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RECOMMENDED PRACTICE  
DNV-RP-F205

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GLOBAL PERFORMANCE ANALYSIS  
OF DEEPWATER FLOATING  
STRUCTURES

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OCTOBER 2010

DET NORSKE VERITAS

# FOREWORD

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## CHANGES

- **General**

As of October 2010 all DNV service documents are primarily published electronically.

In order to ensure a practical transition from the “print” scheme to the “electronic” scheme, all documents having incorporated amendments and corrections more recent than the date of the latest printed issue, have been given the date October 2010.

An overview of DNV service documents, their update status and historical “amendments and corrections” may be found through [http://www.dnv.com/resources/rules\\_standards/](http://www.dnv.com/resources/rules_standards/).

- **Main changes**

Since the previous edition (October 2004), this document has been amended, most recently in April 2009. All changes have been incorporated and a new date (October 2010) has been given as explained under “General”.



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## 1. Introduction

### 1.1 General

A deepwater floating system is an integrated dynamic system of a floater, risers and moorings responding to wind, wave and current loadings in a complex way. The floater motions in shallow water are to a large extent excited and damped by fluid forces on the floater itself. As the water depth increases the interaction/coupling between the slender structures and the large volume floater becomes more important. In this case, a coupled analysis is required to capture the interaction between the two in order to accurately predict the individual responses of floater, risers and mooring. Coupled analysis is now being used by the industry in the design of deepwater floating systems.

In Section 2, definitions of some key terms related to coupled analysis are provided, and the main concepts and characteristics of various floater types and slender structure types are summarised. This is to provide basic understanding of the various floating systems, which is crucial in selecting a coupled analysis strategy and the important input parameters.

Section 3 gives an overview of floater load models and Section 4 gives an overview of load models for mooring and risers. Section 5 describes the traditional decoupled analysis, while Section 6 defines coupled analysis and describes efficient analysis strategies.

### 1.2 Objective

The objective of this document is to provide practical guidance on the key issues in coupled analysis and on how to efficiently perform the analysis.

### 1.3 Scope and application

The Recommended Practice covers the following aspects

- response characteristics of different floating systems
- definitions of ‘coupling effects’, ‘decoupled analysis’ and ‘coupled analysis’
- load models for floater and slender structures
- coupling effects from slender structures to floaters
- necessary input parameters in coupled analysis
- how to efficiently perform coupled analyses.

### 1.4 Relationship to other Rules

This document formally supports and complies with the DNV Offshore Standard “Dynamic Risers”, DNV-OS-F201 and is considered to be a supplement to relevant National Rules and Regulations.

This document is supported by other DNV offshore codes as follows:

- Offshore Standard DNV-OS-C102 “Structural Design of Offshore Ships”.
- Offshore Standard DNV-OS-C103 “Structural Design of Column Stabilised Units”.
- Recommended Practice DNV-RP-C103 “Column Stabilised Units”.
- Offshore Standard DNV-OS-C105 “Structural Design of TLPs”.
- Offshore Standard DNV-OS-C106 “Structural Design of Deep Draught Floating Units”.
- Recommended Practice DNV-RP-C205 “Environmental Conditions and Environmental Loads”.
- Offshore Standard DNV-OS-E301 “Position Mooring” .

Other references:

- Norsok Standard N-003 “Actions and action effects”

## 1.5 Abbreviations

For purposes of this recommended practice, the following abbreviations apply.

CFD	Computational Fluid Dynamics
DOF	Degrees of Freedom
DDF	Deep Draught Floater
DTU	Dry Tree Unit
FE	Finite Element
FD	Frequency Domain
FPSO	Floating Production Storage and Offloading
FTL	Fluid Transfer Lines
GM <sub>L</sub>	Metacentric Height, Longitudinal
GM <sub>T</sub>	Metacentric Height, Transverse
HF	High Frequency
LF	Low Frequency
LTF	Linear Transfer Function
OOL	Oil Offloading Line
QTF	Quadratic Transfer Function
RAO	Response Amplitude Operator
SCR	Steel Catenary Riser
SSVR	Spar Supported Vertical Risers
TD	Time Domain
TLP	Tension Leg Platform
TTR	Top Tensioned Riser
VIM	Vortex Induced Motions
VIV	Vortex Induced Vibrations
WF	Wave Frequency

## 2. Key Definitions and Characteristics of Deepwater Floating Systems

### 2.1 Definitions

For purposes of this recommended practice, the following definitions apply.

#### 2.1.1 Motion time scales

A floating, moored structure may respond to wind, waves and current with motions on three different time scales, wave frequency motions (WF), low frequency motions (LF) and high frequency motions (HF). The largest wave loads on offshore structures take place at the same frequencies as the waves, causing wave frequency (WF) motions of the structure. To avoid large resonant effects, offshore structures and their mooring systems are often designed in such a way that the resonant frequencies are shifted well outside the wave frequency range. Natural periods in surge, sway and yaw are typically more than 100 seconds. Natural periods in heave, roll and pitch of semi-submersibles are usually above 20 seconds. On the other hand, for a tension leg platform (TLP), these natural periods are below 5 seconds where there is little wave energy. Due to non-linear load effects, some responses always appear at the natural frequencies. Slowly varying wave and wind loads give rise to low-frequency (LF) resonant horizontal motions, also named slow-drift motions. Higher-order wave loads yield high frequency (HF) resonant vertical motions, springing and ringing, of tensioned buoyant platforms like TLPs and slender gravity based structures (GBS).

#### 2.1.2 Coupling effects

Coupling effects refer to the influence on the floater mean position and dynamic response from slender structure restoring, damping and inertia forces. These force contributions are elaborated as follows.

Restoring:

- 1) Static restoring force from the mooring and riser system as a function of floater offset
- 2) Current loading and its effects on the restoring force of the

- mooring and riser system
- 3) Seafloor friction (if mooring lines and/or risers have bottom contact)

Damping:

- 4) Damping from mooring and riser system due to dynamics, current, etc.
- 5) Friction forces due to hull/riser contact.

Inertia:

- 6) Additional inertia forces due to the mooring and riser system.

In a traditional de-coupled analysis, item 1) can be accurately accounted for. Items 2), 4) and 6) may be approximated. Generally, items 3) and 5) cannot be accounted for. A coupled analysis as described previously can include consistent treatment of all these effects.

### 2.1.3 De-coupled analysis

In a de-coupled analysis the equations of the rigid body floater motions are solved in time domain, but the effects of the mooring and riser system are included quasi-statically using non-linear springs, i.e. quasi-static restoring force characteristics. All other coupling effects, e.g. contributions from damping and current loading on the slender structures, need to be given as input to the analysis based on a separate assessment.

### 2.1.4 Coupled analysis

In a coupled analysis the complete system of equations accounting for the rigid body model of the floater as well as the slender body model for the risers and mooring lines are solved simultaneously using a non-linear time domain approach for dynamic analyses. Dynamic equilibrium is obtained at each time step ensuring consistent treatment of the floater/slender structure coupling effects. The coupling effects are automatically included in the analysis scheme.

## 2.2 Main characteristics of floaters

A common feature of all types of floaters is that they utilise excess buoyancy to support deck payload and provide slender structure tensions. Depending on the area and the sea state, ocean waves contain 1<sup>st</sup> harmonic wave energy in the period range of 5 - 25 s. For a floating unit the natural periods of motions are key features and in many ways reflect the design philosophy. Typical motion natural periods of different floaters are presented in Table 2-1.

Floater Mode	Natural periods (seconds)			
	FPSO	DDF	TLP	Semi
Surge	> 100	> 100	> 100	> 100
Sway	> 100	> 100	> 100	> 100
Heave	5 – 12	20 – 35	< 5	20 – 50
Roll	5 – 30	50 – 90	< 5	30 – 60
Pitch	5 – 12	50 – 90	< 5	30 – 60
Yaw	> 100	> 100	> 100	> 100

A common characteristic of all floater types is that they are “soft” in the horizontal plane, with surge, sway and yaw periods generally longer than 100s. The fundamental differences among the floaters are related to their motions in the vertical plane, i.e. heave, roll and pitch. The floater motions in the vertical plane are decisive for the choice of riser and mooring systems.

### 2.2.1 FPSO response characteristics

A floating production storage and offloading unit, FPSO, can

be relocated, but is generally positioned at the same location for a prolonged period of time. The unit normally consists of a ship hull, with turret, and production and drilling equipment on deck. For FPSOs, due to their large superstructures and their active or passive weather-vaning ability, wind forces are often dominant relative to current forces. FPSOs normally experience significant LF response in the horizontal plane. They may be particularly sensitive to surge excitation due to the low viscous hull damping. This sensitivity is reduced with increasing water depth since the damping contributions from mooring lines and risers increase.

FPSOs are flexible with respect to selection of deep water mooring systems. For catenary mooring systems, the WF motions can introduce dynamic mooring forces, which tend to increase in deep water due to larger transverse drag forces. Taut mooring systems are not subjected to the same level of transverse motions, thus acting 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. Fishtailing is the unstable coupled yaw and sway motions excited by wind and current. It is associated with the horizontal stiffness of the mooring system. For riser systems, flexible risers and compliant metallic risers are usually applied due to the significant WF motions.

FPSOs may have one or several moonpools, and the water motion in the moonpool can influence the vessel motions. Viscous damping has a strong influence on this water motion. Slamming and green water on deck are other non-linear effects that may influence FPSO response in rough weather.

Combination of wind generated waves and swell with different headings are a challenge and must be taken into consideration. This applies to turret moored vessels as well as vessels with spread mooring. A critical condition is the combination of head sea and beam swell. Significant roll accelerations may occur and thus have impact on topside structure and equipment, riser system and mooring system etc.

Selection of proper roll damping is important in the prediction of FPSO responses.

Floating systems involving multiple floaters have been designed and installed. A typical field architecture may consist of a spread-moored FPSO and a dry tree unit (DTU), e.g. Spar, TLP or barge, connected by fluid transfer lines (FTLs). The offloading system (e.g. CALM buoy) can be a few kilometres away from the FPSO and connected to the FPSO through oil offloading lines (OOLs). These complex multi-floater systems bring additional challenges to both model testing and numerical analyses. From the analysis point of view, the following issues are of importance:

- consistency in phasing of waves and loads
- wind-generated waves, swell and current with different headings
- additional coupling effects due to FTLs and OOLs
- possible hydrodynamic interactions between floaters.

If the two floaters (FPSO and DTU) are close enough to each other, hydrodynamic interactions related to wave effects can be of importance. This requires a hydrodynamic analysis of the two floaters as an integrated system with 12 degrees of freedom using diffraction/radiation theory.

All the above effects may be included in a computer simulation program designed for multiple floaters and their associated slender structures.

### 2.2.2 TLP response characteristics

A TLP differs fundamentally from other floater concepts in the sense that it is the tendon stiffness rather than the waterplane stiffness that governs the vertical motions. The TLP is a soft spring in surge, sway and yaw motions, but stiff in heave, roll and pitch motions.

A TLP generally experiences WF motions in the horizontal plane that are of the same order of magnitude as those of a semi-submersible of comparable size. In the vertical plane, however, the TLP will behave more like a fixed structure with practically no WF motion response. WF forces are directly counteracted by the tendon stiffness forces.

Higher order sum-frequency wave forces may introduce springing or/and ringing responses in the vertical modes. These effects may give significant contributions to the tether responses.

Set-down is the kinematic coupling between the horizontal surge/sway motions and the vertical heave motions. Set-down is important in the calculation of airgap, tether forces and riser system responses such as stroke.

The TLP riser system typically consists of top tensioned risers, flexible risers or compliant metallic risers such as steel catenary risers.

### 2.2.3 DDF response characteristics

A *Deep Draught Floater (DDF)* is characterised by small heave motions. An example of a DDF is a Spar platform. The main hull of a Spar is a cylinder with a central moonpool for a riser system in tension. The hard tank provides buoyancy and the part below may consist of a shell structure (Classic Spar), or a truss structure (Truss Spar) with a soft tank at the keel and added mass/damping plates in between. The Spar has a large area exposed to current forces, which is usually the dominant environmental load. LF vortex induced motions (VIM) may increase the effective drag leading to even higher mean current forces. By adding strakes on the Spar hull, the vortex induced cross-flow oscillation can be reduced by considerable amount. However, the strakes will increase the added mass and the drag forces on the Spar.

The small heave motions of a DDF allows the use of rigid top-tensioned vertical risers. The riser tension is normally provided by either air cans attached to the upper part of the risers, or by tensioners integrated to the hull. Spars using air can supported risers are characterized by having free modes of motion only. Their heave natural period is usually above the range of wave periods. Spars with tensioner supported risers experience greater coupling in heave, since the heave restoring and heave eigenperiod are influenced by the riser system. This means that a heave damping assessment is crucial for the prediction of the Spar heave response.

Current fluctuations may induce significant excitation forces on a DDF. Depth correlation is a central issue when determining the level of such excitation.

Air-gap and moonpool effects should be considered for Spar analysis and design.

Due to low WF motions, a DDF is generally not subjected to large dynamic mooring line forces. This has to be evaluated in relation to the actual location of the fairleads and the increase in horizontal WF motion towards the waterline.

### 2.2.4 Semi-submersible response characteristics

A *semi-submersible* is usually a column-stabilized unit, which consists of a deck structure with large diameter support columns attached to submerged pontoons. The pontoons may be ring pontoons, twin pontoons or multi-footing arrangement.

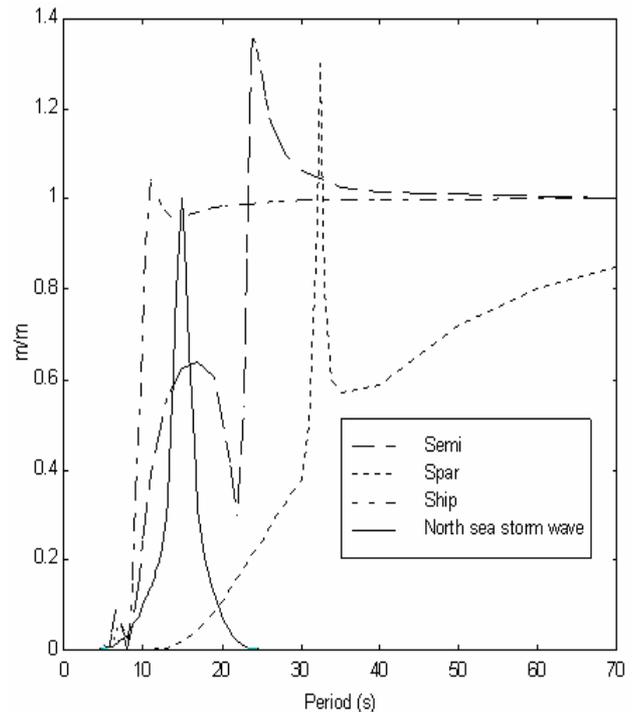
Semi-submersibles have small waterplane areas, which give natural periods (in vertical modes) slightly above 20 seconds, usually outside the range of wave periods except for extreme sea states. This implies that a semi-submersible has small vertical motions compared to a monohull floater. However, its behaviour in extreme weather requires flexible, compliant metallic riser systems or a hybrid arrangement for this concept.

A semi-submersible may be equipped with a variety of mooring systems similar to a FPSO.

The semi-submersible is very sensitive to weight changes; i.e. it has low flexibility with respect to deck load and oil storage.

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

The semi-submersible is characterized by having free modes of motion only, which means that all natural periods are above the range of natural wave periods, see Table 2-1. Despite this fact, the wave frequency motions are not insignificant, especially in extreme conditions, as indicated in Figure 2-1.



**Figure 2-1**  
Heave transfer functions for different floaters and storm wave spectrum

Large semi-submersibles with displacement of 100000 tonnes or more are generally less sensitive to WF action. LF responses may be more dominating in roll and pitch motions.

Wave impact underneath the deck due to insufficient air-gap may influence the global motions and local structural responses for semi-submersibles.

Catenary moored semi-submersibles may experience significant dynamic mooring forces due to WF responses similar to those of a FPSO.

## 2.3 Main characteristics of slender structures

### 2.3.1 Mooring systems

Mooring systems are compliant systems. They provide resistance to environmental loading by deforming and activating reaction forces. Mooring systems work as spring mechanisms where displacement of the floater from a neutral equilibrium position causes a restoring force to react to the applied loading. The tension spring effect of mooring lines derives from two mechanisms:

- hanging catenary effect – from gravity acting vertically on the line
- line elastic effect – from elastic stretch over the length of the line.

Mooring systems with these two mechanisms are called catenary moorings and taut moorings, respectively.

### 2.3.1.1 Catenary moorings

Catenary moorings are defined by standard catenary formulations, which relate the following parameters: submerged weight of the suspended lines, horizontal mooring load, line tension and line slope at fairlead. The compliance to allow for wave-induced floater motions is ensured by a combination of geometrical change and axial elasticity of the lines. The large line geometrical changes make catenary mooring systems subject to significant dynamic effects due to transverse drag load. The mooring lines in catenary mooring systems are commonly composed of steel rope and chain segments. Sometimes clump weights and buoys are used to achieve the desired line configurations.

### 2.3.1.2 Taut moorings

In a taut mooring system the lines are nearly straight between the anchor and fairlead. The vertical forces are taken up as anchor and vessel reactions directly. The compliance to allow for wave-induced floater motions is provided mainly by line elasticity.

The transverse geometric changes in taut mooring systems are not as large as in catenary systems, thus dynamic effects due to transverse drag loads are moderate.

Synthetic ropes have recently been proposed and used as mooring lines in a taut mooring system to provide required elasticity and low weight. Compared to steel, synthetic ropes exhibit more complex stiffness characteristics (e.g. hysteresis), which may induce important dynamic effects.

### 2.3.1.3 Tendons

TLP tendons bear much similarity to the mooring lines in a taut mooring system. However, the fundamental difference is that TLP tendons are usually made of large dimension steel tubes that are hardly compliant in the axial direction. The TLP system acts as an inverted pendulum. The station-keeping forces are governed by tendon length and the pretension. Tethers made of composite material are presently being qualified and will extend the use of TLPs into even deeper waters.

## 2.3.2 Riser systems

Depending on the mechanism of how floater motions are absorbed by the riser system, the risers can be divided into the following three categories:

- top tensioned risers
- compliant risers
- hybrid risers.

They are described in the following three sections.

### 2.3.2.1 Top tensioned risers

Vertical risers supported by top tension in combination with boundary conditions that allows for relative riser/floater motions in the vertical direction are referred to as top tensioned risers (TTRs). A TTR is normally constrained to follow the horizontal floater motions at one or several locations. Ideally, the applied top tension should maintain a constant target value regardless of the floater motions. Hence, the effective tension distribution along the riser is mainly governed by functional loading due to the applied top tension and the effective weight. The relative riser/floater motion in vertical direction is commonly termed stroke. Applied top tension and stroke capacity are the essential design parameters governing the mechanical behaviour as well as the application range. For floaters with rather small heave motions such as TLPs, Spar platforms, deep draught floaters and semi-submersibles, TTRs can be an attractive riser solution.

TTRs operated from semi-submersibles and TLPs are

equipped with a separate hydraulic heave compensation system (i.e. tensioner) to account for the floater motions and at the same time maintain a constant target value for the applied top tension. Bending moments are mainly induced by horizontal floater motions and transverse loading due to current and wave action. A pronounced peak in the bending moment distribution is normally seen close to the wave zone.

Recently, Spar Supported Vertical Risers (SSVR) have been proposed and designed for Spar platforms. Top tensions applied to the SSVRs are provided by tensioners on the Spar.

An alternative solution for providing top tension to Spar risers is by means of buoyancy modules (air cans) attached along the upper part of the riser inside the moonpool. Several supports may be placed along the riser system to constrain riser transverse motions. Except for the friction forces there are no constraints in riser longitudinal motions. This allows the riser system to move vertically relative to the Spar hull. Bending moments in risers operated from a Classic Spar are mainly due to the resulting horizontal hull motions as well as hydrodynamic loading from the entrapped water in the moonpool. Pronounced peaks in the bending moment distribution are normally found at the support locations.

The static and dynamic behaviour of top tensioned risers is largely governed by the applied top tension. The effective weight of the riser system defines the lower bound for the applied top tension to avoid compression in the riser at static position. Moreover, a significant higher top tension must be applied to account for imperfect tensioner arrangements and allow for redundancy in case of partial loss of top tension. Increased top tension can also be applied to reduce the probability of collision in riser arrays and limit the mean angles in bottom of the risers. The applied top tension is commonly specified in terms of excess over the effective weight of the riser system, and referred to as overpull. The required overpull is system dependent with a typical range of 30-60%.

Steel pipes have traditionally been applied for floaters in modest water depths. With attached buoyancy modules, steel risers may be applied for deep water floaters. Titanium and composite risers are suggested for deep water applications in order to keep the top tension requirement at an acceptable level.

The cross-sectional composition depends on the functional applications. Export, import and low pressure drilling risers are normally single tubular risers. Multi-tube cross-sections are typically found in high-pressure drilling and workover risers as well as production risers.

Taper joints, flex-joints or ball-joints may be applied to reduce bending stresses at the riser termination at seafloor. Flex-joint or ball-joint may be applied to reduce bending stresses at riser termination at floater. Taper joint may also be applied at the keel of Spar and other deep draught floaters.

### 2.3.2.2 Compliant riser systems

Compliant riser configurations are designed to absorb floater motions by change of geometry, without the use of heave compensation systems. The required system flexibility is normally obtained by arranging non-bonded flexible pipes in one of the following 'classical' compliant riser configurations; steep S, lazy S, steep wave, lazy wave, pliant wave or free hanging (catenary).

Such solutions will for conventional water depths require a pipe with large capacity regarding tensile loading and external/internal pressure combined with low bending stiffness and low critical radius of curvature, e.g. high 'volume' stiffness combined with high bending flexibility.

The desired cross sectional properties are normally obtained by the introduction of a flexible layered pipe where each layer has a dedicated function. The number of layers and properties of each layer are selected to meet the design requirements and are hence tailor-made for each actual installation. The vast

majority of flexible pipe designs are non-bonded allowing for relative motions between the layers.

In deep water, it is also possible to arrange metallic pipes in compliant riser configurations. Steel Catenary Risers (SCR) have been installed in the Gulf of Mexico as well as Brazilian fields (see e.g. Phifer *et al* 1994). Steel and titanium risers in Lazy Wave configurations have been proposed for semi-submersibles and TLPs in deep water. A Lazy Wave configuration with increased horizontal extension termed Long Wave is proposed for the application of metallic risers for deep water FPSO in North sea conditions (Karunakaran *et al* 1996). In such applications it may also be considered to apply pre-bent pipe sections to reduce the dynamic curvature at critical locations along the riser, i.e. hog and sag bends. Single pipe cross-sections are typically applied for compliant riser configurations.

Compliant riser systems will in general experience significantly larger static and dynamic excursions when compared to top tensioned risers. The floater motion characteristics will in many situations be decisive for the dynamic tension and moment variation along the riser, e.g. TLPs, Semi-submersibles and ships. Environmental load effects will consequently also be of greater concern for compliant configurations. Critical locations on compliant risers are typically the wave zone, hog-and sag bends, touch down area at seafloor and at the terminations to rigid structures.

Termination to rigid structures are an essential design issue for compliant riser configurations. Possible solutions are carefully designed bend stiffener, ball joint or flex joint. The primary design requirement is to limit bending curvature and pipe stresses. The secondary design requirement is to minimise forces on the supporting structures.

#### 2.3.2.3 Hybrid riser systems

There is significant potential for hybrid riser configurations, combining the properties of tensioned and compliant risers in an efficient way. Most proposed designs are based on combining a self-supported vertical riser column, i.e. tensioned riser, with a flexible riser at upper end for connection to the floater.

The vertical column is normally governed by a bundle of steel risers. Control umbilicals may also be integrated in the bundle. A buoyancy module at the upper end provides the required tension in the riser column. The upper end of the vertical column is connected to the support floater by several flexible risers.

A major advantage of such designs is that the vertical column is a self-supporting structure. The system can be designed to withstand significant dynamic floater motions since flexible risers are used for connecting the floater to the riser column.

However, hybrid riser systems tend to be quite complex structures with special design challenges. Prediction of the column response in severe current conditions requires careful evaluation of the hydrodynamic coefficients for the riser bundle. Evaluation of possible VIV response of the individual tubular in the riser bundle must also be conducted.

A special design issue for such systems is the control of the horizontal floater position relative to the upper column end to avoid excessive loading in the flexible risers. Integrity of the subsea buoyancy module is another vital design issue.

#### 2.3.2.4 Fluid transfer lines

Floating/submerged pipes used for transportation of fluids between two floaters are known as Fluid Transfer Lines (FTLs). FTLs are normally low-pressure flexible pipes or hoses. However, use of metallic FTLs has also been proposed. Buoyancy modules may be applied to achieve a desired configuration for floating as well as submerged FTLs.

Analyses need to be performed to ensure that FTLs can operate safely within defined operational conditions and withstand extreme environmental loading in disconnected conditions

without significant damage. To operate permanently, FTLs need to comply with design requirements for risers.

Load effect analyses of FTLs can be challenging. This is particularly the case for floating FTLs, which are highly compliant due to low effective tension. Furthermore, special load models are required to describe variable drag and added mass of such systems as the pipe moves in and out of the water when exposed to loading from waves and floater motions. Simultaneous excitation from floater motions at both ends is required for consistent load effect assessment for rather short FTLs. The critical areas for excessive bending/curvature will normally be close to the floater attachments.

#### 2.3.2.5 Umbilicals

Umbilicals will normally have complex cross-sectional designs displaying pronounced nonlinear stiffness characteristics, e.g. moment/curvature hysteresis. Umbilicals may be arranged in the classic compliant riser configurations or clamped to a compliant or top tensioned riser. The latter solution is commonly termed 'piggy-back' and will require special modelling considerations in the global load effect analyses, e.g. evaluation of hydrodynamic coefficients and stiffness properties for a double symmetric cross-section. Umbilicals are otherwise treated similar to compliant riser systems in the global load effect analysis.

### 2.3.3 Slender structure nonlinearities

Despite the differences in design, function and application areas for the slender structures discussed in the previous sections (top tensioned riser, compliant risers, fluid transfer lines and mooring lines/cables), physical behaviour and governing parameters for the response characteristics are quite similar. Such structures are commonly also termed as tensioned structures to reflect that the effective tension is the overall governing parameter for the global configuration, i.e. geometry, and transverse stiffness. A common overall analysis framework can be applied in load effect analyses of slender structures.

Mooring lines and cable/chain systems are not influenced by bending stiffness. The other systems have a physical bending stiffness that should be considered in the load effect analyses. Understanding the important non-linearities of slender structures is critical for system modelling as well as selection of adequate global analysis approach. Non-linearities will also be decisive for the statistical response characteristics for systems exposed to irregular loading. An essential issue is how non-linear properties of the slender structure and hydrodynamic loading mechanisms transform the wave frequency Gaussian excitation, i.e. waves and 1st order floater motions into non-Gaussian system responses. Important non-linearities to be carefully considered can be summarised as:

- 1) Geometric stiffness, i.e. contribution from effective tension to transverse stiffness. Tension variation is hence a non-linear effect for slender structures.
- 2) Hydrodynamic loading. Non-linearities are introduced by the quadratic drag term in the Morison equation expressed by the relative structure-fluid velocity and by integration of hydrodynamic loading to actual surface elevation.
- 3) Large rotations in 3D space. This is relevant for systems with bending stiffness undergoing two-axial bending.
- 4) Material and component non-linearities.
- 5) Contact problems in terms of seafloor contact and hull/slender structure contact (varying location of contact point and friction forces).

The relative importance of these non-linearities is strongly system and excitation dependent. Non-linearities due to item 1) and 2) will, at least to some extent, always be present. Item 3) is relevant for systems with bending stiffness undergoing two-axial bending due to in-plane and out of plane excitation, while

4) and 5) are more system specific non-linear effects. Material non-linearities are important for flexible risers and umbilicals, e.g. hysteretic bending moment/ curvature relation due to interlayer stick/slip behaviour, and synthetic mooring lines (axial force/elongation hysteresis). Component non-linearities are experienced for several riser system components such as flex-joint, tensioner, bending stiffener etc.

It should be noted that external hydrostatic pressure is not considered to be a non-linear effect as hydrostatic pressures normally will be handled by the effective tension/ effective weight concept (Sparks 1984) in computer programs tailor made for slender structure analysis (e.g. Engseth *et al* 1988, O'Brien *et al* 1988).

### 3. Floater Load Models

#### 3.1 General

Floater motions are commonly split into LF, WF and HF motion components. The WF and HF motions are mainly governed by inviscid fluid effects, while viscous fluid effects are relatively important for LF motions. Different hydrodynamic effects are important for each floater type, and must be taken into account in the analysis and design. An overview of these load effects is presented in Table 3-1. Some of the effects can be linearised and included in a frequency domain approach, while others are highly non-linear and can only be handled in time-domain. In comparison with frequency domain analysis, the advantage of a time domain analysis is that it can easily capture higher order load effects. In addition, a time domain analysis can predict the maximum response without making assumptions regarding the response distribution.

In this RP only the hydrodynamic loads that have an effect on the global motions of the floater and its slender structures will be considered. This means that wave in deck loads, slamming loads and green water loads will not be dealt with here.

	<i>FPSO</i>	<i>Semi</i>	<i>DDF</i>	<i>TLP</i>
Wave frequency loads	X	X	X	X
Low frequency loads	X	X	X	X
Loads in moonpool	X		X	
Mathieu instability			X	
Hull vortex shedding			X	
Wave in deck loads		X	X	X
Slamming loads	X	X		X
Green water loads	X			
High frequency loads				X

#### 3.2 Hydrostatic loads

The structure weight and buoyancy force balance is the starting point for hydrodynamic analyses. Influence from risers and mooring pretensions is part of this load balance.

Usually this effort is trivial, but important for the success of subsequent hydrodynamic analyses. Buoyancy of large volume structures is calculated directly from the wetted surface geometry described by the radiation/diffraction model. In cases where a dual model, including Morison elements is applied, this may also be handled automatically by the computer program as long as the actual location and dimensions of the Morison elements are implemented.

The moonpool needs some special considerations if the moonpool area is large and reduces the waterplane area significantly. In the case of a Spar with air-can supported riser system, using a model with closed bottom of the hard tank or at keel level will result in too high waterplane stiffness.

Applying the correct metacentric height ( $GM_L$ ,  $GM_T$ ) in the

analyses is just as important as the location of the centre of buoyancy. Influence from potential free surface effects (slack tanks) needs to be taken into account while determining the metacentric height.

The additional restoring effects due to the reaction from the buoyancy cans on the riser guides also need to be taken into account.

Stiffness contributions from moorings lines and risers are assumed to be taken into account by the direct FE formulation in the analyses.

The mass distribution of the floater may either be entered as a global mass matrix, or from a detailed mass distribution (e.g. FE model). The input coordinate system varies depending on software and may be referred to the vertical centre of gravity, or the water plane. Input of roll and pitch radii of gyration is very often a source of error in computer programs. Applying the correct reference axis system is usually the challenge in this context.

#### 3.3 Wave loads

##### 3.3.1 General

The floaters are usually large volume structures and thus inertia-dominated. This implies that radiation/ diffraction analyses need to be performed with a suitable analysis tool. Some floaters, such as semi-submersibles and truss Spars, may also require a Morison load model for the slender members/braces in addition to the radiation/diffraction model.

A linear radiation/diffraction analysis will usually be sufficiently accurate. The term 'linear' means that the velocity potential is proportional to the wave amplitude, and that the average wetted area of the floater up to the mean water line is considered. The analysis gives first order excitation forces, hydrostatics, potential wave damping, added mass, first order motions in rigid body degrees of freedom and second order mean drift forces/moments. The mean wave drift forces only dependent on first order quantities, and can therefore be calculated in a linear analysis.

Several wave periods and headings need to be selected such that the motions and forces/moments can be described as correctly as possible. Cancellation, amplification and resonance effects must be properly captured. Modelling principles related to the fineness of the panel mesh must be adhered to, e.g.:

- diagonal length in panel model  $< 1/6$  of smallest wave length analysed
- fine panel mesh to be applied in areas with abrupt changes in geometry (edges, corners)
- finer panel mesh towards water-line in order to calculate accurate wave drift excitation forces.

For radiation/diffraction analyses of FPSOs and Spars attention should be paid to the existence of "irregular frequencies". These frequencies correspond to short internal waves in the numerical model and do not have any physical meaning. It is a deficiency of the mathematical model used. At these frequencies a standard sink/source technique may give unreliable values for added mass and damping. Methods exist to identify the irregular frequencies. Software SESAM:WADAM provides features for removing irregular frequencies so that reliable results are obtained for the whole frequency range.

Hydrodynamic interactions between multiple floaters in close proximity may also be solved using radiation/diffraction software through the so-called multi-body options. The  $n$  floaters are solved in an integrated system with motions in  $n \times 6$  DOFs. An example of a two-body system is a LNG-FPSO and a side-by-side positioned LNG carrier during offloading operations where there may be a strong hydrodynamic interaction between the two floaters. The interaction phenomena may be

of concern due to undesirable large relative motion response between the two floaters. This may cause damage to the ship hull and the offloading system. A collision between the FPSO and the LNG carrier is also possible. An important interaction effect is a trapped standing wave between the floaters that can excite sway and roll motions. Additional resonance peaks also appear in coupled heave, pitch and roll motions. The discretization of the wetted surfaces in the area between the floaters must be fine enough to capture the variations in the trapped wave. Another effect is the sheltering effect which leads to smaller motions on the leeside than on the weather side. A detailed analysis of relative motions of two floaters closely spaced is presented by Kim *et al* (2003).

The calculation described above for first order motions and second order forces/moments is usually the starting point to determine the global performance of a floater. The simultaneous effects of current, wind and waves are described in Sections 5 and 6.

### 3.3.2 Wave frequency loads

The output from a frequency domain analysis will be transfer functions of the variables in question, e.g. exciting forces/moments and platform motions per unit wave amplitude. The first order or linear force transfer function (LTF) is usually denoted  $H^{(1)}(\omega)$ . The linear motion transfer function,  $x_{WA}^{(1)}(\omega)$  also denoted Response Amplitude Operator (RAO), gives the response per unit amplitude of excitation, as a function of the wave frequency,

$$x_{WA}^{(1)}(\omega) = H^{(1)}(\omega)L^{-1}(\omega)$$

where  $L(\omega)$  is the linear structural operator characterizing the equations of motion,

$$L(\omega) = -\omega^2[M + A(\omega)] + i\omega B(\omega) + C$$

$M$  is the structural mass,  $A$  the added mass,  $B$  the wave damping and  $C$  the stiffness, including both hydrostatic and structural stiffness. The equations of rigid body motions are, in general, six coupled equations for three translations (surge, sway and heave) and three rotations (roll, pitch and yaw).

The frequency domain method is well suited for systems exposed to random wave environments, since the random response spectrum can be computed directly from the transfer function and the wave spectrum in the following way:

$$S_R(\omega) = |x_{WA}^{(1)}(\omega)|^2 S_\eta(\omega)$$

where

$\omega$  = angular frequency ( $= 2\pi/T$ )

$x_{WA}^{(1)}(\omega)$  = transfer function of the response

$S_\eta(\omega)$  = wave spectrum

$S_R(\omega)$  = response spectrum

Based on the response spectrum, the short-term response statistics can be estimated.

The method limitations are:

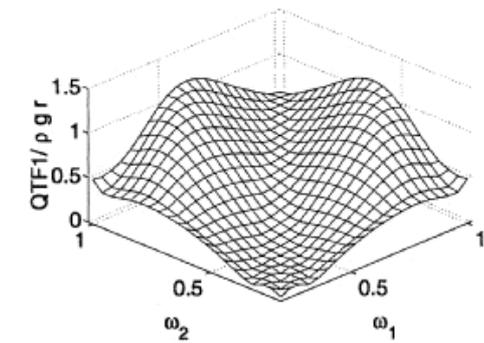
- requires linear equations of motion
- linear assumption is also employed in the random process theory used to interpret the solution. This is inconvenient for nonlinear effects like drag loads, time varying geometry, horizontal restoring forces and variable surface elevation. However, in many cases these non-linearities can be satisfactorily linearised.

Frequency domain analysis is used extensively for floating units, including analysis of both motions and forces. It is usually applied in fatigue analyses, and analyses of more moderate environmental conditions where linearization gives satisfactory results. The main advantage of this method is that the computations are relatively simple and efficient compared to time domain analysis methods.

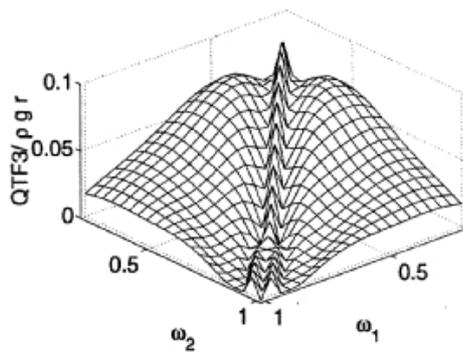
The radiation/diffraction analysis for a floating structure with a moonpool should be treated with some care. Moonpool effects are most relevant for turret moored ships and Spar platforms. Depending on the dimensions of the moonpool, the heave motion RAO may be strongly influenced. The motion of the water in the moonpool has a resonance at a wave frequency corresponding to the eigenfrequency of an oscillating water column,  $T_n = 2\pi\sqrt{h/g}$  where  $h$  is the height of the water column and  $g$  is the acceleration of gravity. Neglecting viscous damping of the water motion in the moonpool will result in unrealistic large motions and free surface elevation in the moonpool close to resonance. Discretization of the wetted area of the moonpool must be done with care in order to capture the flow details.

The moonpool effect can be treated in two ways. One approach is to consider the water column motion as a generalized mode. Another approach is to consider the motion of a massless lid floating on the water column and solve a two-body problem. In both cases additional viscous damping should be introduced. The damping level can be determined from model tests.

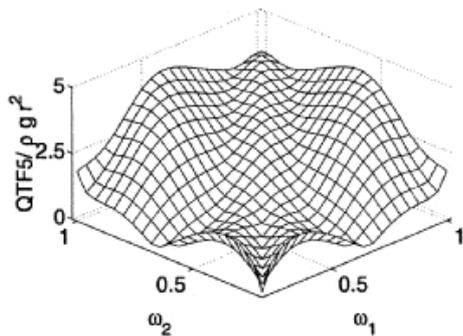
Correlation with model tests regarding WF loads and responses is generally considered good for standard floater types. One exception might be a concept like a mini-TLP with a truss structure on top of the main column and a high degree of drag loading as the wave passes the structure.



(a) Surge QTF



(b) Heave QTF



(c) Pitch QTF (ref VCG)

Figure 3-1  
 Difference frequency QTF for 228 m classical Spar. From Haslum (1999).

### 3.3.3 Low frequency loads

Low frequency motions of a moored floating structure are caused by the slowly varying wave drift force. This is a second-order wave force, proportional to the square of the wave amplitude. In a random sea-state represented by a sum of  $N$  wave components  $\omega_i, i = 1, N$  this force oscillates at difference frequencies  $\omega_i - \omega_j$  and is given by the expression

$$q_{WA}^{(2-)}(t) = \text{Re} \sum_{i,j}^N a_i a_j H^{(2-)}(\omega_i, \omega_j) e^{i(\omega_i - \omega_j)t}$$

where  $a_i, a_j$  are the individual wave amplitudes and  $H^{(2-)}$  is the quadratic transfer function (QTF) for the difference frequency load. The QTF is here presented as a complex quantity with amplitude  $|H^{(2+)}|$  and phase  $a^{(2+)}$ .  $\text{Re}$  denotes the real part. Commercial computer tools exist for calculating the difference frequency QTF. This is a second-order problem requiring discretization of the free surface in addition to the floater body surface.

The QTFs depend on the first order motions  $X_{WA}^{(1)}$ .

The QTF also depends on the directions of propagation  $\beta_i$  of the wave components. For short-crested sea-states this means that it may be necessary to solve the complete bi-chromatic and bi-directional second-order problem.

#### 3.3.3.1 Mean drift force

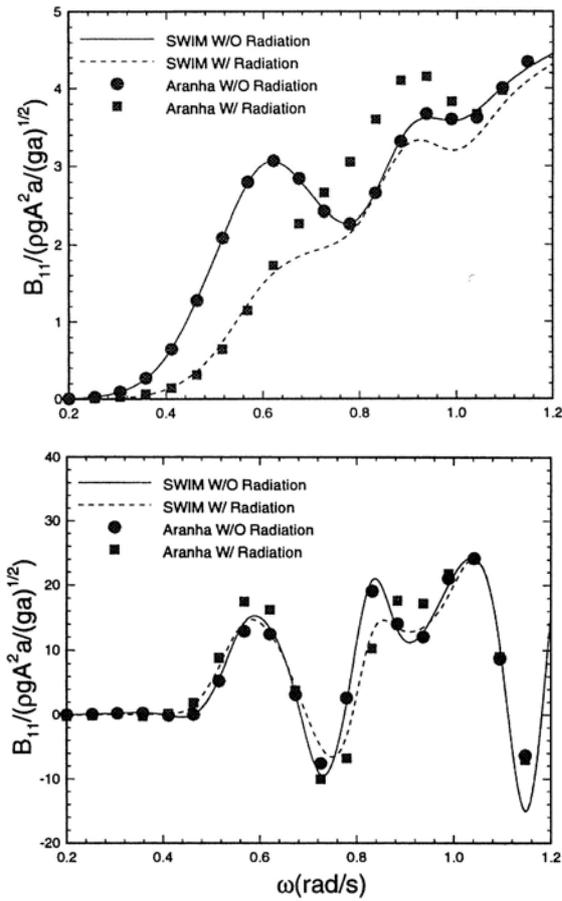
The mean drift force is obtained by keeping only diagonal terms ( $\omega_i = \omega_j$ ) in the sum above. The mono-chromatic drift force is defined by

$$F_d(\omega_i) = \frac{1}{2} a_i^2 \text{Re}[H^{(2-)}(\omega_i, \omega_i)]$$

The bi-directional mean drift force  $F_d(\omega; \beta_i, \beta_j)$  can also be calculated from first order velocity potentials.

The horizontal components (surge, sway) and the moment about the vertical axis (yaw) can be calculated in a robust manner by a far-field method, also called the momentum method. The mean drift force/moment in heave, roll and pitch must be calculated by integrating the 2<sup>nd</sup> order mean wave pressure over the wetted surface of the structure. This usually requires a finer discretization of the geometry. The vertical mean drift force is usually only of interest for structures with small water plane area and catenary mooring (Semis). To check that the pressure integration and momentum method provide the same results is an excellent check of numerical convergence.

For low frequencies, i.e. long waves, diffraction effects are small and the wave drift force is zero. Conversely, at high frequencies, the structure reflects the waves completely and the drift force has a finite asymptotic value. In between these asymptotic cases, the drift force has peaks associated with resonance effects in heave, roll and pitch or in the case of a multi-column platform, interference effects between the columns.



**Figure 3-2**  
Surge wave drift damping coefficient for Spar (upper) and semi submersible (lower). Ref. [13]

Special considerations have to be made for multi-vessel systems when calculating individual mean drift forces. The momentum approach gives only the total drift force on the global system. Direct pressure integration of second-order fluid pressure on each body is required.

### 3.3.3.2 Newman's approximation

In general all frequencies in the  $\omega_i \omega_j$ -plane may contribute to the second order difference frequency wave forces  $q_{WA}^{(2-)}$ .

As the second order wave forces are small, their most important contribution is in the vicinity of resonance. For a floater with low damping, the force components with difference frequencies close to the natural frequency are the most important for the response. Difference frequencies equal to the natural frequency  $\omega_N$  represent two lines in the  $\omega_i \omega_j$ -plane:

$$\omega_i = \omega_j \pm \omega_N.$$

If the natural frequency of the floater is very low, which is the case for horizontal motions, these lines are close to the 'diagonal'  $\omega_i = \omega_j$ . One can then take advantage of Newman's approximation (Newman 1974), which states that the off-diagonal elements in the full QTF matrix can be approximated by the diagonal elements, i.e.

$$H^{(2-)}(\omega_i, \omega_j) \cong \frac{1}{2} [H^{(2-)}(\omega_i, \omega_i) + H^{(2-)}(\omega_j, \omega_j)]$$

Another requirement is that the QTF function is smooth in the region close to the diagonal. Figure 3-1 shows that the surge

QTF satisfies this requirement, while the heave QTF does not.

Using Newman's approximation to calculate slow-drift forces significantly reduces computation time since a linear analysis is sufficient. The diagonal elements  $H^{(2-)}(\omega_i, \omega_i)$  can be calculated from first-order velocity potential alone. Hence there is no need to calculate the second order velocity potential.

Newman's approximation usually gives satisfactory results for slow-drift motions in the horizontal plane since the natural period is much larger than the wave period. For slow-drift motions in the vertical plane, e.g. the heave/pitch motions of a DDF, Newman's approximation may underestimate the slow-drift forces and in such case the solution of a full QTF matrix is required.

For some floater concepts such as TLPs, Newman's approximation has been commonly accepted and used in calculation of slow drift forces/moments due to its efficiency in comparison with the computation of the full matrix of quadratic transfer functions (QTF). However, for new floater concepts, caution should be exercised when applying Newman's approximation. It is recommended that the full QTF matrix is computed. It is especially the case for floaters with relatively large and shallow pontoons/bases in relation to the columns. LF roll and pitch will be the key responses to focus on.

### 3.3.3.3 Wave drift damping

An important potential flow effect for low frequency motions is the wave drift damping force. The wave drift damping force is defined as the increase in the second-order difference frequency force experienced by a structure moving with a small forward speed in waves. By expanding the difference frequency force in a Taylor series in terms of the forward velocity, and retaining the linear term only, the wave drift damping is proportional to the forward velocity. The wave drift therefore behaves like a linear damping, provided that the increase with forward speed is positive. This is usually the case. In some special cases, however, the wave drift damping may be negative (see Figure 3-2). When the slow-drift frequency is much smaller than the wave frequency, the slow-drift velocity varies little over a few wave periods and can be interpreted as an apparent forward speed. The wave drift damping force can therefore also be defined as the first order correction of the mean drift force in terms of the slow drift velocity  $v = \dot{x}$  of the floating structure. Usually, only the mean wave drift damping is considered, based on an expansion of the mean drift force  $F_d$ .

$$F_d(\omega, \dot{x}) = F_d(\omega, 0) - B(\omega)\dot{x} + O(\dot{x}^2)$$

where

$$B(\omega) = -\left. \frac{\partial F_d}{\partial \dot{x}} \right|_{\dot{x}=0}$$

For single- and multi-column structures (Spar, TLP, Semi), software SWIM (1999) provides calculation of the full bichromatic wave drift damping

$$G(\omega_i, \omega_j) = -\left. \frac{\partial}{\partial \dot{x}} H^{(2-)}(\omega_i, \omega_j; \dot{x}) \right|_{\dot{x}=0}$$

For floaters like TLPs and Spars it is sufficient to consider wave drift damping for uncoupled translational modes of motion (surge, sway). But for FPSOs undergoing large slow drift yaw motions as well, the complete 3x3 wave drift damping matrix for coupled surge, sway and yaw damping is needed. In the general case the coupled wave drift damping forces ( $F_{dx}$ ,  $F_{dy}$ ) and moment  $M_{dz}$  in the horizontal plane is given by

$$\begin{pmatrix} F_{dx} \\ F_{dy} \\ M_{dz} \end{pmatrix} = \begin{pmatrix} B_{xx} & B_{xy} & B_{xz} \\ B_{yx} & B_{yy} & B_{yz} \\ B_{zx} & B_{zy} & B_{zz} \end{pmatrix} \begin{pmatrix} \dot{x} \\ \dot{y} \\ \dot{\theta} \end{pmatrix}$$

where  $\dot{x}, \dot{y}$  are the surge and sway velocities and  $\dot{\theta}$  is the yaw angular velocity. A numerical method for calculating three-dimensional wave drift damping matrix  $B_{ij}$  for general offshore structures was presented by Finne *et al* (2000).

For column-based structures (TLP, Spar) there is an approximate method that is widely used. The formula is called Arahna's formula (Arahna 1996),

$$B(\omega) = \frac{\omega^2}{g} \frac{\partial F_d}{\partial \omega} + \frac{4\omega}{g} F_d$$

The formula does not include radiation effects from wave induced motions and should be used with care for non wall-sided structures like an FPSO (see Figure 3-2). The formula can be generalised to the case of combined surge-sway motion and waves from an arbitrary direction  $\beta$  (see Molin, 1993). No such simple formula exists for yaw wave drift damping.

For most deepwater floaters wave drift damping of low frequency heave, roll and pitch motions can be neglected.

Wave drift damping can also be applied to quantify the effect of current on wave drift forces. Wave drift forces are sensitive to the superposition of a current, which affects the way wave energy is scattered by the floating structure. Assuming the current is weak enough so that flow separation does not occur, potential theory can be applied. Flow separation does not occur if the following condition holds (deep water)

$$\frac{U_c}{\omega A} < 1$$

where  $U_c$  is the current speed,  $\omega$  is the wave frequency and  $A$  is the wave amplitude. The drift force in waves and current can be simply related to the drift force in waves only by:

$$F_d(\omega, U_c) = F_d(\omega, 0) + B(\omega)U_c + O(U_c^2)$$

where  $B(\omega)$  is the wave drift damping (see 3.3.3). If waves and current propagate in the same direction, the drift force is increased.

A simple example can be used to quantify the effect of current on the mean drift force. Taking  $U_c = 1$  m/s, a wave with a period of 10 seconds and assuming this corresponds to a peak in the mean drift force as a function of frequency ( $\partial F_d / \partial \omega = 0$ ), the use of Arahna's formula above gives a 25% increase in the drift force. When  $\partial F_d / \partial \omega > 0$ , the increase is even larger.

### 3.3.4 High frequency loads

Second-order wave forces in a random sea-state oscillating at the sum-frequencies  $\omega_i + \omega_j$  excite resonant response in heave, roll and pitch of TLPs.

#### 3.3.4.1 Second order wave loads

Due to its stiff tendons tension leg platforms experience vertical mode (heave, roll, pitch) resonance at relative low eigenperiods  $T_N$ . The heave eigenperiod is given by

$$T_3 = 2\pi \sqrt{\frac{L(M + A_{33})}{EA}}$$

where  $EA/L$  is the tendon stiffness,  $M$  is the structure mass and  $A_{33}$  is the heave added mass. Typical resonance periods are in the range 2–5 seconds. Waves in this range do not carry

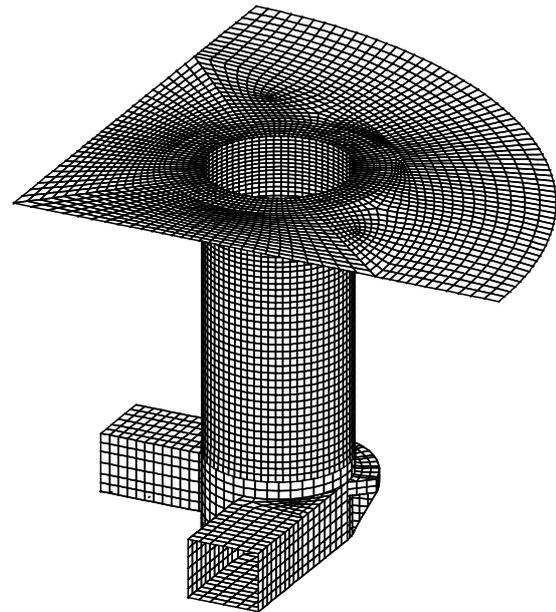
enough energy to excite such structures in resonant response. However, since the wave-body system is inherently non-linear, the structure will also be excited by waves of periods  $2T_N, 3T_N$ , etc. which in a typical sea-state carry more energy. This non-linear transfer of energy to higher order (super-harmonic) response of the structure can equivalently be described by saying that regular waves of frequency  $\omega$  excite the structural response at  $2, 3\omega$ , etc. The high-frequency stationary time-harmonic oscillation of a TLP is called *springing*.

Computer tools are available (i.e. WAMIT) for calculating the sum-frequency quadratic force transfer functions (QTF)  $H^{(2+)}(\omega_i, \omega_j)$ . The high-frequency, or sum-frequency force in a random sea-state is given by

$$q_{WA}^{(2+)}(t) = \text{Re} \sum_{i,j}^N a_i a_j H^{(2+)}(\omega_i, \omega_j) e^{i(\omega_i + \omega_j)t}$$

The most important aspects to be considered for springing analyses are:

- discretization (mesh) of wetted floater surface geometry
- discretization of free surface and its extension
- number of frequency pairs in the QTF matrix
- damping level for the tendon axial response



**Figure 3-3**  
Discretization of one quarter of TLP hull and free surface for calculation of second order sum-frequency wave loads.

Discretization of wetted floater surface and free surface is governed by the second-order sum-frequency incoming wave length which for a given frequency is one quarter of the first-order linear wavelength. Requiring on the order of 6 panels per second-order wavelength, gives as a rule of thumb, that the dimension of the panels on the wetted surface of the structure in a second-order analysis should not be larger than  $gT^2/150$ , where  $T$  is the period of the incoming wave. Special requirements apply to the discretization of the free surface, related to the convergence of the free surface integral over an infinite domain. Even stricter requirements may apply to the discretization when calculating sum-frequency wave elevation. Detailed recommendations should be given in computer program user manuals.

### 3.3.4.2 Higher order wave loads

Deepwater TLPs can experience large resonant high frequency transient response, called *ringing*. Ringing exciting waves have a wavelength considerably longer than a characteristic cross section of the structure (e.g. diameter of column). Therefore, long wave approximations can be applied for higher-order load contribution. A recommended ringing load model is a combination of full three-dimensional first- and second-order wave diffraction together with a third-order slender body contribution (Faltinsen *et al* 1995). Hence, the exciting ringing force can be written as

$$q(t) = q_{WA}^{(1)}(t) + q_{WA}^{(2+)}(t) + q_{FNV}^{(3)}(t)$$

where  $q_{WA}^{(1)}(t)$  and  $q_{WA}^{(2+)}(t)$  are based on the first and second-order force transfer functions.

General cubic transfer functions are not yet available so the third-order term,  $q_{FNV}^{(3)}(t)$  is an approximation using a slender body assumption and is limited to circular column geometries in the wave zone. The effect of pontoons on the third-order term is not included. A validation of this ringing load approach was reported by Krokstad *et al* (1998). Since ringing is a transient phenomenon, the response must be solved in time domain. However, a linear structural model can be applied.

### 3.4 Wind loads

Wind loading is important for prediction of global motion response of floaters. Accurate modelling of the wind effects is therefore essential. For some floating systems the wind loads can be the dominating excitation.

The global wind loads acting on a floating structure consists of two components, a static part resulting in a mean offset and mean tilt, and a fluctuating component due to wind gusts which mainly excite the low frequency motions in surge, sway and yaw. For some floater concepts the roll and pitch motions are also influenced.

Due to its importance, the wind loading is usually determined based on wind tunnel tests. These tests are very often conducted early in the design process. In case of significant changes to the deck/topside structures during detail design, these wind tunnel tests may have to be repeated. For minor deck/topside changes, updates of the wind loading may be performed by spreadsheets.

Wind tunnel tests usually cover a sufficient number of wind directions such that interpolations can be made in subsequent coupled analyses. The influence of heel may have to be taken into account if the resulting heel angle is critical and the wind loading increases considerably with heel angle. This is also needed for floating stability calculations.

The gust wind-loading component is simulated by the wind gust spectrum. A number of wind spectra exist. It should be emphasised that a wind spectrum is selected that best represents the actual geographical area the floater is located. Wind spectra are generally described with a number of parameters making it relatively easy to make input errors. Checking of wind spectrum energies and shapes is therefore considered essential. The most commonly used wind spectra are the API and NPD spectra. Details on these gust wind spectra may be found in the relevant literature. The existence of wind squalls requires special attention in those areas it is occurring.

The wind velocity may be a magnitude higher than the floater velocity. The use of relative velocity formulation compared to wind velocity alone will therefore have marginal influence. It is, however, recommended to use the relative velocity formulation also for wind loading. In coupled analyses the aerodynamic damping contribution is usually insignificant. This is due to the larger damping contributions from the slender struc-

tures overriding the aerodynamic damping. For correlation with model test results with only wind loading, the aerodynamic damping should be estimated and taken into account.

### 3.5 Current loads

Calculation of current loads is challenging due to the fact that the current depends on local topographic conditions with often strong variability in magnitude and direction with depth. Only measurements can provide sufficient background for determination of design current speeds and directions. The current may induce vortex induced motions (VIM) of the floater as well as vortex induced vibrations (VIV) of the slender structures and has to be carefully considered.

A steady current gives rise to a steady force in the horizontal plane and a yaw moment. For small displacement floaters in deep water or floaters with a large number of slender structures, the current loading on the slender structures may dominate the total steady force. It is therefore of importance to apply the correct drag coefficients with due attention to the excitation as well as the damping contribution. Sensitivity checks with different sets of drag coefficients are therefore recommended. Some recommendations on the selection of drag coefficients are included in Appendix A.1.

The influence of current on the mean wave drift force is dealt with in 3.3.3.

### 3.6 Vortex-induced loads

Vortex shedding may introduce cross-flow and in-line hull motions commonly termed vortex-induced-motions (VIM).

Cross flow oscillations are considered most critical due to the higher oscillation amplitude compared to the in-line component.

Hull VIM is important to determine as it will influence the mooring system design as well as the riser design. Both extreme loading and fatigue will be influenced. VIM is a strongly non-linear phenomenon and it is difficult to predict accurately. Model testing has usually been the approach to determine the hull VIM responses. More details can be found in Appendix A.1.

Floaters with single columns like Spars and multicolumn deep draught floaters are most likely to be exposed to VIM oscillations. Therefore, these types of floaters are designed with vortex shedding suppression devices like strakes. The inclusion of strakes makes it challenging to perform CFD simulations as it will require simulation of 3-dimensional effects, and this increases the simulation time considerably. One alternative to CFD simulations is to use results from a bare cylinder and use empirical data to estimate the reduction in oscillation amplitude due to the strakes. Full-scale data is, however, the ultimate solution and should be used to correlate with analytical predictions.

The most important parameters for hull VIM are:

- $A/D$  ratio ( $A$  = transverse oscillation amplitude,  $D$  = hull diameter)
- $V_r$  – reduced velocity ( $= U_c / (f_n D)$ ,  $U_c$  = current velocity,  $f_n$  = eigenfrequency in transverse direction,  $D$  = hull diameter).

Typically VIM oscillations will be small and in-line with the current flow for  $V_r < 3\sim 4$ . For  $V_r > 3\sim 4$  the hull will start to oscillate transverse to the current flow and increase in magnitude compared to in-line. Another important effect from the transverse oscillations is that the mean drag force increases. This is also confirmed by model tests and full scale measurements.

The in-line drag coefficient can be expressed as:

$$C_d = C_{do}[1 + k(A/D)]$$

where

$C_{do}$  = initial drag coefficient including influence of strakes  
 $k$  = amplitude scaling factor.  
 $A/D$  = cross-flow amplitude/hull diameter

The amplitude scaling factor is normally around 2. For a reduced velocity around 5,  $A/D$  can be up to 0.7-0.8 if the hull has no suppression devices such as strakes. Strakes effectively reduce the VIM response down to  $A/D \sim 0.3 - 0.4$ .

The coupled analysis approach can be an effective way of checking out the responses in moorings and risers by introducing the known (analytical, model tests, or full-scale) in-line and cross-flow oscillations as forces/moments onto the floater.

Since the vortex shedding is more or less a sinusoidal process, it is reasonable to model the cross-flow force imposed on the hull as harmonic in time at the shedding frequency  $f_s$ . VIM lock-in occurs when the vortex shedding frequency locks on to the eigenfrequency  $f_n$ . The vortex shedding is dependent on the Strouhal number, and is defined by  $f_s = S U_c / D$ , where  $S$  is the Strouhal number. The Strouhal number is typically equal to 0.2 for a circular cylinder. In general the transverse (lift) force may be written

$$q_{VIM}(t) = \frac{1}{2} \rho U_c D C_L \sin(2\pi f_s t)$$

where  $C_L$  is the lift force coefficient. The oscillating in-line force is given by the same expression, except that the oscillation frequency is twice the vortex shedding frequency  $f_{I-L} = 2f_s$ .

The in-line VIM response may be in the order of 0.2 times the cross-flow VIM response. Hence, the hull VIM response curves are typically in the shape of a skewed '8' or a crescent (half moon).

## 4. Slender Body Load Models

This section will give an introduction to commonly used load models for analysis of risers and mooring lines of relevance for slender structure analysis in connection with coupled/de-coupled system analyses. For a more detailed discussion of special load models for risers (e.g. slug flow, multi-pipe modelling, riser component modelling, temperature effects etc.) reference is made to e.g. API RP 2RD and DNV OS-F201.

### 4.1 Forced floater motions

Forced floater motions represent a primary dynamic loading on riser and mooring systems. Floater motions are applied as forced boundary displacements at fairleads of mooring lines and at all relevant supports of riser systems, e.g. multiple transverse riser supports for Spar platforms.

Floater motions may be specified in terms of motion time histories or floater transfer functions depending on the floater motion analysis strategy as discussed in the following.

#### 4.1.1 Time series representation

Time series is the most general format for representation of floater motions in slender structure analyses. Simultaneous time series for translations and finite rotations at one location on the floater gives a unique representation of the rigid body floater motion at any location on the floater. Special attention should however be given to the definition of finite rotations to ensure consistency.

Simultaneous wave time series will in addition be required for consistent generation of wave kinematics in the slender structure analysis.

Floater motion time series can be obtained from coupled/de-coupled analyses or measurements (model tests or full-scale). A major advantage of the time series format is that it allows for consistent description of different frequency regimes in the

floater motions (i.e. correlation in time is maintained). The floater motions produced by coupled/de-coupled analyses will contain combined WF and LF components (e.g. FPSO, TLP, Spar). TLP motions may in addition contain HF components while Spar motions may contain hull VIM components. The latter will however be in the LF regime due to lock-on to surge/sway eigenfrequencies.

The described approach is applicable to nonlinear as well as linearised TD analyses, but can not be applied in FD analyses.

#### 4.1.2 Transfer function representation

Slender structure analyses have traditionally been performed considering dynamic excitation from WF floater motions represented by floater motion transfer functions (RAOs). LF motions are considered as a quasi-static effect and accounted for by an additional representative offset, i.e. in addition to mean floater position for the actual environmental condition. For Spar platforms, this will also involve an additional heel/tilt to account for LF motions.

It should however, be noted that the described approach is only applicable to slender structures that do not respond dynamically to LF floater motions. Combined WF and LF forced floater motions should be considered if the slender structure dynamics is significantly influenced by LF excitation.

The RAO representation of the floater motions is applicable in TD as well as FD analyses.

### 4.2 Fluid kinematics

Fluid kinematics may comprise a significant dynamic loading on the upper part of deep water riser systems. Direct wave loading on mooring lines is however normally of less importance, except if buoys close to the surface are used to obtain the desired mooring line configuration.

#### 4.2.1 Wave kinematics

Undisturbed wave kinematics is normally based on Airy wave theory. Wheeler stretching may be applied to compute wave kinematics in the wave zone. For further details, see e.g. Gudmestad (1993).

#### 4.2.2 Disturbed kinematics

The presence of the floater gives rise to changes in the fluid kinematics. This disturbance may be determined by the use of radiation/diffraction analysis. The outputs from such analysis are RAOs for disturbed kinematics consistent with the floater motion RAOs. For floaters and risers located close to e.g. columns/pontoons, this disturbance must be accounted for in design.

#### 4.2.3 Moonpool kinematics

Kinematics of the entrapped water in the moonpool area can in principle be treated in the same way as the disturbed wave kinematics, i.e. in terms of transfer functions for moonpool kinematics consistent with the hull motion transfer functions. This approach requires that the entrapped water is included in the hydrodynamic model used to compute the floater motion characteristics. Such calculations will, however, require a very careful modelling to achieve a realistic picture in case of complicated moonpool geometry and/or multiple risers in the moonpool. Special attention should be focused on possible resonant modes of the entrapped water, see also 3.3.2.

A simplified model for the moonpool kinematics can be obtained by assuming that the entrapped water follows the hull motions rigidly. This formulation is applicable for FD as well as TD analysis. The latter approach allows for consistent treatment of moonpool kinematics due to simultaneous WF and LF floater motions.

Assuming that the entrapped water rigidly follows the hull motions, the hydrodynamic loading in the normal (to pipe axis)

direction can be expressed as:

$$f_n = \frac{1}{2} \rho C_D^n D_h |u_H - \dot{x}_n| (u_H - \dot{x}_n) + \rho \frac{\pi D_b^2}{4} \dot{u}_H + \rho \frac{\pi D_b^2}{4} (C_M^n - 1) (\dot{u}_H - \ddot{x}_n)$$

where  $u_H, \dot{u}_H$  are the hull velocity and acceleration components normal to the riser.

The riser motions relative to the moonpool are to a large extent governed by how the riser is supported inside the moonpool. For a Spar, the riser motions in the transverse moonpool direction will typically be constrained at several supports along the riser. The excitation forces are hence not very sensitive to the  $C_D$  and  $C_M$  values due to the small relative motion between the fluid and the riser (see equation). The “Froude Krylov” term, i.e. the inertia term due to fluid acceleration, is in this case the dominating contribution to the excitation force.

### 4.3 Hydrodynamic loading

The hydrodynamic loading on slender structures is usually expressed by the Morison equation in terms of the relative fluid-structure velocities and accelerations. The fluid velocities and acceleration vectors can be found by considering relevant contributions from wave kinematics (regular or irregular, undisturbed or disturbed), current (constant velocity or velocity and acceleration) or moonpool kinematics.

Hydrodynamic loading in normal and tangential pipe directions is usually computed independently according to the so-called cross-flow (or independence) principle. The Morison equation for a circular cross section is expressed as:

$$f_n = \frac{1}{2} \rho C_D^n D_h |u_n - \dot{x}_n| (u_n - \dot{x}_n) + \rho \frac{\pi D_b^2}{4} C_M^n \dot{u}_n - \rho \frac{\pi D_b^2}{4} (C_M^n - 1) \ddot{x}_n$$

$$f_t = \frac{1}{2} \rho C_D^t D_h |u_t - \dot{x}_t| (u_t - \dot{x}_t) + \rho \frac{\pi D_b^2}{4} C_M^t \dot{u}_t - \rho \frac{\pi D_b^2}{4} (C_M^t - 1) \ddot{x}_t$$

where:

$f_n$	= Force per unit length in normal direction
$f_t$	= Force per unit length in tangential direction
$\rho$	= Water density
$D_b$	= Buoyancy diameter
$D_h$	= Hydrodynamic diameter
$u_n, \dot{u}_n$	= Fluid velocity and acceleration in normal direction
$\dot{x}_n, \ddot{x}_n$	= Structural velocity and acceleration in normal direction.
$C_D^n, C_M^n$	= Drag and inertia coefficients in normal direction
$u_t, \dot{u}_t$	= Fluid velocity and acceleration in tangential direction
$\dot{x}_t, \ddot{x}_t$	= Structural velocity and acceleration in tangential direction.

$C_D^t, C_M^t$  = Drag and inertia coefficients in tangential direction

For a discussion of the Morison formulation for double-symmetric cross sections (e.g. riser bundles, piggyback umbilicals etc.) reference is made to DNV OS-F201.

### 4.4 Marine growth

Marine growth on slender structures will influence the loading in terms of increased mass, diameter and hydrodynamic loading.

Site dependent data for marine growth are normally specified in terms of density, roughness and depth variation of thickness. The marine growth characteristics are basically governed by the biological and oceanographic conditions at the actual site. The relative density of marine growth is usually in the range of 1 – 1.4 depending on the type of organisms.

The thickness of marine growth to be included in design analyses will, in addition, be dependent on operational measures (e.g. regular cleaning, use of anti fouling coating) as well as structural behaviour (e.g. less marine growth is normally considered for slender structures with significant dynamic displacements).

In FE analyses, it is recommended to increase mass, buoyancy diameter and drag diameter according to the specified depth variation of marine growth. In addition, the hydrodynamic coefficients should be assessed with basis in the roughness specified for the marine growth.

## 5. De-coupled Response Analysis

De-coupled analysis solves the equations of the rigid body floater motions. The floater load models are the same as in the coupled analysis. However, de-coupled analysis differs from coupled analysis in the solution strategy and slender structure representation.

### 5.1 Static analysis

#### 5.1.1 Still water condition

The static configuration is often the first challenge with coupled analyses. The computer programs have different approaches for e.g. inclusion of risers and mooring lines. Checking the static configuration is a must and has to be validated prior to executing the dynamic analyses. The use of graphics for verification of the static configuration is recommended.

#### 5.1.2 Quasi-static mean response

The first task in a global response analysis is to identify the steady response, or the static position of the structure. The mean wave, wind and current forces/moments determine the static position.

##### 5.1.2.1 Mean wave drift forces

In high sea states there is a considerable viscous contribution to the mean drift force from fluid forces in the splash zone. A simple expression can be derived for the viscous mean drift force on a vertical surface piercing cylinder by applying Morison's formula and regular wave kinematics of Airy wave theory:

$$\bar{q}_{visc} = \frac{2}{3\pi} \rho g k C_D A^3$$

where  $k$  is the wave number and  $A$  is the wave amplitude of the regular wave,  $C_D$  is the drag coefficient and  $D$  is the diameter of the cylinder. It is worth mentioning that while the potential flow drift force is quadratic in the wave amplitude, the viscous

contribution is cubic.

### 5.1.2.2 Steady wind forces

The steady wind forces and moments on the part of the structure above the free surface can be written in a general form as

$$\bar{q}_{wi} = \rho_a c_w(\beta) L^2 U_w^2$$

where  $\rho_a$  is the density of air,  $c_w$  is a directional dependent drag coefficient,  $\beta$  is the angle between the wind velocity and the x-axis,  $L$  is the characteristic length scale and  $U_w$  is the wind velocity experienced by the structure. Empirical or experimental data for the drag coefficient  $c_w$  is necessary. CFD calculations can be carried out to determine  $c_w$ . Aquirre & Boyce (1974) presented data for wind forces on offshore drilling platforms. Isherwood (1973) presented drag coefficients for ships.

### 5.1.2.3 Steady current forces on floater

A steady current gives rise to a steady force in the horizontal plane and a vertical moment. Empirical formulas are most often used to calculate current forces and moments on floating offshore structures.

Viscous current forces on offshore structures that consist of slender structural parts can be calculated using the strip-theory approximation. This applies to columns and pontoons of semi-submersibles and of TLPs. The current velocity is decomposed into one component  $U_{cN}$  in the cross-flow direction of the slender structural part and one component in the longitudinal direction. The latter component causes only shear forces and is usually neglected. The cross-flow velocity component causes high Reynolds number separation and gives rise to an inline drag force

$$F_c^N = \frac{1}{2} C_d D U_{cN}^2$$

where  $C_d$  is the sectional drag coefficient. There may be hydrodynamic interaction between structural parts. If a structural part is placed in the wake behind another part, it will experience a smaller drag coefficient if the free stream is used to normalize the drag coefficient. Such shielding effects should be considered when calculating the steady current forces.

Empirical formulas are also used to calculate current forces and moments of FPSOs. The drag force on an FPSO in the longitudinal direction is mainly due to skin friction forces and it can be expressed as

$$F_{cx} = \frac{1}{2} \rho S U_c^2 C_d(Rn, \beta)$$

The drag coefficient is a function of the Reynolds number  $Rn$  and the angle  $\beta$  between the current and the longitudinal axis of the ship. See Hughes (1954).

The transverse current force and current yaw moment on an FPSO can be calculated using the cross-flow principle. The assumption is that the flow separates due to cross-flow past the ship, that the longitudinal current components do not influence the transverse forces on the cross-section, and that the transverse force on a cross-section is mainly due to separated flow effects. The transverse current force on the ship then can be written as

$$F_{cy} = \frac{1}{2} \rho \left[ \int_L dx C_D(x) D(x) \right] U_c^2 \sin \beta | \sin \beta |$$

where the integration is over the length of the ship.  $C_D(x)$  above is the drag coefficient for flow past an infinitely long cylinder with the cross-sectional area of the ship at position  $x$ .  $D(x)$  is the sectional draught.

The viscous yaw moment due to current flow is simply

obtained by integrating the moments due to sectional drag forces along the ship. It is important to note that the vertical moment has an additional inviscid part, called the Munk moment,

$$M_c = U_c^2 \cos \beta \sin \beta (A_{11} - A_{22})$$

where  $U_c$  is the current velocity in a direction  $\beta$  with the x-axis and  $A_{11}$  and  $A_{22}$  are the added mass coefficients in the x- and y-directions. The viscous current loads are similar to the viscous wind forces. A discussion on current loads on offshore structures is given in Faltinsen (1990).

## 5.2 Frequency domain analyses

### 5.2.1 General

A frequency domain motion analysis is usually the basis for generating transfer functions for frequency dependent excitation forces (1<sup>st</sup> and 2<sup>nd</sup> order), added mass and damping (potential & viscous). It might also be possible to work with motion RAOs, but this is considered more cumbersome when transferring into the time domain.

In a frequency domain analysis, the equations of motions are solved for each of the incoming regular wave components for a wave frequency analysis, and for each of the sum- or difference-frequency combinations for a second-order analysis (high- or low frequency response).

### 5.2.2 Wave frequency response

The output from a traditional radiation/diffraction frequency domain analysis will typically be excitation forces/moments, added mass/moments and potential damping and motion RAOs. If a dual (inclusive Morison loading) model/analysis has been made this will usually be added directly into the results. Inclusion of a Morison model may also encompass linearised finite wave amplitude effects and viscous damping contributions.

Some computer programs may also have the option of using disturbed wave kinematics for calculation of loads on slender (Morison) structures located adjacent to large volume elements (radiation/diffraction).

The frequency domain analysis will require a balanced system with weights, buoyancy and pretensions in equilibrium. The same applies to the boundary conditions like hydrostatic, mooring and riser stiffness.

Selection of wave periods for the wave frequency analysis is usually done with basis in:

- peak period in wave spectrum
- location of rigid body eigenperiods
- geometrical considerations (diameters of columns, spacing between columns, wave headings, ships length/width, etc).

The main objective is to describe the actual RAOs with a sufficient number of wave periods and wave headings.

This is a linear analysis and the output will be given as response amplitude per unit wave amplitude for:

- WF excitation forces/moments (6 DOF)
- added mass/moments (6 DOF)
- damping forces/moments (6 DOF)
- WF motion RAOs (6 DOF).

Post-processing of the frequency domain results to determine short term, or long term responses is not detailed here as it is considered to be well established and not directly relevant for coupled analyses. In this context it should be noted that the WF responses are usually marginally influenced by coupling effects hence minor differences in responses between fully coupled and frequency domain WF responses is expected.

### 5.2.3 Low frequency response

The low frequency or slow-drift motions can be estimated by solving a linearised equation of motion in the frequency domain for each frequency (or difference frequency pair) similar to the wave frequency response. The exciting force/moment is the difference frequency quadratic force transfer function  $H^{(2-)}$ .

While for wave-frequency response, most of the damping is provided by the radiation of free surface waves, several other damping effects come into play for the slow drift response of moored floating structures. As the motion frequency decreases, the structure radiates less and less wave energy, hence for most practical slow-drift problems radiation damping is negligible. Damping of slow-drift motions comprise:

- i) wave drift damping
- ii) drag forces on mooring lines and risers
- iii) viscous loads on the hull (skin friction and form drag)
- iv) variation of the wind loads with the velocity of the structure
- v) friction of the mooring lines on the sea-floor

Several of these damping effects are non-linear, and the total damping used in frequency domain estimation of slow-drift response must be determined by stochastic linearization. Damping contributions i), ii) and iii) as function of significant wave height  $H_s$  for an FPSO is shown in Figure 5-1.

#### 5.2.3.1 Wave drift damping

The constant wave drift damping to be used in a frequency domain analysis can be taken as

$$\bar{B}_{ij} = 2 \int_0^{\infty} B_{ij}(\omega) S(\omega) d\omega$$

where  $B_{ij}$  is the wave drift damping coefficient and  $S(\omega)$  is the wave spectrum.

#### 5.2.3.2 Mooring line damping

Drag forces on mooring lines strongly contribute to slow-drift damping. The wave frequency motions of the floater have been shown to strongly increase the low frequency damping. Compared with the line diameter, the transverse motion amplitude is quite large. Hence the flow is well separated and the drag force can be expressed as

$$dF = \frac{1}{2} \rho C_d D (u + U) |u + U| ds$$

where  $u$  is the wave frequency velocity and  $U$  is the low frequency velocity.  $D$  is the characteristic diameter of the mooring line. Since  $U \ll u$ , the linearised drag force is

$$dF = \frac{1}{2} \rho C_d D |u| U ds$$

Mooring line damping can be estimated by relating the damping to the energy dissipated along the line by the drag forces. Assuming that the mooring line behaves in a quasi-static way, the damping can be related directly to the RAO's of the floater as shown by Huse (1986).

#### 5.2.3.3 Viscous hull damping

The contribution to damping from viscous forces acting on the floater is often the most difficult to quantify and its part of the total damping may differ significantly from one structure to another. For an FPSO in surge motion linear skin friction dominates the viscous forces while for a TLP or semi-submersible

quadratic drag dominates.

The linear skin friction can be estimated by assuming the hull surface to be a flat plate in turbulent flow. But analytic results should be used cautiously. Viscous damping is usually based on decay model tests.

For a TLP or semi-submersible viscous damping can be simplified by reducing the problem to the case of two-dimensional cylinders performing a combination of low frequency and wave frequency motions. This is also relevant for an FPSO in slow sway or yaw motions. The  $KC$  number ( $KC = 2\pi a/D$  where  $a$  is motion amplitude and  $D$  is diameter) for flow around the hull is in the range 0 to 5. Special care is required when selecting drag coefficients in this regime. It is common to use an 'independent flow' form of Morison equation, where the drag forces due to wave frequency and low frequency motions are separated, so that two drag coefficients are required. The low frequency drag force is then given by

$$dF = \frac{1}{2} \rho C_{dU} D U |U| ds$$

where  $U$  is the slow-drift velocity.

#### 5.2.3.4 Wind damping

The wind forces and moments on a moored offshore platform are expressed in terms of directional wind drag coefficients  $C_w$  and the relative (between wind and platform motions) wind velocity  $\bar{U}_w$ . Since the wind force has a large steady component  $\bar{U}$ , a standard linearization procedure gives the wind damping coefficient

$$B_{wind} = 2C_w \bar{U}_w$$

#### 5.2.3.5 Sea floor friction

Soil friction leads to reduced tension fluctuations for the portion of the mooring table in contact with sea floor, causing an increase of the line stiffness. Simulations have shown that low frequency tensions and damping forces are barely influenced by presence of soil friction, but it has some effect on wave frequency tensions (Triantafyllou *et.al.* 1994).

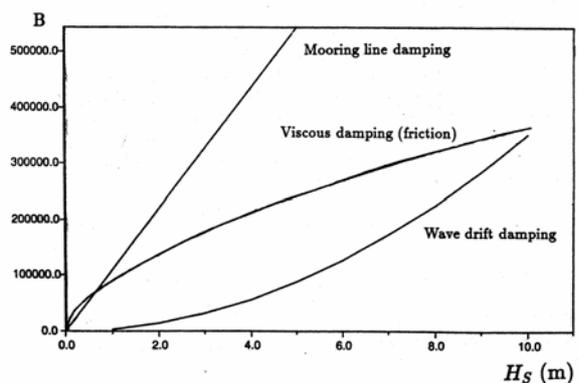


Figure 5-1 Comparison of different slow-drift damping components as function of wave height (Molin 1993).

### 5.2.4 High frequency response

Springing response is usually solved in the bi-chromatic frequency domain for each sum-frequency pair using the sum-frequency force transfer function  $H^{(2+)}$  and a similar linear structural operator  $L$  as for the wave frequency response, except for the damping,

$$x_{WA}^{(2+)}(\omega_i + \omega_j) = H^{(2+)}(\omega_i, \omega_j) L^{-1}(\omega_i + \omega_j)$$

Added mass can be taken as the linear high frequency asymptotic limit. A key element of a springing response analysis is to estimate damping of the high-frequency vertical motions. The following damping contributions should be considered

- i) radiation damping due to radiated free surface waves (can be neglected),
- ii) viscous damping due to hull skin friction and eddy making damping,
- iii) structural damping,
- iv) flex element damping,
- v) soil damping.

### 5.3 Time domain analyses

#### 5.3.1 Formulations

The equations of motion for a freely floating or moored structure can be written as:

$$M \ddot{x} + B \dot{x} + D_1 \dot{x} + D_2 f(\dot{x}) + Kx = q(t, x, \dot{x})$$

where

- $M$  = mass matrix,  $m+A(\omega)$ , including added mass.
- $m$  = structural mass matrix
- $B$  = potential damping matrix  $B = B(\omega)$
- $D_1$  = linear damping matrix, including wave drift damping
- $D_2$  = quadratic damping matrix
- $f$  = vector function where each element is given by  $x_i|x_i|$
- $K$  = position-dependent hydrostatic stiffness matrix
- $x$  = position vector
- $q$  = exciting force vector

The exciting force on the right hand side is

$$q(t, x, \dot{x}) = q_{WA}^{(1)} + q_{WA}^{(2)} + q_{CU} + q_{WI} + q_{ext}$$

where

- $q_{WA}^{(1)}$  = first order wave excitation force
- $q_{WA}^{(2)}$  = second order wave excitation force
- $q_{CU}$  = current drag force
- $q_{WI}$  = wind drag force
- $q_{ext}$  = any other forces (specified forces and forces from station-keeping and coupling elements, etc.)

The wave frequency (WF) motions are excited by the first order wave excitation force. The low-frequency (LF) motions are excited by the slowly varying part of the second order wave excitation force, the wind drag force and the current drag force. The high-frequency (HF) motions are excited by the sum-frequency second-order wave excitation force.

#### 5.3.2 Retardation functions

Another form of the equations of motions can be obtained by introducing the retardation function

$$h(t) = \frac{1}{2\pi} \int_{-\infty}^{\infty} [b(\omega) + i\omega a(\omega)] e^{i\omega t} d\omega$$

which is the Fourier transform of the added-mass and damping,  $a(\omega) = A(\omega) - A_{\infty}$  and  $b(\omega) = B(\omega) - A_{\infty}$ .  $A_{\infty}$  is the asymptotic value of the added mass at infinitely high frequency. The high frequency limit of the wave damping is zero.

The equations of motion can be written as:

$$(m + A_{\infty}) \ddot{x} + D_1 \dot{x} + D_2 f(\dot{x})$$

$$+ \int_0^t h(t-\tau) \dot{x}(\tau) d\tau = q(t, x, \dot{x})$$

The frequency dependent added mass and damping can be obtained from a three-dimensional panel program.

Solving the integral-differential equation above may be very time consuming due to the strict requirements on time steps necessary to capture the wave frequency motions. A common approach is to use a multiple scale approach and separate the wave-frequency part from the low-frequency part. The wave-frequency part is usually solved in the frequency domain, which requires the motions to be linear responses to waves. This means that the quadratic damping  $D_2$  is set to zero and the stiffness  $K$  is constant.

The exciting force is separated in a wave-frequency part  $q_{WA}^{(1)}$  and a low-frequency part  $q^{(2-)} = q_{WI} + q_{WA}^{(2-)} + q_{CU} + q_{ext}$ .

The first and second-order (sum frequency) wave frequency response  $x_{WA}^{(1)}$  and  $x_{WA}^{(2+)}$  are usually solved in the frequency domain while the low frequency response  $x_{LF} = x_{WA}^{(2-)}$  is solved in the time-domain

$$(m + A(0)) \ddot{x}_{LF} + D_1 \dot{x}_{LF} + D_2 f(\dot{x}) + Kx_{LF}$$

$$= q_{WI} + q_{WA}^{(2-)} + q_{CU} + q_{ext}$$

It should be noted that there are standard procedures and commercial software for calculating the wave-frequency response in the time-domain.

#### 5.3.3 Slender structure representation

Restoring from mooring lines and risers is normally represented in terms of a tabulated quasi-static restoring force as a function of displacement. This information is used as a 'look-up' table for restoring forces for a given floater position in the de-coupled analyses. Linear interpolation is normally applied between tabulated values. The restoring force characteristics can be provided by static analyses of each mooring line/riser using state-of-the-art slender analysis computer tools or more simplified calculations, e.g. catenary solutions.

It is important to observe that slender structure dynamics is not included in de-coupled analysis. This means that damping of LF floater motions due to slender structures is not included in de-coupled analyses. As discussed in 2.2, this effect is important for most deep water concepts.

Furthermore, current loading is normally not considered in the restoring characteristics, as this would have required recalculation of the restoring force characteristics for each current condition. The total force on the floater from current loading on the slender structures can be substantial for deep water systems. It is important to note that this force in most cases is not accounted for by the restoring force characteristics.

Seafloor/slender structure frictional forces can not be represented by the use of restoring force characteristics. This is

because frictional effects depends on the displacement history and hence they are impossible to include via restoring force characteristics.

### 5.3.4 Slender structure/floater coupling effects

Separate assessment of slender structure/floater coupling effects is required due to the simplified representation of slender structures in de-coupled analyses.

The force on the floater from current loading on the slender structures can be assessed by static analyses using a standard slender analysis computer tool or by a coupled model of the total system.

The contribution from current loading on the slender structures can be applied directly as an additional force on the floater in de-coupled analyses.

LF slender structure damping can be assessed with basis in coupled analyses (see 6.3.1) or from model tests. For an outline of techniques for estimation of an equivalent linear damping, reference is given to Ormberg *et al* (1998) and Astrup *et al* (2001). The estimated damping coefficient can be included directly in the linear damping matrix for the floater in the de-coupled analysis.

It should however be emphasised that the LF damping from the slender structures for some systems is sensitive to the environmental excitation (wave height, period, current etc). Estimates of LF damping should therefore preferably be based on the same environmental condition as considered in the de-coupled floater motion analysis.

## 6. Coupled Response Analyses

### 6.1 General methodology

The floater, risers and mooring system comprise an integrated dynamic system responding to environmental loading due to wind, waves and current in a complex way. The floater motions may contain the following components:

- Mean response due to steady current, mean wave drift and mean wind load
- WF response due to 1<sup>st</sup> order wave excitation
- LF response due to wave drift, wind gusts and viscous drift
- HF response (TLP)
- Hull VIM (Classic/Truss Spar, Mini TLP, DDF)

These response components will consequently also be present in the slender structure response. Furthermore, the WF, LF and HF components are generally described as stochastic processes.

The purpose of a coupled analysis is to accurately predict the floater motions as well as the mooring and riser system response with due regard of floater/slender structure coupling effects. Such analyses need to be conducted for numerous stationary design conditions to cover extreme conditions, fatigue load cases, accidental conditions as well as temporary conditions. Furthermore, analysis of several modifications of the design should be foreseen as a part of the design process. Hence, computational efficiency and numerical stability is a key issue in practical design analyses of floating offshore installations.

A coupled dynamic model of a floating installation can in principle be obtained by introduction of the rigid body floater model in a FE model of the complete mooring and riser system. Such models can be quite complex, and a 'master-slave' approach is an efficient technique for connecting relevant mooring lines/tethers/risers to the floater. The availability of beam- and bar elements in the FE code is essential for efficient modelling of the slender structures.

Solution of this coupled system of equations in time domain

using a non-linear integration scheme will ensure consistent treatment of floater/slender structure coupling effects, i.e. these coupling effects will automatically be included in the solution.

The floater load model accounts for the mass, hydrostatic stiffness, frequency dependent added mass and damping as well as excitation from wind, waves and current on the floater. Note that the floater model applied in coupled analyses is in general identical to the floater model applied in de-coupled analyses (see Section 5). The differences are the solution strategy and the slender structure representation. The slender structure analysis computer program is the most computationally intensive process and hence governs the efficiency and stability of the coupled analyses.

The dynamic loading from wind and waves is modelled as stationary stochastic processes in a coupled analysis. Simulations of 3-6 hours will be required to obtain extreme response estimates with sufficient statistical confidence. This is of particular importance for response quantities with significant LF components.

The general modelling capabilities of FE slender analysis codes can be applied to establish models of complex offshore installations. It is in principle straightforward to establish coupled models of systems with two or more moored floaters, possibly interconnected by e.g. flow lines. Hydrodynamic interaction between the floaters may be accounted for in the hydrodynamic radiation/diffraction analyses to establish wave forces on the floaters. See 3.3.1 for further details.

### 6.2 Coupled system analysis

Floater response as well as detailed mooring line/riser response can be computed by coupled analysis using a detailed model of the total system. This approach is usually termed '*Coupled System Analysis*' reflecting that all relevant system responses are computed directly by the fully coupled response model. This approach may be suitable for rather simple systems where adequate mooring line and riser response can be predicted by fairly simple FE models.

*Selected modelling* may be applied to gain computational efficiency for more complex systems. In this approach, detailed models of identified critical slender structures are included in the coupled model (e.g. 1-2 detailed riser models for critical slots); otherwise a rather coarse slender structure model is applied to reduce the computational efforts. This model will hence behave as a coupled system analysis as described above for the selected critical slender structures.

### 6.3 Efficient analysis strategies

Coupled analyses normally demand substantial computational efforts. More efficient computation schemes are therefore needed for use in practical design analyses.

Several strategies can be proposed to achieve computational efficiency. All strategies have in common that the floater motion and slender structure analyses are carried out separately, with the exception of selected modelling stated earlier. The first step is always a floater motion analysis. Computed floater motions are then applied as loading in terms of forced boundary displacements in subsequent slender structure analysis. Critically loaded risers and/or mooring lines can then be analysed one by one in the slender structure analyses. This scheme contributes to computational flexibility as well as a significant reduction in computation time.

#### 6.3.1 Coupled floater motion analyses

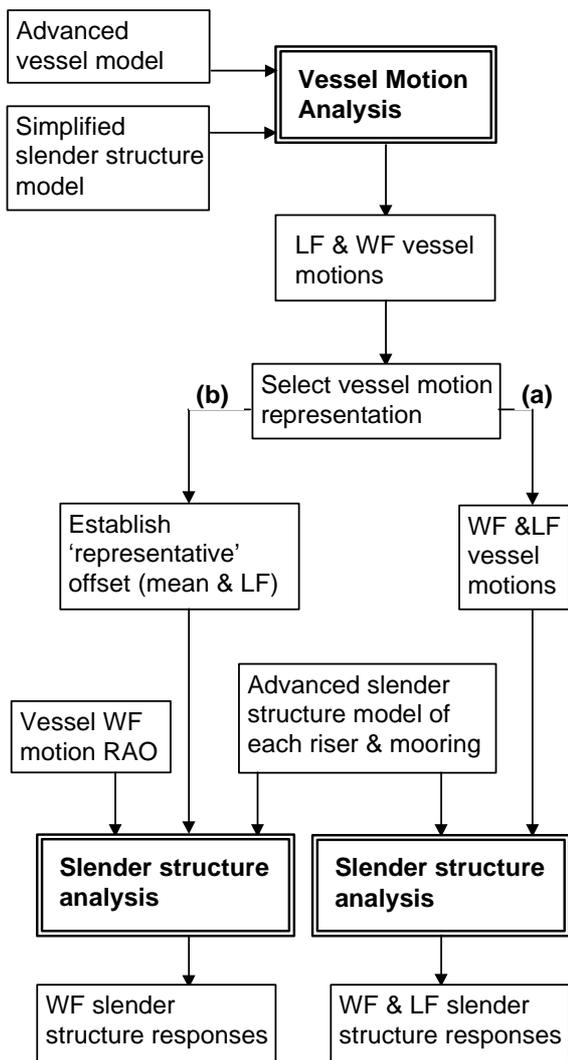
##### 6.3.1.1 Methodology

Coupled floater motion analysis in combination with subsequent slender structure analysis is generally recommended to achieve a more efficient and flexible analysis scheme. A flow chart for this approach is shown in Figure 6-1. Through careful

modelling, this approach is capable of predicting floater motions and detailed slender structure response with the same precision as the complete coupled system analysis.

The primary purpose of a *coupled floater motion analysis* is to give a good description of floater motions; detailed slender structure response is secondary. It can therefore be proposed to apply a rather crude slender structure FE model (e.g. crude mesh, no bending stiffness in risers, etc) in the coupled analysis still catching the main coupling effects (restoring, damping and mass). The numerical solution technique as well as floater force model is however identical to the approach applied in the coupled system analysis as described in 6.2.

This approach gives a significant reduction in computation time due to a reduced number of degrees of freedom in the coupled analyses. Computation times close to real time have been experienced for quite complex FPSO and Spar systems (Ormberg *et al* 1998, Astrup *et al* 2001)



**Figure 6-1**  
**Analysis Strategy 1: Coupled floater motion and slender structure analysis**

Different alternatives for interfacing coupled motion analysis with subsequent slender structure analysis is shown in the flow chart in Figure 6-1. The most direct way to proceed is to apply time series of floater motions (typically combined WF and LF motions) computed by the coupled floater motion analysis as boundary conditions in the slender structure analyses (branch a). This approach will also capture possible LF slender structure dynamics as well as influence from LF response (possibly

quasi-static) on the WF response. Such effects may be important for some deepwater mooring lines and riser designs.

Traditional assumptions can alternatively be applied considering WF floater motion as dynamic excitation while LF floater motions are accounted for by an additional offset (branch b). The slender structure is consequently assumed to respond quasi-statically to LF floater motions.

### 6.3.1.2 Modelling considerations

The principle applied to establish an adequate simplified slender structure model will depend on the actual system layout as well as the required output from the analyses. The primary requirement is to give an adequate representation of the coupling effects. However, it is also often desirable to establish some key results for the mooring and riser system directly as output from the coupled floater motion analyses. Such information can be used to identify critically loaded slender structures to be analysed in more detail.

In most situations, it is convenient to include all mooring lines, tethers and risers in the FE slender structure model. The FE model of each slender structure component is simplified to the extent possible using a rather rough mesh and omission of bending stiffness for most parts of the riser system. This will allow for output of key slender structure responses, e.g. mooring line tensions at fairlead, riser top tensions, tensioner stroke etc., directly from the coupled floater motion analysis.

More detailed riser responses requiring a refined FE model of the riser system are carried out separately in dedicated riser analyses to save computation time and increase the analysis flexibility. Examples are modelling of special components such as taper joints as well as refined mesh for adequate calculation of moment, shear and curvature in critical areas, e.g. touch-down area for SCRs.

Further simplifications are possible if the primary purpose of the coupled analyses is the prediction of floater motion response. These simplifications may involve the use of equivalent models for groups of mooring lines, tethers and risers. This will give a less transparent overview of the slender structure response but may be fully acceptable for description of the floater motion response with due regard of coupling effects.

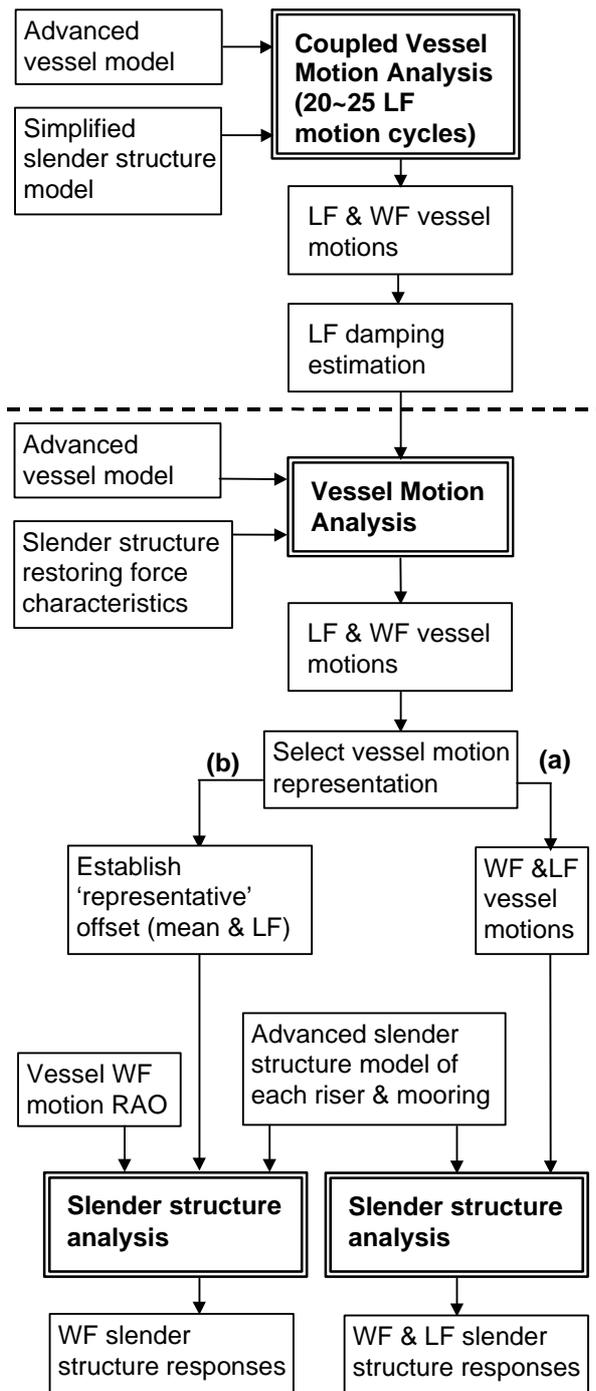
With some modeling experience, it is not difficult to model a slender structure yielding efficient computations while keeping the coupling effects. Some additional guidance is given in Appendix A for verification of the slender structure model.

### 6.3.2 Combined coupled / de-coupled analyses

Efficient computations can also be obtained using de-coupled analyses in combination with coupled analyses. The main idea is to estimate coupling effects (typically LF damping) from a rather 'short' coupled floater motion analyses. The estimated coupling effects are used as input to subsequent de-coupled analysis. The efficiency of the de-coupled analysis allows for 'long' simulations to achieve the required statistical confidence in the results.

It should however be emphasised that the coupling effects from the slender structures for some systems are sensitive to the environmental excitation. Estimates of coupling effects should therefore preferably be based on the same environmental condition as considered in the de-coupled floater motion analysis.

The described approach is convenient for analysis of turret moored FPSOs as all relevant dynamic coupling effects are described by the LF surge damping. Coupling effects from current loading on the slender structures can be assessed from a static coupled analysis. An outline of this approach applied to a FPSO is shown in Figure 6-2. Refer to Ormberg *et al* (1997,1998) for further details.



**Figure 6-2**  
**Analysis strategy 2: Vessel motion and separate slender structure analysis**

The methodology has also been applied to the analysis of Spar platforms (Astrup *et al* 2001). However, less accurate results were reported due to the complex Spar platform response. Refer to Astrup *et al* (2001) for further details.

Combined use of coupled/de-coupled analyses is generally considered less accurate than coupled floater motion analyses. It is however considered as a useful supplement for coupled analysis, especially for screening purposes and sensitivity studies.

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## APPENDIX A SELECTION OF DRAG COEFFICIENTS

### A.1 General

In global analysis of deepwater floater concepts, the selection of drag coefficients ( $C_d$ ) is essential for calculation of viscous drag forces on large volume structures, e.g. Spar hull and TLP columns, as well as slender structures, e.g. risers, mooring lines and tendons.

For a specified body shape, the drag coefficients depend on the following parameters:

- Reynolds number  $Re = UD/\nu$  ( $U$  = characteristic free stream velocity,  $D$  = characteristic dimension of the body,  $\nu$  = kinematic viscosity coefficient)
- Keulegan-Carpenter number  $KC = U_m T/D$  for oscillatory planar flow with velocity  $U_m \sin[(2\pi/T)t + \varepsilon]$  past a fixed body
- Roughness number  $\Delta = k/D$  ( $k$  = characteristic cross-sectional dimension of the roughness on the body surface)

The drag coefficient is generally a function of these 3 parameters  $C_d = C_d(Re, KC, \Delta)$ .

In general, most marine structures in operational conditions are subjected to high Reynolds number flow. For example, the Reynolds number for a current with velocity 1 m/s past a TLP column with diameter 15 m is  $1.4 \cdot 10^7$  at 20°C. Similarly for a riser with diameter 0.3 m and the same current velocity yields  $Re = 2.8 \cdot 10^5$ .

Recommended values for drag coefficients are given in Refs. [A1-1], [A1-2] and [A1-3].

### A.2 Drag coefficients for slender structures

For slender structures in deep water it is crucial to apply proper drag coefficients for calculating damping contributions as well as current and wave loading.

For deepwater regular waves, the horizontal fluid particle velocity decays exponentially ( $e^{2\pi z/\lambda}$ ) with water depth. At water depth larger than the wave length  $\lambda$  there is hardly any wave disturbance, so that the contribution to the fluid particle velocity from the waves can be neglected. A slender structure can therefore be divided into two zones along its length: the free-surface zone where both waves and current are acting, and the zone far from the free surface where only current is acting. Rationally, different drag coefficients should be applied for different water depth because the flow characteristics are different.

In steady current flow, the KC number is not of relevance and the drag coefficients are only functions of the Reynolds number and the roughness number, i.e.,  $C_d(Re, \Delta)$ . The two-dimensional drag coefficients for circular cylinders of various roughness as a function of  $Re$  are given in DNV-RP-C205 Figure 6-6, Ref. [A1-1]. Other valuable references are [A1-1] and [A1-2]. Increasing the roughness alters both the magnitude and the shape of the  $C_d$  curves and this has to be taken into account in areas with marine growth. A normal approach for implementing the marine growth is to scale up the drag coefficient as follows:

$$C_{d\text{ growth}} = C_d [1 + 2(\Delta T_{\text{growth}}/D)]$$

where  $\Delta T_{\text{growth}}$  is marine growth thickness and  $D$  the bare diameter of the slender structure.

In the free-surface zone with both waves and current actions,  $C_d$  also depends on the KC number.

Table A-1 presents some model testing and full scale results. The following observations are made:

- Larger scatter in model scale  $C_d$  ( $Re < 1.2 \cdot 10^3$ ) compared

to full scale  $C_d$

- Model scale  $C_d$  is generally higher than full scale  $C_d$  (especially for  $Re < 100$ ).
- $C_d$  dependence on  $Re$  is evident
- Correlations with model tests need to take into account the  $C_d$  dependence on  $Re$ .

Ideally, when performing a coupled analysis, the coefficient dependence on  $Re$ ,  $KC$  and roughness number should be implemented by choosing coefficients from tables and curves during the analysis. However, present state of the art within coupled analyses usually does not make use of this option.

Based on the above it is difficult to come up with simple recommendations on which drag coefficients to be used. One clear recommendation is to check a range of coefficients since they influence both the excitation side as well as the damping side of the equations.

Table A-2 gives some guidance on typical  $C_d$  values for Reynolds numbers in the range  $10^4 - 10^7$ . These drag coefficients are two-dimensional, normal to the longitudinal slender structure axis and without effects of marine growth or any influence from VIV (increased drag due to cross-flow vibrations) included.

Strakes, or fairings may be needed for parts of the risers (e.g. SCRs and TTRs). For those designs, special evaluations (strake numbers/ heights/pitch,  $A/D$  ratio,  $V_r$ ) have to be made to determine appropriate drag coefficients.

### A.3 Drag coefficients for large volume structures

For complex hull forms, model testing has to be performed to determine the current drag coefficients. Directional variability is usually strong with respect to forces and especially moments. One example is the importance of estimating the correct yaw moment due to current loads acting on a FPSO.

For simple hull forms like columns and pontoons on TLPs and Semis, appropriate  $C_d$  values may be determined from published literature, e.g. Ref. [A1-1]. These large columns/pontoons will usually operate at high Reynolds numbers and low  $KC$  numbers. It is important to note and take into account the increase in  $C_d$  for  $KC < 20$  (Ref. [A1-1]).

Most tabulated drag coefficients are two-dimensional. The reduction in  $C_d$  due to three dimensional effects can be taken into account by a reduction factor  $\kappa$  given as a function of the ratio  $L/D$  ( $L$  = length of member,  $D$  = cross-sectional dimension),  $C_d^{3D} = \kappa C_d^{2D}$ . See e.g. Ref. [A1-1], Table 6-2.

Spar hulls have strakes attached to the main shell hull. The strakes give an increase in drag on the hull and may be included as follows:

$$C_d = C_{do} + 2 \cdot C_{\text{strake}} \cdot h/D$$

- $C_{do}$  = initial drag coefficient for a bare cylinder (typical 0.7 – 0.9)
- $C_{\text{strake}}$  = 2  $D$  drag coefficient for a strake (typical 2.0)
- $h$  = strake height (typical 0.1  $D$  – 0.15  $D$ )
- $D$  = spar hull diameter

In cases with strong loop current and extensive cross flow motions there will be an amplification in the in-line drag coefficient which is described by:

$$C_{dVIM} = f(V_r, A/D)$$

where  $V_r$  is the reduced velocity (see 3.6 Vortex-induced loads). Carefully planned and conducted model tests are usually the best option for establishing the  $A/D$  and  $C_{dVIM}$  values as a function of  $V_r$ . Careful interpretation of model test results is also a key issue. If full scale measurements (platform excur-

sions and current velocities) are available, this will certainly add valuable information and validation.

<b>Table A-1 Measured model and full scale <math>C_d</math> for wire, chain and risers</b>		
$Re$	$C_d$	Notes
<b>Wire, model scale</b>		
11 – 140	2.0 – 1.0	D = 0.65 – 3 mm, towing
13 – 120	1.1 – 0.9	Scale: S200 – S55
120 – 14000	0.8 – 1.1	D = 1.1 – 3.8 mm, towing
<b>Chain, model scale</b>		
13 – 110	3.0 – 2.5	D = 1.05 mm, towing
13 – 120	2.5 – 1.8	Scale: S200 – S55
<b>Risers, model scale</b>		
120 – 1100	1.4 – 1.15	Equiv. Risers, S200 – S55
100	2.0	Single J, Scale S150
<b>Wire, full scale</b>		
104 – 1.4·10 <sup>4</sup>	1.1 – 0.95	D = 1.1 – 38 mm, drop tests
1.4·10 <sup>4</sup> –1.1·10 <sup>5</sup>	1.05 – 0.90	D = 78 mm, towing
10 <sup>5</sup>	0.83	D = 147 mm, vel. = 1 m/s
<b>Chain, full scale</b>		
1.4·10 <sup>3</sup> –10 <sup>4</sup>	2.7 – 2.1	D = 30 mm, KC = 163 – 306
10 <sup>4</sup> –1.3·10 <sup>4</sup>	2.7 – 2.2	D = 30 mm, towing
1.3·10 <sup>4</sup> – 1.1·10 <sup>5</sup>	2.5 – 1.7	D = 65 mm, towing
1.05·10 <sup>5</sup>	1.4	D = 140 mm, vel. = 1 m/s
<b>Risers, full scale</b>		
1.1·10 <sup>5</sup>	1.15	D = 200 mm

<b>Table A-2 Typical two dimensional drag coefficients, <math>C_d</math> for <math>Re = 10^4 – 10^7</math>.</b>	
Type	$C_d$ range
Wire, six strand	1.5 – 1.8
Wire, spiral, no sheathing	1.4 – 1.6
Wire, spiral with sheathing	1.0 – 1.2
Chain, stud (relative chain diameter)	2.2 – 2.6
Chain studless (relative chain diameter)	2.0 – 2.4
Metallic risers	0.7 – 0.9
Flexible risers	0.8 – 1.0

#### A.4 Inertia and drag coefficients for heave plates

Special attention must be given to the heave plates for a Truss Spar. Heave plates contribute by increasing the vertical added mass and damping forces. The contribution to added mass can be calculated by modelling the plates in a sink/source analysis.

Singularities will be introduced if sinks/sources are distributed on each side of a thin plate with thickness much smaller than the characteristic panel size. To avoid such problems, the heave plates can be made artificially thicker. The heave added mass is not sensitive to the thickness for small thickness values and the contribution to the sway/surge added mass is small compared to the contribution from the hard tank. An alternative is to model the heave plate as a dipole sheet if the wave diffraction computer program [A1-5] has this option.

The heave added mass for a square plate in an infinite fluid is given by the formula  $M_a = 0.145\rho\pi b^3$  where  $b$  is the horizontal dimension of the plate. (Ref. [A1-1]).

The vertical motion of the Spar is very small compared to the horizontal dimension of the heave plate. This means that the KC number for periodic motion

$$KC = 2\pi \frac{z_{WA}}{b}$$

( $z_{WA}$  is the heave motion) is very small while the Reynolds number  $Re$  is in the order of  $10^6$ . The drag coefficient in this flow regime is very sensitive to KC and increases strongly with decreasing KC, but is rather insensitive to  $Re$ . A formula for the drag coefficient of a long, thin plate strip is given in Ref. [A1-6],

$$C_d = 15KC^{-1/2} \exp(1.88/Re^{0.547})$$

3D effects will reduce the drag coefficient. Heave plate drag coefficients have been found to be in the order of 5-10 for existing Truss Spar designs. It is recommended that model tests or CFD calculations are performed to verify drag coefficients to be used for the heave plates.

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