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The effect of aircushion division on the motions of large floating offshore structures

by

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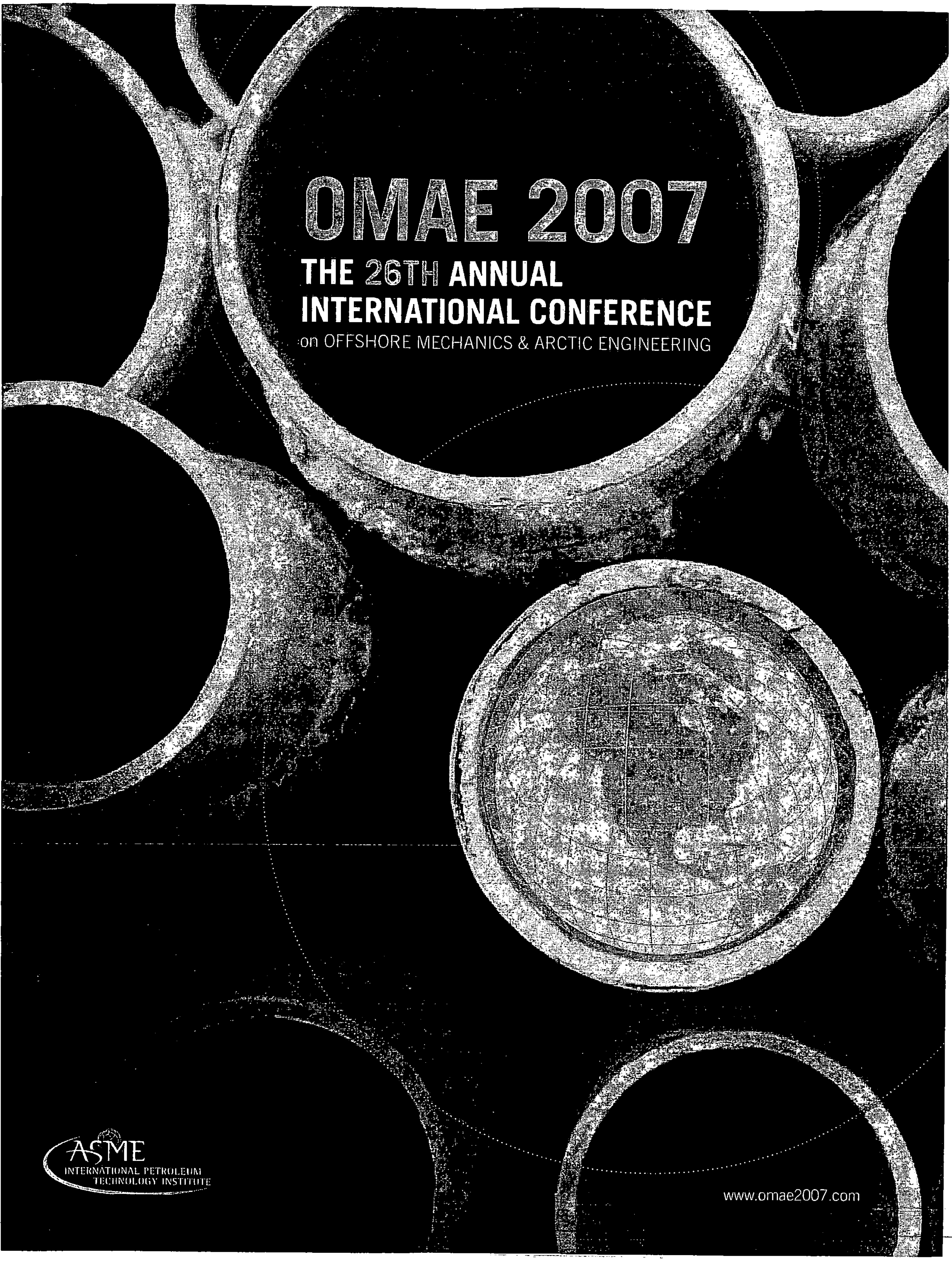

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THE EFFECT OF AIRCUSHION DIVISION ON THE MOTIONS OF LARGE FLOATING STRUCTURES

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ABSTRACT

The effect of aircushion division on the motions of large floating structures is studied by means of calculations based on a linear three-dimensional potential method. A linear adiabatic law is used to describe the air pressures inside the cushions. The water surface within the aircushions and the mean wetted surface are modelled by panel distributions representing oscillating sources. The behaviour of different types of aircushion supported structures is described and compared with that of a rectangular barge having the same dimensions. Successively, the aircushion theory, motion characteristics, wave frequency forces and moments, mean second order drift forces and surrounding wave fields are discussed. The results show that aircushions significantly influence the stability and behaviour of large floating structures.

KEYWORDS

Floating structures; aircushion; compressibility; stability; motion behaviour; wave forces; drift forces; wave field; VLFS.

INTRODUCTION

The use of aircushions to support floating structures has been known for a long time in the offshore industry. Among the first large structures which were partially supported by air were the Khazzan Dubai concrete oil storage units installed in the Persian Gulf in the early 70's, see Burns et. al [1].
In most applications the draft of the structure was decreased by pumping compressed air underneath the construction to allow transportation over a shallow water area as described by Kure et. al. [3].

At Delft University of Technology, the behaviour of large aircushion supported structures in waves has been studied by Pinkster et. al. [4-6] . The existing linear three dimensional diffraction code DELFRAC was modified to take into account the effect of one or more aircushions under a structure at zero forward speed in waves. Model tests were performed by Tabeta [7] and served to validate the results of the computations.

In the present paper a short review is given of the main elements underlying the computational method. The stiffness coefficients and stability of aircushion supported structures is described and a brief discussion is included of the behaviour of different configurations of aircushion supported structures. Successively the motion characteristics, wave frequency forces and moments, mean second order drift forces and surrounding wave fields of the different configurations are presented.

AIRCUSHION THEORY

The volume change in the aircushion is reversible and describes a polytropic process of the form:

$$PV^{\kappa} = \text{constant} \quad (1)$$

The pressure in the aircushion due to waves and oscillations of the structure can be expressed by:

$$P(t) = P_0\left(\frac{V_0}{V(t)}\right)^{\kappa} \quad (2)$$

in which:
V_0 = Initial volume of the aircushion
$V(t)$ = Volume of the aircushion ($V_0 + \Delta V$)
P_0 = Initial cushion Pressure ($P_a + P_c$)
$P(t)$ = Pressure inside the aircushion
κ = gas law index (1.4 for air)

In the above, P_a is the atmospheric pressure, P_c is the pressure due to the support of the structure and ΔV is the volume variation of the cushion.

The given non-linear expression for the pressure was rewritten in a linear form as the general calculations are also based on linear methods. Equation (2) can be made linear by a Taylor expansion of $\left((V_0+\Delta V)^{-\kappa}\right)$ around point $(\Delta V = 0)$, assuming that the volume variations are small compared to the total volume of the cushion, this results in the following equation as was shown by Ikoma et. al. [2]:

$$P(t) = P_0 - \kappa P_0 \frac{\Delta V}{V_0} \tag{3}$$

The spring stiffness of all (N_{AC}) aircushions together is equal to the sum of the individual cushions. The total spring coefficient as given below is derived from the previous equation with use of $V_0 = h_c\, A_c$, in which h_c is the cushion height and A_c the cushion area:

$$c_{33,c} = \sum_{i=1}^{N_{AC}} \kappa\, P_0 \frac{A_{c,i}}{h_{c,i}} \tag{4}$$

It should be noted that the spring coefficient in Eq. (4) is for aircushions only, i.e. the spring stiffness of the buoyant part of the structure is not taken into account in this expression. The contribution of the structure will be discussed in the next section.

Aircushion supported structure
The previous section described the heave stiffness of aircushions only. Henceforward the buoyant part of the floating body is also taken into account.

Due to the fact that the air underneath the structure is enclosed by water instead of a rigid construction, the heave stiffness of the cushions will be less than described in Eq. (4). The cushion height influences the compressibility of the enclosed air, the polytropic process as presented in Eq. (1) can therefore be written as:

$$\left(\frac{P(t)}{P_0}\right)^{1/\kappa} \cdot h_c = \text{constant} \tag{5}$$

The air pressure P_0 is equal to the atmospheric pressure P_a in case the structure is fully supported by its floaters. The cushion pressure can be described as follows:

$$P(t) = P_a + \rho\, g\, T_c \tag{6}$$

In which T_c is the vertical distance of the free surface in the cushion below the mean sea level.

When ε is defined as a small dimensionless number representing the compressibility of the aircushion, the aircushion itself is compressed by $\varepsilon\Delta T$ in case the structure moves down. Substitution of Eq. (6) in Eq. (5) finally results in:

$$\left(1+\frac{\rho g}{P_a}T_c\right)^{1/\kappa} \cdot h_c = \left(1+\frac{\rho g}{P_a}\left(T_c + (1-\varepsilon)\,\Delta T\right)\right)^{1/\kappa} \cdot \left(h_c - \varepsilon\Delta T\right) \tag{7}$$

The right hand side of the expression can be rewritten with use of a Taylor expansion around $\Delta T = 0$, resulting in the compressibility factor of the aircushion:

$$\varepsilon = \frac{\rho\, g\, h_c}{\kappa P + \rho\, g\, h_c} \tag{8}$$

where P is $P(t)$ as defined in Eq. (6).

Stiffness coefficients and stability
The aircushion supported structure can be modelled as a mass spring system shown in Fig. 1.

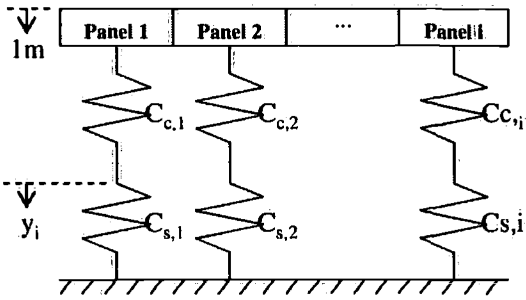

Figure 1: Mass spring system of an aircushion supported structure.

The structure is supported by water and air. Air underneath the construction is in its turn supported by the surrounding water. Displacing the structure in any of the three vertical modes heave, roll or pitch will change the volume of an aircushion thus inducing pressure changes. In order to determine the heave spring stiffness of the structure, both air and water can be modelled as springs with stiffness $C_{c,i}$ and $C_{s,i}$ respectively, resulting in a general expression of the heave stiffness:

$$c_{33} = \rho\, g\,(A_w - A_c) + c_{33,c} - \frac{c_{33,c}^2}{c_{33,c} + \rho\, g\, A_c} \tag{9}$$

in which A_w is the total waterline area of the structure. The first term represents the hydrostatic restoring force of the buoyant part of the structure, the second and third part are contributions of the aircushion.

In order to determine the stability of the floating body with multiple cushions, the displacement of the centre of buoyancy (B) has to be determined, see Fig.2. In case of small heeling angles (ϕ) the vertical displacement of B can be neglected. Both the structure and the cushions are subjected to a buoyancy force, the distance $B\,B_\phi$ of the structure is:

$$BB_{\phi,s} = 2\frac{\int_0^L \int_{y_{c,max}}^{b/2} y_s^2 \tan\phi\, dy_s\, dx_s}{\nabla} \tag{10}$$

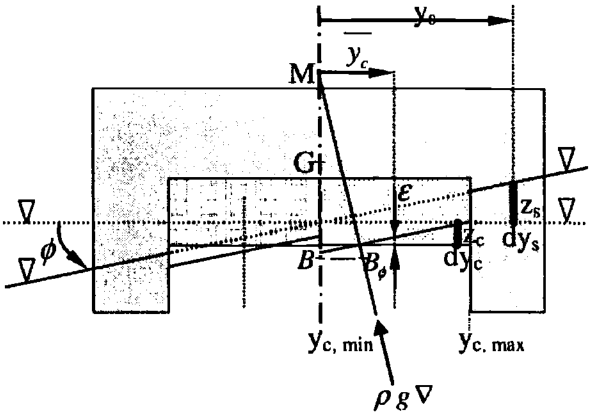

Figure 2: Stability of a structure with two aircushions.

The distance BB_ϕ of the cushion can be calculated in a similar way:

$$BB_{\phi,c} = \frac{\int_0^L \int_{y_{c,\min}}^{y_{c,\max}} \overline{z_c}\,\overline{y_c}\,(1-\varepsilon)\,dy_c\,dx_c}{\nabla} \tag{11}$$

in which:

$\overline{z_c}$ = mean increase of the cushion height $\left(\overline{z_c} = \frac{y_{c,\max}+y_{c,\min}}{2}\tan\phi\right)$

$\overline{y_c}$ = centre of the cushion in y-direction

resulting in an expression of the BM-value for aircushion supported structures:

$$BM_T = \frac{\iint_{A_s} y_s^2\,dy_s\,dx_s + \sum_{N_{AC}}\left\{(1-\varepsilon)\left(\frac{y_{\max}+y_{\min}}{2}\right)^2 \iint_{A_c} dy_c\,dx_c\right\}}{\nabla} \tag{12}$$

In order to determine the transverse stability (GM_T) the horizontal components of the air pressure on the skirts of the structure should also be taken into account. These air pressures result in an additional heeling moment around the centre of gravity of the structure as can be seen in Figure 3. This figure shows an aircushion supported structure in which the excess air pressure in each cushion is constant, so $P_1 = P_{1,H1} = P_{1,H2} = P_{1,H3}$ and $P_2 = P_{2,H1} = P_{2,H2} = P_{2,H3}$, the additional heeling moment due to an angle ϕ is:

$$X_{4c} = \sum_{i=1}^{N_c} B_i \tan\phi \cdot P_{i,H3}\left(\overline{OG} + T_{c,i} - \varepsilon\,\bar{z}_{c,i}\right)\cdot L_i \tag{13}$$

In which:

B_i = width of cushion i

L_i = length of cushion i

$P_{i,H3}$ = pressure in cushion i minus the atmospheric pressure

$T_{c,i}$ = initial draft of cushion i

$\varepsilon\bar{z}_{c,i}$ = additional mean daft of cushion i due to heeling angle ϕ

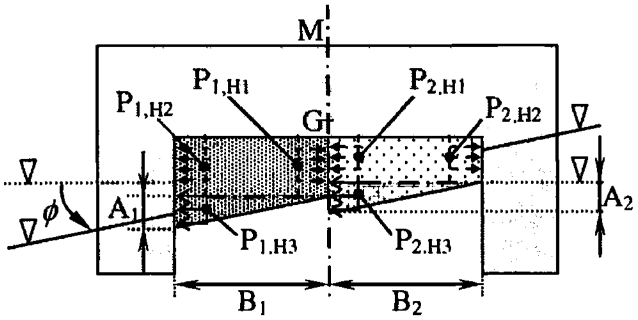

Figure 3: Horizontal air pressures inside the cushions of a structure supported by two aircushions.

With use of Eq. (12) and (13) the GM-value for aircushion supported structures with multiple cushions can be calculated as follows:

$$\overline{GM_T} = \frac{\iint_{A_s} y^2\,dy_s\,dx_s + \sum_{N_{AC}}\left\{\begin{array}{c}(1-\varepsilon)\left(\frac{y_{\max}+y_{\min}}{2}\right)^2 \iint_{A_c} dy_c\,dx_c \\ -\rho g A_c T_c\left(\overline{OG}+T_c\right)\end{array}\right\}}{\nabla} - \overline{BG} \tag{14}$$

The rotational stiffness coefficients are expressed as follows:

$$C_{44} = \rho g\left(\iint_{A_s} y^2\,ds_s + \sum_{N_{AC}}\left\{\begin{array}{c}(1-\varepsilon)\left(\frac{y_{\max}+y_{\min}}{2}\right)^2 \iint_{A_c} ds_c \\ -\rho g A_c T_c\left(\overline{OG}+T_c\right)\end{array}\right\} - \overline{BG}\nabla\right) \tag{15}$$

$$C_{55} = \rho g\left(\iint_{A_s} x^2\,ds_s + \sum_{N_{AC}}\left\{\begin{array}{c}(1-\varepsilon)\left(\frac{x_{\max}+x_{\min}}{2}\right)^2 \iint_{A_c} ds_c \\ -\rho g A_c T_c\left(\overline{OG}+T_c\right)\end{array}\right\} - \overline{BG}\nabla\right) \tag{16}$$

The non-zero coupled restoring coefficients are:

$$c_{34} = c_{43} = \rho g\left(A_w - A_c\right) y_{s,cf} + \left(c_{33,c} - \frac{c_{33,c}^2}{c_{33,c} + \rho g A_c}\right) y_{c,cf} \tag{17}$$

$$c_{35} = c_{53} = \rho g\left(A_w - A_c\right) x_{s,cf} + \left(c_{33,c} - \frac{c_{33,c}^2}{c_{33,c} + \rho g A_c}\right) x_{c,cf} \tag{18}$$

$$c_{45} = c_{54} = \rho g\left(A_w - A_c\right) x_{s,cf}\, y_{s,cf} + \left(c_{33,c} - \frac{c_{33,c}^2}{c_{33,c} + \rho g A_c}\right) x_{c,cf}\, y_{c,cf} \tag{19}$$

where x_{cf} and y_{cf} are coordinates of the centre of the water plane relative to the origin of the axis system. The subscripts 'c' and 's' are for the cushion and structure respectively. In case both structure and cushions are rectangular shaped then x_{cf} and y_{cf} can be expressed as:

$$x_{cf} = \left(\frac{x_{\max}+x_{\min}}{2}\right), \qquad y_{cf} = \left(\frac{y_{\max}+y_{\min}}{2}\right) \tag{20}$$

NUMERICAL APPROACH

The interaction between the aircushions, the structure and the surrounding water are based on a three dimensional potential theory. The rigid part of the structure is modelled in the usual way by means of panels representing pulsating sources distributed over the mean wetted surface of the construction.

The free surface within each aircushion is modelled by panels representing oscillating source distributions laying in the mean free surface of each cushion. The mean surface level of individual cushions may be substantially different from other cushions and the mean water level outside the structure.

All panels of the free surface within an aircushion are assumed to represent a body without material mass but having added mass, damping, hydrostatic restoring and aerostatic restoring characteristics. Each free surface panel has one degree of freedom being the vertical motion. The total number of degrees of freedom ($D.O.F.$) therefore amounts to:

$$D.O.F. = 6 + \sum_{c=1}^{C} N_c \tag{21}$$

In which:

N_c = number of panels in cushion c

The number 6 represents the six degrees of freedom of the rigid part of the structure. The equations of motion can in this case be written as:

$$\sum_{j=1}^{D.O.F.} \left\{ -\omega^2 \left(M_{nj} + a_{nj} \right) - i\omega b_{nj} + c_{nj} \right\} x_j = X_n, \quad n = 1, 2, ..., D.O.F. \tag{22}$$

In which:

M_{nj} = mass coupling coefficient for the force in the n-mode due to acceleration in the j-mode. Zero for cushion panels.
a_{nj} = added mass coupling coefficient
b_{nj} = damping coupling coefficient
c_{nj} = spring coupling coefficient
x_j = mode of motion
X_n = wave force in the n-mode

The wave forces X_n, the added mass and damping coupling coefficients a_{nj} and b_{nj} are determined in the same way as is customary for a multi-body system.

The contribution of the total potential due to the discrete pulsating source distributions over the structure and the free surface of the aircushions can be expressed as:

$$\phi_j\left(\overline{X}\right) = \frac{1}{4\pi} \sum_{s=1}^{N_t} \sigma_{sj}\left(\overline{A}\right) G\left(\overline{X}, \overline{A}\right) \Delta S_s \tag{23}$$

In which:

N_t = total number of panels of the structure and free surfaces of all cushions
$\overline{X}$ = X_1, X_2, X_3 = a field point
$\overline{A}$ = A_1, A_2, A_3 = location of a source
$G\left(\overline{X}, \overline{A}\right)$ = Green's function of a source in $\overline{A}$ relative to a field point $\overline{X}$
ΔS_s = surface element of the body or the mean free surfaces in the aircushions
σ_{sj} = strength of a source on surface element s due to motion mode j
$\phi_j\left(\overline{X}\right)$ = potential in point $\overline{X}$ due to j-mode of motion

The unknown source strengths σ_{sj} are determined based on boundary conditions placed on the normal velocity of the fluid at the centres of the panels:

$$-\frac{1}{2}\sigma_{nj}\left(\overline{X}\right) + \frac{1}{4\pi}\sum_{s=1}^{N_t}\sigma_{sj}\left(\overline{A}\right)\frac{\partial}{\partial n}\left(\overline{X}, \overline{A}\right)\Delta S_s = \frac{\partial \phi_j}{\partial n_m}, \quad m = 1, 2, ..., N_t \tag{24}$$

BEHAVIOUR OF DIFFERENT TYPES OF FLOATING STRUCTURES

The behaviour of different types of aircushion supported structures was calculated and compared with that of a conventional rectangular barge. Both the barge and aircushion variants had the following main particulars:

Length	150.0	m	KG	5.0	m
Breadth	50.0	m	k_{xx}	15.0	m
Draught	5.0	m	k_{yy}	42.0	m
Displacement	38437.5	t	k_{zz}	42.0	m

The height of all cushions is 5 m and the ambient air pressure was taken equal to 100 kPa. Different configurations of the structure resulted in different natural frequencies and stability

Table 1: main particulars of the structures, natural frequencies and stability

Structure type / name		Cushions	Cushion Size		ω_{n3}	ω_{n4}	ω_{n5}	GM$_T$	GM$_L$
		[-]	Length [m]	Breadth [m]	[rad/s]	[rad/s]	[rad/s]	[m]	[m]
1 cushion	(1AC)	1 x 1	150	50	0.68	n/a	n/a	-2.5	-2.5
2 cushions	(2AC)	2 x 1	75	50	0.68	n/a	0.65	-2.5	224.5
3 cushions	(3AC)	3 x 1	50	50	0.68	n/a	0.68	-2.5	266.5
4 cushions	(4AC)	2 x 2	75	25	0.68	0.73	0.65	22.7	224.5
12 cushions	(12AC)	6 x 2	25	25	0.68	0.73	0.71	22.7	291.8
24 cushions	(24AC)	3 x 8	18.8	16.7	0.69	0.78	0.72	27.4	295.4
75 cushions	(75AC)	15 x 5	10	10	0.69	0.8	0.74	29.8	298.8
Pontoon		n/a	n/a	n/a	0.69	0.8	0.74	39.2	372.5
Combi 1		1 x 1	140	40	0.69	0.77	0.82	19.3	128.6

aspects as given in table 1. A graphical representation of the 1AC and 12AC cushion variants is given in Fig. 8.

All structures, except the 24AC configuration, are modelled by square panels of 2.5 x 2.5 m. The total number of panels is equal for all structures. In case of the single cushion variant the rigid structure was modelled by 320 panels and the cushion itself by 1200 panels. Due to the deviating length-width ratio of the 24AC the individual cushions were modelled by 25 panels, resulting in a total of 600 panels for all 24 cushions together.

The whole waterline area of the structures 1AC to 75AC is covered by aircushions. The negative GM-values result from the fact that a single cushion covers the whole waterline in longitudinal or transverse direction. The wall thickness of the skirts was equal to zero. Due to small heeling angles the centre of buoyancy will not shift in these cases. Accordingly the buoyancy force acts through a fixed point at half draught of the structure and the GM-value corresponds to the distance between the centre of buoyancy and the centre of gravity.
The structures with a negative GM-value are unstable, but nevertheless have been included to show the effect of different aircushion configurations on the behaviour of the structure. In these cases additional stability can be gained by giving the skirts a thickness, this is the case for the structure referred to as 'Combi 1'. The rigid skirts surrounding 'Combi 1' have a thickness of 5 m resulting in an aircushion of 140 x 40 m. In general it can be seen in table 1 that the stability of a floating body decreases when the structure is supported by aircushions.

The motions of the various structures are given in Fig. 4, for sake of brevity only results for heave, roll and pitch are shown since these motions are most affected by the aircushions.
A change in the cushion configuration has little effect on the surge and sway motions of the structure except from the shift of the peaks at the roll and pitch motions. These local peaks are the result of the roll-sway and pitch-surge coupling.
The heave motions for all structures are approximately equal. Heave motions are relatively unaffected by aircushions. There is one exception when the wave length corresponds to the length of the cushion, in this case the pressure inside the cushion does not change and the heave motion approaches zero as can be seen in Fig. 4.
Roll motions are nearly zero in case a single cushion covers the total breadth of the structure, this is due to the fact that no natural roll frequency is present for these bodies. When the waterline beam is divided by multiple cushions the roll motions decrease with cushion width and the natural frequency shifts to the right. In case of small cushions like the 75AC, the roll motions approach those of the pontoon.
The same conclusions can be drawn for pitch motions, though in this case the length of the cushions has to be considered. The pitch motions of multiple aircushion configurations are larger than those of the conventional barge. Generally, aircushion supported structures have a small pitch damping compared to a conventional barge. For these reasons the pitch motions are largest for the 2AC and 4AC variants. Additionally, the figure clearly shows that the natural pitch frequency increases when the skirts are given a thickness.

Heave forces in head and beam seas are presented in Fig. 5. The values are approximately the same with the exception of the results at high frequencies. The small heave forces at low frequencies are due to compressibility effects of the aircushions. When the wave length corresponds to (a multiple of) the cushion size the heave force approaches zero for head and beam seas respectively, this is the case with the 1AC variant at 0.65 and 0.90 rad/s.
Roll moments in beam seas are smallest in case the cushion covers the total width of the structure. The moments are almost similar for structures having cushions of equal breadth, but they are significantly higher when the waterline beam is divided by multiple cushions.
Pitch moments in head seas are generally lower for the aircushion variants, though they significantly increase with decreasing cushion length.

The mean drift forces in Fig. 6 show that the effect of the cushion configuration is largest in head seas. For cushion lengths smaller than 25 m, drift forces are almost equal to those of the pontoon while other multiple cushion variants with larger cushions show higher peaks at 0.65 – 0.70 rad/s. In addition, at higher frequencies the drift force for structures with less than 12 cushions is small compared to that of the pontoon.

Moreover, the drift force reaches a minimum when the wavelength is equal to the cushion length, this is the case for the single cushion variant at 0.65 rad/s and for the 2AC and 4AC at 0.90 rad/s.
The figure also shows that the drift force in head seas is equal for the 1AC, 2AC and 4AC for waves smaller than 75 m (0.90 rad/s), the wavelength corresponds in this case to the cushion length of the two and four cushion variants. In general it can be concluded that for different structures, the mean drift force in a considered direction is approximately equal for wavelengths smaller than the length of the smallest cushion, providing that all bodies have similar dimensions and are totally supported by air.

Figures 7 and 9 show the surrounding wave field as well as the height of the waves inside the cushions. The wave heights are given for different types of structures in terms of non-dimensional response amplitude operators (RAOs).
For beam waves with a wavelength equal to the width of the structure (1.10 rad/s) the waves are transmitted underneath the structure. The aircushion does not absorb energy from the waves, i.e. the waves can travel freely underneath the structure resulting in a small wake behind the floating body. The reflected waves at the front are also small as could be expected from the drift forces given in Fig. 6.
The difference in the surrounding wave field between the pontoon and the aircushion variants is even more evident in head seas. For all wave frequencies, the incident waves are more distorted by the pontoon than by the single aircushion variant. The wave field surrounding the four cushion variant is similar to the one of the single and two cushion variants, parenthetically this is the case for all wavelengths smaller than the cushion length of 75 m.
Less waves are transmitted into the cushion when the skirts are given a thickness, moreover the front skirt attenuates the waves resulting in lower values underneath 'Combi 1' compared to the single cushion variant.
In addition, the wave field and drift forces in oblique seas are presented in figures 10a and 10b. The wave frequency is 0.95 rad/s corresponding to a wavelength of 68 m approximately equal to the diagonal distance between the side skirts of the structure. Again, the surrounding wave field is less disturbed in case the length of the cushions in the considered direction is equal to the wavelength.

CONCLUSIONS

The results shown in this paper indicate that the behaviour of large floating structures is significantly influenced by the use of aircushions. A single aircushion supported structure shows the best results, it has small roll and pitch motions, the wave field is less distorted resulting in low second order mean drift forces, and the wave frequency forces and moments are small. The effect of the aircushions on the drift force and the surrounding wave field is largest in head seas. The presented cushion configurations showed that the mean drift forces can be reduced in case the structure is supported by large aircushions. The advantages of an aircushion supported structure decrease when multiple cushions are used.

The results indicate that the behaviour of large floating structures partly or wholly supported by aircushions can be predicted by means of three dimensional linear potential theory. Besides, the computational method proved to be a suitable tool to optimize cushion configurations for a particular application.

Finally, the results have shown that an aircushion supported structure can be a good alternative for large floating structures.

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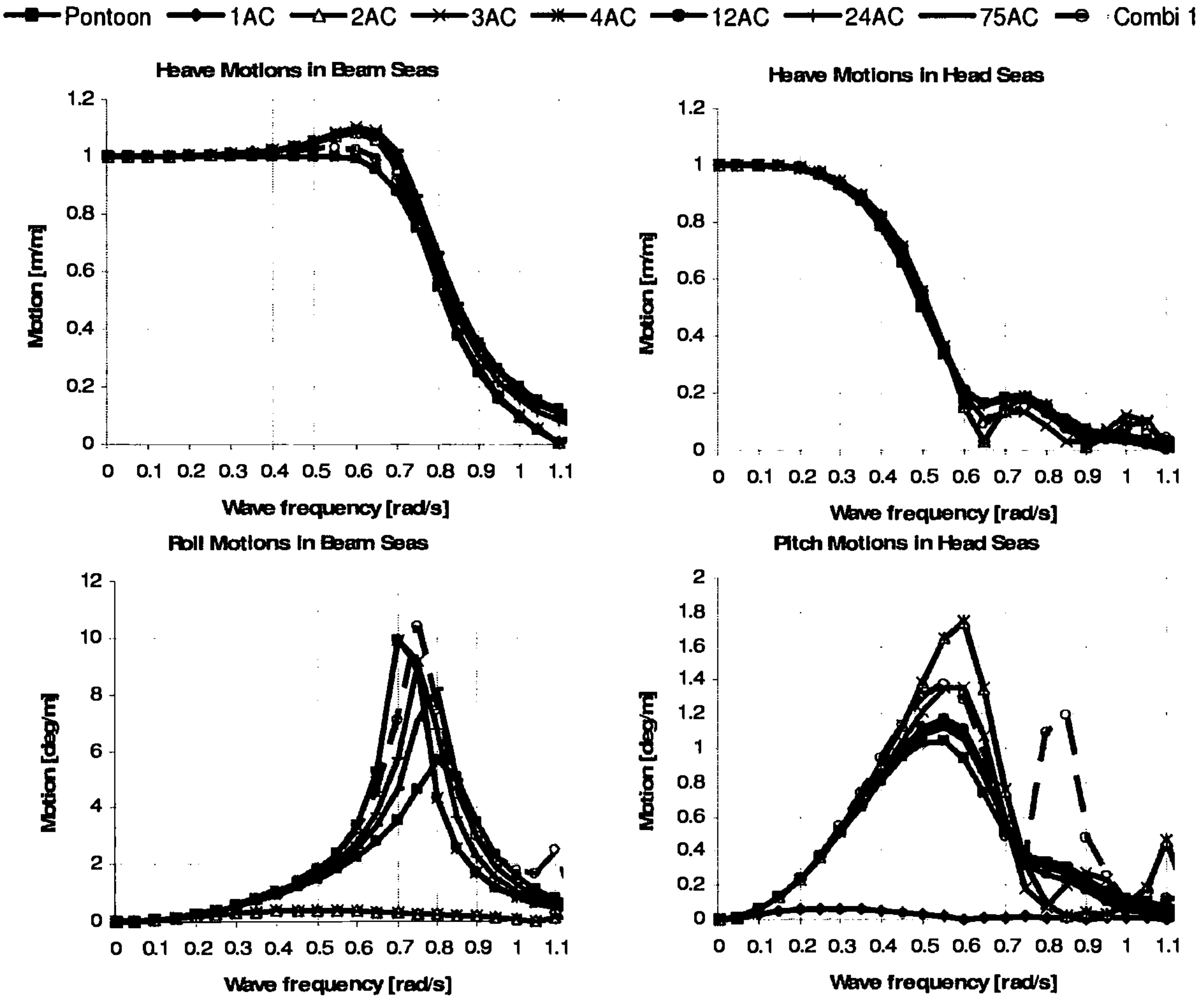

Figure 4: Motions of a pontoon and aircushion supported structures in beam and head waves.

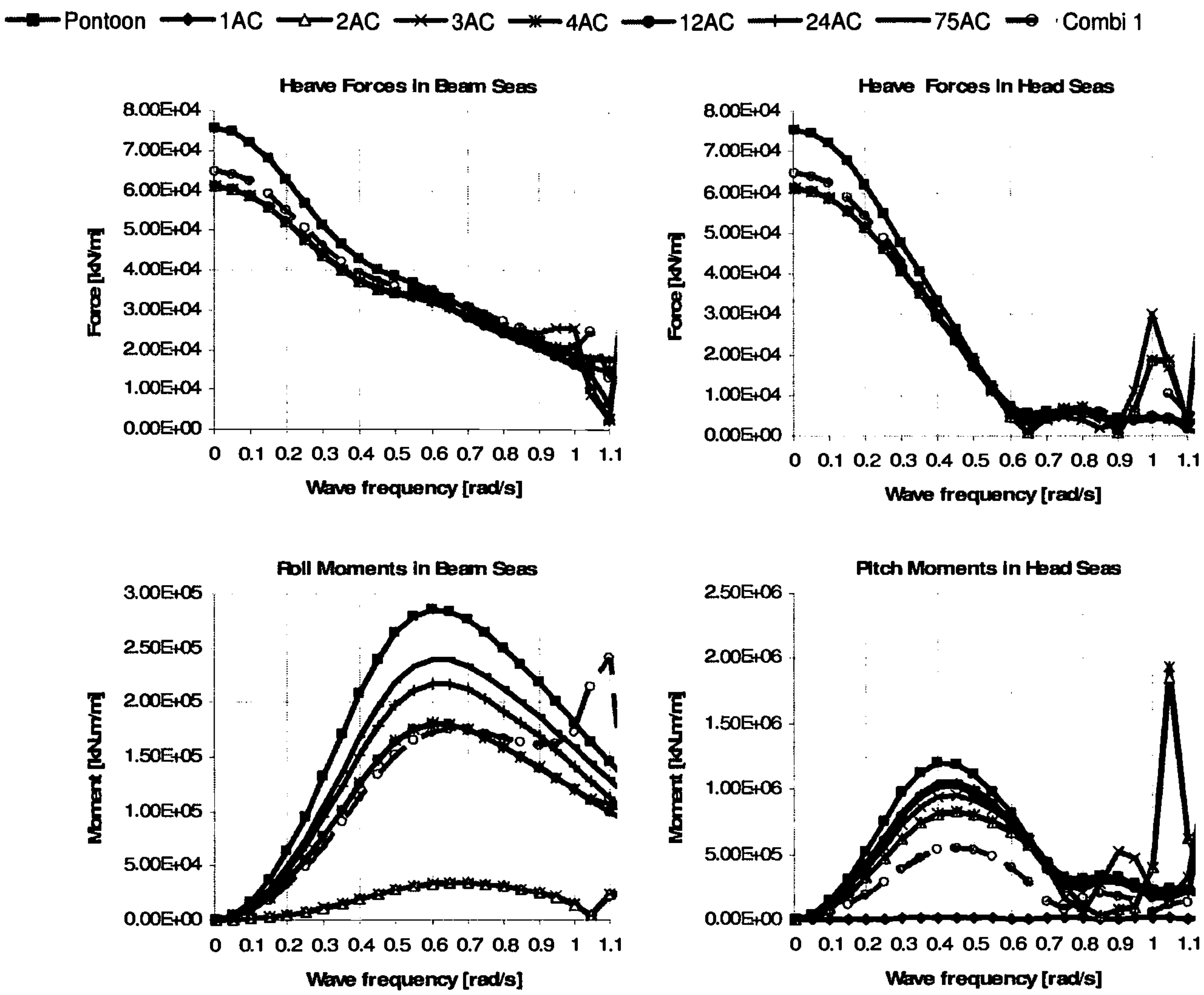

Figure 5: Wave frequency forces and moments on a pontoon and aircushion supported structures

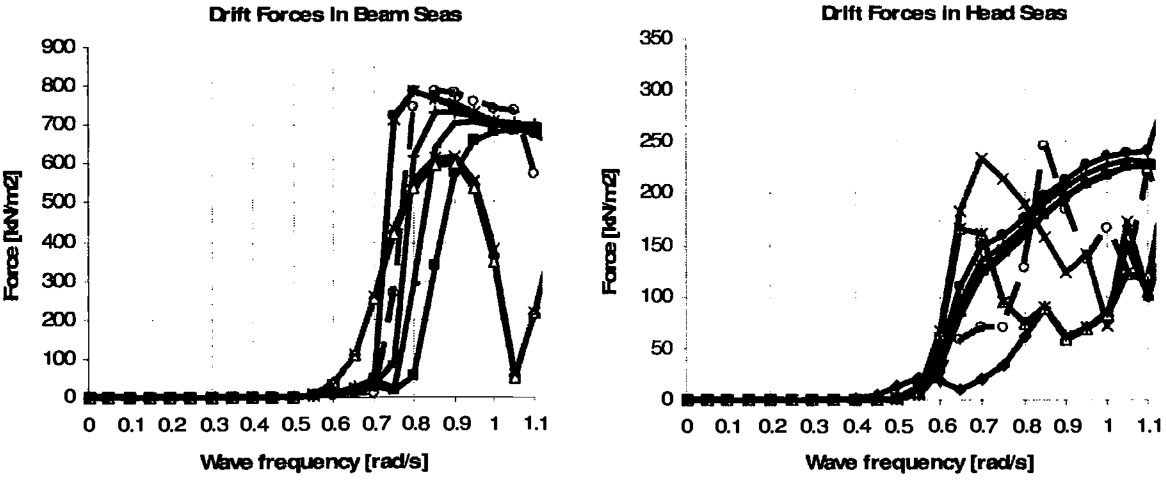

Figure 6: Mean drift forces on a pontoon and aircushion supported structures

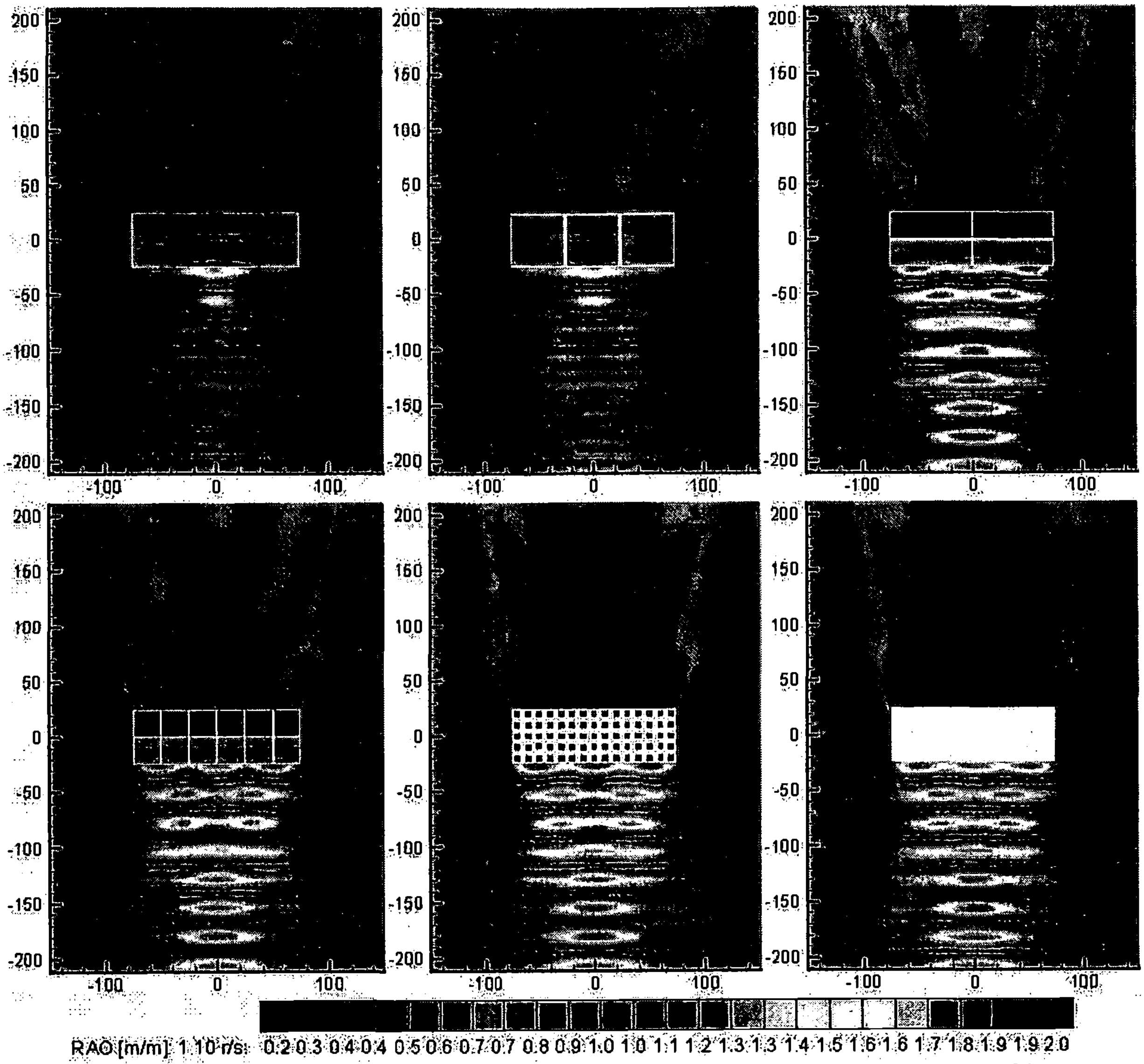

Figure 7: Wave fields surrounding a pontoon and different types of aircushion supported structures in case of beam waves with wave frequencies of 1.10 rad/s ($\lambda = 50\,m$). Respectively the following cases are presented: 1AC, 3AC, 4AC, 12AC, 75AC and a pontoon.

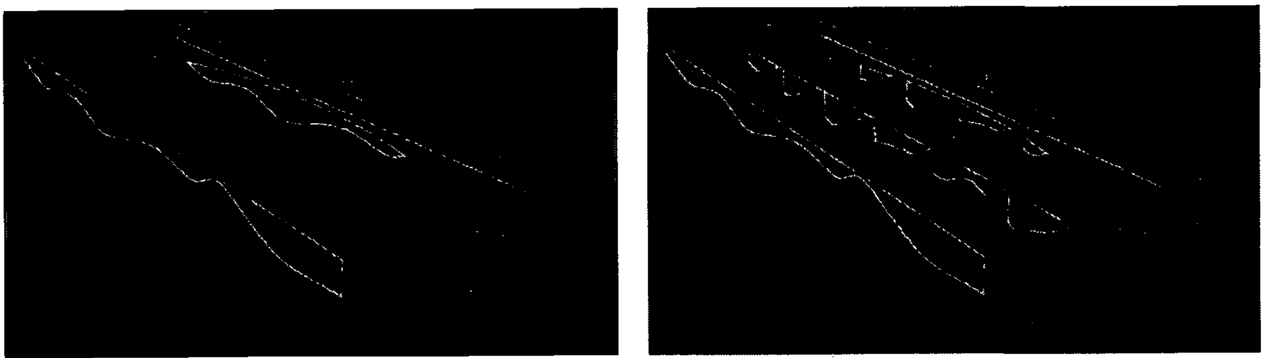

Figure 8: Graphical representation of the single (1AC) and twelve (12AC) cushion variants.

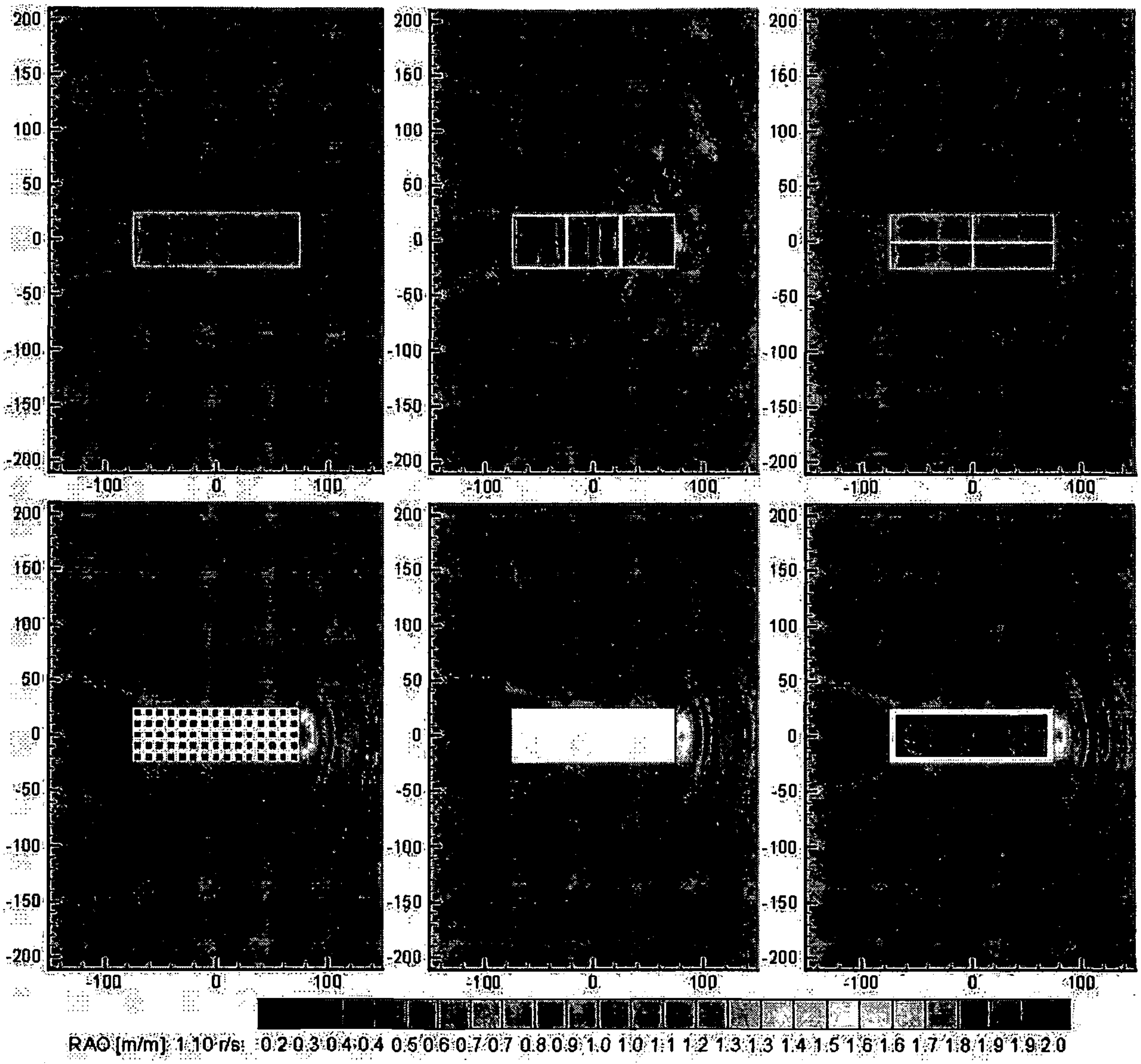

Figure 9: Wave fields surrounding a pontoon and different types of aircushion supported structures in case of head waves with wave frequencies of 0.90 rad/s ($\lambda = 75\ m$). Respectively the following cases are presented: 1AC, 3AC, 4AC, 75AC, a pontoon and 'Combi 1'.

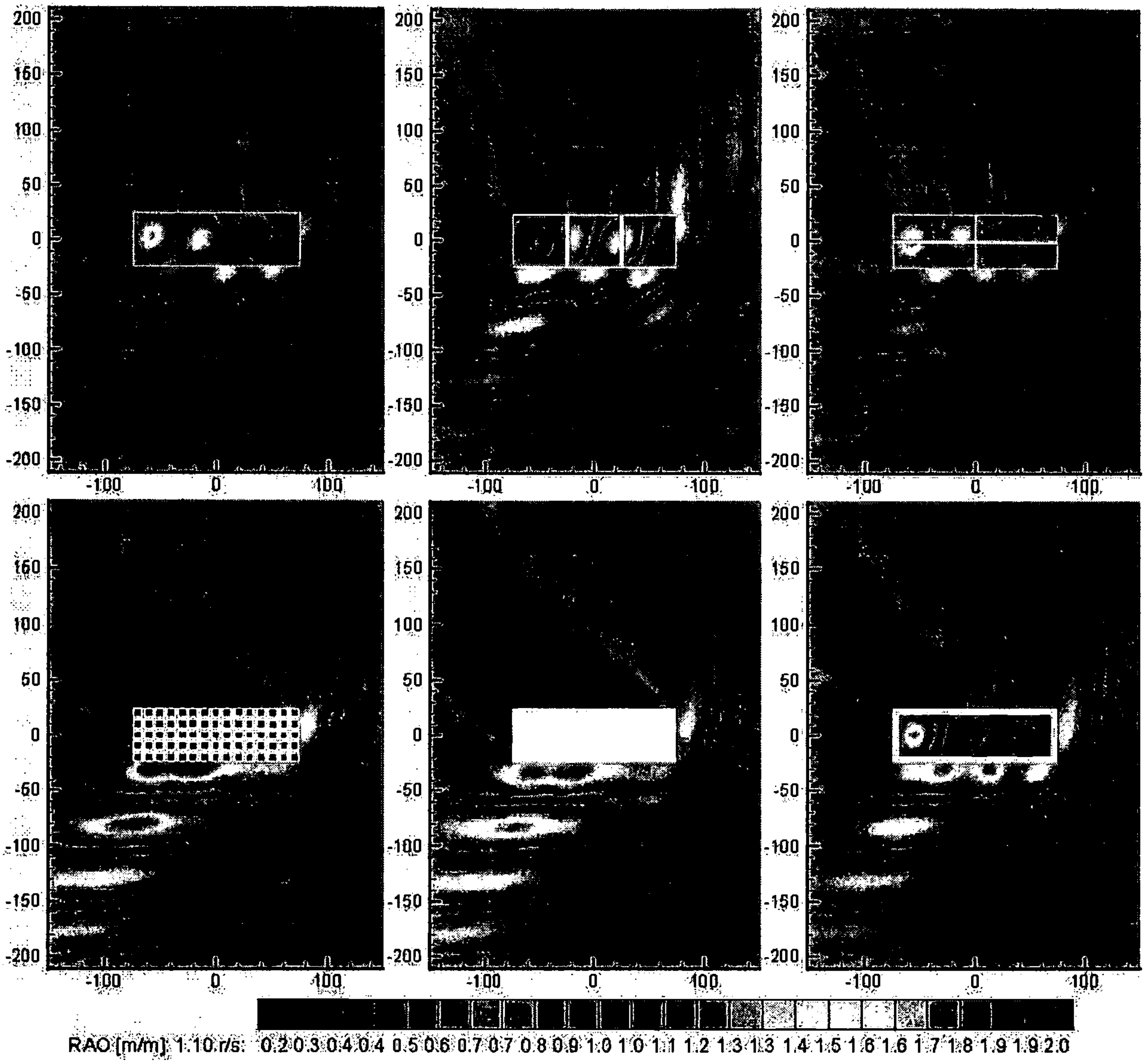

Figure 10a: Wave fields surrounding a pontoon and different types of aircushion supported structures in case of oblique waves with frequencies of 0.95 rad/s ($\lambda = 68\,m$). Respectively the following cases are presented: 1AC, 3AC, 4AC, 75AC, a barge and 'Combi 1'.

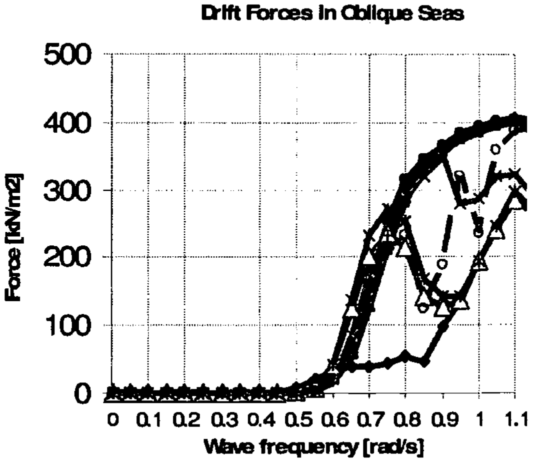

Figure 10b: Mean drift forces on a rectangular barge with and without aircushions.