ABSTRACT

This paper describes a series of model tests aimed at gaining insight in the tension variations in the export risers and mooring lines of a CALM buoy. The test result were therefore not only analysed carefully, but were also used as input and to validate a numerical tool that computes the coupled motions of the buoy and its mooring system.

The tests were carried out at a model scale of 1 to 20. Captive tests in regular and irregular waves were carried out to investigate non-linearities in the wave forces on the buoy for example from the presence of the skirt. Decay tests were carried out to determine the damping of the buoy's motions and to obtain the natural periods. Finally, tests in irregular waves were carried out.

The dynamics of the mooring system and the resulting damping have a significant effect on the buoy's motions. A numerical tool has been developed that combines the wave-frequency buoy motions with the dynamical behaviour of the mooring system. The motions of the buoy are computed with a linearised equation of motion. The non-linear motions of the mooring system are computed simultaneously and interact with the buoy's motions.

In this paper, a comparison is shown between the measurements and the simulations. Firstly, the wave forces obtained with a linear diffraction computation with a simplified skirt are compared with the measured wave forces. Secondly, the numerical modelling of the mooring system is checked by comparing line tensions when the buoy moves with the motion as measured in an irregular wave test. Thirdly, the decay tests are simulated to investigate the correctness of the applied viscous damping values. Finally, simulations of a test in irregular waves are shown to validate the entire integrated concept.

The results show that:
1. The wave-exciting surge and heave forces can be predicted well with linear diffraction theory. However, differences between the measured and computed pitch moment are found, caused by a simplified modelling of the skirt and the shortcomings of the diffraction model.
2. To predict the tension variations in the mooring lines and risers (and estimate fatigue) it is essential that mooring line dynamics are taken into account.
3. The heave motions of the buoy are predicted well.
4. The surge motions of the buoy are predicted reasonably well.
5. The pitch motions are wrongly predicted.

INTRODUCTION

The dynamic behaviour of anchor lines has been a subject of study in model tests and numerical simulations for a long time. So far, studies mainly focused on the coupling between anchor line dynamics and low frequency motions of for example FPSO's. However, for floaters like a buoy the damping due to the dynamical behaviour of the mooring system is important for
the first-order motions, especially near the natural periods of the buoy motions.

Mooring buoys are located at many places around the world. The specific CALM (Catenary Anchor Leg Mooring) buoy in the tests is used as offloading point for shuttle tankers so that the shuttle tanker need not connect with an FPSO. The buoy can be exposed to considerable wave actions resulting in large motions and fatigue problems in risers and mooring lines. The model tests have been carried out at a large model scale of 1:20. The purpose of these large-scale tests was to investigate the first order buoy motions, as input to the fatigue assessment of underwater export risers between an FPSO and the buoy in mild wave conditions. A relatively large buoy model was used in order to be able to accurately model short-period waves. This is also an advantage with respect to minimisation of scale effects, especially at the buoy skirt. The FPSO was not modelled but the risers were attached to the basin wall. Due to the large scale the mooring system and risers, designed for a water depth of 1200 metres, had to be truncated to fit in water with a depth of 200 metres (10 metres basin water depth). The tests consisted of captive tests in regular and irregular waves to obtain the wave exciting forces, decay tests to estimate the damping and natural periods of the system and tests to determine the behaviour of the combined buoy and mooring system in waves.

Simulations of buoy motions have a long history. Remery and Kokeel [1] investigated buoy motions using analytical expressions for the Froude-Krylov forces on the buoy and its skirt. Forces due to the disturbance of the waves were estimated from the distribution of the added mass over the skirt. Drag coefficients were used to estimate the viscous damping of the buoy and mooring system. The results showed that the effect of the skirt can not be neglected in the wave forces. The present tests served to calibrate and validate a numerical tool that combines the dynamical behaviour of the mooring system with both second-order (low-frequency) and first-order (wave frequency) motions of the floater of interest. A combined linear equation of motion for the buoy and a non-linear equation of motion for the mooring lines are integrated simultaneously. As a pre-processing step, the tool computes the wave-exciting forces, added mass and damping values in the frequency domain and transfers these to the time domain.

The paper is organised as follows: first, a description is given of the tests and the numerical model. Second, the results of the captive tests are compared with results of a linear diffraction analysis. Third, the correct numerical modelling of the mooring system is checked by giving the buoy the motion that was measured during the test and by comparing the resulting mooring line and riser forces with the measured ones. Fourth, the decay tests are simulated to see if the theoretical damping values correspond to the actual motion decay. Finally, the numerical tool is used to determine the motions of the combined buoy and mooring system in irregular waves. A comparison is made between measured and computed motions and mooring line and riser forces. Additionally, results are also shown of the full-scale mooring system.

**NOMENCLATURE**

- \( \rho = \) density of water [kg/m³]
- \( \mathbf{A} = \) added mass matrix [kg-kgm²]
- \( a_n = \) normal added mass [kg]
- \( a_t = \) tangential added mass [kg]
- \( \mathbf{C} = \) hydrostatic spring matrix [N/m-Nm/rad]
- \( C_{Dn} = \) normal drag coefficient
- \( C_{Dt} = \) tangential drag coefficient
- \( C_{In} = \) normal mass coefficient
- \( C_{It} = \) tangential mass coefficient
- \( \mathbf{D} = \) diameter of line element [m]
- \( \mathbf{F} = \) external force vector [N-Nm]
- \( F_{Dn} = \) normal drag force [N]
- \( F_{Dt} = \) tangential drag force [N]
- \( \mathbf{K} = \) matrix of retardation [kN-m-rad-s]
- \( L = \) length of line element [m]
- \( \mathbf{M} = \) mass matrix [kg-kgm²]
- \( t = \) time [s]
- \( \mathbf{X} = \) motion vector [m-rad]
- \( \dot{x}_n = \) normal velocity of line element [m/s]
- \( \dot{x}_t = \) tangential velocity of line element [m/s]

**DESCRIPTION OF THE MODEL TESTS**

The CALM buoy was tested in MARIN’s offshore basin at a model scale of 1:20. The main particulars of the buoy are given in table 1. Figures 1 and 2 show the buoy before and during the tests. The buoy was fitted with a skirt to increase the pitch and heave damping.

**Full-scale and truncated mooring system**

In this paper, reference is made to the ‘full-scale mooring system’ and the ‘truncated mooring system’. The full-scale mooring system corresponds to the original and desired mooring and riser system of the buoy in a water depth of 1200 metres. In the basin, a water depth of 200 metres could be modelled. Therefore, the mooring system had to be truncated just above the basin floor such that the static characteristics of the truncated system were as close as possible to those of the full-scale system. Figure 3 shows a top view of the full-scale mooring system. The full-scale mooring system consists of six short mooring legs (1-3 and 7-9), three long mooring legs (4-6) and two export risers. The mooring legs consist of a heavy top
chain, a steel wire mid-section and a heavy bottom chain. The long mooring legs have a longer mid-section than the short ones. The export risers are made of flexible material and have extra buoyancy in the middle part. The static position of the risers is sketched in figure 4. To obtain a truncated mooring system with characteristics as close as possible to the full-scale system, the following approach was used:

**Risers:**
1. The full-scale riser system was cut-off just above the seabed. Only a short stretch of the two risers of approximately 200 metres could therefore be modelled. This results in a too low pretension because some weight is missing.
2. To obtain the correct pretension at the fairlead, masses were hung at the lower end of the two risers. These masses were connected with a horizontal steel wire to a second mass which was then again connected with a steel wire to the basin wall

The truncated riser system is shown in figure 5. The main particulars of the risers are shown in table 2. The angles between the risers and keel of the buoy are fixed (clamped condition).

**Long mooring legs:**
1. The long mooring legs (4-6) were cut off just above the seabed. Therefore, only the heavy top chain and part of the steel wire mid section could be modelled.
2. To compensate for the missing mass of the heavy bottom chain and the resulting too low pretension, heavy masses were hung at the lower end of the lines. These masses were connected with a steel wire to a second mass, which was again connected to the basin wall. This system is therefore similar to the one used for the risers.

**Short mooring legs:**
1. The ratio of the actual water depth (1200 m) and the tested water depth (200 m) was used to dimension the truncated short mooring legs (factor 0.167).
2. The length of the short mooring lines was reduced proportionally. This was done by removing the bottom chain and part of the steel wire mid section. A linear spring was used to attach the mooring line to the basin floor. The spring stiffness was such that the stiffness of the truncated line equals the stiffness of the original line.
3. To compensate for the missing mass of the bottom chain and the steel wire (resulting in a too low pretension at the fairlead) extra mass was placed inside the buoy to reach the required draft. The mass of the buoy was therefore increased from the original 955 tons to the tested 1299 tons.

**Model test description and results**
An extensive set of tests was carried out. First, captive tests were carried out in which the buoy model without the mooring system was fixed to the carriage with a six-components frame. Second, decay tests were carried out with and without the mooring system to estimate the damping of the buoy and to obtain the natural periods. Finally, a set of tests was carried out in irregular head seas with several mean wave periods and wave heights. In these tests, the motions of the buoy and the tensions in the risers and mooring lines were measured.

The aim of the captive tests was to investigate the nonlinearities in the wave-exciting forces and moments on the buoy. For that purpose, the forces were measured in a set of regular and irregular waves with different wave heights. Figures 6, 7 and 8 show the measured surge force, heave force and pitch moment with respect to the centre of gravity of the buoy. Also included in this figure are the results of a linear diffraction calculation, based on potential flow. The theory behind this is described by for example Huijsmans [2] and Prins [3]. The diffraction calculation was done with a simplified skirt. The panel model is shown in figure 25. The real skirt has holes in it and has an effective area of 95.5 m². The skirt in the diffraction analysis was modelled solid and with a smaller diameter such that the effective area was the same as for the real skirt. The diffraction model is based on a source-distribution approach. Therefore, opposite panels cannot be located too close to one another because they induce unrealistically large water velocities. Therefore, the height of the skirt was increased to 0.625 m. Figures 6 and 7 show that the response of the surge and the heave force is fairly linear and can be predicted well with a linear diffraction calculation. Figure 8 shows that the response of the wave-exciting pitch moment contains some nonlinearities. The overall behaviour and order of magnitude of the computed pitch moment is the same as the measured pitch moment. However, the differences are considerable. These differences can for the larger part be attributed to the simplified treatment of the skirt in the calculation. The effects of the skirt are local and far from the buoys centre (long lever) and therefore have a large effect on the pitch moments, but negligible effects on the surge and heave force.

The decay tests served to obtain the natural periods of the buoy motions and the damping at the natural period. Due to the presence of the skirt, the damping values exceed the potential damping found in the linear diffraction calculation without skirt. The extra damping can then be fitted into the numerical model.

The tests in irregular seas served to gain insight in riser tension variations, but also to validate the numerical model. In this paper, simulations are shown for the following two tests:
A JONSWAP spectrum was used with a peak factor of $\gamma = 4$. All tests were done in head seas, which means the waves propagate in the direction of the risers (see figure 3).

### NUMERICAL MODELING OF COMBINED MOTIONS OF BUOY AND MOORING LINES

It is assumed that the motions of the buoy can be described with a linearised equation of motion in the time domain. In the linearisation procedure, it is assumed that the wave-exciting forces on the buoy can be separated from the radiation forces (added mass and damping forces) due to the buoy motion. The wave-exciting forces can thus be computed for the mean position of the buoy. The forces on the buoy due to the mooring system and risers are taken into account in a non-linear way. This means the dynamics of the mooring system are fully taken into account. In the simulation the actual position of the mooring lines and risers and the dynamic tension variations are computed. These steps are sketched in the figures below:

1. **Step 1: Forces on a fixed buoy in waves**
   - $F_z$
   - $F_x$
   - $M_y$
2. **Step 2: Forces on an oscillating buoy in calm water**
   - $z(t) = A \sin \omega t$
3. **Step 3: Dynamical, non-linear mooring forces**

The motions of the buoy are described by the following equation of motion, see Ogilvie [4]:

$$\begin{align*}
(M + A)\ddot{X} + \int_{-\tau}^{\tau} K(t - \tau)x(\tau) d\tau + C \dot{X} &= F(t, X)
\end{align*}$$

Where $M$ is the mass matrix, $A$ the frequency-independent added mass matrix, $K(t)$ the retardation function matrix and $C$ the matrix with the hydrostatic spring terms, so excluding the stiffness of the mooring system. The retardation functions are the Fourier transform of the frequency dependent damping coefficients:

$$K(t) = \frac{2}{\pi} \int_{0}^{\infty} B(\omega) \cos \omega t d \omega$$

The right-hand-side forcing function $F$ contains all forces on the buoy which are not accounted for by the mass and added mass forces, the wave making damping forces (retardation functions) and the hydrostatic restoring forces, like:

1. Wave forces on the buoy
2. Viscous damping forces
3. Mooring forces

With a linear diffraction code, the wave forces, wave making damping and added mass values are computed in the frequency domain. Before the simulation starts, the frequency domain values are transformed to the time domain with knowledge of either the wave spectrum (using a random phase method) or the wave elevation (for example when a model test is simulated and the calibrated wave is used). Alternatively, measured wave forces can be inserted.

The viscous damping forces cannot be computed by the diffraction code since it is based on potential flow and viscosity has therefore been neglected. However, the viscous damping forces can be estimated from the decay model tests by subtracting the potential damping from the measured damping.

The mooring forces form the most complicated aspect in the equation of motion because they themselves depend on the motions of the buoy. The forces in the mooring lines and risers are computed fully dynamically. Drag forces and added mass are taken into account in the following way:

- **Normal drag force**: $F_{Dn} = -\frac{1}{2} \rho C_{Dn} DL \dot{x}_n |x_n|$
- **Tangential drag force**: $F_{Dt} = -\frac{1}{2} \rho C_{Dt} DL \dot{x}_t |x_t|$
- **Normal added mass**: $a_n = \rho (C_{in} - 1) \frac{\pi}{4} D^2 L$
Tangential added mass: \( a_t = \rho(C_{D_t} - 1) \frac{D^2}{4} L \)

Relative velocities due to fluid motion are not taken into account.

In the simulations presented in this paper, the following drag and mass coefficients were used:

<table>
<thead>
<tr>
<th></th>
<th>( C_{Dn} )</th>
<th>( C_{D_t} )</th>
<th>( C_{In} )</th>
<th>( C_{It} )</th>
</tr>
</thead>
<tbody>
<tr>
<td>chain</td>
<td>2.4</td>
<td>0.8</td>
<td>3.1</td>
<td>1.7</td>
</tr>
<tr>
<td>riser</td>
<td>1.3</td>
<td>0.2</td>
<td>2.0</td>
<td>1.2</td>
</tr>
<tr>
<td>steel wire</td>
<td>1.3</td>
<td>0.4</td>
<td>1.6</td>
<td>1.2</td>
</tr>
</tbody>
</table>

These coefficients are based on model tests.

To compute the dynamic position of the entire mooring line at each time step in the simulation, a lumped mass method is used, see for example Derksen and Wichers [5]. In this lumped mass method, the mass of the mooring line is lumped to a finite number of nodes that are connected by linear springs, corresponding to the axial stiffness of the line. Bending stiffness is not taken into account. In the simulation, the mooring forces in the equation of motion are kept constant over a communication interval (typically 0.1 seconds). During this interval, the equation of motion is integrated with a Runge-Kutta method and the position and velocity of the buoy at the new time level is obtained. The new position of the buoy is then used in the lumped mass method to compute the position of and the tension in the mooring lines at this new time level, after which the procedure is repeated until the simulation reaches its desired time level. Two equations of motion are solved which communicate on fixed time levels:

2. Simulate the decay test of buoy with mooring system to check the damping and natural periods of surge, heave and pitch.
3. Simulate tests in waves.

**Step 1: prescribed motions**

Step 1 does not yet involve an integration of the equation of motion. The motions of each fairlead are computed from the measured motions and inserted into the numerical model that computes the dynamics of the mooring lines. For this purpose test number 1 has been used (Hs=2.25, Tp=10 s). A comparison is made between the measured and computed top tensions in the mooring lines and risers. Figures 10 through 13 show snap shots of the measured and computed tensions. Also shown in these figures are the results from a static computation in which the static load excursion curve of the mooring lines or risers has been used to obtain the variations in the top tension. The following observations can be made:

1. The variations of the tensions in the mooring lines are small when compared to the pretension.
2. The prediction of the variation of the tension in line 2 (line with a linear spring) is very good when computed dynamically. When computed statically, a phase difference occurs.
3. The prediction of the variation of the tension in line 5 (with two heavy masses) is reasonably well when computed dynamically. The simulation is somewhat more peaked and shows some high-frequency oscillations which cannot be observed in the test. This is likely caused by inaccuracies induced by the heavy masses swinging above the basin floor. The static simulation clearly gives a much too small tension variation.
4. The prediction of the horizontal top tension of the clamped risers shows a similar behaviour: The dynamic simulation gives a more peaked signal than the test. The variations in the static simulation are too small.

These results show that the numerical modelling of the mooring system is accurate enough when the dynamics of the truncated system are fully taken into account. The results presented in the remainder of this paper are therefore all computed including the dynamics of the mooring lines.

**Step 2: Decay tests**

Step 2 involves the simulation of the decay tests for the surge, heave and pitch motions. For that purpose, the equation of motion is solved without wave forces, but with an additional external force \( F_{ext} \) given by:

\[
F_{ext} = \begin{cases} 
  F_{max} \frac{t}{t_{max}} & \text{if } t \leq t_{max} \\
  0 & \text{if } t > t_{max}
\end{cases}
\]
So the external force builds up linearly towards its maximum value and then disappears. Note that this is similar to how the decay test is carried out in the test basin.

In the equation of motion, additional heave and pitch damping values have been used that are determined from the decay tests without the mooring system. Since a surge decay test without the mooring system is not possible, the surge damping has been tuned such that the motion decay in the simulation corresponds to the motion decay in the test. Figures 14, 15 and 16 show the time traces of the surge, heave and pitch motions in the decay tests. The following conclusions can be drawn:

- The period of the surge motion in the simulation is somewhat too small. A difference of nine percent with the test is found. This is either caused by small differences between the tested and modelled surge stiffness or by a too small low-frequency surge added mass.
- The period of the heave motion in the simulation is the same as in the model tests.
- The damping that is estimated from the heave decay test is sufficiently accurate. The decay in the test and in the simulation behaves the same way.
- The differences between the tested and simulated pitch decay are large. Both period and decrease of amplitude differ. The differences may be explained as follows:
  - The risers have bending stiffness and are clamped to the buoy. This gives extra rotational stiffness that is not accounted for in the numerical tool. In the tool, bending stiffness and clamps cannot be modelled yet.
  - The effect of the skirt on the pitch added mass is significant. In the simulations, the skirt has been modelled in a very simplified manner.

Due to the differences in the pitch decay, it is not expected that the pitch motions in waves can be computed correctly.

**Step 3: Buoy and mooring system in irregular waves**

The final step is to simulate a test in irregular waves. For that purpose, test number 2 has been taken (\(H_s=2.25\) m, \(T_p=14\) s). The simulation is carried out in two ways:

1. Using the wave-exciting forces and moments measured in the captive test.
2. Using the wave-exciting forces and moments predicted by linear potential theory (making use of the calibrated wave train).

The following table shows the standard deviation of the measured and computed time traces with wave forces:

<table>
<thead>
<tr>
<th></th>
<th>sdv surge force [kN]</th>
<th>sdv heave force [kN]</th>
<th>sdv pitch moment [kNm]</th>
</tr>
</thead>
<tbody>
<tr>
<td>calculated</td>
<td>476.2</td>
<td>1170.4</td>
<td>303.0</td>
</tr>
<tr>
<td>measured</td>
<td>492.7</td>
<td>1234.5</td>
<td>383.2</td>
</tr>
</tbody>
</table>

Figures 17-19 show snapshots of the surge, heave and pitch motions. The following observations can be made:

- The prediction of the surge motion is reasonable. The prediction using the measured wave forces is slightly better than the one using computed wave forces.
- The prediction of the heave motion is good. No large difference between the two predictions is found.
- The prediction of the pitch motion is not good. This was expected since the decay test could also not be reproduced. The simulation that uses the computed wave forces predicts a smaller pitch motion than the simulation that uses the measured wave forces, and comes closer to the measured motion. However, this is because the wave-exciting pitch moment is underestimated in the diffraction calculation near the natural period of pitch (8.9 s), see figure 8.

Figures 20 and 21 show the top tension in the upper and lower riser. The following conclusion can be drawn:

- The variations in the riser tension are predicted well, even when the pitch motions are too large (simulation with measured wave forces). Since the riser-buoy connections are close to the centre of the buoy, pitch motions do not lead to large vertical offsets of the riser.

Finally, some simulations were done with the full-scale mooring system. The same mass distribution of the buoy was assumed to be able to compare the results directly. The motions of the buoy and the forces in the risers are shown in figures 22, 23 and 24. The following observations were made:

- The tension variations in the risers are larger for the full-length system than for the truncated system.
- A simulation without mooring dynamics gives much too low tension variations.
• The effect of the mooring dynamics on the pitch motion is considerable.

SUMMARY AND CONCLUSIONS

A CALM buoy has been tested in the model test basin. The resulting motions and tensions in the mooring lines have been compared with results of numerical simulations. In the simulations, the coupling between the motions of the buoy and the non-linear dynamic motions of the mooring system has been taken into account. The simulations were partially successful with respect to predicting the motions of a buoy in irregular waves. The following observations were made:

• The wave-exciting pitch moments on the buoy are greatly affected by the presence of the skirt. The skirt introduces non-linear viscous effects that can not be accounted for by a linear diffraction code based on potential theory.
• The surge and heave wave forces are far less influenced by the skirt and can be well predicted by a linear diffraction code.
• To predict the tension variations in the risers and the mooring lines it is essential that their behaviour is computed fully dynamically in stead of statically.
• The heave and surge motions of the buoy in waves can be predicted well.
• The pitch motions cannot be predicted well when the viscous effects on the skirt are neglected. Moreover, bending stiffness of the risers will have some influence on the buoy motions. This was not modelled in the simulations.

Future research should therefore focus on the following:
- Near future:
  • Investigate why pitch is not predicted well with simple tools
- Far future:
  • Prediction of viscous skirt forces by Computational Fluid Dynamics. It is not sure whether the separation of wave forces and radiation forces (due to buoy motions) as applied in this paper is valid for the buoy its pitch motions. It may be that the equation of motion of the buoy and its mooring system should be solved by a viscous flow solver in the time domain.
### Table 1: Main particulars of the tested buoy

<table>
<thead>
<tr>
<th>Particular</th>
<th>Unit</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Diameter without skirt</td>
<td>m</td>
<td>20</td>
</tr>
<tr>
<td>Diameter with skirt</td>
<td>m</td>
<td>24</td>
</tr>
<tr>
<td>Position skirt above keel</td>
<td>m</td>
<td>1.25</td>
</tr>
<tr>
<td>Skirt area</td>
<td>m²</td>
<td>90.5</td>
</tr>
<tr>
<td>Mass</td>
<td>tons</td>
<td>1299</td>
</tr>
<tr>
<td>Draft</td>
<td>m</td>
<td>5.9</td>
</tr>
<tr>
<td>Pitch radius of inertia</td>
<td>m</td>
<td>5.7</td>
</tr>
<tr>
<td>Vertical position CoG above keel</td>
<td>m</td>
<td>5.1</td>
</tr>
<tr>
<td>Longitudinal position CoG</td>
<td>m</td>
<td>0.96</td>
</tr>
</tbody>
</table>

### Table 2: Main particulars of the tested risers

<table>
<thead>
<tr>
<th>Particular</th>
<th>Unit</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Length</td>
<td>m</td>
<td>200</td>
</tr>
<tr>
<td>Diameter</td>
<td>m</td>
<td>0.5</td>
</tr>
<tr>
<td>Axial stiffness EA</td>
<td>kN</td>
<td>4.4E6</td>
</tr>
<tr>
<td>Bending stiffness EI</td>
<td>kNm²</td>
<td>1.1E5</td>
</tr>
<tr>
<td>Under water weight</td>
<td>N</td>
<td>1854</td>
</tr>
<tr>
<td>Top angle wrt horizontal</td>
<td>deg</td>
<td>63/68.5</td>
</tr>
</tbody>
</table>

### Figure 1: The buoy before the tests

### Figure 2: The buoy during the tests

### Figure 3: Top view full-scale mooring system

### Figure 4: Static position full-scale risers

### Figure 5: Truncated riser system
Figure 6: Response (amplitude and phase) of wave-exciting surge force

Figure 7: Response (amplitude and phase) of wave-exciting heave force

Figure 8: Response (amplitude and phase) of wave-exciting pitch moment

Figure 9: Wave spectra
Figure 10: Top tension in line 2 with prescribed motion

Figure 11: Top tension in line 5 with prescribed buoy motion

Figure 12: Horizontal top tension in the upper riser with prescribed buoy motion

Figure 13: Horizontal top tension in the lower riser with prescribed buoy motion

Figure 14: Surge decay test with mooring lines and risers

Figure 15: Heave decay test with mooring lines and risers
Figure 16: Pitch decay test with mooring lines and risers

Figure 17: Surge motion in irregular seas, $H_s=2.25$ m $T_p=14$ s.

Figure 18: Heave motion in irregular seas, $H_s=2.25$ m $T_p=14$ s.

Figure 19: Pitch motion in irregular seas, $H_s=2.25$ m $T_p=14$ s.

Figure 20: Top tension in upper riser in irregular seas, $H_s=2.25$ m $T_p=14$ s.

Figure 21: Top tension in lower riser in irregular seas, $H_s=2.25$ m $T_p=14$ s.
REFERENCES


