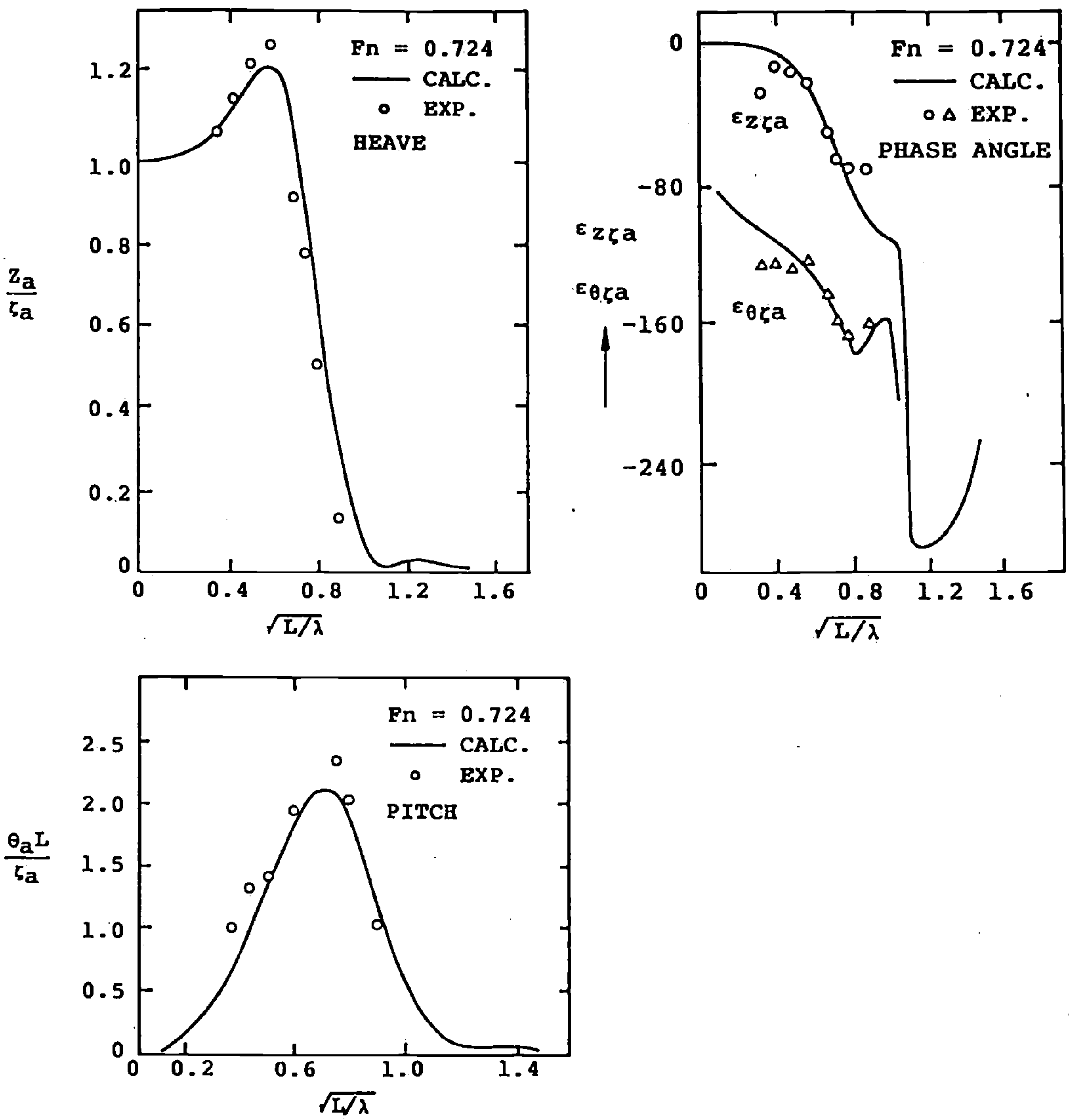


FIGURE 10. FREQUENCY RESPONSE FUNCTIONS OF A HARD-CHINE SEMI PLANNING HULL FORM.

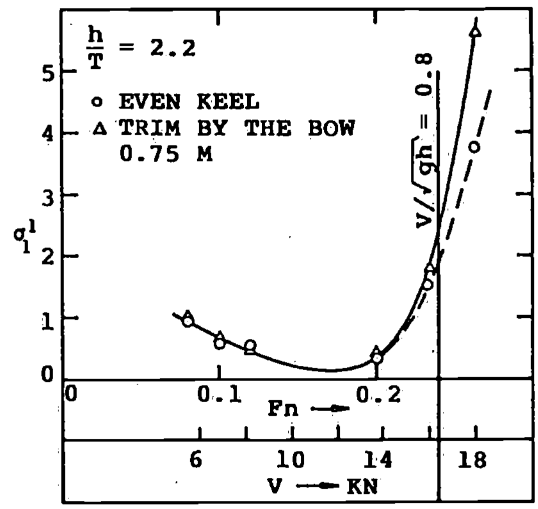


FIGURE 11. STABILITY ROOTS OF A RO RO PASSENGER FERRY IN SHALLOW WATER. (h = 12.5M) L=126.1M, B=22.7M, T=5.69M