The behaviour of a large air-supported MOB at sea

J.A. Pinkster & E.J.A. Meevers-Scholte

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Special Issue on

VERY LARGE FLOATING STRUCTURES (VLFS)

PART II

GUEST EDITORS

R. C. Ertekin

J. W. Kim

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Editorial

I am pleased to announce that, from 1st January 2001, Professor Torgeir Moan will replace me as European Editor for the Journal. I very much welcome Torgeir's promotion. He is widely known and well respected in both academia and industry, and brings a deep understanding and practical awareness to many aspects of ship and offshore structure response and design. He plays a leading role in the use of reliability-based approaches that are close to my heart and which provide an extremely useful framework for code development in particular and, more recently, design development.

I have some regrets in handing over the reigns after some 12 years in the post, I took over from Professor Douglas Faulkner at Volume 2. However, more recently, I have found it challenging to fit in the post of Editor with some of the jobs I am involved in and cannot, because of this, always give the Journal the attention it deserves.

One of the frustrations I have felt over the years has been the seeming absence of manuscripts from ISSC members. Despite even specific encouragement at times, this situation does not seem to have changed.

In closing, I wish Torgeir and the Journal future success and I will continue to do what I can to support this.

Paul A. Frieze
18 Strawberry Vale, Strawberry Hill, Twickenham, Middlesex TW1 4RU, UK

Since Paul succeeded the founding editor, Professor Douglas Faulkner 12 years ago, he has made significant efforts as editor of the journal. On behalf of the other members of the editorial board and the readers of the journal, I would like to thank Paul for his contributions to making the Journal of *Marine Structures* such an important forum for disseminating information from research on marine structures to the research community and, not least, designers and other users.

In cooperation with my colleagues, Alaa Mansour and Tetsuta Yao and our Editorial Board, I will do my best to further develop the journal to be a valuable

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in their fields. It is hoped that these papers will improve our understanding of the behavior of very large floating structures.

R. Cengiz Ertekin[a]
Jang Whan Kim[a]
Koichiro Yoshida[b]
Alaa E. Mansour[c]
[a]*Department of Ocean Engineering, SOEST, University of Hawaii at Manoa, 2540 Dole Street, Holmes Hall 402, Honolulu, HI-96822-2303, USA*
[b]*Tokai University, Japan*
[c]*University of California at Berkeley, USA*

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The behaviour of a large air-supported MOB at sea

J.A. Pinkster*, E.J.A. Meevers Scholte

Delft University of Technology, Mekelweg 2, 2628 CD, Delft, Netherlands

Abstract

The behaviour of an air-supported concept of a large floating structure is investigated by means of model tests and computations based on three-dimensional potential theory. For this study, the ONR–MOB structure is used as a target vessel since, based on results of other studies, comparable data are available for other concepts for large floating structures. In this paper some results of model tests carried with a 1:200 scale model of an air supported MOB concept are compared with the results of computations. The comparison shows that by and large the motions, relative motions of water in the cushion and the midship bending moment are reasonably well predicted. It is confirmed that the midship bending moment is reduced by the air cushion. For short wavelengths, more detailed numerical modelling of the structure is necessary. Results of model tests are given of the resistance in still water are given. © 2001 Elsevier Science Ltd. All rights reserved.

Keywords: MOB; Floating structures; Air cushion

1. Introduction

For some years studies are being carried out with respect to the mobile offshore base (MOB), a large floating platform that is intended to provide a forward-deployable logistics facility for military hardware. In Remmers et al. [1] an overview is given of the Mission Requirements and Performance Measures of such a platform and the main concepts explored to date are reviewed. Most are based on large semi-submersibles which may or may not be connected depending on the particular concept. Based mainly on structural considerations, in all cases, the semi-submersibles are intended to

* Corresponding author. Tel.: 15-278-3598; fax: 15-278-1836.
E-mail address: j.a.pinkster@wbmt.tudelft.nl (J.A. Pinkster).

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be separated when sea conditions increase beyond some limit. It was indicated that single, monohull or monolithic structures were not considered to be viable in part due to the large structural loads which would occur in survival sea conditions.

This paper is concerned with the behaviour in waves of a monolithic MOB-type structure with main dimensions corresponding to a length of 1500 m and a breadth of 200 m which is partially supported by a single air cushion bounded on all sides by a vertical skirt. The skirt extends sufficiently far into the water to ensure that no air loss takes place and consequently, in theory, no constant supply of air from fans is necessary. This concept for a MOB has been selected for investigation specifically to study not only the motion behaviour in waves but also to study the effect of a large air cushion on the structural loads. The underlying idea being that, at first approximation the air pressure in the cushion, although time dependent in waves, is spatially equal. This property can be used to obtain a considerable reduction in, for instance, the midship vertical bending moment in waves compared to a conventional monohull form.

An additional consideration for the application of an air cushion is related to the fact that the presence of the air cushion on the underside of the structure reduces the frictional surface. Even though the rigid skirts may have a relatively large resistance, the absence of frictional resistance over the very large bottom could lead to favourable overall resistance characteristics. This is of importance for the mobility of such a structure.

For many years, much attention has been paid to the development of relatively small, fast waterborne sea transport based on air cushion technology as applied to ACV and SES craft (see, for instance, [2–4]). A large number of these vessels are in service world-wide and much experience in air cushion technology has been gained as a result.

The use of air cushions to support very large floating structures, although only used in few applications, has been known for a long time in the offshore industry [5–7]. In these cases, the application of air cushions has mainly been related to a temporary increase in the buoyancy of large bottom-founded structures for the purposes of transit from a shallow building dock to deeper water.

In the 1970s the Seatek Slo-Rol system was introduced to reduce the wave-induced motions of jack-up platforms in the floating mode. As a result of the application of this system, the transverse and longitudinal stability of the platform are reduced thus bringing the natural roll and pitch periods outside the range of the wave frequencies. As a result the angular motions in waves are reduced as are the dynamic loads in the jack-up legs in the wet tow mode.

Iwata et al. in [8,9] studied the motions in waves of a catenary moored structure partially supported by an air cushion. Some model experiments were carried out in a wave tank in order to validate their numerical method for predicting the behaviour in waves.

In recent years, as part of the ongoing MOB studies, a pneumatically stabilized platform has been investigated for application as a permanent maritime platform in an open sea environment [10].

At the Delft University of Technology the behaviour in waves of large structures wholly or partially supported by air cushions has been studied using three-dimensional diffraction computations and model tests [11,12]. In the present paper, a short review is given of the main elements of the theory underlying the computational method. Following this a brief description is given of an air supported MOB concept developed based on such computations. Subsequently, a model test program is described and results of model tests in still water, and in regular and irregular head waves will be given. Where appropriate, results are compared with the results of computations.

2. Theory

2.1. Behaviour of air and water

The theory is given for an air cushion-supported construction consisting of one rigid body and one or more air cushions which may or may not be interconnected. The air cushions are passive and there is no air leakage or induction. The air cushions are bounded by the rigid part of the construction which extends sufficiently far below the mean waterlevel within an air cushion in order to ensure that no air leakage will occur.

The wave frequency air pressure variations within a cushion are determined by the change in cushion volume through the linearized polytropic gas law

$$\Delta p = -\Delta vol * n_g * (p_0 + p_c)/vol, \tag{1}$$

where vol is the mean air volume in cushion, p_0 the atmospheric pressure, p_c the mean excess of pressure in cushion, Δvol the wave frequency volume change in cushion, Δp the pressure variation relative to mean cushion pressure and n_g the gas law index.

For wave frequency pressure variations adiabatic conditions are assumed. In that case $n_g = 1.4$.

The air cushions and the rigid part of the structure are partly bounded by water. The interaction between the air cushions, the structure and the surrounding water are determined based on linear three-dimensional potential theory.

The fluid motions in regular waves with frequency ω in a point X with earth-bound co-ordinates X_1, X_2, X_3 are described by the total potential Φ as follows:

$$\Phi(X_1, X_2, X_3, t) = \phi(X_1, X_2, X_3)e^{-i\omega t}. \tag{2}$$

The potential ϕ satisfies Laplace's equation, the linearised boundary conditions on the free surface outside the body, the boundary condition at the sea-floor and, excepting the undisturbed incoming wave potential, the radiation condition. On the rigid part of the body surface a no-leak condition has to be satisfied while at the free-surfaces of the air cushions the potential must satisfy the no-leak condition at the unknown, moving free-surface and also the requirement of a spatially equal but time-dependent pressure in each cushion. These requirements are not automatically met so besides the incoming wave potential, additional potentials are introduced which represent pulsating source distributions over the mean wetted surface of the rigid part of the structure and over the mean free-surface of the cushions.

The complex potential ϕ follows from the superposition of the undisturbed wave potential ϕ_0, the wave diffraction potential ϕ_d, the potentials associated with the 6 d.o.f. motions of the rigid part of the construction ϕ_j and the potentials associated with the vertical motions of the free-surface within each cushion, ϕ_c.

$$\phi = -\mathrm{i}\omega\left\{(\phi_0 + \phi_d)\zeta_0 + \sum_{j=1}^{6} \phi_j x_j + \sum_{c=1}^{C} \frac{1}{S_c}\iint_{S_c} \phi_c \zeta_c \, \mathrm{d}S_c\right\} \tag{3}$$

where ϕ_0 is the potential of the undisturbed incoming wave, ϕ_d the diffraction potential, ϕ_c the potential associated with vertical motions of the free-surface in the c-cushion, x_j the rigid body motion in the j-mode, ζ_0 the amplitude of the undisturbed incoming wave, ζ_c the vertical motion of free-surface in c-cushion, S_c the free-surface area of c-cushion and C the total number of independent, non-connected cushions.

In the above equation the undisturbed wave potential ϕ_0 and the diffraction potential ϕ_d together describe the flow around the captive structure under the assumption that the free surfaces within each of the air cushions is also rigid and non-moving. The potentials ϕ_j are associated with the flow around the structure oscillating in still water under the assumption that the free surface within each air cushion is rigid and fixed.

The potentials ϕ_c are associated with the flow around the captive structure as induced by the vertical motions ζ_c of the free surface within each cushion.

2.2. Numerical approach

When considering a conventional rigid body, it is customary to determine the wave forces on the captive structure based on the undisturbed wave potential ϕ_0, the solution of the diffraction potential ϕ_d and the added mass and damping of the structure oscillating in any one of the six modes of motion in still water based on the solution of the motion potentials ϕ_j. The motions of the structure are then determined by solving a 6 d.o.f. equation of motion taking into account the wave forces, added mass and damping and restoring terms.

With a construction partially supported by one or more air cushions different approaches may be followed in order to determine the motions of the structure, the pressures in the cushions and other relevant quantities such as the water motions within an air cushion.

In a direct method the motions of the structure, the free-surface behaviour within the air cushions and the cushion pressures are obtained as the solution of a multi-body or multi-degree-of-freedom problem with added mass, damping and spring coupling effects. No data is obtained on the wave forces or added mass and damping of the structure including the effects of the air cushions. A second method can also be followed in which the wave forces and added mass and damping including effects from the air cushions are determined as the solutions of separate multi-body or multi-d.o.f. problem [11]. In that case the motions of the structure in waves are determined as the solution of a normal 6 d.o.f. equation of motion.

For both methods the rigid part of the structure is modelled in the usual way by means of panels representing pulsating sources distributed over the mean underwater part of the construction.

The free surface within each air cushion is also modelled by panels representing source distributions lying in the mean free surface of each cushion. This level of the mean free surface may be substantially different to the mean waterlevel outside the structure and also different for each cushion.

Each panel of the free surface within an air cushion is assumed to represent a body without material mass but having added mass, damping, hydrostatic restoring and aero-static restoring characteristics. Each free surface panel (body) has one degree of freedom being the vertical motions of panel n within cushion c.

It will be clear that properties such as added mass coupling and damping coupling exists between all free surface panels and between free surface panels and the rigid part of the structure. The effect of the air cushion is shown in the aero-static coupling between the motions of the free surface panels themselves and between the free surface panels and the rigid part of the structure.

The hydrodynamic problem is solved using a zero-order panel method. Results are obtained on the structural motions, air cushion pressures, structural loads, hydrodynamic pressures and mean second-order wave drift forces which are computed based on the far-field theory [13].

3. The air cushion MOB concept

For the present study a preliminary concept of an air-supported MOB was chosen. The main particulars of the concepts were derived from the ongoing ONR study i.e. the length amounts to 1500 m full scale, the displacement is in excess of 1 million tons and the structure has to have good motion characteristics in extreme sea conditions with significant wave height in the region of 15 m and mean period around 20 s.

Taking this data into account preliminary computations were carried out to determine the wave induced motions and midship girder loads for a number of different concepts. These concepts were all basically of simple shape consisting of vertical side walls or skirts, with or without a pontoon running the full length of the lower end of the skirts. One of the aims was to produce a form which would have a natural pitch period in excess of 30 s full scale so as to avoid any resonant pitch motions in high sea states. Furthermore, the structure should have sufficient lateral and longitudinal static stability.

As far as the static stability is concerned, for a structure with vertical side walls and supported partially by a single air cushion the computation of the stability is straightforward. The longitudinal and transverse metacentric height are determined from the following classic stability equation:

$$GM = KB + BM - KG, \tag{4}$$

in which KB is the vertical position of the center of buoyancy of the total displaced volume relative to the base line point K, BM follows from the moment of inertia of the waterline of the (wall-sided) skirts and the total displaced volume, and KG is the vertical position of the center of gravity of the structure.

The above relationship can be derived by considering the air-supported MOB to be a conventional ship of equal draft as the MOB but having a liquid cargo, i.e. the volume of water between the skirts. Taking into account the stability loss due to the free surface of the 'cargo' results in the above equation for the GM value. The righting moment per unit heel or trim follows from the product of the relevant GM value and the displaced weight of the structure. In this particular case, since heeling or trimming does not change the air volume, the air cushion properties in terms of compressibility do not play a roll so there are no scale effects arising from different compressibility of the air cushion in the model and at full scale. If more than one air cushion were employed compressibility effects in the static stability could be present.

The finally chosen configuration has a length of 1500 m and a breadth of 200 m and consists of a parallel mid-body and straight-sided fore- and aftbody with a half waterline angle of 30°. The total skirt depth measured from the lower deck amounts to 42.4 m. The superstructure consists of a box with a height of 24.6 m above the lower deck giving a total depth of 67 m.

The 30° waterline angle was selected as being a reasonable value from the point of view of resistance. The wall thickness of the skirts amount to 3.0 m. This was selected in order to achieve sufficient static stability in heel and trim. At the base of the skirt a pontoon with a crossection of 9.0 m × 12.0 m is fitted. In conjunction with the static stability of the structure for heel and trim, the pontoon also plays a roll in the natural roll and pitch period besides being of importance from the point of view of affording space for propulsion systems and storage.

The draft of the structure is varied by changing the air volume. For the present study, results for three drafts are presented i.e., transit drafts of 7.0 and 10.5 m for still water resistance tests and an operational draft of 20 m full scale for tests in waves. The total displacement at a draft of 20 m amounted to 1.408 million m^3.

At the draft of 20.0 m the air pressure in the cushion depresses the free surface in the cushion 3.81 m relative to the free surface outside the structure. At this draft 69% of the displacement is borne by the air cushion. At a draft of 7.0 m, the free surface depression in the cushion amounts to 5.0 m and 82% of the displacement is borne by the air cushion. At the draft of 20.0 m the transverse GM value amounted to 24.92 m and the longitudinal GM value to 1302.4 m. Based on the results of this preliminary analysis a model of the MOB structure was constructed.

4. Model tests

4.1. The model

Model tests were carried out with a 1:200 scale model of a MOB which was built of plywood sheeting. The main dimensions of the structure at full scale and for the model scale are given in Table 1 and a body plan is given in Fig. 1.

Table 1
Main particulars of MOB in full scale and in model

Quantity	Units	Full scale	Model
Length	m	1500	7.50
Breadth	m	200	1.00
Draft	m	20	0.10
Depth	m	67	0.335
Displacement	m^3	1,408,000	0.176
KG	m	43.15	0.216
GM(transv.)	m	24.92	0.1246
GM(long.)	m	1302.4	6.512
k_{xx}	m	61.41	0.30705
k_{yy}	m	428.48	2.1424
k_{zz}	m	428.48	2.1424
Roll period	s	33.8	2.39
Pitch period	s	29.0	2.04

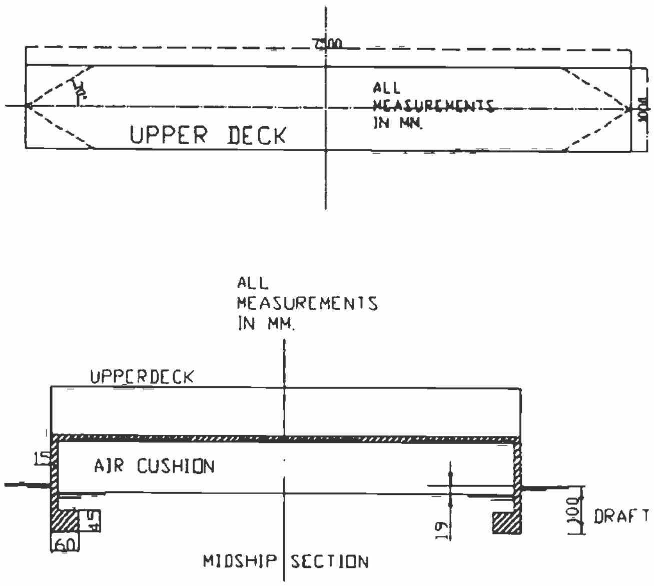

Fig. 1. General arrangement of MOB model.

With a view to the measurement of the midship bending moment and vertical shear force in waves, the model was built in two sections. These were joined at the midship by means of force transducers. The air tightness of the structure was maintained by the introduction of a latex membrane attached to the inside of the transverse cut in the structure. The stiffness of the midship connection incorporating the force transducers was such that the lowest natural frequency in the longitudinal bending mode amounted to 5 Hz or 31.4 r/s. This was sufficiently high to ensure absence of significant dynamic magnification effects in the measured forces since the highest wave frequency amounted to approximately 1.3 Hz or 8 r/s. The model was constructed without provisions for means to reduce scale effects related to the air cushion stiffness. This was motivated by the consideration that an important motion mode is the pitch motion. This will determine to a large extent whether in waves the relative motions will be so large as to induce air losses in the cushion. The pitch motions are relatively unaffected by the air cushion stiffness as was noted with respect to the longitudinal and transverse static stability. At a later stage, if more refined data are required, however, such means will be employed. At the present stage, model test results obtained without this feature are correlated with computations based on the same assumption.

4.2. Test facility

The model tests were carried out in towing tank no.1 of the Department of Ship Hydromechanics of the Delft University of Technology. This facility measures 140 m × 4.25 m × 2.5 m. For the tests the waterdepth amounted to 2.31 m which corresponds to 462 m full scale. The basin is fitted with a towing carriage and a flap-type wave maker with regular and irregular wave capabilities. The speed control of the carriage has been specially designed for accurate low-speed motion.

4.3. Resistance tests

Model tests were carried out in still water to determine the resistance characteristics of the MOB model for a range of speeds at transit draft. Considering the unusual shape of the model, especially in the bow and stern areas where the oncoming water has to flow around and under the wall-sided skirt, considerable flow separation occurs in these areas. As such it was not deemed necessary to apply turbulence stimulation devices such as those which are normally applied for towing tests with conventional ship models.

At this stage no attempt has been made to streamline the bow and stern sections in order to minimize flow separation. The resultant resistance is assumed to be an indication of the upper limit of the resistance which can be improved upon by critical evaluation of fore and aft form details. Resistance tests were carried out for a draft of 7.0 m and at 10.5 m for a speed range of 2.9–20.3 knots full scale. The maximum speed corresponds to $F_n = 0.0816$. The results of the tests are given in terms of the model and full scale resistance in Fig. 2. The model data were extrapolated to full scale using the frictional coefficient of the ITTC 1957 line and a correction for blockage effects. The blockage effect correction results is a small increase in the full scale speed.

4.4. Tests in waves

4.4.1. Measurements in waves

All tests in waves were carried out in head waves at zero forward speed. The model was moored by a soft-spring arrangement at the bow and the stern. At the points of attachment longitudinal force transducers were attached to the deck of the model in order to measure the mean longitudinal drift force. Other measurements included the

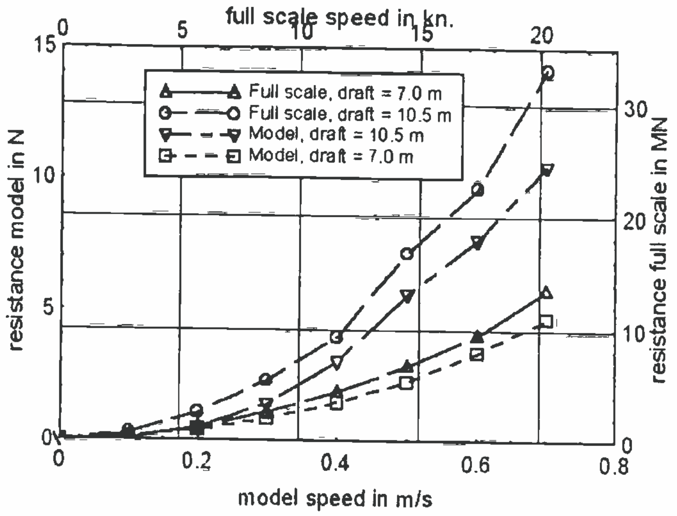

Fig. 2. Resistance of model and full scale MOB.

pitch, heave and surge motions, the cushion pressure, the midship shear force and bending moment and the relative wave elevation within the cushion at three locations i.e, at the bow, the midship and the stern.

4.4.2. Tests in regular waves

Tests were carried out in regular waves for the full scale draft of 20.0 m for a range of wave frequencies and wave amplitudes. At model scale the wave frequencies ranged from approx. 1.3–8 r/s which corresponds to 0.09–0.56 r/s. In terms of full-scale wave periods these ranged from 11 to 70 s and in terms of the ratio of wave length to ship length from $\lambda/L = 3.0$ to 0.13. The chosen wave periods seem extraordinarily high but it should be remembered that for such a huge structure the most energetic response in terms or motions and structural loads will be in the very long wave range. Long-wave periods were also chosen in order to straddle the natural pitch period of the structure.

Nominal wave heights of the regular waves corresponded to 4, 6, 12 and 18 m full scale, with the higher wave amplitudes being applied to the longer wave periods. The range of wave heights was chosen in order to investigate non-linear effects in the motion responses. Results of regular wave tests are given in figures in terms of RAOs of the various quantities. Results are compared with results of computations and, where appropriate, with results of RAOs obtained from tests in irregular waves. In all cases model and full-scale values are given by appropriate axes.

4.4.3. Tests in irregular waves

Tests were carried out for a range of irregular wave conditions which were based on sea conditions relevant for the ONR–MOB study. These sea conditions start off at the high end of the operational conditions for cargo transfer and go right through to the extreme survival conditions. The following irregular wave conditions were tested Table 2.

Table 2
Sea conditions for tests in irregular waves (full-scale values)

Significant wave Height (m)	Zero-crossing period (s)	Peak period (s)
4.1	12.9	14.6
6.1	10.8	11.0
6.4	10.8	11.3
7.9	11.2	11.5
8.7	12.5	14.1
12.2	14.3	19.8
15.6	16.1	20.5

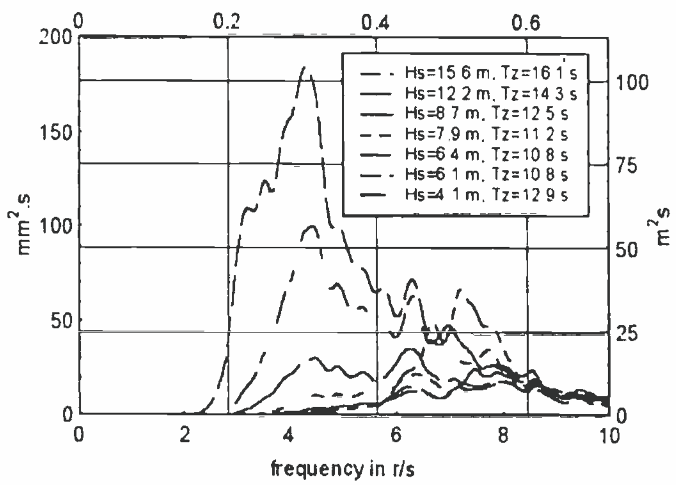

Fig. 3. Spectra of irregular waves.

The wave spectra are shown in Fig. 3. In this figure the model frequency and spectral scales are given on the lower-and left-hand axes, respectively, and the full scale values on the upper and right-hand axes.

The tests in irregular waves were carried out for a full scale duration of 3.50 hrs. corresponding to 15 min model time. The results of the measurements were analysed to yield statistical data in terms of mean, RMS, maximum and minimum values. Using spectral analysis the RAOs of measured quantities were also obtained.

5. Computations

Computations were carried out based on the theory briefly outlined in Section 2. One quadrant of the panel model used for the computations is shown in Fig. 4. The total number of panels for the rigid part of the strucure amounted to 768 while the total number of panels for the free surface in the air cushion amounted to 724. The number of panels forms a limitation to the wave frequencies for which accurate

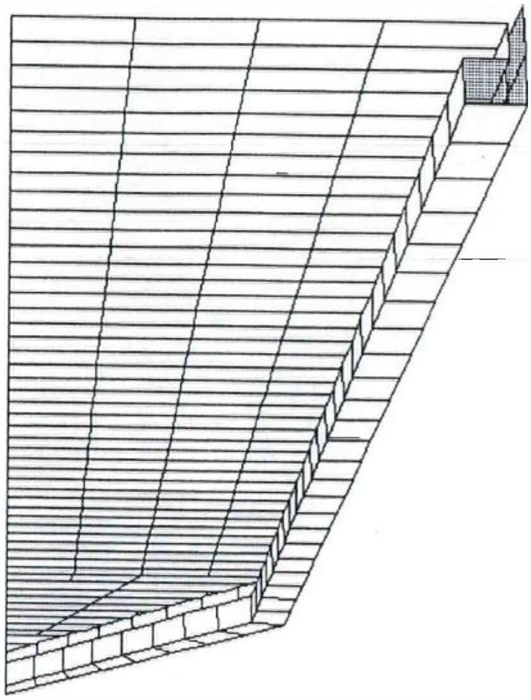

Fig. 4. One quadrant of panel model of MOB.

results can be obtained from these computations. In this particular case, results are accurate for frequencies up to about 8 r/s on model scale. At this time it was not deemed necessary to increase the number of panels drastically in order to compute the behaviour in relatively short waves since these would be very small anyway. Also, at this model scale the results for shorter waves could not be validated by the model tests due to the frequency limitations of the wave generator. For this to be possible, a significantly larger physical model will need to be tested. At such times it will also be necessary to increase the frequency range of the computations by increasing the number of panels.

6. Discussion of results

6.1. Resistance tests

The resistance has been measured for speeds up to 20.3 knots. The results are shown in terms of model and full-scale resistance curves in Fig. 2. As mentioned previously, the extrapolation from model to full scale was carried out using the conventional method whereby the model viscous resistance is deducted from the measured resistance, the remainder is scaled up using Froude's law and the full scale viscous resistance is added to obtain the total. It should be borne in mind that the hull form being considered here is unconventional and the results should be viewed with caution and considered as being indicative rather than definitive. It is however clear that the draft of the structure has great influence on the resistance the value being more than doubled by increasing the draft from 7.0 to 10.5 m. At the smaller draft the resistance at the highest speed of 20.3 knots amounts to about 13.4 MN. This results in a value of the effective power (product of resistance and speed) of approx. 140 MW or 190,000 EHP. The actual installed power depends on the type and characteristics of the propulsion units chosen and on the layout. A possible layout could be a row of podded propeller units attached below the pontoons over the length of the parallel midbody. With a length of some 1150 m available on each side there is ample room to set up thrusters in such a way as to minimize thruster–thruster interference effects.

6.2. *Tests in regular waves*

The results of the tests in regular waves, the results of computations and the results of spectral analysis of the tests in irregular waves are compared in the form of RAOs of the motions, relative motions and midship vertical bending moment in Figs. 5–10.

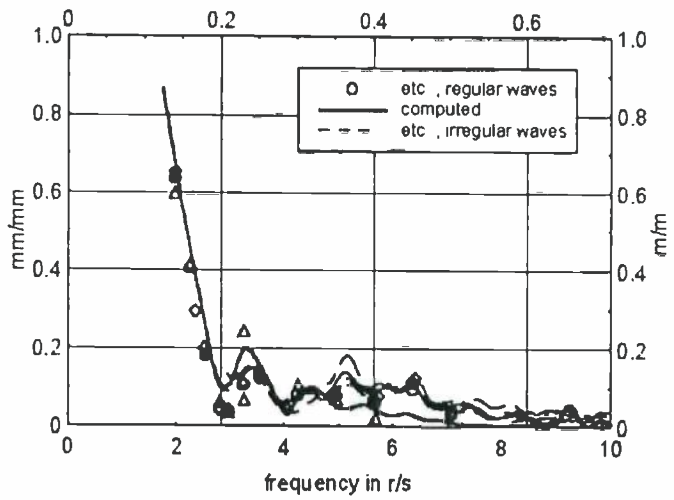

Fig. 5. RAO of surge in head seas.

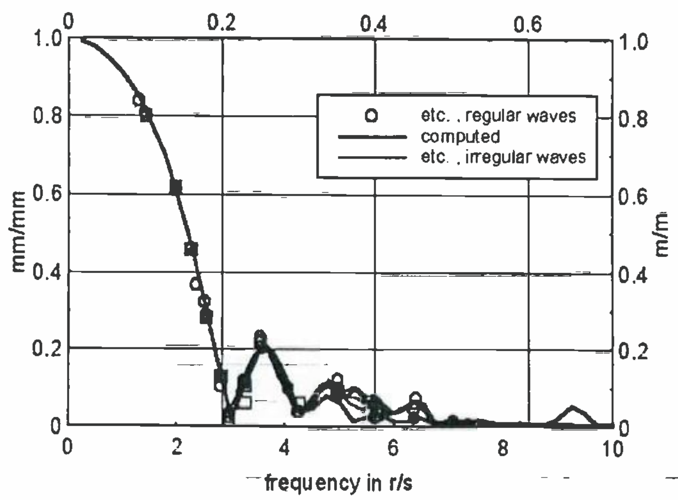

Fig. 6. RAO of heave in head seas.

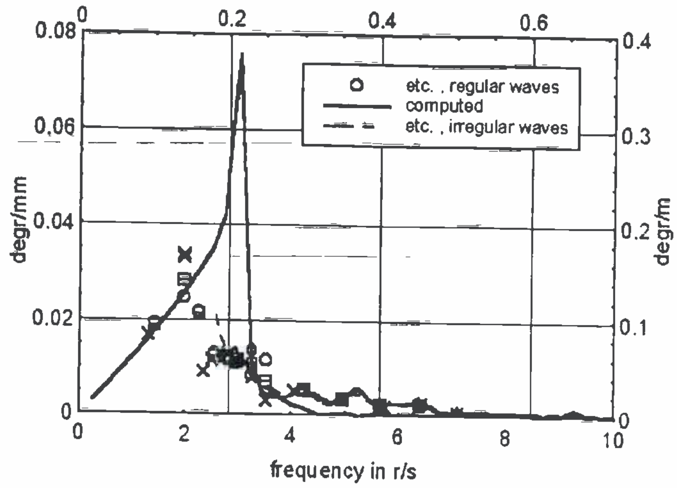

Fig. 7. RAO of pitch in head seas.

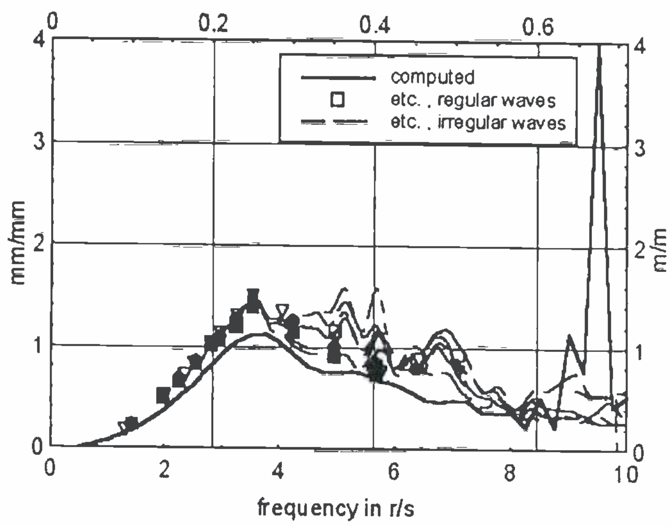

Fig. 8. RAO of relative wave elevation at midship.

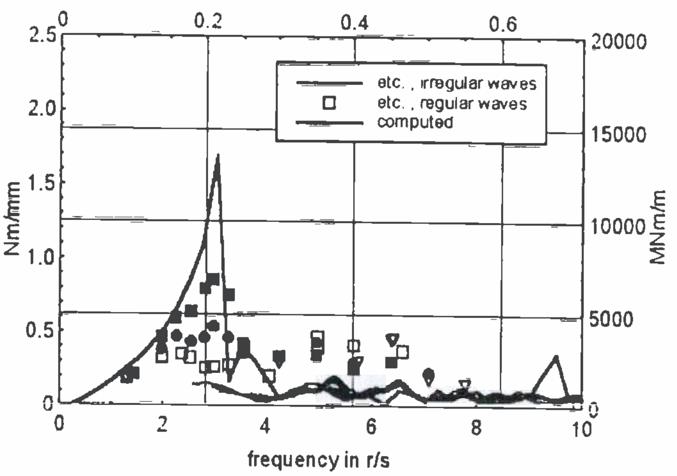

Fig. 9. RAO of midship vertical bending moment in head seas.

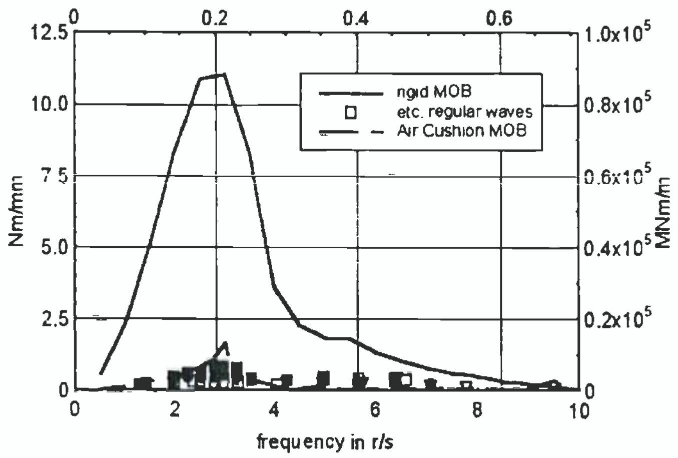

Fig. 10. Comparison of RAO of bending moment with a rigid MOB.

The surge motions given in Fig. 5 show a satisfactory correlation between all results. At higher frequencies the computations underestimate the surge motions. The values at these frequencies are however low.

The heave motions are shown in Fig. 6. The heave response is very similar to that of a conventional hull form. Due to the air cushion which supports a large part of the structural weight and is relatively stiff, heave exciting forces are fully transmitted to the structure even though equalization effects are present. In conventional hulls this effect also occurs in the heave exciting force but in such cases pitch excitation arising from pressure differences over the bottom is also present. The zeros in the heave response are related to the ratio between length of the structure and the wave length and occur at shorter frequency intervals as the frequency increases. In general the results agree well at higher heave response values. Again some differences are seen at the higher wave frequencies at which the computed values are somewhat lower than the measured data.

The pitch response functions are shown in Fig. 7. The characteristics of the pitch response are very similar to those of semi-submersibles. This is due to the fact that only the skirts of the structure contribute to the pitch exciting moments and the cushion has no significant effect. Also, the stability and the pitch moment of inertia are only determined by the skirts and the structural mass distribution. By choosing an appropriate skirt configuration, the natural pitch period can be adjusted to high values (low frequencies) so that the pitch resonance frequency is outside the range of wave frequencies. As can be seen from the wave spectra shown in Fig. 3 compared with the pitch response function, very little wave energy is present at the natural pitch period even in the most severe sea condition. Differences in pitch motions between computations and measurements are larger than is the case with the surge and heave motions. The high response value at pitch resonance predicted by computations is not realized in the model tests. The peak in the measured pitch response is shifted to lower frequencies and remains well below the computed peak value. The lower measured pitch response at resonance is to be expected since at these very low frequencies viscous damping effects, which are not accounted for in the computations, are of importance in reducing the resonance effects.

The RAO of the relative wave elevation measured at the midship location is shown in Fig. 8. Over the major part of the frequency range the measured and computed data correlate quite well even though the computed data stay below the measured data. At higher wave frequencies the computed data shows a sharp peak. The existence of such a peak does not seem to be confirmed by the model test data. Computations with more panels on the free surface within the air cushion confirmed the peak at higher frequencies which seems to be related to standing wave effects in the cushion free surface. More detailed analysis will be required here.

The midship vertical bending moment is shown in Fig. 9. In this case there is a good agreement between the computed RAO and the values obtained from spectral analyses of the irregular wave tests. The RAO obtained from the regular wave tests are lower than the computed values at low frequencies near the natural pitch frequencies while they are higher at higher wave frequencies. At the lower wave frequencies the RAO from regular wave tests also show a significant dependence on the regular wave amplitudes as is evident from the greater scatter at these frequencies. Regarding the magnitude of the midship bending moment response function, an indication can be found by comparing the above data with computed results for the same structure under the assumption that the air cushion free surface is also rigid. In Fig. 10 the same data as presented in Fig. 9 is given alongside the computed response function for a completely rigid structure of the same dimensions and geometry. The comparison shows that the initial supposition that a large air cushion can serve to reduce the midship bending moment is fully supported for this case.

6.3. Tests in irregular waves

The RMS and maximum single amplitude of the measured heave and pitch motions and the midship bending moment are given in Tables 3 and 4, respectively. As could be expected, the heave and pitch motions are small. Interpretation of the bending moment values can only be carried out when viewed in relation to the strength characteristics of the structure. No structural design work has been carried out to date so this comparison cannot be made at this time.

During the tests in irregular waves it was ascertained that for the 20 m draft no air loss took place even under the highest sea condition. This could easily be verified by measurement of the draft of the structure before and after each test. A special test was carried out in order to establish the minimum draft at which no air loss would occur for the highest sea condition.

To this end the draft of the structure was reduced as far as possible by introducing air in the cushion until air loss took place in still water. The draft in this condition amounted to 5 m. Subsequently, the wave generator was started and the highest irregular wave condition with $Hs = 15.60$ m was generated. At the first few highest waves, air loss took place at the bow and aft of the forward shoulder by being vented outwards from under the skirt. The structure settled down to an increased draft without

Table 3
RMS values in irregular waves (full scale values)

Sign. wave height (m)	RMS of heave (m)	RMS of pitch (deg)	RMS of moment (MN m)
4.1	0.05	0.013	912
6.1	0.07	0.019	1,114
6.4	0.09	0.024	1,859
7.9	0.10	0.027	1,539
8.7	0.17	0.046	1,814
12.2	0.33	0.083	2,528
15.6	0.57	0.145	3,282

Table 4
Maximum single amplitude in irregular waves (full-scale values)

Sign. wave height (m)	Max. heave (m)	Max. pitch (deg)	Max. moment (MN m)
4.1	0.19	0.057	3,800
6.1	0.27	0.080	4,920
6.4	0.37	0.117	8,728
7.9	0.36	0.160	5,899
8.7	0.68	0.197	7,290
12.2	1.20	0.435	11,570
15.6	1.55	0.520	15,531

any increased oscillatory motions due to the air venting. The final draft at the end of the test, which had a duration corresponding to 3.5 h full scale, amounted to 14 m.

7. Conclusions

The results of model tests and computations show that to a large extent the motions and bending moments in waves are reasonably well predicted by the computations. Analysis of the behaviour in the shorter wave range will require a higher number of panels. The still water resistance of the air cushion MOB is strongly dependent on the draft so in order to attain a high transit speed the lowest possible draft should be maintained. Results of tests in irregular waves indicate that, for the conditions tested, an air supported MOB shows excellent motion characteristics. Midship bending moments are confirmed to be significantly reduced by the air cushion.

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