The Specialist Committee on Deep Water Mooring

Final Report and Recommendations to the 22nd ITTC

1. MEMBERSHIP AND MEETINGS

The membership of the ITTC Specialist Committee on Deep Water Mooring was as follows:

Dr. Christian Aage, Denmark (Chairman)
Prof. Michael M. Bernitsas, USA
Prof. Hang S. Choi, Korea
Mr. Liviu Crudu, Romania
Mr. Kazuo Hirata, Brazil
Prof. Atilla Incecik, UK
Prof. Takeshi Kinoshita, Japan
Mr. Simen Moxnes, Norway (Secretary)
Dr. John J. Murray, Canada (Secretary)

Due to his change of employment Dr. Murray resigned from the Committee in 1997 and was replaced by Prof. Bernitsas. Dr. Murray's duties as Secretary of the Committee were taken over by Mr. Moxnes.

Four Committee meetings have been held during the work period:

- Trondheim, Norway, September 1996, in connection with the 21st ITTC.
- Tokyo, Japan, April 1997, at University of Tokyo after the OMAE'97 Conference.
- Newcastle upon Tyne, England, December 1997, at University of Newcastle. A joint meeting was held with the ITTC Committee on Loads and Responses.
- Galati, Romania, October 1998, at ICEPRONAV S.A. In connection with the meeting a Workshop on Deep Water Mooring and Related Topics in Offshore Engineering was arranged in a co-operation between the University "Dunarea de Jos", ICEPRONAV and the Committee.

2. COMMITTEE TASKS AND CONTENTS OF THE REPORT

In 1996 the 21st ITTC requested the following tasks to be carried out by the Committee:

"Evaluate techniques and recommend procedures for the experimental and numerical simulation of moored vessels in wind, wave and currents."

The number of deep water moored offshore vessels is growing rapidly, and hydrocarbon fields in water depths down to 3000 m are now seriously considered for floating production development. Physical and numerical modelling of such vessels with their extensive mooring lines and risers is a challenge to the ITTC Community. So the preparation of uniform model testing and computational procedures is a highly relevant matter.

The Committee has had to accept that a uniform coverage of the tasks above could not be obtained during the work period. While the numerical simulation of moored vessels has been rather well developed and documented in publications, this has not been the case for the experimental, physical model testing of moored vessels in very deep water.

This state-of-the-art is reflected in the contents of the report. It is the Committee's hope that the work can be continued and completed.

3. DEEP WATER MOORED VESSELS

This section will give a presentation of the different types of vessels and mooring systems that are presently used or planned for use in deep water developments. Moored vessels in this context will normally be floating production units. Exploratory drilling units working in deep waters are generally using dynamic positioning for station keeping.

The work of the ITTC Deep Water Mooring Committee has been focused mainly on methods for prediction of global responses, i.e. floater motions and positioning system loads. Vessels, mooring systems, their interaction, environmental excitation forces and the resulting responses will be discussed.

3.1 Vessels

The following four main types of floating production units, shown in Figure 1, will be considered:

- Monohulls
- Semisubmersibles
- Spar buoys
- Tension leg platforms.

The four concepts above represent different design philosophies. There will of course exist solutions that do not fit directly into any of the listed groups. A thorough discussion of these concepts should, however, cover the most important aspects related to their use as floating production systems.

Common for all floating production units is that they utilize excess buoyancy to support deck payload. They will therefore be weight sensitive to some extent. This has, however, no direct impact on dynamic behaviour and will not be the main focus of the present discussion.

Dependent on the area and the sea state, ocean waves contain 1st order energy in the range 5-25 s. For a floating vessel the natural periods of the different modes of motion are

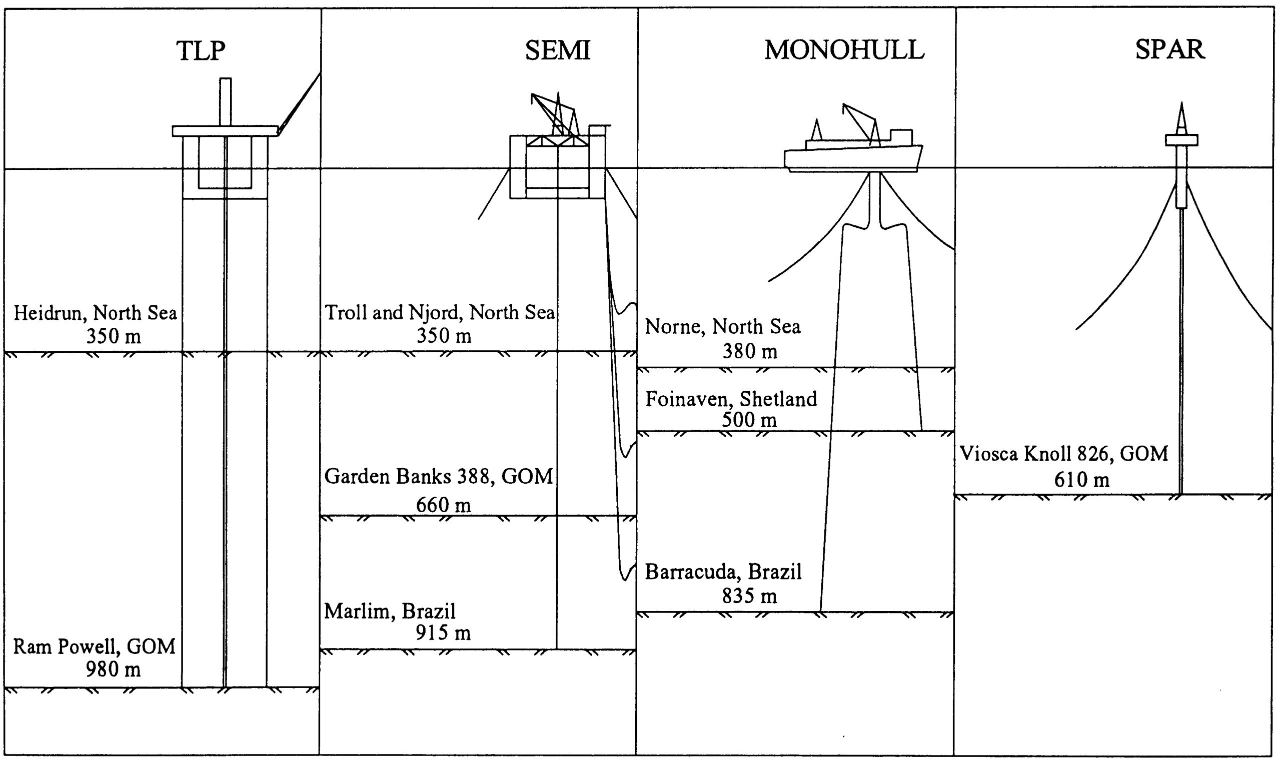


Figure 1 – The four main types of deep water moored vessels used for floating production (Frieze et al., 1997).

therefore of primary interest, and in many ways reflect the design philosophy. Typical natural periods for the different vessels are presented in Table 2 in section 3.6.

A common characteristic of all the floaters is that they are “soft” in the horizontal plane, which implies that they have natural periods in excess of 100 s in surge, sway and yaw. The fundamental differences between the vessels are therefore related to their vertical motions, heave, roll and pitch. This will be discussed further in section 3.6.

Monohulls. The monohull is a ship-shaped structure characterized by a high pay-load capacity. Natural periods of heave, roll and pitch are all within the wave frequency range, leading to relatively large vertical wave frequency motions. This necessitates the use of flexible riser systems. To reduce environmental loads, partly or full weather-vaning is required for permanently moored monohulls.

Semisubmersibles. The semisubmersible consists of three main structural elements, deck, columns and pontoons. Number and shape of columns and pontoons vary with the different designs. Semisubmersibles have small waterplane areas, which give natural periods longer than 20 s, outside the range of 1st order wave forces except for extreme sea states. This implies small vertical motions compared to the monohull. Flexible riser systems are required, however, for this concept as well. The semisubmersible is very weight sensitive, i.e. it has a low flexibility with respect to deck load and oil storage.

Spar buoys. The Spar buoy is a long cylindrical structure floating vertically with a large draft. Very small vertical motions make the use of rigid vertical risers possible. The Spar has a large oil storage capacity.

The Deep Draft Floater has also been proposed as a concept for deep water. It can be described as a semisubmersible with a very deep draft. Its dynamics and payload characteristics are similar to those of the Spar buoy.

Tension leg platforms. The tension leg platform, or the TLP, is a multicolumn structure moored to the seabed by vertical tethers. It is thereby restrained from moving vertically and rigid risers may be used. The TLP is very weight sensitive.

The TLP differs fundamentally from the other vessels. It may be argued whether the TLP is in fact a floating vessel, since it is the tether stiffness and not the waterplane stiffness that governs the vertical motion.

The Sea Star and the Tension Buoyant Tower are other concepts that utilize a vertical mooring system.

3.2 Mooring Systems

Only passive station-keeping systems will be discussed here. The station-keeping systems can be divided into two main groups, compliant and rigid moorings:

- Compliant: Catenary moorings
  Taut moorings

- Rigid: Vertical moorings.

This section will discuss the fundamental physics and dynamic characteristics of the different mooring systems. Various system layouts will be presented briefly at the end.

The primary function of the station-keeping system is to counteract the horizontal environmental forces so that the floating vessel remains within specified position tolerances. At the same time the system must be compliant enough to allow for the wave frequency motion, except for tethers in the vertical direction. The environmental forces act in the horizontal plane, and the resulting horizontal forces must be transferred to the seabed. Since

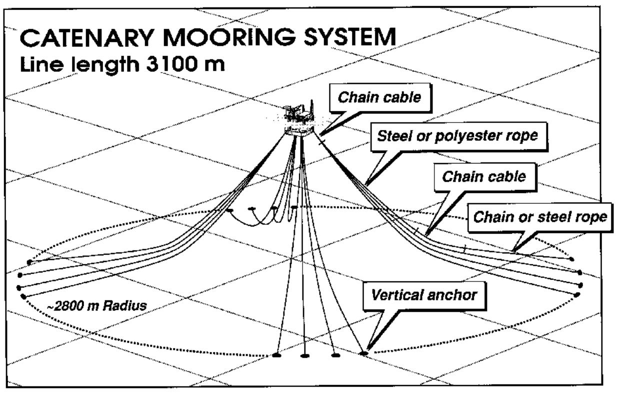


Figure 2 – Components of a catenary mooring system at 1200 m water depth.

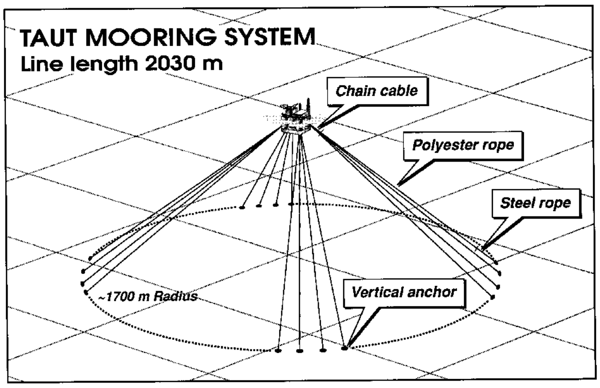


Figure 3 – Components of a taut mooring system at 1200 m water depth.

these forces are acting at different levels, they will introduce a moment M_s that is proportional to the water depth:

$$M_s = F_h\, d \tag{1}$$

where F_h is the horizontal force and d is the water depth.

A mooring line cannot transfer any moment, so the station-keeping moment must be balanced by a vertical force couple. The fundamental difference between catenary and taut moorings is given by the way this station-keeping moment is balanced.

Catenary moorings. As illustrated in Figure 4, the buoyancy-corrected weight of the suspended part of the mooring line must balance the station-keeping moment in a catenary mooring system:

$$M_s = w\, s\, a \tag{2}$$

where w is the buoyancy-corrected weight per unit length, s is the line length, and a is the horizontal distance from the fairlead to the centre of gravity of the suspended line. The vertical force at the top will be equal to the buoyancy-corrected weight of the suspended line.

The compliance to allow for wave frequency motions is ensured by a combination of geometrical and axial elasticity of the lines. The large geometrical variations make these systems susceptible to significant dynamic effects, mainly due to transverse drag forces.

Taut moorings. In a taut mooring system the lines are light compared to the line tension, $w\, s \ll T$, and the lines will be nearly straight between the anchor and the fairlead. In this case the vertical forces are taken up as anchor and vessel reactions directly, as seen in Figure 5. The vertical force F_v is determined by:

$$M_s = F_h\, d = F_v\, a \tag{3}$$

Note that the vertical force will decrease with increasing mooring line length.

For a taut mooring line the compliance to allow for wave frequency motions must be provided entirely by the axial elasticity. Taut mooring systems do not experience large transverse geometric changes to the same extent as catenary systems, and dynamic effects due to drag loads are therefore moderate. To obtain sufficient elasticity and low weight, synthetic ropes are often utilized in taut mooring systems. Such materials exhibit a more complex behaviour than steel, including for example hysteresis, which may give rise to important dynamic effects.

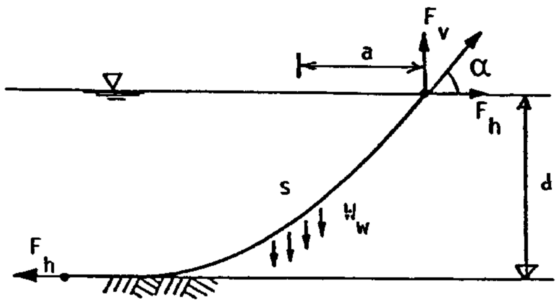


Figure 4 – Force balance in a catenary mooring (Fylling, 1992).

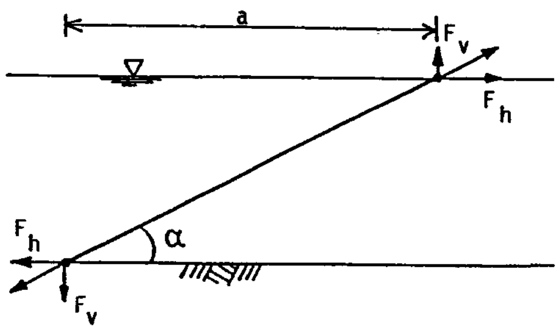


Figure 5 – Force balance in a taut mooring (Fylling, 1992).

Vertical moorings. Apparently a TLP structure could be considered as a taut mooring system with vertical mooring lines. There is however one fundamental difference. TLP tethers are usually made of steel tubes with such massive dimensions that they cannot be considered as being compliant in the axial direction. The TLP system acts as an inverted pendulum. The station-keeping forces are governed by the tether length and the pretension, the latter being typically 20% of the floater displacement weight.

Configurations. The configuration of mooring systems regarding grouping and spread of the lines differs from one concept to the other. There is a tendency, however, to group the lines. This provides more spacing for risers and a better system behaviour in damaged condition. Some typical configurations of catenary and taut mooring systems are illustrated in Figures 2 and 3. TLP tethers will always be vertical in the initial configuration. Any deviation from this will cause strong couplings between horizontal translations and horizontal axis rotations, e.g. surge and pitch.

3.3 Interaction of Vessel/Moorings/Risers

In the discussion of the interaction of the vessel, the mooring system and the risers, focus will be on the global response of the vessel. The interaction effects can be divided into the following groups:

- Stiffness forces
- Damping forces
- Inertia forces
- Mean forces
- Excitation forces.

The most obvious interaction effect is the horizontal stiffness imposed on the vessel by the mooring system, which is the primary function of the mooring system. In a good design the stiffness contribution from the riser system should be relatively small.

An unwanted side effect of the mooring system is the coupling between horizontal displacement and rotation, e.g. between surge and pitch. The vertical components of the mooring line forces introduce heeling and pitching moments. For vessels with small waterplane stiffness and long moment arms, like semisubmersibles, this coupling effect is significant.

The damping contribution from the mooring lines and risers will be significant, especially in deep water. Since low frequency motion is partly resonant, this damping is of great importance. The level of damping generated by a mooring line is controlled by several parameters, for example:

- The effective drag coefficient, dependent on possible velocity induced vibration.
- The top end wave-frequency excitation.
- The mean tension level, given by the pretension and the low-frequency motion.

Damping is the motion parameter where the interaction between different response types is the strongest. For a given mooring configuration the damping effects will generally increase with water depth.

The inertia forces from the mooring and riser system will generally not have a significant influence on the global vessel response. As the water depth increases this could change, but compared to the drag-induced forces the effect is expected to be relatively small.

The presence of current imposes drag forces on the mooring and riser systems, and as the water depth increases, the exposed area and thereby the mean forces will increase. The current drag force is a function of the current velocity, which varies in both magnitude and direction with depth. Ocean currents are varying in time, as well. Assuming perfect correlation between these fluctuations at different layers, the excitation imposed on mooring lines and risers can be significant in deep water. Assuming no correlation, the forces due to fluctuation will tend to level out over the line length.

In deep water the current forces can often become the dominant environmental loads on the mooring system, contributing up to 75% of the mean drift forces on a floating production system.

3.4 Installation and Removal

The installation and removal of deep water structures by use of crane ships with long lifting cables becomes increasingly difficult as the water depth increases. A typical dynamical problem is the possibility of resonance due to the relation between elasticity of the lifting cables and the mass and added mass of the structure. The natural period of the system can coincide with the wave period, and therefore the dynamics of the cable and the dynamics of the crane in operation should be considered in the modelling of the system. Literature about the subject is scarce and published full-scale measurements are lacking. Due to the complex geometry of these structures, the numerical calculation of added mass and damping is not always an easy task. For a composition of plates, pipes with a variety of combination and relative distances and positions that can interfere with each other, the added mass and damping coefficients are difficult to estimate.

3.5 Sources of Excitation

A thorough discussion of excitation and response of floating vessels is outside the scope of this Committee's tasks. The Loads and Responses Committee will cover these aspects in more detail. However, a brief overview of excitation mechanisms will be given in this section.

For a deep water mooring system, all the environmental loads, except the current forces, can be considered as acting at the water surface. Only current has a variation with depth, and current forces may be of significant magnitude all the way to the bottom. This is reflected in the level of discussion. Resulting responses of the different types of vessels as well as the relative importance of the different excitation mechanisms are discussed in section 3.6.

Waves. Wave forces are usually divided into the following categories:

- 1^{st} order forces at wave frequency (WF)

- 2^{nd} order forces
  - mean wave drift forces
  - forces at sum frequencies (HF)
  - forces at difference frequencies (LF)
- Higher order forces
  - wetted surface effects
  - ringing
  - viscous (non-potential) drift forces.

Wind. Wind forces may be divided into the following three categories:

- Mean wind loads
- Fluctuating wind loads, due to fluctuations or gusts in the wind field
- Vortex induced vibrations (VIV), due to structure/wind interference.

Current. As for the wind, current forces may be divided into the following three categories:

- Mean current loads
- Fluctuating current loads
- Vortex induced vibrations.

The nature of vortex induced vibrations is actually a strong coupling between excitation and response. It will mainly affect mooring lines and risers. The primary effect with respect to global vessel response is the increased line and riser drag due to transverse oscillations. This will increase both the mean forces and the dynamic forces.

The current velocity vector varies in both time and space. Considering the large volume occupied by a deep water moored structure, maybe several km^3, the complete description of a current field for a specific structure is very complicated. Ideally it should comprise the following information for a volume grid with sufficient resolution:

- Mean velocity and direction
- Depth dependent velocity profile
- Time dependent velocity profile
- Direction and depth dependent fluctuation spectrum
- Information on spatial correlation.

Environmental specifications do not usually comprise such information. A typical specification defines a unidirectional depth dependent constant-velocity profile with a given return period.

Ice. Structures in arctic waters will be subjected to ice. The ice loads will act directly on the vessel and produce a mean drift force in the drifting direction of the ice field. Any fluctuations will most likely have very long periods.

3.6 Resulting Responses

The 1st order wave forces will be the dominant dynamic loads, orders of magnitude larger than any other dynamic loads. The global behaviour of offshore structures may therefore be classified by the motion characteristics of each rigid body mode of motion under wave frequency excitation. Modes having a natural period below the wave periods are usually denoted as restrained, while modes with a natural period above the wave periods are denoted as free.

Table 1 - Modes of motion of deep water moored vessels

Vessel	Modes of motion: Free (F) or Restrained (R)					
	Surge	Sway	Heave	Roll	Pitch	Yaw
Ship	F	F	F	F	F	F
Semi	F	F	F	F	F	F
Spar	F	F	F	F	F	F
TLP	F	F	R	R	R	F

Table 2 - Natural periods of deep water moored vessels

Vessel	Natural periods (s)					
	Surge	Sway	Heave	Roll	Pitch	Yaw
Ship	> 100	> 100	5-12	5-30	5-12	> 100
Semi	> 100	> 100	20-50	30-60	30-60	> 100
Spar	> 100	> 100	20-50	50-100	50-100	> 100
TLP	> 100	> 100	< 5	< 5	< 5	> 100

Table 1 summarizes the global behaviour of the four main types of deep water moored vessels, while Table 2 lists the typical natural periods of their six modes of motion. The discussion will focus on:

- Motions: Mean offset, WF and LF
- Mooring system forces:
  Mean, WF, LF and HF (for TLPs)
- Depth sensitivity.

The motions of importance to a floating production system, moored in deep water, are:

- Horizontal translation
- Horizontal rotation
- Vertical translation
- Vertical rotation.

The horizontal translation and rotation (surge, sway and yaw) must be limited due to the capabilities of the riser system. The maximum allowable offset will typically be 10% of the water depth.

The vertical translation (heave) determines whether rigid risers may be used or not. The use of rigid risers is often desirable since it makes the use of dry wellheads possible.

Vertical rotation (roll and pitch) is a limiting factor in the function of the on board process equipment.

Depth sensitivity is mainly connected with the horizontal motions, whereas the vertical motions are almost independent of the water depth. Figure 6 shows the relative magnitudes of the different components of horizontal motions for typical moored structures.

The following observations regarding depth sensitivity are more or less valid for all moored structures. Except for very shallow water the wave frequency motion can be considered as being independent of the water depth. Mean offset and low frequency motions will tend to increase with increasing water depth for a given mooring configuration. This is due to decreasing horizontal stiffness of the mooring system. Mean offset and low frequency motions thereby tend to be more and more important for the extreme offset as the water depth increases.

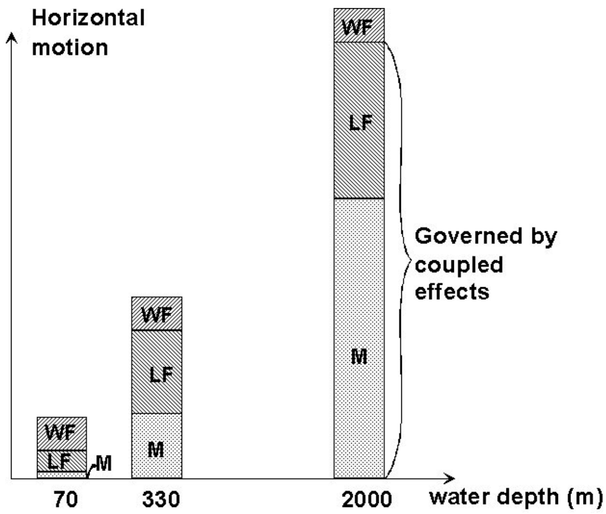


Figure 6 - Horizontal motions of moored structures at different water depths, composed of wave frequency motions (WF), low frequency motions (LF), and mean offset (M).

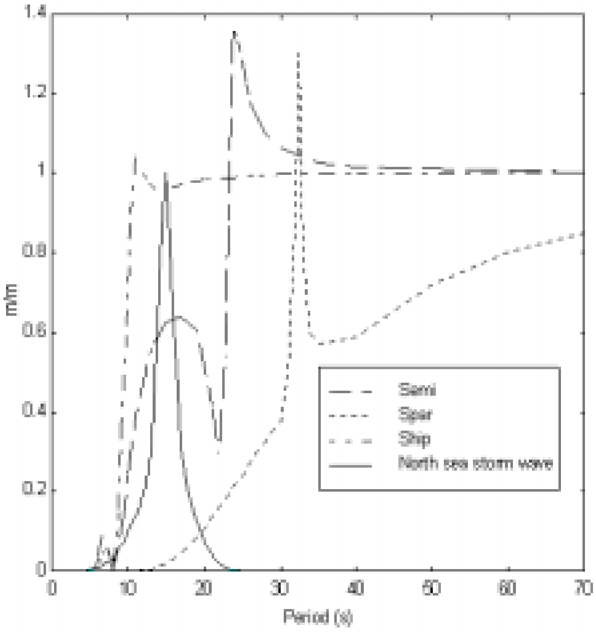


Figure 7 – Heave transfer functions of a semi, spar and ship with a North Sea storm wave spectrum.

Monohulls. Due to the large superstructures of the floating production units, and their active or passive weather-vaning ability, the wind forces will often be dominant relative to the current forces, at least for shipshaped floaters.

The natural periods of all vertical modes of motion of a monohull are in the 1st order wave frequency range. This implies significant wave frequency motions, as indicated in Figure 7.

Monohulls will experience significant LF response in the horizontal plane only. Ship-shaped floaters may be particularly sensitive to surge excitation since the viscous hull damping is very low. This sensitivity is reduced with increasing water depth since the damping contribution from mooring lines and risers increases.

Fishtailing, an unstable coupled yaw and sway motion excited by wind and current, is a particular challenge in the design of monohull moored vessels. The horizontal stiffness of the mooring system is a governing parameter, and fishtailing may thereby be a growing problem with increasing water depth.

For catenary systems the wave frequency motions will introduce dynamic mooring forces that will tend to increase in deep water due to increased transverse drag forces. Taut mooring systems are not subjected to the same level of transverse motions, and they will thereby act more quasi-statically. Dynamic forces will tend to decrease with increasing water depth for such systems, since the elastic length of the mooring lines increases.

Semisubmersibles. Compared to the ship-shaped floaters the current forces will be larger on semisubmersibles due to the bluff shapes of their underwater columns and pontoons. Wind loads will still dominate the mean drift forces, except in calm areas with strong currents.

The semisubmersible is characterized by having free modes of motion only, which means that all natural periods are above the range of natural wave periods. Despite this fact, the wave frequency motions are not insignificant, especially in extreme conditions, as indicated in Figure 7.

Large semisubmersibles at 100,000 t displacement or more are naturally less sensitive to WF action, and for such structures the LF response may dominate the roll and pitch motions.

Catenary moored semisubmersibles may also experience significant dynamic mooring forces due to WF response. The discussion performed for the monohulls is also valid here.

Spar buoys. With a typical draft of 200 m the spar buoy has a very large area exposed to current forces and with a cylindrical shape leading to separated flow, the current force is usually the dominant mean drift force on a spar buoy. Low-frequency vortex induced vibrations may increase the effective drag leading to even higher mean current forces.

The spar buoy is characterized by having free modes of motion, only. The natural heave period is well above the range of 1st order wave periods. In addition, the spar buoy has a low level of vertical wave excitation due to its large draft, which exploits the fact that the 1st order wave motions and dynamic pressures decay exponentially with depth. This results in very small heave motions, as seen in Figure 7, which makes the use of rigid vertical risers possible.

Low frequency motions will dominate the response with respect to horizontal translations as well as horizontal and vertical rotations. Current fluctuation may be a significant motion excitation force on a spar buoy. Depth correlation is a central issue when determining the level of such excitation.

Due to very low wave frequency motion, the spar buoy is generally not subjected to large dynamic mooring line forces.

Tension leg platforms. The TLP is basically free in the horizontal plane (surge, sway and yaw), but restrained in the vertical plane (heave, roll and pitch).

The TLP will thereby experience wave frequency motions in the horizontal plane that are of the same order of magnitude as those of a semisubmersible of comparable size. In the vertical plane, however, the TLP will behave as a fixed structure with practically no wave frequency motion response. Wave frequency forces are directly compensated by the stiffness of the tether system.

Higher order wave forces at different sum-frequencies may introduce resonant (springing) or transient (ringing) responses in the vertical modes. These effects may give significant contributions to the tether loads.

Due to the tether system the TLP will move along a spherical surface. This gives rise to the set-down effect, which is a kinematic coupling between the horizontal surge and sway motions and the vertical heave motion. Set-down is of importance to the wave airgap.

4. NUMERICAL MODELS

Numerical modelling is getting increasingly important for verification of moored systems as the water depth increases. This because no physical model test facilities can accommodate all water depths and mooring line spreads that are being developed today. Even if large enough model basins do exist, the designer will usually benefit from a combined approach, using both numerical and physical modelling. Furthermore, a theoretical understanding of the behaviour of the complex vessel/mooring/riser system is necessary for the development of hybrid model testing methods that can take care of the inevitable truncations of the mooring systems in the model basin.

For these reasons, numerical approaches are continuously progressing, in this area as in many others. Numerical modelling of deep water mooring systems poses large demands on computer power, and many numerical problems still need to be solved.

4.1 Dimensional analysis

It is not easy to describe the overall behaviour of mooring systems connected to offshore structures, because the behaviour is affected by diverse factors that are specific to the system under consideration. A convenient way to understand the underlying mechanism may be to use dimensional analysis. It is well-known that the results of this method are not unique but manifold, depending on the aim of the analysis and also on the choice of reference parameters. For example, Webster (1995) obtained the dimensional relation for the tension of a uniform mooring line, which is undergoing imposed sinusoidal motions at the top in association with mooring damping, as follows:

$$\frac{T}{wH} = f\begin{bmatrix} \frac{t}{\tau}, \frac{l}{H}, \frac{T_0}{wH}, \frac{a}{H}, C_d\frac{D_h}{D_s}, C_m\frac{A_h}{A_s}, \frac{I_s}{A_s^2}, \\ \frac{\tau}{2\pi}\sqrt{\frac{g}{H}}, \rho_f\frac{g}{w}, \frac{E_s A_s}{wl}, \frac{\sqrt{A_s}}{H}, \frac{\rho_f}{2}\frac{U_c^2}{wD_s} \end{bmatrix} \quad (4)$$

Since the meaning and role of these non-dimensional parameters are comprehensively explained in Webster (1995), we cite herein only those parameters related to the water depth H. The geometric parameter l/H is the ratio of the mooring length to the water depth called the "scope" of the mooring line. The parameter T_0/wH represents the static pretension of the mooring line divided by the buoyancy compensated weight of a length of mooring line equal to the water depth. These two parameters govern the geometry of the mooring line when no motions are imposed.

Furthermore a/H denotes the motion amplitude relative to the water depth, $\frac{\tau}{2\pi}\sqrt{\frac{g}{H}}$ the ratio of the period of the excitation to that of a pendulum of length H, whereas the characteristic cross-sectional dimension relative to the water depth is given by $\sqrt{A_s}/H$. In the case of deep water, it is obvious that these three parameters tend to zero and lose their effective roles.

On the other hand, the dynamic parameter E_sA_s/wl is indirectly related to the water depth corresponding to the relative stiffness of the mooring line. It is recognized that the dynamic tension of the mooring line is significantly influenced by this parameter, and that it normally increases as the water becomes deeper. Hence, the relative stiffness plays a leading role in deep water, as found by Papazoglou et al. (1990 a). This is particularly the case for taut mooring systems, which are being increasingly deployed in deep water. Taut mooring systems are characterized by having very large values of T_0/wH and E_sA_s/wl.

4.2 Mathematical modelling

The behaviour of a mooring system is directly influenced by the floater at the top, with equations of motion as expressed by Lee et al. (1998):

$$\begin{aligned}\boldsymbol{M}\dot{\boldsymbol{v}} + \boldsymbol{C}_{RB}(\boldsymbol{v})\boldsymbol{v} + \boldsymbol{C}_A(\boldsymbol{v}_r)\boldsymbol{v}_r &= \boldsymbol{F}(t)_{memory} \\ &+ \boldsymbol{F}(t)_{viscous} + \boldsymbol{F}(t)_{wind} + \boldsymbol{F}(t)_{current} \\ &+ \boldsymbol{F}(t)_{wave} + \boldsymbol{F}(t)_{mooring} + \boldsymbol{F}(t)_{thrust}\end{aligned}$$

with $\quad \dot{\boldsymbol{\eta}} = \boldsymbol{J}(\boldsymbol{\eta})\boldsymbol{v}$ (5)

where $\boldsymbol{M}$ is the mass matrix including added masses, while $\boldsymbol{C}_{RB}(\boldsymbol{v})\boldsymbol{v}$ and $\boldsymbol{C}_A(\boldsymbol{v}_r)\boldsymbol{v}_r$ are the Coriolis force and the centripetal force, respectively. The velocity $\boldsymbol{v}$ is referred to the body-fixed coordinates, which are transformed from the earth-fixed coordinates through the relation given above with the help of the rotation matrix $\boldsymbol{J}(\boldsymbol{\eta})$. The relative velocity vector is denoted by $\boldsymbol{v}_r$.

External forces consist of the wave radiation force, the viscous damping, the wind force, the current force, the wave exciting force, the mooring force and the thruster force, if the mooring system is assisted by a dynamic positioning system. The wave radiation force contains time memory effects, which can be estimated most conveniently from the wave damping, as explained in Choi et al. (1994).

The horizontal floater motions can be divided into the following three components classified by their time scale:

- Mean offset resulting from the steady current force, mean wind and mean wave drift forces.
- Low frequency offset resulting from the slowly-varying wind and wave drift forces.
- Wave frequency motion in direct response to the waves.

The mean offset of a moored system is determined statically. Generally speaking, the low frequency offset can also be predicted by the same method as a sequence of steady offsets because of the long periods. However, the wave frequency excursion can only be appropriately predicted by a dynamic approach.

In accordance with these motions, the mooring line displays quasi-static and/or dynamic behaviour, which in turn affects the floater in a coupled fashion. Depending on the assumptions and simplifications made, different methods have been developed. A variety of references are available dealing with different kinds of floaters and cables, combined or separate. In this section, we confine our discussion to numerical models of the mooring line systems.

The first step in the numerical modelling is to model the mooring line as either a continuous line or a lumped-mass-spring system. The statics and dynamics of cables are classical subjects treated in several books and papers. Irvine (1981) gives a broad overview and a historical perspective. More recently, Triantafyllou (1994) and Howell (1991) treated the continuous cable without considering bending stiffness. Garrett (1982) derived the flexible slender structure model by using beam or rod elements including the bending stiffness. Paulling and Webster (1986) expanded this theory to include the stretch of the cable together with various loads acting on it. The more popular lumped-mass-spring model has several advantages in terms of straightforwardness, economy and versatility at the expense of accuracy. Among many others, Huang (1994) developed a numerical method for dynamic analysis of marine cables. Huang (1992) and Chucheepsakul et al. (1995) derived a variational formulation for marine cables based on the work-energy principle. The next question is to which extent the material and strength properties are included, such as axial, bending and torsional stiffness, structural damping, linear or non-linear elasticity, uniformity and homogeneity of the line. Depending on this, a class of different methods is developed.

4.3 Quasi-static Modelling

Neglecting the dynamic excitation due to waves, the variation of current in time and the vortex-induced vibration, mooring lines may be treated statically. A number of different approaches have been undertaken to solve the problem. The main issue hereby is the dimensionality, the elasticity and the current drag, as seen in Huang and Vassalos (1993).

Mooring lines basically show a three-dimensional shape, but the assumption of co-planar configuration reduces the complexity of geometric descriptions. The static solution is often insensitive to elastic deformations and thus the cable may be modelled as inelastic in many practical situations. The governing equations of an inelastic cable are of course simpler than those of an elastic cable. The current drag acting on the cable is proportional to the square of the relative velocity, and hence it is non-linear.

The principal components act in the normal directions both in the plane and out of the plane. The latter components are caused by vortex shedding and invoke the need to treat the problem dynamically in three dimensions. The simplest model is to assume an inelastic co-planar cable without external forces, which leads to the classical catenary equation.

Most mooring chains may be well approximated by the catenary equation. But the traditional catenary equation is cumbersome for evaluating the force-deflection relation, because it requires a number of intermediate steps. Flory (1997) suggested a new form of the catenary equation to overcome this difficulty. By using a co-ordinate system based on the undeflected position of the cable, the catenary top excursion can be directly calculated when the external force is imposed.

Huang and Vassalos (1993) developed a semi-analytical method, which predicts the static behavior of a three-dimensional cable under a given distribution of point loads, for example due to current. They derived an exact solution as a function of the internal force vector at the end point along the cable. Since the internal force is determined as the solution to the problem, this scheme must be implemented iteratively.

Huang (1992) and Chucheepsakul et al. (1995) analyzed the quasi-static behavior of marine cables based on a hybrid formulation, in which the variational principle is applied for the virtual horizontal displacement coupled with an equilibrium equation in the tangential direction. With this formulation, they investigated the effect of axial deformations on the configuration and tension of the cable and found that, the displacement and strained arc length increase with decreasing extensibility in the case of given top tension, while the tension variation decreases. Chucheepsakul et al. (1996) reformulated the problem in polar co-ordinates in order to circumvent the limitation involved with the use of rectangular co-ordinates that may cause problems when the slope at the bottom end of the cable becomes very small.

4.4 Dynamic Modelling

It is generally believed that the maximum attainable value of the dynamic tension in deep water happens within the wave frequency range, and thus is invoked by linear motions of a floater at the top. Therefore, the need to incorporate a dynamic approach for mooring applications in deep water has been addressed in several works, e.g. by Bergdahl and Rask (1987) and Fylling et al. (1987). This work has eventually led to a modification of the design rules and guidelines of various organizations (DNV, 1989 and API, 1995). Based on the conclusions drawn by a task group on this topic, API (1995) strongly recommended using a dynamic analysis for permanent mooring systems in deep water. The task group undertook a parameter study to compare predicted line tensions obtained from quasi-static and dynamic analyses for various conditions.

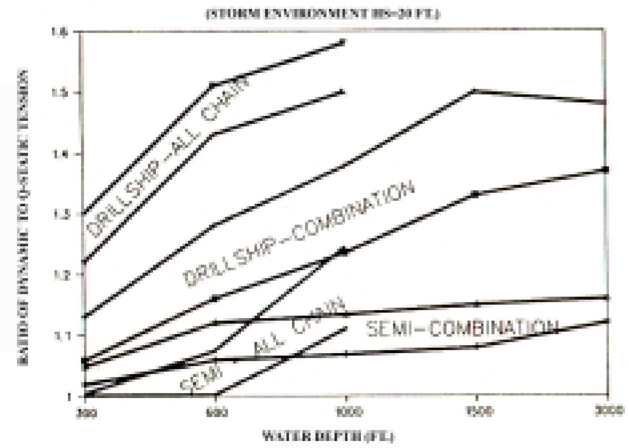


Figure 8 - Parameter study of dynamic relative to static line tensions for a drillship and a semisubmersible with all-chain and combination moorings (Kwan, 1991).

Figure 8 from Kwan (1991) shows the maximum tension from the dynamic analysis divided by the maximum tension from the quasi-static analysis for a drillship and a semi-submersible, respectively, connected to an all-chain or to a combination mooring system in a storm environment. In this figure, the largest and smallest ratios in three wave headings have been selected. The results indicate that a drill ship with an all-chain mooring system is the most dynamically excited system, and that its tensions increase by 20-60%. The combination mooring system is amplified by 10-50% for a drillship. It should be noted that the dynamic amplification is much less for the semisubmersible, due to its inherent damping.

4.5 Non-linear Time Domain Analysis

The dynamics of the mooring cable is normally involved with non-linear loadings and interactions among slowly varying drift motions, fast varying wave induced motions, and vortex induced vibrations, which makes the dynamic analysis a difficult task. There are four primary non-linear effects that have an important influence on the mooring line behaviour:

- Non-linear stretching of the line. The longitudinal stiffness of the line is a function of the tension level.

- Change in over-all geometry associated with the shape deformation of the line.
- Fluid loading proportional to the square of the relative velocity.

- Bottom effect. The interaction between the line and the seafloor is not fully clarified.

The bottom effect has been investigated by Wung et al. (1995), who developed a numerical tool for the anchor-chain-soil interaction and carried out centrifuge tests. They concluded that a significant amount of energy is dissipated through the embedded mooring line.

Chatjigeorgiou and Mavrakos (1997) investigated the non-linear effect on mooring tensions at the top including bending effects and time variation of the strain along the mooring line. They compared the numerical results with experimental data as well as with those obtained from simplified methods, in which the dynamic tension was assumed to be constant or its variation along the line was neglected. They found that the fully non-linear model more closely predicts the experimental data. It was also found that contributions arising from time differentiation of the dynamic quantities in the compatibility relations might be significant in predicting the dynamic behaviour of an elastic cable even for small excitation amplitudes.

It is well-known that the dynamic analysis can be made either in the time domain or in the frequency domain. All non-linear terms are properly accommodated in the time domain analysis, whereas they must be more or less approximated by equivalently linearized ones in the frequency domain. Mavrakos et al. (1989) compared experimental results with numerical predictions based on both time and frequency domain analyses for deep water mooring lines with submerged buoys. They found that the experimental data showed good correlation with numerical predictions, particularly with those based on time domain computation. They argued that discrepancies between theory and experiment originate mainly from inaccurate treatment of the interaction between cable and bottom.

Ran et al. (1998) calculated non-linear coupled responses of a moored spar in random waves with and without current both in time and frequency domains. The mooring dynamics were solved based on a generalized coordinate finite element method. In the time domain analysis, Morison's equation was used to estimate mooring line drag force, which acts both as wave excitation force and as viscous damping. In the frequency domain analysis the non-linear forces were stochastically linearized. Comparisons show that the time-domain analysis produces larger wave-frequency and slowly varying responses and also higher mooring tensions, except for wave-frequency top tension, than the frequency-domain analysis. They postulate that it is because the viscous damping is likely to be overestimated by stochastic linearization in the frequency-domain analysis. They also found that the heave response and top tension increase in the presence of current.

Chucheepsakul et al. (1995) investigated the effect of axial deformation on the natural frequencies of the in-plane vibration of marine cables. They used the Galerkin finite element method to obtain the mass and stiffness matrices and then solved the eigenvalue problem. They found that the natural frequencies increase with an increase in the top tension as well as elastic modulus. Newberry and Perkins (1996, 1997) propose a new mechanism for the dynamic tension as a consequence of the coupling between lateral and tangential deformations.

The stability of a non-linear system may be examined by bifurcation theory to produce catastrophe sets in the design space defining regions of qualitatively different system dynamics. Catastrophe sets have been derived numerically by system search for bifurcation without waves by Chung and Bernitsas (1992) and with waves by Bernitsas and Kim (1998). Here it is shown that resonance with the natural frequencies of a mooring system represents only one of the mechanisms that are responsible for large amplitude slow motions of a spread mooring system. The authors indicate a variety of bifurcations that are caused by mean or slowly varying drift forces and suggest the appropriate design criteria. In Bernitsas et al. (1999) a review of mooring design based on catastrophes of slow dynamics is presented.

4.6 Coupled or Uncoupled Models

The low frequency motion of a floater in deep water is significantly influenced by the current load and the damping of slender members like mooring lines and risers. Fully coupled floater/mooring/riser analysis yielding a consistent representation of these coupling effects as described by Ormberg et al. (1997) will require a huge amount of computational power. In order to gain computational efficiency and flexibility, alternative analysis strategies are proposed by Ormberg et al. (1998), using a combination of uncoupled and coupled analysis. They examined three alternatives to the fully coupled system analysis and applied them on a turret-moored tanker with a taut mooring system. Based on the comparison of these results with those obtained by a fully coupled analysis, they concluded that a coupled vessel motion analysis could be a practical alternative to the fully coupled system analysis. A simplified model of the slender structures, moorings and risers, is still catching the main coupling effects in terms of restoring force and damping. Slender structure responses are then evaluated in detail in subsequent slender structure analyses, where critically loaded mooring lines and risers are analysed one by one considering vessel motions as forced support displacements.

4.7 Mooring Damping

As the water depth increases, the damping induced by the mooring lines increases relatively to other sources of damping, affecting the motion response of the vessel considerably. Thus an accurate estimate of mooring induced damping is critical to a realistic simulation of deep water moored vessels. Huse and Matsumoto (1988, 1989) and Huse (1991) have treated this problem by means of the dissipated energy model with an iterative use of mooring equations.

Wichers and Huijsmans (1990) extended the model to a finite element method in order to include the dynamic effects of cables. A direct time-domain simulation was performed by Dercksen et al. (1992).

Webster (1995) carried out a systematic parameter study based on a non-linear dynamic simulation, where both the cross-flow drag and the internal damping of the mooring line were considered. He found that the damping depends strongly on transverse motions and therefore increases drastically with the stiffness of the mooring line, because a stiff cable is more prone to undergo large transverse oscillations.

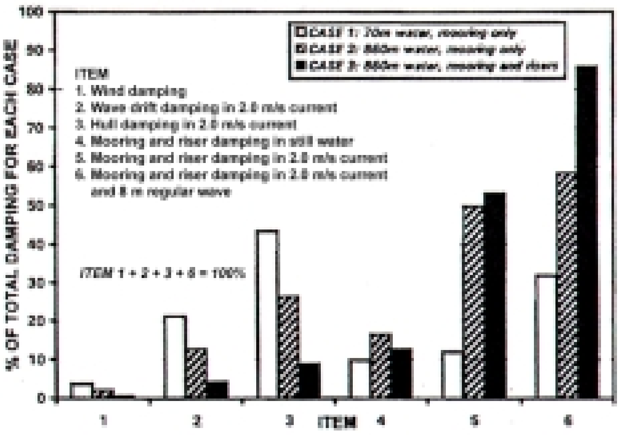


Figure 9 – Low frequency damping from tanker, mooring lines and risers (Hwang, 1998).

Hwang (1998) simulated damped oscillations of a surging tanker in still water, in current, and in current with regular and irregular waves, similarly to decay tests in a model basin to evaluate the mooring damping. Figure 9 shows the contributions of tanker, mooring lines and risers on low frequency damping for water depths of 70 m and 860 m. It is observed that the mooring line damping constitutes 32% of the total low frequency damping in the 70 m shallow water case. This contribution increases to about 58% in the 860 m deep water case.

4.8 Validation

Numerical methods should be verified by a demonstration that the original equations can be recovered from their discretized version, as the grid spacing tends to zero. For linear initial-value problems, stability is the necessary and sufficient condition for convergence in the light of the Lax equivalence theorem. For non-linear problems, however, it is hardly possible to prove the stability and convergence of a numerical method in an analytical form. Therefore, a convergence test is usually conducted by repeated computation on a pre-designed set of different grid systems. For the mooring analysis in deep water, such a systematic approach has not yet been found in the open literature.

On the other hand, there are ample works, in which numerical results are compared with experiment. For example, Papazoglou et al. (1990 b) suggested a scaling procedure for mooring experiments. In order to satisfy the dynamic similitude, they introduced elastic springs in the mooring line model. The procedure was validated by comparison of full-scale and scaled-down numerical results with experimental measurements. Figure 10 shows the dynamic tension amplification at the top of the mooring line and directly below the attached lower buoy. As observed in the figure, the numerical results show good agreement with experiments.

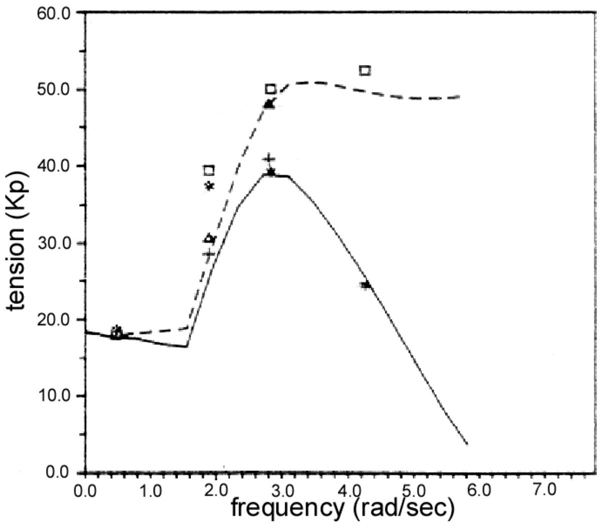


Figure 10 - Dynamic amplification of line tension at the top of the mooring line and at a submerged buoy (Papazoglou et al., 1990 b). Legend:
—— , – – – *numerical results, frequency domain*
+ , □ *numerical results, time domain*
* , ▲ *experimental results.*

4.9 Future Work

As pointed out by Leite and Fernandes (1998), conventional catenary mooring systems are becoming impractical in deep water. Taut mooring systems with polyester synthetic ropes are regarded as an effective alternative and have been deployed more frequently in recent years. Further studies are needed of the dynamic behaviour of this system and of the mechanical behaviour of synthetic ropes under severe environmental conditions (Fernandes et al., 1998).

Turret moorings assisted by dynamic positioning systems are widely employed for drill-ships and FPSOs, so it is strongly recommended to investigate the combined effects of mooring lines and thrusters, as done by Kim et al. (1997), Strander et al. (1997), and Nakamura et al. (1994).

The dynamic behaviour of the embedded mooring line on the sea floor is a long-standing research topic. A description of in-plane and out-of-plane cable motions at the seabed caused by vortex shedding has been made by Pesce et al. (1997). Further investigation into this important subject is recommended.

5. PHYSICAL MODELS

Experimental testing by means of physical scale models in a model basin has been the traditional way of investigating the behaviour of moored offshore vessels for nearly fifty years. Model testing of complete systems is recognized as the most reliable method for verification of floating offshore structures with respect to global responses such as vessel motions and mooring loads. Physical models have the advantage, compared to numerical models, that they to some degree can bridge a gap of incomplete knowledge of the many factors influencing the behaviour of these complex systems. As at sea, all laws of nature are obeyed in the model basin, albeit only in model scale.

The scale effects, that are an unavoidable problem in any physical model test, play a role in model testing of deep water moored vessels as well. But with different weighting of the dominant physical factors, model testing of deep water moored structures makes other demands on test facilities and scale ratios than model testing of floating structures in more moderate water depths. In addition to the correct water depth, a deep water model basin must have large horizontal dimensions, at least five times the water depth in any direction, if full models of catenary mooring systems are to be accommodated. As in any modern seakeeping laboratory, facilities for generating multidirectional waves and wind all over the basin are required. A system for accurate generation of uniform and non-uniform current over the whole water column is an important requirement for deep sea model test facilities.

Physical models of deep water moored systems require very large model basins. Even with a scale ratio of 1:100, which is traditionally considered very small and indeed unacceptable to many model basins, modelling of a full-scale water depth of 3000 m does require a 30 m deep model basin. At MARIN in Wageningen, the Netherlands, a new offshore basin is under construction with a maximum water depth of 30 m in the central pit (Buchner et al., 1999), which will allow model testing of vertical mooring systems (TLPs) at these extreme water depths. But spread mooring systems have a footprint of about five times the water depth, in this case 150 m in all directions. No existing or planned model basin can accommodate a spread mooring system at this water depth with traditional scale ratios.

Three different approaches have been pursued to circumvent this problem. One approach is to do the model testing in large "natural" model basins such as fjords, lakes and rivers. A second approach is to use extremely small scale ratios. A third approach is to truncate the mooring lines and risers at the model basin walls and bottom and then simulate the truncated parts by passive or active mechanisms, the so-called hybrid method.

5.1 Natural Model Basins

Over the years, several laboratories have been using fjords, lakes and rivers as "natural" model basins for special research projects. Very large test dimensions can be achieved by this approach. But due to the non-controllability of the environmental test conditions, waves, wind and current, such facilities cannot be used on a routine basis.

5.2 Ultra Small Scale Model Testing

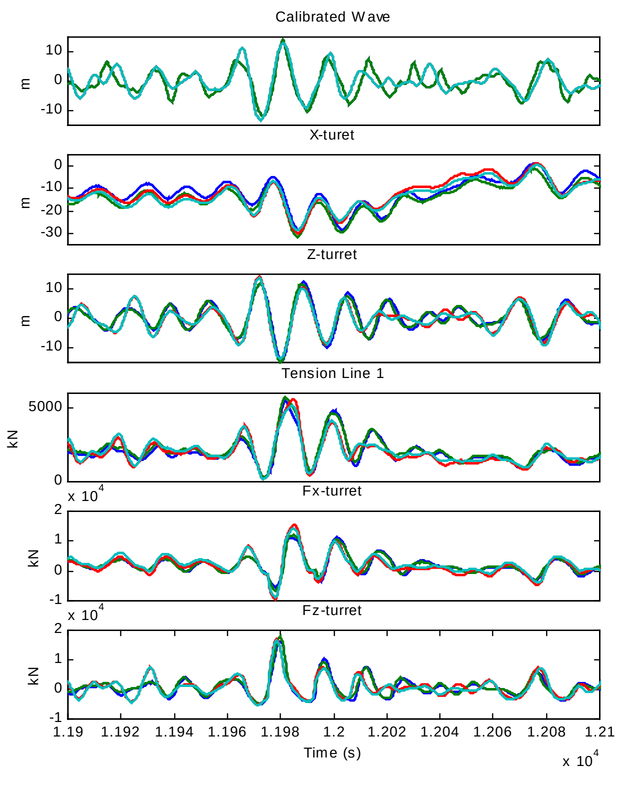
At MARINTEK in Trondheim, Norway, a comprehensive and very interesting test series has been carried out to investigate the feasibility of model testing at ultra small scale ratios (Moxnes and Larsen, 1998).

The Ocean Laboratory model basin at MARINTEK measures 80 m by 50 m with a maximum water depth of 10 m. These dimensions allow for a full modelling of a catenary mooring system up to 1000 m water depth with traditional scale ratios. To extend this limit by the use of ultra small model scales, a comparative model test series was carried out with two models of the same floating production unit

(FPSO) at the model scales of 1:55 and 1:170. Scale ratios around 1:55 have been used for many years in model testing of moored offshore vessels, and their reliability in relation to full scale is well established, which makes the large model a suitable benchmark object.

The FPSO models were turret-moored at a full-scale water depth of 385 m. The practical problems related to constructing an FPSO model with turret and mooring lines at scale 1:170 were considerable. At this scale ratio, 1 tonne full scale becomes 0.2 grammes in model scale, and extreme care is needed in the building, instrumentation and ballasting of the model.

The two models were exposed to identical wave, wind and current conditions. Great care was taken to make the input wave time series identical in both scales, which was actually achieved to a high degree, as seen in Figure 11, where four independent realisations of the



same test are displayed, two at each scale ratio.

Figure 11 - Four independent realisations of the same FPSO model test, two at scale 1:55 and two at scale 1:170. The four time series are almost identical for waves as well as responses. (Moxnes and Larsen, 1998).

The results with respect to global behaviour, vessel motions, mooring line tensions and turret forces are almost identical. It should be underlined, however, that scale effects on the viscous forces and damping may still be significant, but obviously without much influence on the global behaviour for this type of structure.

It can be concluded that, model testing of moored floating offshore vessels in waves can be carried out successfully at a scale ratio of 1:170 with results very similar to those obtained at scale ratio 1:55. Practical problems rather than scale effects seem to be the limiting factor for this type of model testing, but a scale ratio of 1:170 is indeed very close to the practical limit with today's model basin technology.

5.3 Hybrid Model Testing

When even the most extreme scale ratios do not allow the modelling of complete deep water mooring and riser systems, hybrid model testing is an interesting option. The hybrid method takes its name from its combination of physical and numerical models. Hybrid modelling is being developed at several institutions, but few results have been published yet.

Hybrid model testing normally means a combination of physical and numerical models that are working on-line during the model test. The truncated parts of the mooring lines and risers are simulated by mechanisms at the sides and bottom of the model basin. The mechanisms can be either passive or active. A variant of the method is the numerical reconstruction method, where the model test time-history is reconstructed after the test in a numerical simulation model that comprises the parts missing in the physical model.

While hybrid methods are usually introduced because of the limited model basin dimensions, the method also offers an interesting solution to the well-known scaling law conflict. If the parts of the mooring system that are most troubled by scale effects are replaced by actuators coupled to on-line computational models with full-scale properties, the complete system can to some extent be free of scale effects.

Watts (1999) has discussed the principles of hybrid hydrodynamic modelling. Some promising results from an on-going joint industry project at Haslar are presented. It is concluded that hybrid hydrodynamic modelling presents the only viable alternative to the construction of ever deeper, more expensive wave basins.

Buchner et al. (1999) have described the development of passive and active hybrid model mooring systems, presently going on in connection with the construction of the new Offshore Basin at MARIN. The basin measures 45 m by 36 m with an overall depth of 10,5 m. It has a central pit of 30 m. The proposed active hybrid system is seen in Figure 12.

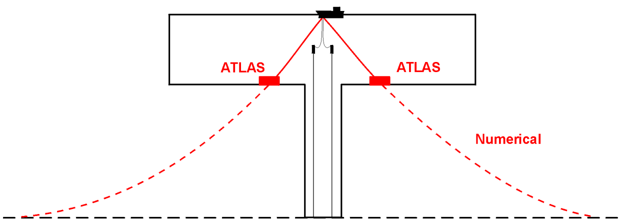


Figure 12 - The active hybrid mooring system being developed at MARIN (Buchner et al., 1999).

Passive Hybrid Systems are mechanisms that simulate the truncated parts of the mooring lines and risers by a system of springs, masses and mechanisms connected to the floater. Clauss and Vannahme (1999) have described a passive cam-controlled model testing mechanism that can simulate arbitrary non-linearities of the truncated mooring lines. Typically, the horizontal mooring stiffness and thereby the low-frequency motions of the vessel can be modelled quite well by passive hybrid systems, whereas mooring damping and mooring line dynamics cannot be modelled correctly.

Active Hybrid Systems simulate the truncated parts of the mooring lines and risers by computer-controlled actuators that must be able to work in model-scale real-time. The motions of the floater and other important system components are measured and fedback into the computer simulation, which delivers the control signals to the actuators. With an active hybrid system, dynamic mooring line behaviour can be simulated, including damping and soil mechanical aspects.

Numerical Reconstruction is an interesting variant of the hybrid modelling technique, where the physical model test and the numerical simulation are decoupled in time. A numerical model is calibrated to reconstruct the time histories from the model test with the truncated system. The calibrated model is then used to extrapolate the behaviour of the complete system to full water depth with all truncated parts of the system included. This can be done by use of coupled analysis tools as described by Ormberg et. al. (1999). Methods for experimental estimation of hydrodynamic coefficients, important for the reconstruction phase, are described by Stansberg et al. (1998).

The numerical reconstruction is more than just a calibrated numerical simulation. The method acts as a correction to and extrapolation of the model test, adding the characteristics of the mooring system that cannot be modelled correctly in the physical model, while still containing the true non-linear or even chaotic behaviour of the floater, which cannot be modelled correctly in the numerical model.

6. EVALUATION OF NUMERICAL AND EXPERIMENTAL TECHNIQUES

6.1 Introduction

This section gives the results of an evaluation carried out in order to establish the state-of-the-art procedures used in numerical and experimental simulation of deep water moored vessels in wind, waves and current.

In this respect the Committee has decided to carry out a survey to identify the current procedures employed by ITTC member organisations as well as non-member organisations involved in the analysis of deep water moored systems. The organisations responding to the questionnaire are listed below.

1. China Ship Scientific Research Centre, China.
2. David Taylor Research Centre, USA.
3. Defence Evaluation and Research Agency, UK.
4. El Pardo Model Basin, Madrid, Spain.
5. Global Maritime Ltd., UK.

6. Hyundai Maritime Research Institute, Korea.
7. ICEPRONAV S.A., Galati, Romania.
8. Ishikawajima-Harima Heavy Industries Co. Ltd., Japan.
9. Krylov Shipbuilding Research Institute, Russia.
10. MARIN, Wageningen, The Netherlands.
11. Marine Design & Research Institute of China, China.
12. Mitsui Zosen, Akishima Laboratories, Japan.
13. MSC International (UK) Ltd., UK.
14. NRC Institute for Marine Dynamics, Canada.
15. Offshore Technology Research Centre, USA.
16. Osaka Prefecture University, Department of Marine System Engineering, Japan.
17. Ship Research Institute, Japan.
18. University of Glasgow, Department of Naval Architecture and Ocean Eng., UK.
19. University of Michigan, Dept. of Naval Architecture and Marine Eng., USA.
20. University of Newcastle, Department of Marine Technology, UK.
21. VBD, Duisburg, Germany.
22. W.S. Atkins Oil and Gas, UK.
23. Yokohama National University, Japan.

6.2 Results of the Questionnaire

The responses to the questionnaire were analysed and the percentage-wise results together with comments *(in italics)* are summarised on the questionnaire form as shown in the following.

1. Which of the following mooring concepts can be modelled using your mooring analysis tools/experimental facilities?

	Num.	Exp.
a) Combination of wire-chain system	60%	68%
b) Combination of wire-chain-submersible buoy system	60%	45%
c) Combination of wire-chain-polyester system	50%	32%
d) Comb. of wire-submersible buoy-polyester-chain system	50%	32%

2. Are your tools suitable for uncoupled as well as coupled vessel mooring system investigation?

a) Yes	83%
b) No	17%

3. Do you consider the effect of loading, stiffness and damping of a riser system on the vessel and mooring system behaviour?

	Num.	Exp.
a) Yes	45%	55%
b) No	55%	45%

4. Which of the following effects do you consider when you carry out uncoupled analysis/experimental investigation of vessel motions of a vessel-mooring-riser system?

	Num.	Exp.
a) Mean current forces on mooring lines	41%	41%
b) Mean current forces on risers	45%	50%
c) Low frequency damping effects due to mooring lines	23%	36%
d) Low frequency damping effects due to risers	23%	41%
e) Mooring mass	50%	59%
f) Riser mass	50%	63%
g) Mooring stiffness	59%	59%
h) Riser Stiffness	45%	50%
i) Please add other effects that you consider: *Riser pretension*	4%	4%

5. In your uncoupled analysis tools, are the motion response equations of the vessel in the vessel-mooring-riser system based on:

a) Linear frequency domain equat.?	60%
b) Non-linear time-domain equat.?	40%

6. Is your dynamic mooring analysis in the uncoupled system based on:

a) Analytical modelling?	32%
b) Lumped-mass modelling?	53%
c) Finite-element modelling?	32%

7. Do you have coupled analysis tools for:

a) Vessel-mooring systems?	100%
b) Vessel-mooring-riser systems?	60%

8. Are your coupled dynamic analysis tools based on:

a) Linear frequency-domain equat.?	53%
b) Non-linear time-domain equat.?	47%

9. How do you obtain maximum/extreme design parameters?

	Num.	Exp.
a) Through deterministic anal.	14%	23%
b) Through statistical analysis	64%	58%
c) Depending on the parameters we may use, both	22%	19%

10. If you have answered (c) at question 9, please give details.

No details given.

11. Have you validated your uncoupled vessel motion and dynamic mooring analysis tools through:

a) Model testing?	24%
b) Numerical simulations?	18%
c) Model testing as well as numerical simulations?	54%
d) Full scale measurements?	4%

12. Have you validated your coupled vessel motion and dynamic mooring analysis tools through:

a) Model testing?	38%
b) Numerical simulations?	15%
c) Model testing as well as numerical simulations?	47%
d) Full scale measurements?	0%

13. In defining the environmental design criteria for deep water moored vessel behaviour and mooring loads, how do you specify wave, wind and current conditions?

	Num.	Exp.
a) 50-year return concurrent conditions	26%	32%
b) 100-year return concurrent conditions	26%	26%
c) 100-year wind and wave plus 10-year current	21%	21%
d) 100-year wave and current plus 10 year wind	11%	21%
e) 100-year wave and associated wind and current	21%	37%
f) 100 year wind and associated wave and current	16%	32%
g) Steeper waves than associated with 100-year wave height	21%	21%
h) 1-year wave and 100 year current	16%	26%
i) N year response	21%	26%

14. Do you assume that waves, wind and current act on the moored system:

a) Colinearly?	47%	51%
b) Non-colinearly?	53%	49%

15. If your answer is (b) at question 14, how do you select the angles between waves, wind and current, and how many different angles do you consider?

Typically two or three; Decide in co-operation with metocean people; Customer specifies; Typically twelve directions specified from the analysis of metocean data.

16. If you are to simulate the behaviour of a moored deep water vessel (depth greater than 1500 metres) experimentally, which of the following test strategies would you choose?

a) Testing a small scale model in an existing basin?	41%
b) Testing a traditional scale model at sea?	0%
c) Testing a complete model of the vessel with simplified models of moorings and risers?	36%
d) Testing elements of the system in existing basins at traditional scale and synthesising the complete system by numerical simulations?	41%
e) Constructing a new experimental facility?	14%

17. Further comments and/or suggestions?

Expression of interest to carry out comparative studies using numerical simulations under the auspices of the ITTC Deep Water Mooring Committee. Further development of hybrid model testing techniques.

6.3 Conclusions of the Survey

The results of the survey indicate the need for the development of numerical tools for the analysis of:

- Mooring systems, which consist of the combination of wire-chain-polyester or wire - submersible buoy - polyester-chain systems.
- Coupled vessel, mooring and risers system behaviour.
- Low frequency damping of mooring lines and risers.

The results of the questionnaire analysis also indicate the need for:

- Validation of uncoupled/coupled analysis tools through physical or hybrid modelling.
- Further development of hybrid model testing techniques.

7. GENERAL TECHNICAL CONCLUSIONS

The number of deep water moored offshore vessels is growing rapidly, and hydrocarbon fields in water depths up to 3000 m are now seriously considered for floating production development. Physical and numerical modelling of such vessels with their extensive mooring lines and risers is a challenge to the ITTC Community.

Physical models of floating vessels moored in extreme water depths require testing at extreme scale ratios. Model testing of moored floating offshore vessels in waves has been carried out successfully at a scale ratio of 1:170 with results very similar to those obtained at scale ratio 1:55. Practical problems rather than scale effects seem to be the limiting factor for this type of model testing, but the scale ratio of 1:170 is close to the practical limit with today's model basin technology.

Even at such extreme scale ratios, however, no existing or planned model basin can accommodate a full model of a catenary mooring system at 3000 m water depth. In addition to the correct water depth, such a basin should have horizontal dimensions at least five times the water depth in any direction. Systems for accurate generation of uniform and non-uniform current as well as multidirectional waves and wind all over the basin will be required. When a basin is too small for a full model, truncations or simplifications of the physical model will have to be made.

Testing in “natural” model basins such as fjords, lakes and rivers can be considered for special research projects. But due to the non-controllability of the environmental test conditions, such facilities cannot be used on a routine basis.

Hybrid models are physical models where the necessary truncations or simplifications of e.g. the mooring system are simulated by computer-controlled mechanisms. A hybrid model can be an interesting solution, when neither a physical nor a numerical model of a complicated system can be made. A reliable hybrid model will require a good numerical model of the behaviour of the truncated parts, which can be operated in model-scale real-time. Even if such hybrid models have not been successfully much developed in the past, the necessary building blocks are at hand.

Numerical models of floating offshore vessels moored in deep water have been developed for many years. Some of the problems involved in the numerical modelling of such complex systems are amplified in extreme water depths, and the relative importance of current-, wave-, and wind-driven forces are different compared to the more usual water depths.

Generally, the more simplified numerical models that work satisfactorily in less complex problems, have had to be abandoned. Dynamic modelling is used instead of quasi-static modelling. Non-linear time-domain analysis is used instead of frequency-domain linear models because of the strongly non-linear behaviour of the anchor lines regarding stretching, catenary geometry, fluid loading and bottom friction. Finally, coupled instead of uncoupled models of vessel, anchor lines and risers are desirable, but also extremely computer demanding. A workable solution to this problem is a de-coupling into slow motion dynamics (manoeuvring and memory effects) and fast motion dynamics (wave frequencies).

8. RECOMMENDATIONS TO THE CONFERENCE

None.

9. RECOMMENDATIONS FOR FUTURE WORK

Work should be continued to incorporate other positioning systems than moorings as well as the hydrodynamic aspects of physical and numerical modelling of systems being designed for deep and ultra deep waters.

Development of recommended procedures, including procedures for hybrid model testing, should be further pursued.

Verification and validation of physical as well as numerical models should be further pursued.

The possibilities of obtaining metocean information and do environmental modelling of deep waters, especially of the currents down to 3000 m water depth should be investigated. Systematic information on current velocities, profiles, directions as function of depth and time, etc. is needed.

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The Specialist Committee on Deep Water Mooring

Committee Chair: Prof. Christian Age
(Technical Univ. of Denmark)
Session Chair: Prof. Grant E. Hearn
(Univ. of Newcastle Upon Tyne)

I. Discussion

Contribution to the Discussion of the Report of the 22nd ITTC Specialist Committee on Deep Water Mooring

by Bas Buchner, MARIN, The Netherlands

First, I would like to congratulate the Committee with its interesting work on this important issue. The report gives a very good overview of the problems and solutions and puts them into a broader perspective.

I had only a few small comments and one question:

Ultra small scale testing (5.2)

In the excellent paper of Moxnes and Larsen (1998) on their ultra small scale model tests, they also give a good overview of the problems related to small scale testing, that would be worthwhile to be incorporated in more detail in the report:

- Wave generation: the waves show stronger non-linearities due to capillary effects ('Wave generation is believed to represent an absolute limit for ultra small scale testing' according to their conclusions).

- Wind generation becomes troublesome.

- There are a number of practical problems related to weights, instrumentation ranges, etc.

With respect to the larger scale effects at ultra small scales, the Committee report states that 'scale effects

may still be significant, but obviously without much influence on the global behaviour for this type of structure'. In my view this last statement is a bit too general: there is only one experiment for one relatively low water depth (385 m). Changes in the water depth, number of mooring lines/risers and environmental conditions can make the effect of scale effects more or less important for the global behaviour of this type of structures.

Development of numerical tools (6.3)

In the list of needs for the development of numerical tools one important aspect is missing in my view: the investigation of the low frequency viscous damping on the hull (coupled surge, sway and yaw).

A lot of work has been carried out in the field of (non)-linear potential flow calculations of wave (drift) loads and responses, but to determine the actual response, its viscous counterpart is of similar importance and complexity. This has not been studied in sufficient detail yet.

Vortex Induced Vibration (VIV)

Can the Committee give its opinion on how the problem of Vortex Induced Vibrations (VIV) of deep water risers needs to be investigated?

II. Committee Reply

Reply of the ITTC Specialist Committee on Deep Water Mooring to Bas Buchner

The Committee would like to thank Dr. Buchner for his kind remarks and for his very relevant and valuable comments. As will be seen in the following reply, the Committee is in close agreement with Dr. Buchner on most of the issues raised.

Ultra small scale testing (5.2)

Wave generation in ultra small scale model testing brings us closer to a point where capillary effects begin to play an important part. However, in the actual test series reported by Moxnes and Larsen (1998) with storm sea-states modelled at a scale ratio of 1:170, capillary effects were not observed to be a problem. The shortest wave components of the model spectra were 25 cm. Some difficulties were observed regarding repeatability and non-linearity in the wave generation due to the very small motion amplitudes of the wave-makers and a relatively long distance between wave-makers and model. The quoted remark is not concerned with the actual test series, but rather aimed at a possible future development with even more extreme scale ratios than those tested here.

Wind generation is indeed difficult in ultra small model testing. The models are generally too small to carry any wind force simulation fans, and external fans can only with some approximation create the true wind boundary layer, turbulence, etc. The wind forces on any wires and other test equipment attached to the model will cause relatively large errors at these scale ratios.

There are a number of practical problems related to weight and instrumentation of very small models. Special model-making and instrumentation methods, bordering on watch-maker technology, will have to be developed. As mentioned in the report, a scale ratio of 1:170 means that 1 tonne in full scale becomes 0.2 grammes in model scale. A scale ratio of 1:300 means that 1 tonne in full scale becomes 0.04 grammes in model scale, which makes model construction very difficult, but not impossible.

Dr. Buchner is right in pointing out the dangers of generalizing the conclusions of one single test series, and the Committee has had no intention of doing so. A test result like this one can only be an indication, not a proof. However, the FPSO is not the easiest touch-stone. The frictional damping on the hull itself is strongly influenced by scale effects, and yet the global motions of the 1:55 model and the 1:170 model are almost identical. There are probably two reasons for this:

- Viscous hull damping constitutes a minor part of the total damping (less than 10%).

- Total damping, including viscous and non-viscous hull damping and viscous damping of mooring lines and risers, is relatively high (over 20% of critical damping). At this level the global behaviour is not very sensitive to small damping variations.

The above discussion is even more valid in deeper water, where the total damping will increase and scale effects on the viscous hull damping therefore be relatively less important. The inevitable scale effects on mooring lines and risers can be minimized by adjustment of the drag diameters.

The tests were all carried out in storm sea-states and the conclusions should not be applied uncritically to lower sea-states, where both total damping and the relative importance of viscous hull damping could be different.

Development of numerical tools (6.3)

Dr. Buchner points out the importance of the low-frequency viscous hull damping in determining the motion response by numerical tools. The Committee agrees, but does not see this as a specific deep water problem. In fact, the relative importance of viscous hull damping decreases with increasing water depth, as mentioned above.

The evaluation of viscous hull damping is a general problem related to ships and floating offshore structures, and it has been discussed in the Report of the 22nd ITTC Loads and Responses Committee.

Vortex Induced Vibration (VIV)

The Committee cannot give a simple answer to Dr. Buchner's question on how to investigate the problem of Vortex Induced Vibration (VIV) of deep water risers. The VIV problem is very complex and is relevant also to mooring lines, cables, pipelines, slender bars, and wind exposed structures. Again, the Committee would like to make a reference to the Report of the 22nd ITTC Loads and Responses Committee, where the experimental and numerical treatment of VIV problems is discussed in detail.