ABSTRACT

Prediction of environmental forces on offshore structures is indispensable in the design and engineering stage. Evaluation and dimensioning of mooring and/or dynamic positioning systems require accurate information on both the mean forces and the dynamic loading. For semi-submersibles the complicated non-streamline geometry was normally prohibitive for calculating such environmental loads. Based on systematic wind tunnel tests, model tests and diffraction analysis a practical calculation procedure has been developed recently. The method is based on a component building block approach with special attention paid to component interaction and lift force effects.

INTRODUCTION

Floating units are used more and more for offshore exploration and production. For deep water floating production is an obvious choice, but also for moderate depths floaters are becoming more attractive. Reduction of pay-load requirements due to flexible risers, subsea completion, multi-phase pumps and decrease of production equipment make floating production an attractive alternative to fixed platforms. Floating systems are also capable to move from one field to another.

Besides the exploration and production facilities, an increasing fleet of vessels is engaged in installation, workover, maintenance and support work. For all these structures the station keeping ability is of prime importance for their operation. Passive systems such as moorings, and active systems such as dynamic positioning are used to prevent drift of the vessel due to waves, wind and current.

For the drift of a vessel basically the average components of the wind, wave and current loads are of importance. Due to the large mass of the vessel and the restoring capacity of its stationing system, the vessel normally shows a resonant response for low frequency excitation. Such excitation may originate from wind gusts, second order wave forces and changes in current. Since the damping of the system is small in this frequency region, the amplitudes of the low frequency motions may be large. Dynamically controlled systems may counteract or dampen these motions. For this purpose mooring systems are equipped with controlled thrust in 'DP assist'-mode of operation.
The direct wave induced motions of the vessel are normally not restricted by the stationing system. The related wave forces cannot be counterbalanced by the system and on the other hand the resulting oscillatory motions are of the same magnitude as the wave amplitude.

As outlined above, the wind, wave and current loads are of prime importance for the station keeping performance of floating structures. Detailed knowledge and computational tools to determine these forces are therefore indispensable for the design, engineering and operation of station keeping systems.

To compute the motions of a floating platform kept on station by means of its mooring system or by controlled thrust, a time step simulation is required. Non-linearities in the mooring characteristics and the control system prevent frequency domain calculations. Concerning the fluid loading, normally the superposition approach is applied to the reactive forces (due to oscillations in calm water) and the exciting forces (due to a captive vessel in waves). The reactive forces are then normally incorporated in the equations of motion. The wave exciting forces, wind and current and other loads are then accounted for in the right-hand side of the equations of motion.

For tanker type ships the number of shape parameters affecting the wind, wave and current loads are limited. Hence, systematic model tests, wind tunnel tests and computations can be used to provide dimensionless shape coefficients for the wind, wave and current forces (Ref. [1]). Applying these coefficients to the actual dimensions of a specific vessel a reasonably accurate prediction of the loads can be given.

Semi-submersible structures show extreme variations in shape and dimensions and therefore the above procedure cannot be followed. The shape of the floaters, the number and cross-section of the columns, the superstructures, to mention only a few, differ greatly over the fleet.

In order to develop a practical computational method for the assessment of wind, wave and current loads on arbitrary semi-submersibles, an extensive research program was carried out in 1990 by the Maritime Research Institute Netherlands. The program formed part of the development of a computer program package (DPSEMI) to simulate the low frequency motions of moored and dynamically positioned semi-submersibles. The program was coordinated by the Netherlands Industrial Council for Oceanology (IRO) and was sponsored by the following companies:

- Shell Internationale Petroleum
- Maatschappij
- Heerema Engineering
- Smit Engineering
- Gusto Engineering
- Statoil
- van Rietschoten & Houwens
- Petrobras.

In this paper the methods to compute the wave, wind and current forces are given, the simulation package and results of some simulations are discussed.

WAVES

In general terms the wave action is characterized by high frequency orbital motions of the water particles, asymptotically decreasing with the depth beneath the surface. The orbital paths of the water particles tend to circles for deep water. The path radius and period at the surface are equal to the surface elevation and the wave period respectively. Typical wave periods in irregular seas range from 4 to 15 seconds while swell waves which originate from remote wind driven seas, have periods ranging from approximately 10 to 20 seconds.

The fluid excited loads on a submerged structure may be described by the effects of gravity, inertia and viscosity, which govern the fluid motion. For large volume structures, the high frequency wave forces are dominated by gravity and inertia and are approximately proportional to the wave amplitude. This justifies the use of numerical methods based on 3-D potential theory. Assuming an ideal and irrotational fluid the flow field can be represented by a velocity potential representing the undisturbed incident wave ('Froude-Krylov' part) and the contribution of the flow disturbance ('diffraction' part). The potential contributions have to fulfill the Laplace equation and the boundary conditions at the vessel, free surface and sea floor. Knowing the potential functions,
the wave excited forces may be determined by integrating the hydrodynamic pressure, following from Bernoulli's theorem, along the wetted surface of the vessel.

Potential theory can also be used to compute the low frequency wave drift forces. Pinkster [2] developed a method based on direct integration of the second order pressures over the wetted hull by using second order boundary conditions and defining a second order velocity potential. Diffraction analyses are rather computer time consuming and are therefore normally carried out in the frequency domain (regular waves and wave groups). For time domain simulations the diffraction results e.g. the linear added mass, damping and wave force transfer functions and the quadratic transfer functions of the wave drift forces, are then transformed into the time domain by means of inverse FFT. For large volume structures such as tankers this has resulted in excellent simulation models [3], [4].

For small volume and slender bodies or structures which can be decomposed in such bodies, diffraction effects are of minor importance. The viscous effects, however, may contribute significantly to the overall forces. Furthermore, the non-linearities caused by viscosity, e.g. drag being a quadratic function of the velocity, spoil the superposition approach as outlined earlier. This means that the fluid reactive and excitation forces cannot be treated separately. For slender semi-submersibles Hooft [5] successfully subdivided the structure in slender and small simple shaped elements, calculating the forces on each element using the Morison approach and summed the forces neglecting interaction effects.

Modern semi-submersibles featuring large cross-sections for floaters and columns are a mixed breed between large and small volume structures. The first order wave induced motions may be calculated reasonably accurate by Hooft's method; on the other hand diffraction effects and interaction between the surface piercing columns are evidently important.

In 1986 Angwin [6] developed calculation procedures for the second order wave forces on semi-submersibles based on earlier work by Kagemoto and Yue [7]. The method follows on Pinkster's direct integration method but it takes advantage of the specific shape of semi-submersibles similar to Hooft's method. For fully submerged elements the effect of the free surface and potential radiation damping is neglected and interaction ignored. For vertical surface piercing members, however, these effects are considered. Results of the approach are satisfactory when compared to 3-D diffraction analysis and model tests, see Fig. 1.

The Hooft/Angwin approach for calculating first order motions and second order wave drift forces is suitable for the simulation of the dynamic behaviour of moored and dynamically positioned semi-submersibles. A reasonable accuracy is combined with an straightforward modelling and a limited computer power consumption. This enables an on-line hydrodynamic analysis in the simulation.

WIND AND CURRENT

The calculation procedure for wind and current loads on semi-submersibles is based on the building block approach. This means that the geometry is divided in a number of standard components with known force characteristics. In this way, steady fluid loads on semi-submersibles with arbitrary geometries can be obtained. The basic features of the developed calculation method are:

- an extensive database describing the standard component characteristics;
- a realistic description of the flow field experienced by the components;
- a calculation model for lift forces and overturning moments on elevated main decks of semi-submersibles.

These features are briefly dealt with in the next sections, for more detailed information one is referred to Van Walree and Willemsen [8].

The standard component database consists of a set of three-dimensional geometrical components. The force characteristics are reflected by the drag, lift and side force coefficients and their centres of effort. These characteristics are mainly obtained from published results of mod-
al experiments. By means of the following types of components an adequate description of offshore structures can be given:
- circular cylinders
- rectangular prisms
- flat plates
- lattice structures
- floater-ship type geometries.

The description of the force coefficients includes effects due to the Reynolds number, surface roughness, flow turbulence, aspect ratio and taper.

For the transfer of a force coefficient of a specific component to an actual force, knowledge is required of the mean dynamic pressure experienced by the component. A component bounded effective flow velocity $u_e$ is defined:

$$U_e^2 = \frac{1}{A} \int \int u^2(x,y,z) \, dydz \quad \ldots \quad (1)$$

where $A$ is the effective flow area and $U(x,y,z)$ is the local velocity vector for the mean $x$ position of the component.

This local velocity vector is determined by using an empirical model accounting for the boundary layer of the fluid, separated flow regions around large and bluff components (main decks) and reduced flow velocities in a component's wake. The empirical model is based on potential flow theory with corrections for viscous effects. Special purpose wind tunnel tests have been conducted for its verification, see Van Walree and Willemsen [8].

Wind loads

It is known that main deck induced lift forces may contribute significantly to the drag and overturning moments due to wind. A calculation model for these wind loads, based on classic aerodynamic theories on lifting surfaces, is used within the wind loads calculation procedures. The features of a semi-submersible platform are a rather linear lift curve with a positive (upwards) lift in level condition, a lift induced drag increase at inclined conditions and a lift increase due to the proximity of the water surface. The basic formulations are as follows:

$$C_L = C_L(\alpha_e + \alpha_0) \quad \text{lift coefficient}$$

$$C_D = \frac{1}{\text{MAR}} C_L^2 \quad \text{drag coefficient} \quad \ldots \quad (2)$$

$$C_L = \frac{2 \text{MAR}}{\text{AR} + 3} f(t/c) \quad \text{lift curve slope}$$

$$\alpha_e = \theta \cos \psi_y - \phi \sin \psi_y \quad \text{effective incidence}$$

where $\alpha_0$ is the upflow correction for the incidence experienced at level conditions, $\text{AR}$ is the aspect ratio of the main deck ($b^2/A$) and $f(t/c)$ is an empirical correction accounting for thickness effects on the lift curve slope. The upflow angle $\alpha_0$ is estimated from the calculated drag distribution along the platform and the air-gap geometry by means of an empirical relation. Similar formulations are used for the lift and drag centre of effort, needed for their contribution to the overturning moment.

The various formulations have been validated by using general experimental data and by conducting special purpose wind tunnel tests at the Dutch Aerospace Laboratories NLR. A good impression of the predicting capabilities of the complete calculation model may be derived from Fig. 2. It is seen that the lift and drag forces and the overturning moment are in good agreement with the aforementioned experimental results.

Since the calculation model is to be used for time domain simulations, dynamic wind load effects have to be taken into account. This is done by generating a wind velocity time series $u_e(t)$:

$$u_e(t) = \int_0^\infty S_e(\omega) \, \omega \cos(\omega t + \phi) \quad \ldots \quad (3)$$

This gust velocity is based on an effective wind spectrum as follows:

$$S_e(\omega) = \frac{X^2(\omega)}{S(\omega)} \quad \ldots \quad (4)$$

where $X^2$ is an aerodynamic admittance function and $S$ is a wind spectrum formulation according to Davenport, Harris, Ochi or others. The instantaneous wind load component $i$ follows from:

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\[ F_i(t) = \frac{1}{2} \rho A_1 C_{p1} (\bar{U} + u_e(t))^2 \]  

\( i = 1-6 \)  

(5)

where \( C_{p1} \) is the total wind force coefficient for the semi-submersible, obtained as outlined above.

The aerodynamic admittance \( X^2 \) takes into account the loss of the effectiveness of wind gusts, which are small compared to the size of the structure, and is determined according to:

\[ X^2(\omega) = \sum_{i=1}^{n} \sum_{j=1}^{n} k_j k_i \int \gamma_{j1}(\omega) \, dA_j \, dA_i \]  

(6)

where each component of the structure contributes a fraction \( k \) to the total mean wind force and \( A_j \) is the projected area of component \( j \).

This formulation is based on a so-called root coherence function which generally has the following form (see Vickery and Pike [9])

\[ \gamma_{j1} = \exp \left[ -\frac{\omega}{2\mu} \sqrt{a^2 \beta^2 + b^2 \alpha^2} \right] \]  

(7)

where \( \bar{U}_{j1} \) is the average mean wind speed between \( \omega \) components \( j \) and \( l \), \( \alpha \) and \( \beta \) are the horizontal and vertical separations of components \( j \) and \( l \) and \( a \) and \( b \) are empirical constants.

Current loads

The calculation model for wind loads is also used for current loads, simply by using the mass density of water instead of air. This is limited to relatively low velocities since no wave making resistance can be taken into account. The calculated force coefficients are valid for quasi-steady relative current velocities, composed of the current velocity and the structure's low frequent motion components. In case of low frequency oscillations about a mean position, extra turbulence will be present in the flow. This may have a rather significant effect on the hydrodynamic damping experienced by the oscillating semi-submersible. These turbulence effects are estimated and incorporated in the component force characteristics.

Shallow water effects are estimated by using an empirical method developed for ordinary ships. The formulation was adapted for use for semi-submersibles and reads:

\[ \frac{C_{p0}}{C_{D0}} = C_1 \left( \frac{B}{T} - 1 \right) C_2 - \frac{D_0 \sum A_i}{1} \cdot \frac{1}{\mu_0^2} \]  

(8)

where \( C_1 \) and \( C_2 \) are empirical constants, \( B \) is a reference width, \( T \) is the draft, \( h \) is the water depth, \( C_{D0} \) is the basic component drag coefficient, \( A_i \) is the component projected flow area and \( D_0 \) is the horizontal drag force without blockage effects.

The Figs. 3 and 4 show a comparison between calculated and experimentally derived current force coefficients in steady flow. It is thought that the agreement is sufficient for engineering simulations.

**COMPUTER MODEL**

Recently, a time domain simulation program for the low frequency behaviour of moored and/or dynamically positioned semi-submersibles in six degrees of freedom has been completed: DPSEMI. This program is based on the main body of the MARIN DP-simulation program for ships, DPSIM, see Nienhuis [10], and the aforementioned calculation model for environmental loads. The basic principles of this program will be reviewed here.

For the determination of the low frequency motions two axis systems are used. The axes \( x-\)y \( z \) are fixed in space with \( x \), vertical and \( y \) from right to left, see Fig. 5. A second axis system \( x-\)y-z \( \) is fixed in the centre of gravity of the semi-submersible and moves parallel to the space-fixed system \( x-\)y-z \( \) with the velocity of the semi-submersible.

The following equations of motion relate the motions of the semi-submersible to the external forces acting on it:
\[ \begin{align*}
m(\dot{u} + qv - rv) &= X, \\
m(\dot{v} + ru - pw) &= Y, \\
m(\dot{w} + pr - qu) &= Z, \\
I_{xx}\dot{p} + qr(I_{zz} - I_{yy}) - I_{xx}\dot{r} &= K, \\
I_{yy}\dot{q} + rp(I_{xx} - I_{zz}) &= M, \\
I_{zz}\dot{r} + pq(I_{yy} - I_{xx}) - I_{xx}\dot{p} &= N \\
\end{align*} \]

where \( m \) and \( I \) are the mass and mass moments of inertia, including added mass components, \( \mathbf{u} = (u,v,w) \) is the translation velocity vector, \( \mathbf{\omega} = (p,q,r) \) is the rotation velocity vector, \( \mathbf{X} = (X,Y,Z,K,M,N) \) is the external force vector and an overdot denotes differentiation with respect to time.

These equations are solved for the motions in the \( x-y-z \) axis system. Three more equations relate these motions to the motions in the space-fixed axis system \( x_0-y_0-z_0 \):

\[ \begin{align*}
\dot{x}_0 &= u \cos \psi - v \sin \psi, \\
\dot{y}_0 &= u \sin \psi + v \cos \psi, \\
\dot{z}_0 &= w \\
\end{align*} \]

\[ \mathbf{X} = \mathbf{X}_c + \mathbf{X}_w + \mathbf{X}_{wv} + \mathbf{X}_T + \mathbf{X}_M + \mathbf{X}_H \]  

The force vector \( \mathbf{X} \) is made up of contributions due to current, wind, waves, thrusters, mooring system and hydrostatic forces:

\[ \mathbf{X} = \mathbf{X}_c + \mathbf{X}_w + \mathbf{X}_{wv} + \mathbf{X}_T + \mathbf{X}_M + \mathbf{X}_H \]  

The hydrostatic forces and moments are determined from the geometry of the underwater structure and will not be treated any further here.

Current forces

The current forces are obtained from a database determined by means of the aforementioned calculation model:

\[ \mathbf{X}_c = \frac{1}{2} \rho_c c_c \left( A_c \left( \phi, \theta, \psi_{cr} \right) \right) \]

where \( \mathbf{X}_c \) is the velocity of the structure through the water, \( A_c \) is a reference area, \( c_c \) is the current force coefficient vector and \( \psi_{cr} \) is the angle between the velocity through the water and the structure's heading.

The determination of the current forces includes the hydrodynamic damping since the relative current velocity vector \( (V_{cr}, \psi_{cr}) \) is used:

\[ \begin{align*}
V_{cx} &= V_c \cos \psi - u, \\
V_{cy} &= V_c \sin \psi - v, \\
V_{cr} &= \sqrt{V_{cx}^2 + V_{cy}^2}, \\
\psi_{cr} &= \arctan(V_{cy}/V_{cx}) - \psi_c \\
\end{align*} \]

where \( V_c \), \( \psi_c \) denote the current velocity and direction respectively, see Fig. 5.

An additional hydrodynamic damping contribution is added, accounting for the yaw velocity \( \psi \) of the structure.

Wind forces

For the determination of the wind forces a similar procedure is followed:

\[ \mathbf{X}_w = \frac{1}{2} \rho_w V_{wr}^2 A_w C_w(\phi, \theta, \psi_{wr}) \]  

where \( V_{wr} \) and \( \psi_{wr} \) are the relative wind velocity and direction respectively.

Wave forces

The mean drift forces \( \mathbf{X}_{wv} \) are computed with the help of the transfer function \( T(\omega, \omega, \psi_{wv}) \):

\[ \mathbf{X}_{wv}(\psi_{wv}) = \frac{1}{2} \int \frac{S(\omega)}{r_0^2} \frac{T(\omega, \omega, \psi_{wv})}{\tau_0^2} \, d\omega \]  

For the dynamic behaviour the drift force, spectral density functions \( S_x(\mu) \) are determined as follows:

\[ S_x(\mu, \psi_{wv}) = 8 \int S(\omega) S(\omega + \mu) \left[ \frac{T(\omega, \omega, \psi_{wv})}{\tau_0^2} \right]^2 \, d\omega \]  

\[ \text{where } S(\omega) \text{ is the spectral density function of the wave force.} \]
where $\psi_{wv}$ is the wave direction, $S(w)$ is the wave spectrum, $\zeta_{w}$ is the wave amplitude and $\mu$ is the frequency of a regular wave group originating from a system of regular waves with frequencies $\omega$ and $\omega + \mu$.

A method devised by Pinkster [2] to generate an exponentially distributed drift force record is based on the fact that

$$\varepsilon = \ln(\text{ran}) \quad \text{(17)}$$

has an exponential distribution with average $-1$ and standard deviation 1, where ran denotes a random number. The time varying wave drift forces are obtained from:

$$X_{wv}(\psi_{wv}) = -X_{A}(\psi_{wv})(1+\varepsilon) + \bar{X}_{wv}(\psi_{wv}) \quad \text{(18)}$$

for the force components and

$$X_{wv}(\psi_{wv}) = -X_{A}(\psi_{wv})\varepsilon \text{ sign(ran-0.5)} + \bar{X}_{wv}(\psi_{wv}) \quad \text{(19)}$$

for the moment components.

The amplitudes of the drift forces depend on the spectral densities at zero frequency $S_{x}(\mu=0)$:

$$X_{A}^{2}(\psi_{wv}) = S_{x}(\mu=0,\psi_{wv}) \frac{\pi}{\Delta t} \quad \text{(20)}$$

where $\Delta t$ is the sample time step.

**Thruster forces**

In the computer model, three types of thrusters are distinguished: azimuthing thrusters, tunnel thrusters and main propellers. The total thrust force is the sum of the individual thrust forces:

$$X_{T} = X_{az} + X_{t} + X_{m} \quad \text{(21)}$$

All these thruster forces are split into a thrust force and an induced hull force. The thrust forces are determined by means of the actual thruster setting in terms of the RPM and P/D ratio and a polynomial representation of open water diagrams for Ka and B-series propeller types.

In this respect, the propeller speed of advance is an important quantity. Due to interaction between the underwater structure, the relative current velocity and induced velocities due to other thrusters a rather disturbed velocity field may be created. A semi-empirical calculation model accounting for these effects has been derived in the past, see Nienhuis [10]. This calculation model is based on an extensive series of model experiments conducted at MARIN.

Besides the thrust force delivered by the propeller, an induced hull force is present. For ships in ahead conditions this is known as the thrust deduction effect. For semi-submersibles the situation is much more complicated due to the interaction between the jet of water generated by the thrusters, the current velocity field and the underwater structure where all three contributions vary in magnitude and direction. The induced hull forces due to these interaction effects can only be determined from detailed model test results.

For use in DPSIM and DPSEMI a number of systematic model tests have been carried out concerning:
- azimuthing thruster interaction;
- tunnel thruster-hull interaction;
- main propeller-stern tunnel thruster-hull interaction;
- main propeller-rudder-hull interaction;
- azimuthing thruster-floater interaction.

These tests were all performed for a number of hull and/or floater shapes and for the full range of current direction (0-360 deg). The results have been converted into empirical models for the determination of the induced hull forces. The general form of these models is as follows:

$$F_{i} = F_{i}(h,T,D,\alpha_{T},V_{cr},\psi_{cr}, \text{hull form}) \quad \text{(22)}$$

where $F_{i}$ is induced hull force component $i$, $h$ is the water depth, $T$ is the thrust force, $D$ is the propeller diameter, $\alpha_{T}$ is the direction of the thrust and $V_{cr}$ and $\psi_{cr}$ are the relative current velocity and direction respectively.

In these models, only the low frequency motion components of the semi-submersible are taken into account.
Mooring forces

The mooring system forces are obtained from the static load curves of the individual mooring lines and the actual position of the semi-submersible. Effects due to first order wave motions and dynamic mooring line damping are not taken into account. These matters are under investigation and may be incorporated in the program in the future.

Control system

In order to perform realistic simulations of the low frequency behaviour of the semi-submersible, a control system must be included. This control system links the position deviations to the thruster forces. In the computer program, arbitrary control systems may be used.

A default control system of the PID-type is included. The required thruster forces \( X_{\text{treq}} \) are then obtained from:

\[
X_{\text{treq}} = \overline{X} + c\Delta x + b\Delta x + i \int \Delta x dt + X_{\text{vf}}
\]

\[
\ldots (23)
\]

where \( \overline{X} \) is the mean environmental force, \( \Delta x \) and \( \Delta x \) are the position deviation and velocity respectively, \( c, b \) and \( i \) are the damping, spring and integrator coefficients respectively and \( X_{\text{vf}} \) is the wind feed forward force vector. Only the low frequency motions in the horizontal plane are considered, so \( X_T = (X,Y,N) \).

The position deviations \( \Delta x \) may be contaminated with an arbitrary noise signal. One of the options is to add the first order wave motions to the noise signal. On the other hand, the position deviation signals may also be filtered in order to remove high frequency components.

The required force vector \( X_{\text{treq}} \) has to be split into the forces to be delivered by each thruster. To this end a non-linear procedure has been implemented which minimizes the following penalty function \( P \):

\[
P = \sum_{i=1}^{n} \left( T_i^2 + P(X_{\text{treq}} - T_i) \right)
\]

\[
\ldots (24)
\]

where \( T_i \) are the thrust forces of \( n \) thrusters, \( P \) is a weight factor vector and \( T_i \) is the force vector due to the thrust forces \( T_i \) and their directions \( \alpha \).

By using this function, a compromise is reached between a low power consumption (\( T_i^2 \) term) and the satisfaction of the positioning requirements in terms of the required forces. Additional penalty functions on the thruster angles \( \alpha \) may be used in order to minimize interactions between azimuthing thrusters.

RESULTS

In order to show the use of the computer program some simulation results are shown here. These results concern a simulation of the dynamic positioning properties of a representative semi-submersible in moderate environmental conditions. The main particulars are shown in Table 1 while Fig. 6 shows the general arrangement.

Two environmental conditions are considered, see Table 2. The first one, CASE1, with unidirectional wind, wave and current and a second one, CASE2, with the current at a direction of 35 degrees relative to the wave and wind direction. The mean environmental loads are plotted in Fig. 7. The Figs. 8 through 13 show time series of the position deviations, the inclination angles, mooring line tensions, thruster forces, directions and consumed power.

Some statistical information on the position deviations is provided in Table 3. The results show in general smaller position deviations for CASE2, especially for the sway and pitch modes of motion.

REFERENCES


Table 1 - Main particulars of semi-submersible

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
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<td>Beam overall</td>
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<tr>
<td>Height from keel to main deck</td>
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Table 2 - Environmental conditions

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Table 3 - Statistical results of position deviation

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Fig. 1 - Comparison mean wave drift forces
Eight column semi-submersible:
wave angle 90 deg, beam seas
Fig. 6 - General arrangement

Fig. 7 - Mean environmental loads due to wind, waves and current

Fig. 8 - Time traces of simulation results CASE1

Fig. 9 - Time traces of simulation results CASE1
Fig. 10 - Time traces of simulation results CASE1

Fig. 11 - Time traces of simulation results CASE2

Fig. 12 - Time traces of simulation results CASE2

Fig. 13 - Time traces of simulation results CASE2