The Specialist Committee on Stationary Floating Systems

Final Report and Recommendations to the 23rd ITTC

1. MEMBERSHIP, MANDATE AND MEETINGS

1.1. Members

The membership of the committee includes:

- Professor Takeshi Kinoshita (Chair), University of Tokyo, Japan,
- Mr. Kazuo Hirata, IPT-Instituto de Pesquisas Tecnologicas, Brazil,
- Dr. B. Colbourne, National Research Council of Canada, Canada,
- Professor J.A. Pinkster, Technical University Delft, The Netherlands,
- Professor Yang Jianmin, Shanghai Jiao Tong University, China,
- Dr. Mun-Keun Ha, Samsung Heavy Industries Co. ltd., Korea,
- Dr. Krish Thiagarajan, The University of Western Australia, Australia
- Mr. Liviu Crudu, ICEPRONAV, Romania

1.2. Mandate

The committee was tasked by the ITTC as follows:

Evaluate techniques and recommend procedures for the experimental and numerical simulation of stationary floating systems in wind, wave and currents including hybrid-testing techniques and deep-water current profiles.

Report on the progress of full dynamic positioning (DP) systems and DP assisted deep-sea mooring, and develop procedures for model testing DP systems.

1.3. Meetings

Four formal meetings of the committee were held as noted below.

1. September 11, 1999, Shanghai China, at the 22nd ITTC.
2. February 22, 2000, Shanghai Jiao Tong University, Shanghai, China.
3. June 2, 2001, IPT-Instituto de Pesquisas Tecnologicas, Sao Paulo, Brazil
4. October 19, 2001, National Research Council of Canada, St. John's, Canada.

2. DEEP WATER PRODUCTION SYSTEMS

2.1. Market trends

It is accepted that the offshore oil industry is moving into deeper water (Brass, 2000) with tens of billions of barrels of oil equivalent discovered in water depths over 500 m. The main areas of activity are the Gulf of Mexico (GOM) and the Atlantic coasts of Brazil and Africa, with other prospects offshore eastern Canada and in the Philippines. Depths currently explored are in excess of 2000 m with ambitions for 3500 m. Production structures, considered for such applications, include FPSOs, TLPs and Spars. In each case, the systems consist of a floating platform, a mooring and a riser system in which the wellhead may be at the bottom, on the vessel, or somewhere in between. Systems without risers and moorings are being considered for future development but it is likely that near-term installations will consist primarily of the basic floater-mooring-riser configuration.

Deep-water concepts are originating primarily from Houston for the GOM and West African areas, and from Petrobras for offshore Brazil. Trends in the US reflect a desire for platforms with dry wellheads and rigid risers. Minimal vertical plane motions are required for a platform to support dry trees with vertical risers. Thus there has been a push towards platform designs that have lower heave motion. This still requires some means of supporting the risers in water (such as buoyancy cans) so the stroke requirement at the top can be minimized. Alternatively, short stroke tensioners can be employed at the top of the wellhead. Recently innovative methods of supporting risers using pneumatic and hydraulic platforms have been explored.

The trend in Brazil, Norway and Scotland has been to flexible risers and platform designs without stringent motion restrictions. Use of flexible risers depends largely on reservoir characteristics, and the location and ease of production of fluids.

Mooring systems for platforms deployed in deep water are the subject of much development. The trend in the GOM is to chain-wire-chain systems, in semi-taut configurations. In Brazil, polyester and other manmade fiber ropes have been tried with some success, and the trend is to increase their use. The benefits of fiber moorings are enormous cost reduction and a reduction in mooring line tension on the platform. However, long term viability of fiber moorings is not clearly established.

Risers have typically been made of marine grade steel, however, titanium alloy alternatives have been explored. The future holds a lot of promise for novel materials for risers, that can provide better corrosion protection and enhanced fatigue life.

2.2. New systems

FPSO and semi-submersible systems have been developed for enhanced safety and operability. While older FPSOs were largely conversion from tankers, recent FPSOs are built to specifications. Hulls are generally block-shaped to maximize volume. Newer semi-submersibles have targeted reduced heave motions, to enhance their capability. Hullform research in the immediate future is expected to be targeted towards dry wellhead platforms and we focus on these types.

The past twenty years have seen the emergence of the Tension Leg Platform (or variants of it), and the Spar platform as the dominant deep-water production concepts.

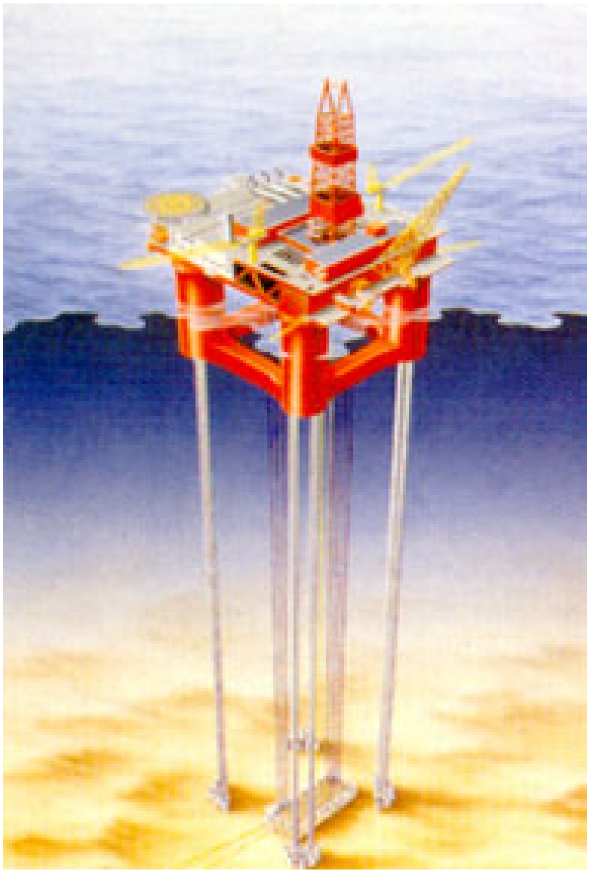

Figure 2.1 A Tension Leg Platform.

The Tension Leg Platform (TLP), (Figure 2.1) is a compliant structure moored to the seabed by vertical tethers that are tensioned by excess buoyancy over weight of the structure. Buoyancy is provided by the vertical columns and the horizontal submerged pontoon structural members. The structure's response is stiff in the vertical direction (heave, pitch/roll) and soft in the horizontal direction (surge/sway and yaw). Heave natural periods are typically 2 to 4 seconds and the surge natural periods 100 to 200 seconds (Chakrabarti, 1997). These platforms are weight sensitive and have limitations on accommodating variable payloads. Floating storage vessels are required in locations where there is no pipeline infrastructure as there is no oil storage capacity on the TLP. The history of TLP installations is shown in Table 2.1. It is noted that some of the TLP listed support flexible risers with subsea wellheads (such as Morpeth seastar).

Table 2.1 TLP – past, present and future (from Ronalds & Lim, 2001).

Name	Location	Water depth	Year of operation
Hutton	UK North Sea	148 m	1984
Jolliet	GOM	537 m	1989
Snorre	Norway	310 m	1992
Auger	GOM	872 m	1994
Heidrun	Norway	345 m	1995
Mars	GOM	896 m	1996
Ram-Powell	GOM	980 m	1997
Morpeth	GOM	509 m	1998
Allegheny	GOM	991 m	1999
Ursa	GOM	1160 m	1999
Marlin	GOM	986 m	1999
Typhoon	GOM	610 m	2001
Prince	GOM	457 m	2001
Brutus	GOM	910 m	2001

The classic spar (Figure 2.2a) comprises a cylindrical floating caisson with a large draft of 180÷200 meters in water depths up to 1830 meters. These dimensions are required to maintain the center of buoyancy above the center of gravity to provide stability, and reduce heave and pitch motions (Chakrabarti, 1997). The first classic spar was built in 1996, and two more were installed in the Gulf of Mexico (Table 2.2). Spars typically have hard tanks for buoyancy and variable ballast near the top of the caisson and soft tanks below for oil storage. Deck area is limited and the structures are sensitive to payload changes. Stationkeeping is provided by 12-16 semi-taut catenary anchor lines. The mooring line fair leads are located at or above mid draft, to minimize mooring line dynamics (Chakrabarti, 1997). The spar is compliant in all degrees of freedom, and typical natural periods are 300÷350 s in surge, 50÷100 s in pitch and 30 s in heave.

Table 2.2 Spars – past, present and future (Finn, 2001).

Name	Diameter	Water depth	Type	Year of operation
Neptune	72 ft	1935 ft	Classic	1996
Genesis	122 ft	2590 ft	Classic	1998
Hoover	122 ft	4750 ft	Classic	1999
Nansen	90 ft	3680 ft	Truss	2001
Boomvang	90 ft	3450 ft	Truss	2001
Horn Mountain	106 ft	5400 ft	Truss	2002
Medusa	94 ft	2500 ft	Truss	2003
Devils Tower	94 ft	5700 ft	Truss	2003
Holstein	148 ft	4350 ft	Truss	2004
Mad Dog	148 ft	4500 ft	Truss	2005

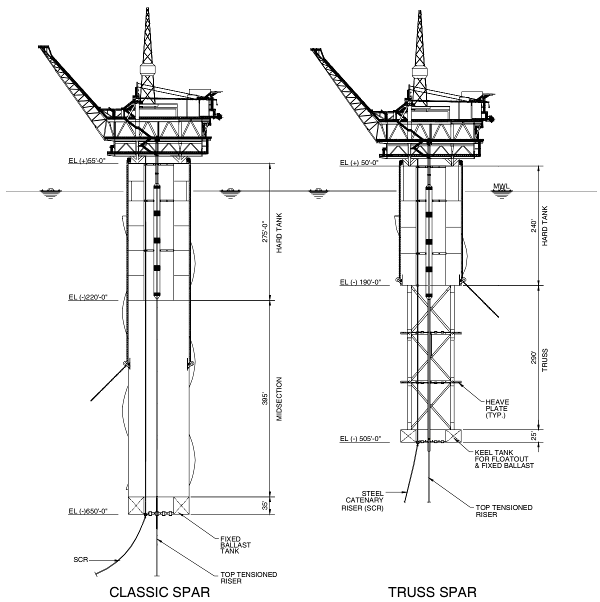

Figure 2.2 Spar platforms. (a) Classic and (b) truss type.

The hybrid-concept truss spars of which several are being built (Table 2.2), retain the good motion behavior of spars, without the penalty of the high hull/topsides weight ratio. Truss Spars are neutrally buoyant, free-floating cylindrical caissons with 12 or 16 lateral, catenary anchor lines for station keeping (Figure 2.2b). A truss structure attached to the base of the caisson provides the ballast distribution without extending the caisson.

Other concepts have been developed. The mini TLP (Figure 2.3) is proposed for marginal and remote field applications. The concept is developed to de-couple the deck dimensions, column spacing and tether separation distance, thus giving more freedom in optimizing each parameter. The Prince TLP supports 5500 t, with a displacement of 13200 t. In comparison, a classic TLP like the Jolliet has a displacement of 16700 t for a similar topsides weight.

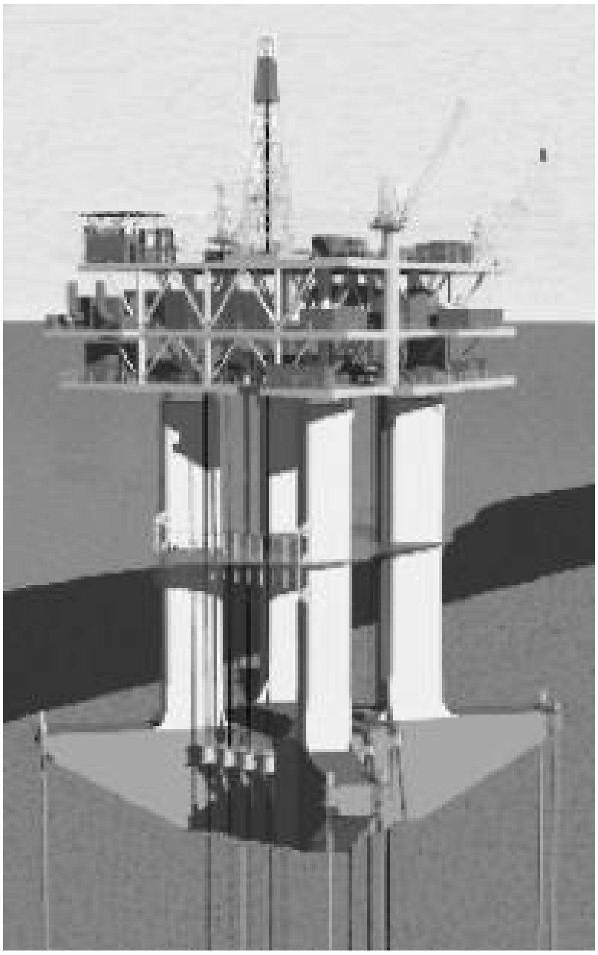

Figure 2.3 Artist impression of the Prince TLP.

2.3. Hybrid model testing and alternatives

Hybrid model testing is a method of analysis where shortcomings in the physical model or experiment are addressed through calculation or numerical models. An analog in conventional modeling is the use of artificial wind loads based on calculation. These can be either input to a numerical model that combines the calculated wind effect with the measured wave basin results, or injected directly into the model test through the application of a force to the model by actuator or deadweight.

Similarly, Hybrid modeling can take two forms. The limitation on a full physical model is most often limitations in water depth and/or extent of the basin that do not allow the full mooring system to be modeled. Inappropriate scaling of current forces or damping in the subsea portion of the system due to improper Reynolds Number scaling in a Froude based experiment will be discussed later but is also indirectly addressed by Hybrid Model Testing.

An example requirement for analysis is an FPSO with a lightweight catenary mooring in 1000 m of water, subject to wind wave and current loads. This case illustrates the two different approaches that may be taken in hybrid model tests. The exact form of hybrid analysis is dependent on the physical facility limitations and/or the numerical modeling capabilities of the organization performing the study. The following two cases are not particular to any organization but serve only to illustrate differences between two approaches.

The first form of Hybrid Model Testing is where partial or systematic model tests are conducted on subsets of a system. Results from the partial model tests are combined in a numerical simulation, which may also incorporate numerical models of parts of the system. For example wave basin tests on the FPSO hull restrained by a horizontal spring mooring would identify first and second order wave responses. Wind drag would be measured in wind tunnel tests. The two sets of data are then used in combination with numerical models of the current drag and the deepwater mooring response to develop a numerical simulation of the entire system response (Loken et al., 1999). The response of the system is derived from the numerical simulation model.

In the second form, the system response is derived from the model test, but numerical models of some subsystems are used to replace physical models. For example, actuators that follow a calculated load-extension curve apply the mooring response. Wind load is generated by a similar actuator, which applies a force time series calculated from the combination of an input wind spectrum and calculated drag coefficients for the above-water portion of the FPSO. The model is then exercised in a wave/current basin and the response of the system is observed in the basin with measurements of the quantities of interest (Watts, 1999).

There are commonalities in the two methods, the first being a physical model of the wave effects on the floating hull. The second is a numerical model of the mooring. Environmental effects may be derived from physical or calculational models. The main difference is whether physical modeling is used as input to a final numerical simulation, or if numerical models are used as input to a final physical simulation.

2.4. Alternatives to Hybrid Model Testing

Small Scale Model Tests. Small-scale model tests have been investigated by MARINTEK (Moxnes & Larsen, 1998) and reported to be reasonable for global responses including motions and mooring system loads. They calculated significant (40%÷100%) differences in viscous damping between a 1:55 scale experiment and a 1:170 scale experiment but, in the end, indicated that these differences appeared to have little effect on the compared results. However the results matched best for large waves at both scales and can be seen to be considerably less well matched for smaller waves. This implies that ultra small scales may be appropriate for predicting extremes but perhaps not so appropriate for operating conditions or predictions of long term responses.

Consistent with this, Moxnes and Larsen indicated that the most significant problem was likely to be wave generation at the small scales. This becomes less of a consideration for extreme or survival conditions where wave heights are generally above 10 m but operating conditions where wave heights might be 5 m or less would cause significant trouble for most basin wavemaker systems, particularly in the presence of a current. Later work, reported on the Marintek website (2002), has indicated that scales between 1:100 and 1:125 are more appropriate.

Larger facilities. Larger facilities are the conceptually easiest solutions for deeper water (if we set the viscous scaling issue aside) but costs associated with such facilities are high. At present 10÷11m depth, is the limit for offshore modeling facilities (excluding deep pits). At scale factors of 60 to 75, full models are limited to approximately 800 m of full-scale water depth.

Deep Pits in Basins. Deep pits are relatively small areas of the basin, deeper than the main part by some significant amount. These pits are useful for TLP structures where the lateral extent of moorings and risers are small (on the order of the dimensions of the structure itself). A pit does not offer benefits for a spread mooring. There is evidence that a pit shelters the lower part of a mooring from hydrodynamic effects, thus leading to different results than might be achieved with a uniformly deep basin (Murray et al., 1991).

Tests in Natural settings. Natural settings such as lakes, fiords or stretches of open ocean have been proposed for model testing deep-water structures. Such locales offer unlimited space and the use of large scales but present logistical difficulties and offer no control over the applied environment. The best use of such a test location would be to verify hybrid or numerical models. A model could be put in place and the environmental action and resulting response measured over a period of time. These conditions could then be simulated in the hybrid or numerical model and the measured and predicted responses compared to validate the models.

2.5. Current Best Practice

Analysis of a system by hybrid model test is conditional on the following:

1. A full model of the system cannot be accommodated in a basin at an acceptable scale.
2. The entire system response cannot be adequately analyzed using numerical models alone.
3. A satisfactory fully dynamic numerical model of the moorings and risers is available.
4. A satisfactory means of applying the mooring response is available for a physical simulation.
5. A numerical model, which allows the model test results to be reconstructed, is available.
6. The post-test numerical reconstruction and the hybrid model test provide consistent results.

Based on recent literature from various test establishments, including MARIN (Buchner et al., 1999), DERA/OTRC and Marintek (Ormberg et al., 1997; Ormberg et al., 1999; Stansberg, 2000), current best practice appears to be that put forward by Marintek. A truncated model test, using a passive mooring, is performed and then replicated with a fully coupled numerical model. The calibrated numerical model is then extrapolated to the deep-water case.

This method could be improved by introducing the idea proposed by Watts (1999) of an active actuator that provides the dynamics of the deep-water mooring in the model test. This should allow results from the model test to be directly applied to full scale or compared with numerical simulation. This would also allow the intermediate step of calibrating the numerical model at the shallow depth to be skipped and should provide a better simulation of LF wave response (and possibly the HF line tensions) in the model test. The DERA scheme would have to be improved by developing an actuator that more-closely replicates a mooring line in point of application and angles of departure. This could either be along the lines of the arm-type actuator proposed by MARIN (Buchner et al., 1999) or one using servo controlled winches (Millan & Lindstrom, 1995) as developed at IMD.

A recommended procedure document for Hybrid Model Testing (attached as Appendix A) was prepared and submitted to the ITTC as part of this review.

2.6. Numerical Modeling

Full numerical models of mooring and riser systems have progressed steadily in the past two decades, gaining wider acceptance as experience grows. There is a general preference for Finite Element Models of mooring lines and risers but there is also some work on spring-mass models, which offer advantages in computational efficiency. Deep-water situations are amenable to linearization, particularly in the mooring statics and dynamics. Commercial Finite Element computer packages are generally accepted in industry as reliable mooring and riser system models. This allows a high degree of confidence that the parts of the system most difficult to model in a physical basin can be modeled numerically.

Motions of the floating element(s) of a deepwater system, subject to wave loading, can also be modeled numerically within assumptions of linearity. Linearity may be stretched by extreme wave cases or by the shape of the floating structure. In either case, results become less reliable and are often confirmed by physical model tests. The main problems with numerical models appear to be in the following areas:

1. Extreme wave loadings where the forcing function becomes highly non-linear.
2. The prediction of second order wave effects where the forcing associated with wave difference frequencies appears to be generally under-predicted.
3. The calculation of damping associated with wave radiation (potential damping) and the interference of incident waves.

Traditionally, numerical analyses have been conducted by un-coupling the mooring and riser system from the floating element and providing a quasi-coupled solution where the vessel motions are assumed independent of the mooring. Recent improvements have been reported with fully coupled numerical models but these are reported to be computationally intensive to the point where they are prohibitively slow (Ormberg et al., 1997).

Reports on full numerical models indicate problems in predicting second order wave effects in mean offsets and low frequency oscillations. This appears to be the only major issue that requires experimental data. With a solution for this issue, full numerical models would offer a viable alternative to Hybrid Modeling.

Loads in long mooring lines and risers are strongly influenced by dynamic tension effects. Predicting dynamic tensions in a numerical model relies on a good dynamic model and a good estimate of the frequency dependent hydrodynamic forces. Such models are difficult to verify at model or full scale.

Coupling between the moorings and the floating structure limits the ability to do independent tests. At any basin the numerical mooring model used in a numerical simulation and that used to derive a truncated mooring response is likely to be the same model. Thus a post-model test numerical reconstruction is not likely to be a fully independent verification of either approach. It may be worthwhile to exercise different models and approaches at different basins to explore differences in results.

2.7. Vortex Induced Vibrations

Vortex shedding from cylinders occurs all the time for all currents. However, problematic vortex shedding occurs only at a certain current and diameter combinations. There is a combination of diameter (D) and current speed (U_c) at which the shedding is stabilized at the frequency f_v. If we define a Strouhal number,

$$St = \frac{f_v D}{U_\infty} \tag{2.1}$$

von Karman's theory suggests f_v corresponds to St = 0.28, while experiments revealed that St = 0.2÷0.3 depending on Reynolds number (see e.g. Blevins, 1990).

Vortex shedding results in oscillatory lift and drag forces on the cylinder, due to varying pressure distributions when the vortex is shed. The frequency of this force (F_{Lv}) coincides with the shedding frequency, and is expressed in terms of a lift coefficient (C_{Lv}) as:

$$F_{Lv} = C_{Lv} \frac{1}{2} \rho U_c^2 D \cos(2\pi f_v t + \alpha) \tag{2.2}$$

where α is the phase angle between the force and shedding pattern. C_{Lv} is typically around 1.35. It is found that vortex shedding does not occur uniformly along the cylinder axis, resulting in an erratic variation of α along the cylinder length. The net effect is that the oscillatory lift force is almost negligible in comparison with other forces, except when the cylinder locks-in.

In terms of drag, the force oscillates at twice the shedding frequency (every vortex shed irrespective of the location is on the lee side of the cylinder) and is added on to the steady drag term.

$$F_{Dv} = C_{Dv}\frac{1}{2}\rho U_{\infty}^{2} D\cos(4\pi f_v t+\beta) \qquad (2.3)$$

where C_{Dv} is typically about 20% of the steady flow drag coefficient.

Vortex-induced vibrations (VIV) occur when the frequency of vortex shedding coincides with the natural frequency (f_n) of the structure. VIV in risers can cause elastic resonant oscillations. Lock-in can be described by the parameter called "Reduced velocity"

$$V_r = \frac{U_c}{f_n D} \qquad (2.4)$$

Reduced velocity is important because it characterizes the riser. The most noticeable form of VIV occurs in the cross flow direction. As current speed varies and the vortex shedding frequency approaches the natural frequency of the cylinder, the vortex shedding locks into the natural frequency of the cylinder, resulting in resonant motions. The lock-in occurs well ahead of the resonant condition ($f_v = f_n$), and does not unlock until well after the resonance has passed.

The "wake capture" or "lock-in" region is best described in terms of the reduced velocity. Currently accepted practice states that cross flow vibration results from:

$4.8 < V_r < 8$ (in current alone)

$3 < V_r < 9$ (in waves and current)

In the lock-in region, cross-flow oscillations have been observed to be a maximum of 1÷2 times the diameter. According to Allen & Henning (2001) maximum vibration amplitudes are about 1 to 1.5 times the cylinder diameter and under forced vibration, the cylinder vibration amplitude is usually around 0.1 to 0.2 times the cylinder diameter. The "self-limiting" nature of VIV is due to the fact that beyond a certain vibration amplitude, the movement of the cylinder destroys the stable vortex shedding pattern causing a decrease in the exciting force.

Apart from cross flow vibration, VIV also occurs along the in-line direction. There are two primary regimes of vibration in the in-line mode,

Inline vibration (first instability) $1 < V_r < 2.2$

(second instability) $2.2 < V_r < 3.5$

The first instability arises due to combined action of:

1. normal vortex shedding giving rise to two oscillations per shedding
2. secondary symmetric vortex shedding which occurs as a result in-line motion of the cylinder relative to the fluid.

This creates a situation where the oscillation frequency is approximately three times the Strouhal frequency. The second instability mode is the case where the oscillation frequency is about twice the Strouhal frequency.

Research has shown the validity of equating energy input from the current to energy dissipated in structural damping at the resonant condition. This leads to a stability parameter, k_s, that determines the magnitude of oscillation.

$$k_s = \frac{2m_e\delta_s}{\rho D^2} \qquad (2.5)$$

where m_e is the equivalent mass per unit length, and δ_s is the logarithmic decrement of structural vibrations, and are well defined in e.g. Sumer & Fredsoe (1997). It is generally shown that the lesser the value of k_s, higher the amplitude, with the first and second instability conditions giving similar magnitude of oscillations. Compared to cross-flow conditions, the inflow oscillation amplitudes are much smaller.

Sumer & Fredsoe (1997) mention that certain researchers (e.g. Tsahalis, 1984) have observed a third region of in-line vibrations that occurs primarily for cylinders with two degrees of freedom. This region coincides with the cross-flow vibration regime, i.e. $4.8 < V_r < 8$, and the amplitude in this region is much higher than that of the first and second instability region. While the cross-flow oscillations occur at the Strouhal frequency, the in-line vibrations continue to occur at twice the Strouhal frequency. Increased flow velocities around the cylinder are believed to be the driver for this type of oscillations. This region of oscillation is not mentioned in some widely used design guides.

Although a large volume of literature exists on VIV, much still remains unknown. For example, until recently, lock-in was understood to be due to hysteretic behavior of the flow. However, Vikestad et al. (2000) gave a description of the role added mass plays in development of the lock-in phenomenon. The change in added mass changes the natural frequency, and sheds new light on the mechanism of lock-in. The consequence may be more applicable for complex configurations such as flexible risers moving in deep water where the currents are varying with depth. Further numerical and experimental analyses backed by full-scale data will enable better VIV prediction, and its effects on fatigue prediction of risers.

2.8. Deep Water Current Profiles

The present trend in extracting oil from deep locations is to use a large number of flexible or rigid risers, sometimes more than a hundred, suspended by the floating system. The influence of current flow on these risers is a concern. Drag damping on the risers affects the motion of the floating structure and interference between the risers can lead to clashing. Vortex induced vibrations give rise to structural problems.

Design verification or scientific studies, depend on accurate current profile reproduction. One difficulty in simulating current in a laboratory is the space and time variation, particularly the full-scale profile along the depth. This variation is in intensity and in direction. The current profiles given in Figures 2.4 to 2.7 show these aspects. Samples represent currents at Campos Basin going South and North-East at the surface, in about 1000 m depth. Figure 2.4 shows variation of current direction with depth, when coming South.

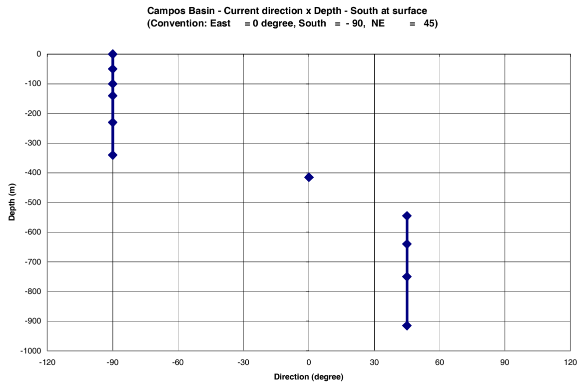

Figure 2.4 Current Direction with Depth.

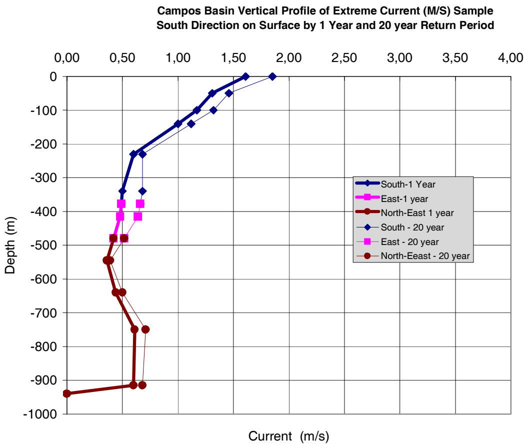

Figure 2.5 South Current Profile.

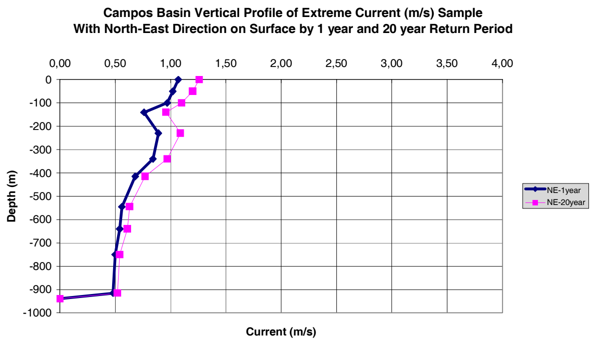

Figure 2.6 Northeast Current Profile.

Figure 2.5 shows extreme current in the South direction at the surface for 1 and 20 year return periods. The variation along depth is according to Figure 2.4. Figure 2.6 shows extreme current in the North-East direction at the surface for 1 and 20 year return periods. In this case the direction is constant along depth. Figure 2.7 shows a comparison between South and NE direction extreme current for the 1-year return period.

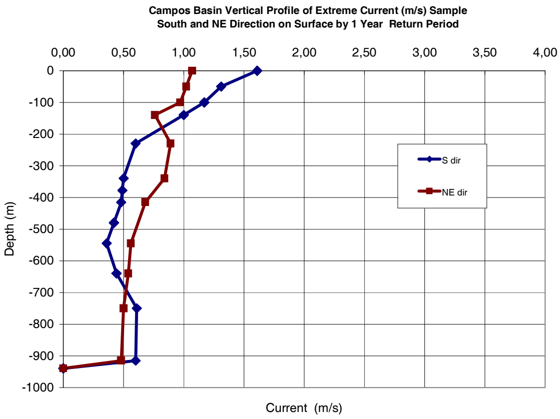

Figure 2.7 Current Profiles, South and North-East.

3. DYNAMIC POSITIONING SYSTEMS

Dynamic positioning is the controlled use of forces generated by purposely fitted thrusters to maintain a vessel position in the horizontal plane against forces generated by the environment. DP systems have been in use since the 60's when they were first used to position small survey vessels at sea. In the 70's much effort was put into developing reliable DP systems for application on drilling ships and drilling semi-submersibles.

With the increasing reliability of such systems and the awareness of the inherent flexibility of DP as a way of maintaining station for limited periods of time as compared with conventional mooring systems, DP is now applied in many activities offshore. It is not limited to keeping a vessel on a fixed location but also to controlling the motions of vessels along predetermined tracks.

3.1. Usage

DP systems provide station-keeping, position-keeping, and course-tracking for the following vessel types.

Deep Water Drilling Vessels. A new generation of very large drillships will be used for exploration in very deep waters. These ships have complex thruster arrangements and use advanced DP systems to control the thrusters. Recent examples are given by Wichers et al. (1998) and by Aalbers & Michel (1998).

Floating production vessels. The use of DP for FPSO vessels started with controlled thrusters to assist in maintaining the heading of turret-moored FPSO's (Nakamura et al,. 1995; Aalbers, 1996). Today there is a trend towards full DP for floating production systems. In the 80's the first full DP production ship, the Seillean, was used in the North Sea, (Davison et al., 1986). This ship is now being used off the coast of South America for production in deeper water. The vessel has been producing oil through a flexible, single riser for extended periods of time. It is expected that as experience and confidence is gained, more full-DP FPSO vessels will appear.

Offshore offloading of oil and/or gas. Purpose-built, DP tankers are being used for offloading operations from permanently moored FPSO vessels. The DP shuttle tanker maintains its position relative to the (moving) stern of the FPSO.

Cable & Pipe laying vessels. Cable laying operations can make use of the ability of DP systems to control the vessel in a tracking mode whereby the set-point of the vessel is moved along the cable route. Some pipe laying vessels have dispensed with conventional wire-rope mooring and anchor handling vessels and are relying exclusively on DP.

Ocean research & exploration vessels, dredging vessels, and ROV. (Hsu et al., 1999;

Spindler et al., 1998; Ishii et al., 1996, 1997). Ocean research & exploration vessels were among the first to make use of DP systems, for carrying out coring operations. For this type of ship the ability to stay accurately on station or to move the vessel accurately along a predetermined track using DP is a necessity for obtaining high-quality and reliable data. Dredging vessels, especially trailing suction dredges, are contracted to clear entrance channels and harbors to a specified depth. Efficient operation requires that the suction units accurately follow a predetermined path. Modern suction dredges are fitted with DP systems capable of performing the tracking operation to the required accuracy.

Large Crane Vessels. Large semi-submersible crane vessels used for offshore construction and removal are fitted with DP systems thus avoiding the interference of mooring systems with pipeline infrastructure in the vicinity of fixed platforms. These systems, although fitted to very large vessels, are required to operate in relatively low environmental conditions.

The duration of time in which a vessel is required to operate in the DP-mode is continually increasing. Vessels are currently operating continually in DP-mode for periods greater than 1 year. This reflects the greatly increased reliability of the on-board systems and is also an important factor in the wider application of such systems.

3.2. Basic functions of DP

A Dynamic Positioning System (DPS) is designed to keep a vessel within specified position and heading limits, and to minimize fuel consumption and wear and tear on the propulsion equipment. The vessel should keep position by operating thrusters against environmental forces caused by wave, wind and current. DPS includes the following basic elements: sensor system, controller and thruster system.

The sensor systems include GPS, acoustic systems, radio systems and taut-wire systems for continuously measuring the position of the vessel. The heading of the vessel is obtained from a gyroscope or compass. After filtering the position data to remove high-frequency disturbances, the measured position is compared with the desired position to obtain the position error on the basis of which the control system will act. Environmental sensors also measure wind speed and direction. This data is used to make real-time estimates of wind forces and moment acting on the vessel. This data can be included in the control algorithm. The DP controller calculates force and moment commands in three control axes (two for position and one for heading) to reduce the errors and then distributes the force and moment commands among the active thrusters in a logical manner.

Many developments have improved the performance of DP systems. Filtering wave frequency motions avoids 'thrust modulation', which is an oscillation of thrust caused by wave frequency motions. Current well-designed DP systems also incorporate numerical models to estimate the low frequency motion of the vessel.

In model test and simulation programs the control theory should include the numerical model and the filter algorithm to be used on the full-scale vessel.

3.3. Designing DP systems

In this report we do not go into the details of the hardware involved in DP systems but only into the physical phenomena and required data. The purpose of this review is to identify shortcomings in knowledge. First we focus on the important physical phenomena.

External forces on the vessel. External forces acting on a DP vessel are due to waves, wind and current. Additional loads arise from items such as risers or, in the case of DP-assisted vessels, from the mooring system. For solitary vessels, environmental forces are derived from knowledge of the open sea conditions. For DP vessels operation in the Offloading mode behind an FPSO, the effects of the presence of the FPSO on the environmental loads on the DP vessel need to be accounted for.

Much of the data on environmental forces has to be obtained from model tests. This applies specifically to current forces and wind forces. Wave forces, of which second order mean and low frequency drift forces are the most important, can be estimated based on 3-dimensional diffraction theory (Pinkster, 1980). Interaction effects between waves and current and the resultant mean and low-frequency forces are generally neglected although these effects can be significant. In the case of semi-submersibles, it has been shown that 3-D diffraction computations can underpredict mean and low frequency drift forces (Dev, 1996).

In cases involving two vessels, for instance, a DP shuttle tanker behind a FPSO, current and wind shielding effects as well as wave shielding effects need to be accounted for.

<u>Operation of the thrusters</u>. Thrusters operate under complex conditions both with respect to the inflow to the thruster and the outflow conditions. Both aspects influence efficient operations of the thrusters. We have to account for, for instance:

Thruster degradation due to current
Thruster-hull interaction
Thruster-thruster interaction
Thruster degradation due to waves including ventilation effects due to emergence.

Aalbers & Michel (1998) give a review of the phenomena affecting thruster effectivity.

Secondly we need to focus on the control algorithms used in DP systems. The control algorithm of a DP system has several functions:

- Filter the position error signal to ensure that the thrusters only counteract low frequency and mean environmental disturbing force components. Thrusters should not react to wave frequency position errors since they lead to unacceptable thrust modulations.
- Account for instantaneous wind forces and moment based on wind speed and direction measurements.
- Determine the horizontal thruster forces and yaw moment necessary to regain position.
- Distribute the required thruster forces over the available thruster system.
- Allocate thrust signals to the individual thrusters

Modern control systems use models of the DP vessel and the thruster characteristics in order to derive optimal control strategies. This makes it essential that knowledge of hydrodynamic properties and phenomena affecting the behavior of DP vessels be available to DP system designers.

In the design process, one of the first steps is to assess the mean environmental forces acting on the vessel (surge and sway forces and yawing moment) for a range of sea-conditions. From this static assessment a first estimate can be made of the necessary number and arrangement of thrusters and the installed power. However, this will not be sufficient for the full design of the system (Aalbers et al., 1995). Simulations, in which static and dynamic effects of the environmental forces, as well as vessel control algorithm and thruster characteristics, are required to develop a complete design. An overall validation of the performance through model tests in which the total system is modeled, is recommended by Wichers et al. (1998) and Aalbers & Michel (1998).

3.4. Simulations

DP capability plot. The capacity and position of thrusters determine DP capability. The following commercial programs are used in the shipyards for checking a vessel's DP capability;

(1) DPSIM developed by MARIN
(2) SIMULA developed by GUSTO Engineering (Aalbers et al., 1995)
(3) DYNACAP developed by MARINTEK
(4) Integrated Vessel Simulator by ABB
(5) Various self-developed simulation programs

These programs require the following data:

(1) Thruster position, angle, thrust, type (azimuth/tunnel, azipod, main screw) and rudder effect.
(2) Hydrodynamics forces; wave drift forces, wind forces and current forces.
(3) Vessel particulars; length, breadth, draft, mass, wind and current area.

Often in the initial stage of design, detailed data on external forces are not available and either the database in the program or estimated values, as per the following method, are used.

(1) Wind forces; Data from the OCIMF or similar ships can be used in the initial stage, to estimate thruster capacity, but wind tunnel test data is more exact.
(2) Current forces; drag and moment acting on the submerged hull from current can also be estimated using OCIMF data.
(3) Wave forces; wave forces are composed of the linear wave excitation force and the non-linear wave force. Considering an irregular wave spectrum, the linear wave force can be expressed as,

$$F_1^{(1)}(t) = \sum_i |\zeta_i| F_{1c} \cos(-\omega_i t + \varepsilon_i) + \sum_i |\zeta_i| F_{1s} \sin(-\omega_i t + \varepsilon_i) \qquad (3.1)$$

The non-linear wave force is formed from interaction of the different wave frequencies, such as $2\omega_i$, $2\omega_j$, $\omega_i + \omega_j$, $|\omega_i - \omega_j|$, but the slow drift damping force will only affect horizontal motions. The DP system considers the low frequency drift force and moment which has a second order transfer function (Pinkster, 1976).

$$F_1^{(2)}(t) = \sum_i \sum_j |\zeta_i||\zeta_j| P_{1ij} \cos(-(\omega_i - \omega_j)t + \varepsilon_i - \varepsilon_j) + \sum_i \sum_j |\zeta_i||\zeta_j| Q_{1ij} \sin(-(\omega_i - \omega_j)t + \varepsilon_i - \varepsilon_j) \qquad (3.2)$$

For small ships, second order wave drift forces are generally not the dominant environmental forces. This is because diffraction effects are small for the same wavelengths. For large ships wave drift forces tend to dominate both the mean and low frequency forces. For this reason, it is of interest to investigate wave-feed-forward systems for large DP vessels.

(4) Mooring Line forces; the horizontal component of tension in mooring system can be described with an empirical polynomial of position variance.
(5) Time memory function; for simulation of motion in the time domain, the time memory function may also be considered.

$$L_{ij}(t) = \frac{2}{\pi}\int_0^{\infty} b_{ij}(\omega) \cos \omega t \, d\omega \qquad (3.3)$$

Simulation Program Motion Equation. The plane motion of moored vessel can be decomposed into the low frequency motion caused by wind, current, second order wave forces and the wave frequency motion cased by first order wave force (Lee & Ha, 1999).

$$\begin{aligned}(M + a_{11}(\infty))\dot{u} = {} & M x_g r^2 - \int_{-\infty}^{t} L_{11}(t-\tau)u(\tau)d\tau + (M + a_{22}(0))vr \\ & -(a_{22}(0) - a_{11}(0))V_c \sin(\alpha_c - \psi)r \\ & + \frac{1}{2}\rho_w LTC_{xc}(\alpha_{cr})V_{cr}^2 + \frac{1}{2}\rho_a A_T C_{xw}(\alpha_{wr})V_{wr}^2 \\ & + F_{1wave}^{(1)} + F_{1wave}^{(2)} - X_{1moor} + T_x\end{aligned} \qquad (3.4)$$

$$(M+a_{22}(\infty))\dot{v} = -(Mx_g + a_{26}(\infty))\dot{r} - \int_{-\infty}^{t} L_{22}(t-\tau)v(\tau)d\tau$$
$$-\int_{-\infty}^{t} L_{26}(t-\tau)r(\tau)d\tau - (M+a_{11}(0))ur$$
$$-(a_{22}(0)-a_{11}(0))V_c\cos(\alpha_c-\psi)r$$
$$+\frac{1}{2}\rho_w LTC_{yc}(\alpha_{cr})V_{cr}^2 + \frac{1}{2}\rho_a A_L C_{yw}(\alpha_{wr})V_{wr}^2$$
$$+F_{2wave}^{(1)} + F_{2wave}^{(2)} - X_{2moor} + T_y \tag{3.5}$$

$$(M+a_{66}(\infty))\dot{r} = -(Mx_g + a_{62}(\infty))\dot{v} - \int_{-\infty}^{t} L_{66}(t-\tau)v(\tau)d\tau$$
$$-\int_{-\infty}^{t} L_{22}(t-\tau)v(\tau)d\tau - (Mx_g + a_{62}(0))ur$$
$$+\frac{1}{2}\rho_w L^2 TC_{xyc}(\alpha_{cr})V_{cr}^2 + \frac{1}{2}\rho_a A_L LC_{xyw}(\alpha_{wr})V_{wr}^2$$
$$+F_{6wave}^{(1)} + F_{6wave}^{(2)} + T_{xy} \tag{3.6}$$

DP Controller and Filter Algorithm. To solve the control problem, the position and heading of the vessel are measured with respect to reference points. From the measured differences, thrust commands are calculated to reduce these differences. There are many linear and nonlinear control laws, such as GUAS by Grovlen (1996), GES by Fossen (1998), GAS by Loria et al. (2000), and optimal control by Chauver et al. (1998). As a different control method, Katebi et al. (1997) and Yamamoto et al. (1999) propose an H-infinite control algorithm.

In this paper, PID/LQR controller and Kalman filter are summarized as the basic control and filter algorithm.

PID Controller. A common control equation is the Proportional-Integral-Derivative (PID) control.

The required thrust can be calculated as:

$$T_x = P_x + I_x\int e_x dt + D_x\dot{e}_x \tag{3.7}$$

Where P_x: proportional gain factor

I_x: integral gain factor

D_x: derivative gain factor

And the required thrust forces and moments are obtained from:

$$\begin{bmatrix} T_x \\ T_y \\ T_{xy} \end{bmatrix} = \begin{bmatrix} D_x & P_x & 0 & 0 & 0 & 0 & I_x & 0 & 0 \\ 0 & 0 & D_y & P_y & 0 & 0 & 0 & I_y & 0 \\ 0 & 0 & 0 & 0 & D_\Psi & P_\Psi & 0 & 0 & I_\Psi \end{bmatrix} \vec{e} \tag{3.8}$$

where

$$\vec{e}^T = [\dot{e}_x, e_x, \dot{e}_y, e_y, \dot{e}_\Psi, e_\Psi, \int e_x dt, \int e_y dt, \int e_\Psi dt] \tag{3.9}$$

LQR Controller. A state variable equation is introduced as:

$$\vec{x}^T = [u, X, v, Y, r, \Psi] \tag{3.10}$$

$$M\,\dot{\vec{x}} = A\vec{x} + B\vec{u} \tag{3.11}$$

$$\vec{u}^T = [T_x, T_y, T_{xy}] \tag{3.12}$$

where $\vec{u}^T = [T_x, T_y, T_{xy}]$ is the required thrust in the x and y directions and the moment for heading.

This LQR controller is similar to the PD controller. The cost function is given by:

$$J = \frac{1}{2}\int (e^T Qe + u^T Ru)dt \tag{3.13}$$

where Q and R are weighting matrices. Increasing the Q matrix reduces the position error. To minimize the cost function, the feedback gain K matrix can be obtained and thrust can be calculated as follows:

$$\vec{u} = -K\,\vec{x} \tag{3.14}$$

Kalman Filter. The purpose of a DP system is to control the slow large motions. It is not necessary to respond to fast small motions since it is practically impossible to suppress this component by thruster action. Thus the filtering of motions is important to increase control efficiency. The Kalman filter is an estimator that minimizes error covariance under the assumption of a statistical knowledge of noise processes.

State variable $\vec{x}^T = [u, X, v, Y, r, \Psi]$ (3.15)

Feedback Input: $\vec{u}^T = [T_x, T_y, T_{xy}]$ (3.16)

System Equation: $\dot{\vec{x}} = A\vec{x} + B\vec{u} + G\vec{w}$ (3.17)

Measurement: $y = C\vec{x} + D\vec{u} + \vec{v}$ (3.18)

By using Kalman filter theory, estimated variables are given by

$$\dot{\hat{x}} = A\hat{x} + Bu + L(y - C\hat{x}) \quad (3.19)$$

where $L = PC^T R_o^{-1}$, P is the solution of the following algebraic Riccati equation.

$$A^T P + PA^T + GQ_o G^T - PC^T R_o^{-1} CP = 0 \quad (3.20)$$

where G is the matrix for disturbance input, Q_o is the covariance of the disturbance of system, $Q_o = E(ww^T)$ and R_o is the covariance of the measurement noise, $R_o = E(vv^T)$.

It is assumed that the system noises are first order wave forces and the measurement noises are a sum of first order wave motion and instrumental noises in position signal. Then Q_o and R_o can be determined as proportional to the variance of first wave forces and motion, respectively.

3.5. Model testing

One may ask why model tests need to be carried out for DP systems. Effects arising from the simultaneous action of waves, wind and current, combined with multiple thrusters working near each other and close to the hull provide significant interaction effects. Insufficient systematic data is presently available to be able to develop reliable simulation tools that incorporate all these interactions. Recent model test programs carried out for new generations of large DP drilling vessels include:

- Resistance and propulsion tests to obtain reliable speed-power predictions
- Wind tunnel tests to determine wind resistance coefficients
- Towing tests to determine the current resistance coefficients
- Thruster-hull interaction tests to determine thrust degradation due to, for instance, Coanda effects or current (Brix, 1978; Nienhuis, 1992; Wichers et al., 1998).
- Thruster-current interactions (Cozijn et al., 1999).
- Thruster-thruster interaction tests to determine the forbidden sectors of operation of thruster acting in the vicinity of other thrusters (Wichers et al., 1998).

The above tests generate the data needed for rational simulation of the total DP system. In addition to wave, wind and current tests in stationary conditions, transient tests in which fast wind or current changes occur are sometimes also investigated (Aalbers & Merchant, 1996; Wichers et al., 1998; Lee et al., 1999; Aalbers & Michel, 1998).

To fully validate the results of simulations, tests in irregular waves, wind and current with fully operating DP system including modern control algorithms and modeled thruster array are recommended.

Model tests with full DP need facilities with wind, current, and waves to provide the external forces. There are no model basins that perfectly match the ocean environment at this time because of the directionality of current and waves.

Instrument and Measurement. Model motion is measured by opto-electronic position measuring systems. These consist of onshore cameras viewing active or reflective targets onboard the model. Velocities are calculated as time derivatives of position. Wave elevation is measured by means of wave probes at various positions in the basin. Azimuth and tunnel thrusters are driven by electric motor. Propeller rate of revolution is measured by tachometers and changed by the dynamic positioning control algorithm. The azimuth angle is also commanded from the DPS algorithm.

Model test procedure.

- A basin fixed co-ordinate system is used for vessel's position tracking and a model fixed system is used for motion estimation.
- Conversion from the model to prototype values follows the Froude scaling law using scaling factors that assume potential flow, i.e. viscous scale effects on the hull due to the difference in model and ship Reynolds number are small. It is assumed that scale effect on wave drift damping can be neglected, and the relative contribution from skin friction damping will be low.
- The cross-angle between the main wave and current directions has limitation owing to the test facilities.
- Force interactions between thrust and hull, thruster and thruster are complicated and have to be well estimated to be in the same range.
- The DP control system and power distribution algorithms usually do not match the actual condition perfectly. However it is desirable that the model system be matched with the full-scale system and algorithm as closely as possible.

3.6. Installations

A ship fitted with a DPS is capable of automatically maintaining position and heading within a specified operating envelope, under a specified environmental condition. A high-grade system such as DP Class 2 or 3 has to operate under the condition of any single fault including complete loss of a compartment due to fire and flood. The grade of DPS has to be decided at the concept design stage, because up-grading later is expensive. Carter et al. (1999) shows the results of investigation of upgrading DPS-3 classification for a drill ship.

System makers and specifications. The following table shows some DP systems and a comparison of their specifications.

Table 3.1 DP Systems.

Item	*Simrad*	*Cegelec*	*Nautronics*
Dynamic Positioning System	SDP 11/12 SDP 21/22 SDP 31/32	DPS901(Simplex) DPS902(Duplex) DPS903(Triple)	ASK5001 ASK5002 ASK5003
Thruster Control System	STC	TCU900	TCS5001
Operator Console	Windows-NT Based PC	VME Bus type SPARC W/S	Windows-NT Based PC
Control Network	Ethernet	World-FIP Fieldbus	Fiber optic control network
Sensor Interface	IO-sys, serial, LAN	IO-sys, serial, LAN	IO-sys, serial, LAN

Considerations for installed DP systems. For the installation of a DPS, the classification societies require detailed information on the system. Following are items that have to be submitted to ABS for certification of DPS-3.

1) General arrangement information for the DP system
- System description
- Position reference systems and environmental monitoring system
- Location of the control system component
- DP alarm system and interconnection with the main alarm system
- Electrical generation PMS and its interconnection with the control system
- Consequence analyzer
- Electrical supply arrangement
- Thruster remote control system
- Automatic DP control and monitoring system
- Environmental force calculations and design safe operating envelope
- Details of the Thrusters
- Thruster force calculations and predicted polar plots.
- Thruster location details
- Failure Modes and Effects Analysis (FMEA)

2) Environmental Sensor and Position Reference System
- position reference system – 2 set ,
- wind sensor – 2 set,
- gyro compass – 2 set (a third independent position reference system + a third gyro compass located in the back-up control station with their signal repeated in the main control station).
- Where 3 position reference systems are required, the control computers are to use signal-processing techniques to validate the data received. When out of range data occurs an alarm is to be given.

3) A Failure Modes and Effects Analysis (FMEA)
- FMEA is to be carried out for the entire DP system. The FMEA is to be sufficiently detailed to cover all the systems major components and is to include but not be limited to the following information.

3.7. Further Research on DPS

DP-JIP (MARIN proposal)

1) Project period: Mar. 2000 – Nov. 2002
2) Contact point: A.B. Aalbers, MARIN (E-mail; a.b.aalbers@marin.nl)
3) Main aspects
- Carry out full-scale monitoring on DP shuttle tanker, drilling vessel measuring items; thruster actions, DP control setting, position, motions, waves, currents, etc.
- Development of an improved control system, using a real time estimator of the environmental forces acting on the vessel. A main element of this system is the prediction of the instantaneous wave drift forces and moment (wave-feed-forward) based on the continuous measurement of the relative wave elevation at six points around the waterline of the vessel. (Pinkster, 1978; Aalbers & Nienhuis, 1987).
- Model tests in MARIN offshore Basin for finding the correlation of the model and full-scale and the effects of using the Environmental Force Estimator on the quality of the DP control.

4) Aims of the program
- Reduction of fuel consumption on DP vessel
- Significant increase of position accuracy
- More reliable positioning in harsh weather and increased limit
- Improved on-board decision support to the DP operator

DP systems under development by Samsung Heavy Industries.

1) DP Class 3: Triple Voting DP
- All the sensors and Position measurement system send data
- To all three DP controllers and Operator stations.
- All three controllers output thruster reference to the thruster voting modules.

2) DP Controller Logic
- PID High gain (Position Error Priority)
- PID Low gain (Fuel Consumption Priority)
- Modern Control Theory (LQG Logic Available)
- Feed-Forward Logic
- Wind Feed-Forward Logic
- Wave Direction Feed-Forward Logic

Weather-vaning DP system. Full-scale tests of a weather-vaning DP system are reported. This system makes use of a CP main propeller and a single bow-thruster to maintain the position of a point on the centerline of a 50 m training vessel while leaving the heading of the vessel free to assume a weather-vaning position. Full-scale tests have shown that such a system allows the vessel to be positioned accurately using almost minimum power. By making the set-point follow a specified path in the horizontal plane, both straight lines and circles, it was shown that the vessel can also efficiently carry out dynamic tracking using this simple system.

Weather-vaning DP can be applied as a back up to conventional DP systems, in cases of failure of the thrusters at one end of the vessel, or as a means of 'Dynamic Anchoring' when it is only required to keep the vessel on station without requiring a specific heading.

4. VERY LARGE FLOATING STRUCTURES

4.1. New concepts of reduction of hydroelastic responses

The feasibility and installable sea area of stationary Very Large Floating Structures (VLFS) are improved by reducing the wave loads on the structures. So far, pontoon-type VLFSs are proposed for comparatively calmer seas such as in a bay or behind a breakwater, even though the semi-submersible-type can be installed under more severe conditions. Several concepts are proposed to reduce the wave load on VLFSs and to improve survivability without gravity type breakwaters.

Semi-submersible VLFS. Yoshida et al. (2001) presented that a 2.2 km VLFS can survive storm conditions of 12.5 m wave height and 15 second wave period. The VLFS is supported by 293 columns of 28 m length and 14 m square section with round corners. The columns are arranged in a staggered pattern. The draft of the columns is 9 m with an air-gap of 12 m. The structure is moored with 33 anchor-chain lines along each longer side and 18 lines along each shorter side.

VLFS with submerged horizontal plate floating breakwaters. Takaki et al. (2000), Kanda et al. (2000) and Fujikubo et al. (2000) presented that a 4.5 km VLFS can survive a storm condition of 10 m wave height and 12 second wave period. On the edge of this VLFS, an oscillation-diminishing horizontal plate, 30 m long is attached and a submerged, horizontal-plate, floating breakwater of 85 m length is installed in front of the VLFS. The submerged horizontal plates dissipate wave energy as follows:

As incident waves propagate into the plate, the flow due to the waves is accelerated around the fore edge of the plate, while strong reverse flow is generated at the aft edge. These two opposite flows collide with each other, and the resulting, wave breaking and wave fission dissipate the incident wave energy.

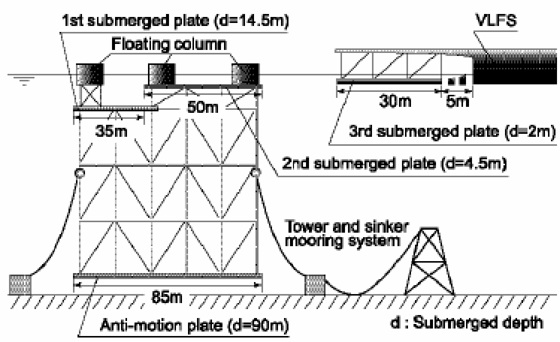

Figure 4.1 Submerged Plate VLFS.

VLFS with OWC wave energy absorption device. Maeda et al. (2000, 2001) presented that a 4.0 km VLFS with Oscillating Water Column (OWC) wave energy devices can survive a storm condition of 10 m wave height and 14 second wave period. They optimized the arrangement of OWC devices and suggested that at least three rows of OWC devices of 90 m total length on the weather side of a VLFS are required. Wave energy can be utilized in this case. Shigemitu et al. (2001) proposed a self-sufficient DUAL PORT with airport and seaport functions. They estimated monthly demand and supply of electricity when the DUAL PORT utilizes not only wave power but also wind, solar and waste power. The total supply covers the total demand in each month.

Behavior of a large air-supported Mobile Offshore Base (MOB) was also studied. In operational mode (with almost zero-forward speed) it is very similar to a VLFS with OWC type wave energy absorbing devices. Pinkster et al. (1999) confirmed excellent motion characteristics and significantly reduced midship bending moments from both model tests and numerical computations.

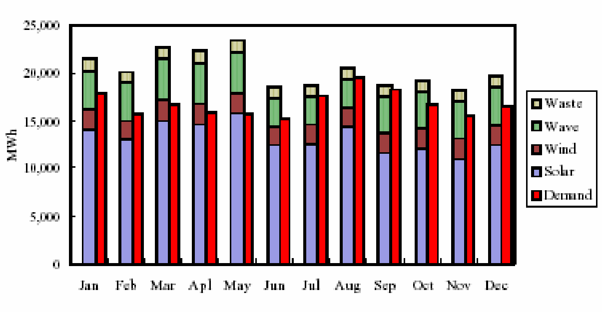

Figure 4.2 VLFS Energy supply and consumption.

4.2. Safety Evaluation of Mooring System

To satisfy functional requirements, such as for airports, the mooring of VLFS is expected to provide very little excursion due to waves and winds. Vertical pile type and jacket type dolphins have been proposed. Kato et al. (1997) showed a reliability analysis method for the mooring system against a Typhoon. A direct simulation for estimating the initial failure (first passage failure) and progressive collapse failure probabilities of multi-point mooring dolphin system of the VLFS was carried out, and its failure probability was obtained.

Drift Scenario. The breakwaters topple or are damaged due to a large typhoon, and the wave height and thus the wave load increase in the breakwater area. The increased wave and wind load induces significant long period movements in the horizontal plane, with a resultant increased mooring reaction force. A load exceeding the allowable load acts on the mooring dolphins. The mooring piles are pulled out or subjected to plastic collapse. They fall and are broken or pulled out to fail the mooring device. The failure of a single mooring device triggers a progressive collapse of other mooring devices, with eventual total failure of the mooring system and drift of the floating structure. The safety of the mooring system is evaluated assuming the severest condition or absence of breakwaters.

Definition of Failure Probability. A mooring device is failed when the mooring reaction force W, due to oscillation of the floating structure, exceeds the yield strength R. The floating structure drifts when all its mooring devices are failed. Failure of a mooring device indicates presence of an event satisfying the following condition:

$$Z_k(t) = W_k(t;X) - R_k > 0 \qquad 0 \le t \le T \qquad (4.1)$$

where X is natural condition parameters, T duration of the natural condition parameters, and R_k the random variable for the final yield strength of mooring device k, and R_k are independent of each other.

A mooring device is failed when a typhoon hits; the floating structure oscillates excessively due to low frequency tuning; and the mooring reaction force W exceeds the yield strength R. Large model experiments were conducted at the Ship Research Institute, Ministry of Land, Infrastructure and Transport (Kato et al., 1997; Saitou et al., 1997) and multi-point mooring system collapse simulations were carried out by Yoshida et al. (1998). These studies show that the mooring system collapses within the duration time of a typhoon (say three hours) once a mooring device is initially failed under severe external conditions. In favor of the safer side, we represent the failure probability of a mooring system by the probability that at least one mooring device in a multi-point mooring system is failed in a typhoon.

The probability of a multi-point mooring system being failed by strong wind and waves in a specified service life is given by the following equation:

$$P_f(T) = \int\int \mathrm{Prob}\left[\bigcup_{k=1}^{m} Z_k(t) > 0, 0 \le t \le T \,\middle|\, X = x_i, R_k = r_k\right] f_X(x_i) f_R(r_k)\, dx_i dr_k \qquad (4.2)$$

where Prob[$A|B = C$] is the probability of A under the condition of $B = C$, and $f_X(x)$ and

$f_R(r)$ are probability density functions of natural condition parameters and final yield strength of mooring device, respectively.

Using the extreme-value distribution of the annual maximum values as the distribution of natural condition parameters, we define the annual reliability as follows:

$$R(T) = 1 - P_f(T) \quad (4.3)$$

The total reliability for N years of service life is approximated by the following equation:

$$R(T) = [1 - P_f(T)]^N \quad (4.4)$$

Estimation of Failure Probability

a) Estimation of Probability of exceedance of Maximum Mooring Reaction Force

The probability of exceedance of the maximum mooring reaction force is defined in the equation below. This involves several required runs of time domain direct simulations corresponding to duration time under the natural environmental conditions x (wind velocity and wave height).

$$\mathrm{Prob}[W > w \mid x] = \frac{i_w}{n+1} \quad (4.5)$$

where $\mathrm{Prob}[W > w]$ is the probability distribution for the maximum mooring reaction force W that is greater than w, n the total number of maximum mooring reaction forces W, and i_w is the number of the maximum mooring reaction forces that are greater than w.

b) Estimation of Conditional Failure Probability

We read the point where the final yield strength $w = R$ from equation (4.5). Alternatively, the probability of exceedance of the maximum mooring reaction forces greater than the constant reaction force is approximated and extrapolated using a Weibull distribution, to derive position of $w = R$.

c) Estimation of Failure Probability

We estimate the failure probability from equation (4.2) by approximating the conditional failure probability with a Weibull distribution and using the product of this approximation and the extreme-value distribution of natural environmental condition parameters.

Direct Simulation

a) Examples of Calculation

Direct simulations were carried out for a mat-type VLFS of 4770 m length assuming JONSWAP type long crested irregular waves of significant wave expressed by Wilson IV and Ochi-Shin type wind as shown in Table 4.1.

Table 4.1 Analysis Conditions.

Site		20 m water depth area in Tokyo bay
Natural Environmental Condition	Wind and wave direction	90°,120°, where wind and wave direction are identical
	Spectrum Type	Fluctuating wind: Ochi-Shin, waves: long crested JONSWAP
	Fluctuating wind Spatial correlation factor	Cross to mainstream: $k_1 = 7.0$ Mainstream: $k_2 = 5.13$
	Duration	3 hours
	Fetch	20 km
	Significant wave and period	Estimation from significant wave method (Wilson IV)
Floating structure	Dimensions	4770×1714×6×2 m (L×B×D×d)
Mooring	Dolphin	Short side: 20 units, Long side: 50 units Regularly-interval configuration of Combined pile dolphin units Design load: 1000 tonf Initial yield load: 2860 tonf (Final resistance)
	Fender	Cellular Fender: SUC3000RH (Constant reaction force: 550 tonf)

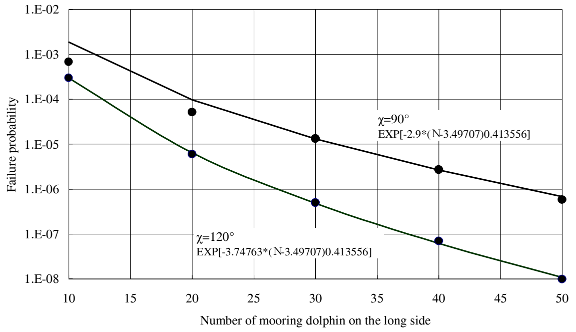

Figure 4.3 Variation of failure probability to a number of mooring dolphins on the long side.

The number of mooring devices on the long and short side was reduced at the same rate and the devices were placed at equal distances to calculate the annual failure probability. The result is shown in Figure 4.3. The lateral axis is the number of mooring devices on the long side. The solid line is the Weibull fit curve for respective cases. A 20% change in the number of mooring devices results in a ten times change in the failure probability. The failure probability increases when the incident angle is 90 degrees.

5. SHALLO WATER ISSUES

In recent years, basins have been required to model systems in shallow water to asses the response of moored systems at limiting conditions. Offshore systems in shallow water are commonly used for offloading or near-shore terminals or extensions of land based facilities. In many cases these involve a moored buoy with a vessel or vessels attached. These systems are in wide use worldwide.

Analysis of the environmental loads on such systems has been difficult because the shallow conditions modify the wave environment in such a way as to render linear numerical models invalid. Model tests have been employed in correctly scaled water depth as a means of assessing the loads and response of such systems. Such models are relatively easy to set up in model basins because of their physical size but suffer from problems of highly non-linear environments and responses, which are less evident in deep water tests.

As a result we have identified a number of problems in environment specification and in the measurement of responses. Although there are methods of dealing with these problems, there is insufficient information on which to base standardization derived from committee consensus. Following is a short description of each of these problems:

<u>Wave height selection</u>. It has been common practice to select the design wave height based on open ocean conditions. The problem then arises in the wave basin that the selected wave height cannot be generated in the specified water depth due to wave breaking arising from depth effects. We believe that some work to establish limiting wave conditions for shallow water conditions is justified.

<u>Spectral Shape</u>. Spectral shape is also modified by depth effects. Open ocean spectra such as the JONSWAP are frequently specified for shallow locations. Despite the ability to model limited fetch conditions, the spectra do not account for limited depth. The TMA spectral formulation (Hughes, 1984), which includes the effect of depth and converges to the JONSWAP for infinite depth provides a better representation for shallow water. However this spectral formulation is not widely known in the client community and some work in evaluating this (or another shallow water) spectrum would be beneficial.

<u>Non-linearity of response</u>. Buoy responses, in shallow water, are not well predicted using linear numerical models. Because these models are frequently used as a check on model testing data, there are significant discrepancies between numerical and tank models. The committee could evaluate the causes of such discrepancies, some of which are related to wave effects in the incident environment and some of which may be related to pressure ef-

fects on the responding body. Such an evaluation could demonstrate the expected deviations from linear theory to be expected in model testing and point the way for improvements in numerical models.

Mooring modeling. Shallow water catenary mooring models are more difficult than deeper water cases because there is usually a rapid transition from weight effects to axial tension effects that increase the non-linearity of the system. This requires accurate models of the weight distribution and more importantly a rational scheme for modeling the elasticity and the distribution of elasticity in the mooring elements. This aspect of the models would benefit from an analysis of the current state of the art and a recommendation on best available methodology.

Although two of these problems are related to wave generation, the ultimate effect is in the response of the moored system, which is, in our experience, quite different from an equivalent moderate or deep-water case. Because most of these effects are associated with non-linearities in the driving wave environment or in the response of the system, the problems do not lend themselves well to conventional numerical models.

6. CONCLUSIONS

The offshore oil industry is moving into deeper water. FPSO, semi-submersible, TLP and Spar platform have been developed for deep applications and several new systems are under study for the near future. Hybrid model testing, using a number of possible approaches, such as a truncated model using a passive mooring, or other alternatives are proposed for very deep water cases.

Physical and numerical modeling of vortex induced vibrations of risers are important topics for deepwater applications. Space and time variability in deep water current profiles will influence the design of the risers. Currents are shown to vary in intensity and in direction.

Dynamic positioning is presently applied in many activities offshore. In the DP system design process, we first make assessment of mean environment forces acting on the vessel, and then the first estimate of thruster arrangement and installed power can be made. The next step is to simulate the dynamic effects of the environmental forces including the vessel, control algorithm and thruster characteristics. Finally, in an overall validation of the performance, through model tests is recommended.

Very large floating structures have been extensively studied in the USA and far east countries. New concepts for reduction of hydroelastic responses, such as semi-submersible, VLFS with submerged horizontal plate floating breakwaters and VLFS with OWC type wave energy absorbing devices are currently being studied. A Safety evaluation method for VLFS is proposed by using a direct simulation and model tests for the initial failure and for progressive collapse failure probabilities of multi-point mooring dolphin system.

7. RECOMMENDATIONS TO THE CONFERENCE

Adopt the Procedure “Hybrid Mooring Simulation Model Test Experiments” 7.5-02-07-03.4.

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The Specialist Committee on Stationary Floating Systems

Committee Chair: Prof. Takeshi Kinoshita (University of Tokyo)
Session Chair: Ir. George F.M. Remery (MARIN)

I. DISCUSSIONS

I.1. Discussion on the Report of the 23rd ITTC Specialist Committee on Stationary Floating Systems: Procedures for hybrid testing

By: R.H.M. Huijsmans, MARIN, The Netherlands

The recommendation of the committee is to evaluate the procedures for hybrid testing.

The aim of such a procedure is to describe a rational and accurate way to do model testing for water depth beyond what is reasonably possible in today's model test basins. An alternative method for hybrid testing is under review by an international consortium under the DEEPSTAR project. The alternative approach is sketched as follows.

Model testing is performed on a 'truncated' mooring system for limited water depth such that the horizontal and vertical restoring characteristics of the truncated mooring system are representative for the real deep water situation.

A computer simulation programme is then tuned such that the calculated floater motion response and loads in the mooring/risers in the truncated system are as closely as possible to the model test results.

After tuning the computer simulations, the final design is calculated for the mooring system at the full water depth.

The new committee should be advised to study the advantages and disadvantages of the procedure described by the committee and the procedure investigated by the DEEPSTAR consortium.

I.2. Discussion on the Report of the 23rd ITTC Specialist Committee on Stationary Floating Systems: Hybrid modelling

By: Martin R. Renilson, QinetiQ, United Kingdom

First I would like to thank the Committee for a very interesting and useful report.

I was particularly interested in the comments on hybrid modelling. The report appears to imply that this is a fully understood technique, which is acceptable and should be used by all member organizations, and that no further work in this area is required. Is this true, and if not can the Committee spell out what work still needs to be done to give member organisations confidence with this technique?

Also, although wind energy devices are mentioned briefly under the heading of VLFS, there is not very much about this topic, which I believe will become very important to the ITTC in the future. Does the Committee think this should be addressed in more detail in the future?

II. COMMITTEE REPLIES

The Committee thanks both Dr. Renilson and Dr. Huijsmans for their comments and the opportunity to clarify some points.

II.1. Reply of the 23rd ITTC Specialist Committee on Stationary Floating Systems to R.H.M. Huijsmans

Dr. Huijsmans identifies a Hybrid method, under study by DeepStar, which is similar to one of the approaches identified in our report. We would be very pleased to be able to evaluate the results of the DeepStar study with a view towards improving general understanding in constructing Hybrid Models.

Areas of particular interest would be the ability to correctly model both horizontal and vertical forces in the "truncated " mooring and the issue of how well computer simulations can be made to match the physical calibration experiments. Tuning computer models may provide reasonable interpolation but is less certain for extrapolation. This work will have to be undertaken by the 24th ITTC and hopefully the appropriate committees will be able to incorporate the outcomes of the DeepStar work.

II.2. Reply of the 23rd ITTC Specialist Committee on Stationary Floating Systems to M.R. Renilson

The point, raised by Dr. Renilson, that we have implied that Hybrid Modeling is a fully understood technique might have been drawn from the fact that we presented a Draft Standard Procedure. A recommended procedure was requested in our mandate and we felt that it provided a good summary of the current state of the art. However, we did not intend to imply that the subject is mature – in fact quite the opposite. There are currently many different approaches. The Procedure is written around, what we judged to be, the best of current practice. The primary area that requires further development is adequate representation of the full dynamics of mooring line response in the hybrid model.

The committee would not propose that Floating Wind Energy devices should be addressed except in the sense that this type of structure may impose unique experimentation or modeling requirements. To date we are not aware of any reports of particular issues, although they undoubtedly demand good models of combined wind and wave environments. Combined wind-wave modeling has been identified as an issue for the ITTC and may become a more important issue for future ITTC consideration, specifically related to such structures.