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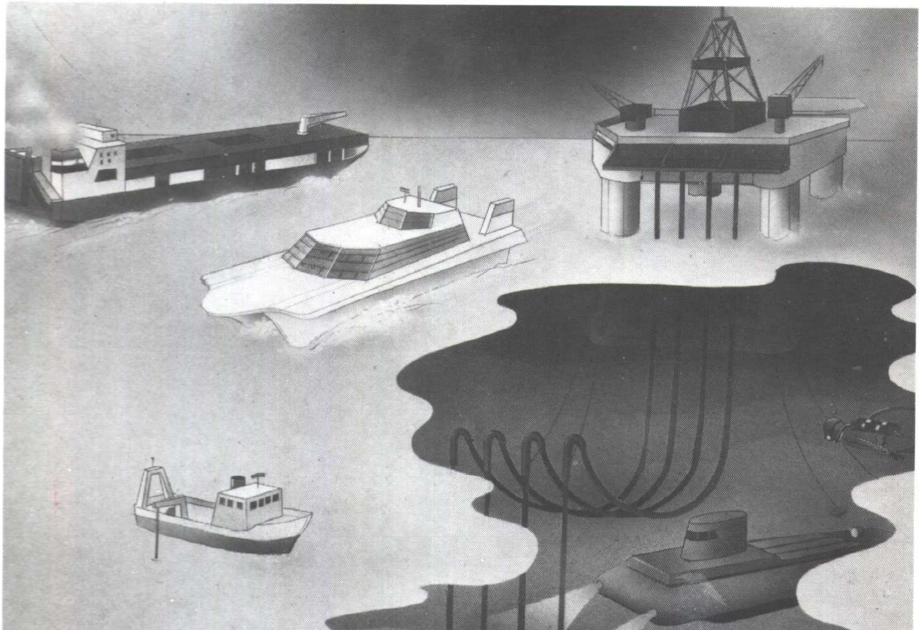
# SEA LOADS ON SHIPS AND OFFSHORE STRUCTURES



O.M. Faltinsen

# SEA LOADS ON SHIPS AND OFFSHORE STRUCTURES

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# CONTENTS

# 1 INTRODUCTION

Knowledge about wave induced loads and motions of ships and offshore structures is important both in design and operational studies. The significant wave height (the mean of the highest one-third of the waves) can be larger than 2 m for 60% of the time in hostile areas like the North Sea. Wave heights higher than 30 m can occur. The mean wave period can be from 15 to 20 s in extreme weather situations and it is seldom below 4 s. Environmental loads due to current and wind are also important. Extreme wind velocities of 40 to 45 m $s^{-1}$ have to be used in the design of offshore structures in the North Sea.

Fig. 1.1 shows five examples of offshore structures. Two of them, the jacket type and the gravity platform, penetrate the sea floor. At present, fixed structures have been built for water depths up to about 300 m. Two of the structures, the semi-submersible and the floating production ship, are free-floating. The tension leg platform (TLP) is restrained from oscillating vertically by tethers, which are vertical anchorlines that are tensioned by the platform buoyancy being larger than the platform weight. Both the ship and the semi-submersible are kept in position by a spread mooring system. An alternative would be to use thrusters and a dynamic positioning system. Pipes (risers) are used as connections between equipment on the sea floor and the platform.

Ships serve a large variety of purposes. Examples are transportation of goods and passengers, naval operations, drilling, marine operations, fishing, sport and leisure activities. Fig. 1.2 shows three types of ships: a monohull, a SWATH and a SES. The monohull is exemplified by a LNG (liquid natural gas) carrier with spherical tanks. SWATH stands for small-waterplane-area, twin-hull ship and consists of two fully-submerged hulls that are connected to the above water structure by one or several thin struts. Between the hulls there may be fitted fins or foils as in Fig. 1.2. SES (surface effect ship) is an air-cushion supported high-speed vehicle where the air-cushion is enclosed on the sides by rigid sidewalls and on the bow and stern by compliant seals. By high speed we mean high Froude number (Fn). This is defined as $\mathrm{Fn} = U/(Lg)^{\frac{1}{2}}$ ($U$ = ship speed, $L$ = ship length, $g$ = acceleration of gravity). A ship is considered a high-speed marine vehicle when $\mathrm{Fn} > \approx 0.5$. From a

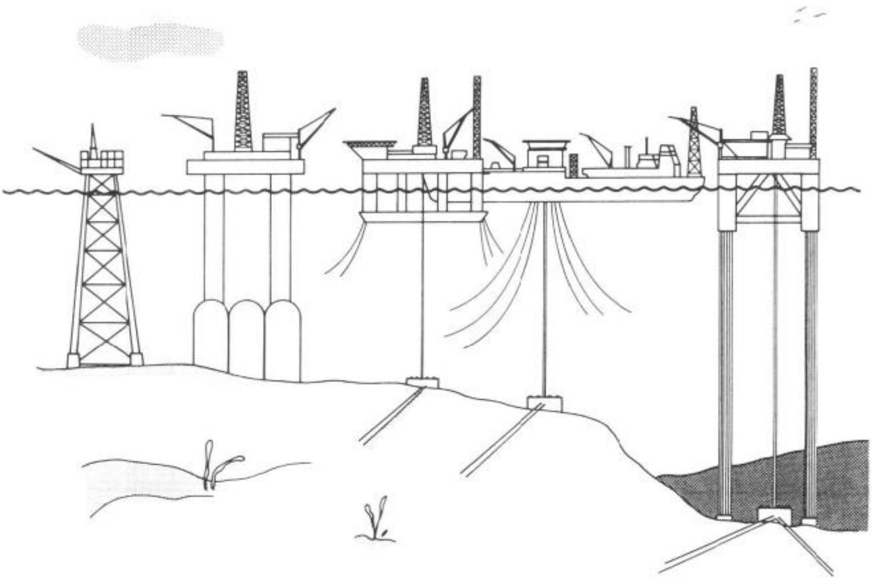


Fig. 1.1. Five types of offshore structures. From left to right we have, jacket, gravity platform, semi-submersible, floating production ship, tension leg platform (TLP). (Partly based on a figure provided by Veritec A/S.)

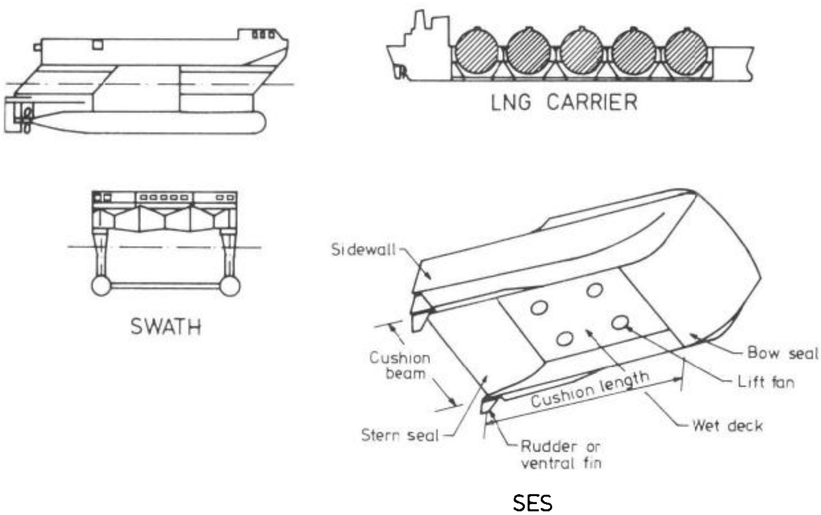



Fig. 1.2. Three types of ships. SWATH (small-waterplane area, twin-hull ship), LNG (liquid natural gas) carrier, SES (surface effect ship).

hydrodynamical view point one can distinguish between ships at zero, normal and high speed. SWATH concepts have been designed for both normal and high-speed applications.

Most of the applications presented in the main text will deal with ships at zero or normal speed and with offshore structures. Applications to high-speed marine vehicles will be given by exercises. We will discuss both wave-induced loads and motions, with motions being the result of integrated hydrodynamic loads on the structure. In the introduction we will give a survey of important wave load and seakeeping problems for ships and offshore structures. Before doing that we need to define the motions.

## DEFINITIONS OF MOTIONS

Motions of floating structures can be divided into wave-frequency motion, high-frequency motion, slow-drift motion and mean drift. The oscillatory rigid-body translatory motions are referred to as surge, sway and heave, with heave being the vertical motion (see Fig. 1.3). The oscillatory angular motions are referred to as roll, pitch and yaw, with yaw being rotation about a vertical axis. For a ship, surge is the longitudinal motion and roll is the angular motion about the longitudinal axis.

The wave-frequency motion is mainly linearly-excited motion in the

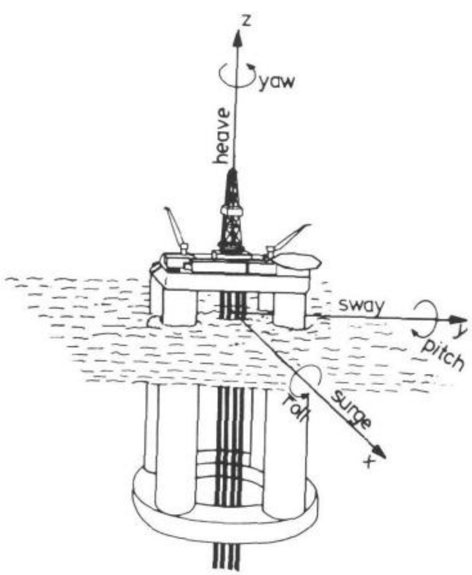
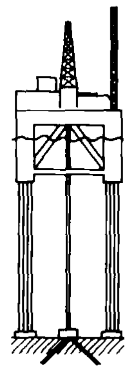
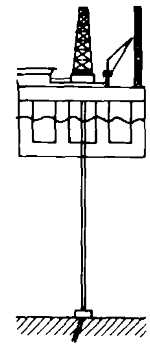
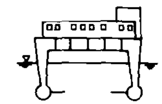



Fig. 1.3. Definition of rigid-body motion modes. Exemplified for a deep concrete floater.

Table 1.1. *Resonant heave oscillations of ships, offshore structures and high speed vehicles*

| | | | | | |
|---|---|---|---|---|---|
| *Vessel*: |  |  |  |  |  |
| | SES (surface effect ship) | TLP (tension leg platform) | Monohull ship Catamaran | Semi-submersible | SWATH (small waterplane area twin hull ship) |
| *Natural heave period*: | <1 s | 2–4 s | 4–16 s[a] | >20 s | >20 s |
| *Restoring force*: | Air compressibility | Elasticity of tethers | Waterplane area | Waterplane area | Waterplane area |
| *Dominating excitation mechanism around the natural heave period*: | Linear wave forces due to high encounter frequency between ship and waves | Non-linear sum frequency wave forces | Linear wave forces | Swell (long waves) | Linear wave forces due to low encounter frequency between ship and waves |
| *Important damping*: | 'Ride Control' | Viscous effects | Wave radiation | Viscous effects | Foil control |

[a] Rough estimate: $\sqrt{(L/1.5)}$, where $L$ is ship length in metres.

wave-frequency range of significant wave energy. High-frequency motion is significant for TLPs and is often referred to as 'ringing' or 'springing' and is due to resonance oscillations in heave, pitch and roll of the platform. The restoring forces for the TLP are due to tethers and the mass forces due to the platform. The natural periods of these motion modes are typically 2–4 s which are less than most wave periods. They are excited by non-linear wave effects.

Similar non-linear effects cause slow drift and mean motions in waves and current. Wind will also induce slow drift and mean motion. Slow drift motion arises from resonance oscillations. For a moored structure it occurs in surge, sway and yaw. The restoring forces are due to the mooring system and the mass forces due to the structure. Typical resonance periods are of the order of 1 to 2 minutes for conventionally moored systems.

Heave is an important response variable for many structures. Table 1.1 illustrates the range of the natural heave periods of different types of marine structures. These include SES, TLPs, monohull ships, catamarans, SWATH ships and semi-submersibles. The table indicates how the natural heave oscillations can be excited. For instance for the SES-hull it occurs due to high encounter frequency between the ship and the waves, while for the SWATH it occurs due to low encounter frequency between the ship and the waves. The table also shows what types of restoring forces can cause heave resonance. For the SES it is the compressibility effect of the air in the cushion. For the monohull ship, catamaran, SWATH and semi-submersible it is due to change in buoyancy forces. This is related directly to the waterplane area of the vessels. Finally we see in Table 1.1 either the most important physical source of natural heave damping or how one artificially increases the damping by control systems.

For the SES it is the heave accelerations and not the heave motions that are important. If no 'ride control' is used, acceleration values of $1.5g$ can occur in relatively calm sea. If the natural heave period is 0.5 s, it means the heave amplitude is $\approx 0.1$ m.

A semi-submersible is designed to avoid resonance heave motion and the maximum heave motion in severe sea states will be less than half the maximum wave amplitude.

## TRADITIONAL SHIP PROBLEMS

Examples of important seakeeping and wave load problems for ships are illustrated in Fig. 1.4. In particular, vertical accelerations and relative vertical motions between the ship and the waves are important responses. Accelerations determine loads on cargo and equipment and are an important reason for seasickness. The relative vertical motions can be used to evaluate the possibility and damage due to slamming and water on deck. (Slamming means impact between the ship and the water.) For a ship it is important to avoid slamming as well as water on deck because of the resulting local damage of the structures.

Rolling may be a problem from an operational point of view of fishing vessels, crane vessels, passenger ships and naval vessels. Means to reduce

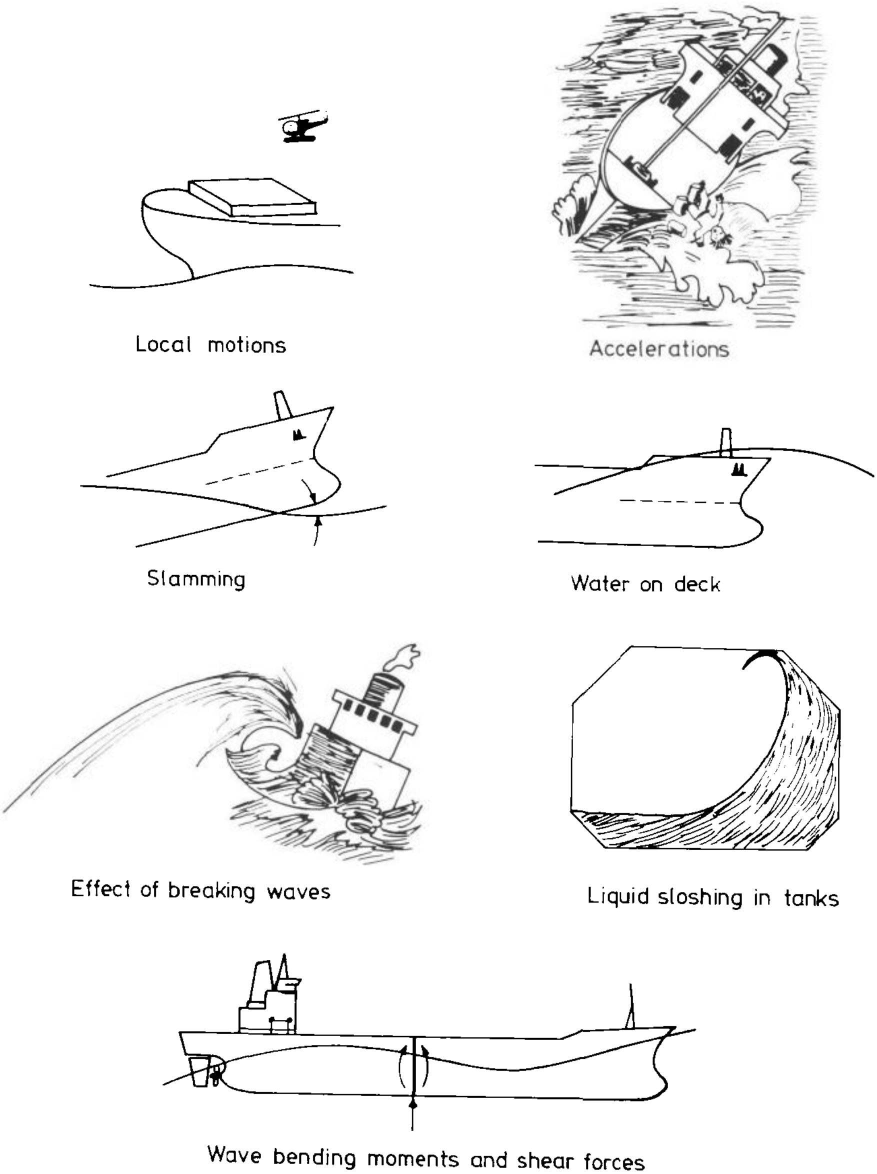



Fig. 1.4. Examples of important seakeeping and wave load problems for ships.

the rolling of a ship are therefore of interest. Examples are bilge keels, anti-roll tanks and active fins. For smaller ships, rolling in combination with either wind, water on deck or motion of the cargo can cause the ship to capsize. Another important reason for capsizing of smaller ships is breaking waves. Several accidents off the Norwegian coast have been explained by breaking waves. Following sea can cause different critical capsizing situations. If the wave profile is stationary relative to the ship, the ship may be statically unstable in roll relative to the waterline defined by the wave profile. The ship may also lose its directional stability in following waves. This can happen when the frequency of encounter between the ship and the waves is small. The result is an altered course relative to the waves. This situation is called 'broaching' and is most critical with respect to capsizing of ships with small static stability.

Liquid sloshing in tanks may be a problem for bulkships, combination ships oil–bulk–ore (OBO), liquid natural gas (LNG) carriers and tankers loading at offshore terminals. There are two reasons why the fluid motion in a tank can be violent. One is that a natural period for the fluid motion in the tank is in a period domain where there is significant ship motion. The other reason is that there is often little damping connected with fluid motion in a tank. If the excitation period is close to a natural period for the tank motion, a strong amplification of the fluid motion in a tank will occur. Liquid sloshing can cause high local pressures as well as large total forces. Both effects may be important in design.

For larger ships, wave-induced bending moments, shear forces and torsional moments are important. More specific problems are whipping and springing. Whipping is transient elastic vibration of the ship hull girder caused for instance by slamming. Springing is steady-state elastic vibration caused by the waves and is of special importantce for larger oceangoing ships and Great Lake carriers. Springing is due to both linear and non-linear excitation mechanisms. The linear exciting forces are associated with waves of small wavelengths relative to the ship length.

Ship motions and sea loads can influence the ship speed significantly due to voluntary and involuntary speed reduction. Voluntary speed reduction means that the ship master reduces the speed due to heavy slamming, water on deck or large accelerations. Involuntary speed reduction is the result of added resistance of the ship due to waves and wind and changes in the propeller efficiency due to waves. The importance of involuntary speed reduction is exemplified in Fig. 1.5. It shows the results of computer calculations for a container ship at a given sea state. The significant wave height $H_{\frac{1}{3}}$ is 8.25 m. The waves are assumed longcrested with different propagation directions relative to the ship. The ship has a length of 185 m. The actual speed at constant engine power is given for different wave headings together with the design speed in still water at the same engine power. For instance in head seas the ship speed is 8 knots (4.1 m $s^{-1}$) compared to 16.2 knots (8.3 m $s^{-1}$) in still water. Depending on the wave direction, the actual ship speed may be lower than that shown in Fig. 1.5. This is due to voluntary speed reduction. Information like this may be used to choose optimum ship routes based on relevant criteria like the lowest fuel consumption or the shortest time of voyage.

Criteria for acceptable levels of ship motions have been discussed in the Nordic co-operative project 'Seakeeping performance of ships' (NORDFORSK, 1987). Considerations have been given to hull safety, operation of equipment, cargo safety, personnel safety and efficiency. General operability limiting criteria for ships are given in Table 1.2. Criteria with regard to accelerations and roll for special types of work and for passenger comfort are given in Table 1.3. The limiting criteria for fast small craft are only indicative of trends. A fast small craft is defined as a vessel under about 35 metres in length with speed in excess of 30 knots. A reason why the vertical acceleration level for fast small craft is set higher than for merchant ships and naval vessels, is that personnel can tolerate higher vertical acceleration when the frequency of oscillation is high.

## OFFSHORE STRUCTURE PROBLEMS

For drilling operations heave motion is a limiting factor. The reason is that the vertical motion of the risers has to be compensated and there are limits to how much the motion can be compensated. An example of a

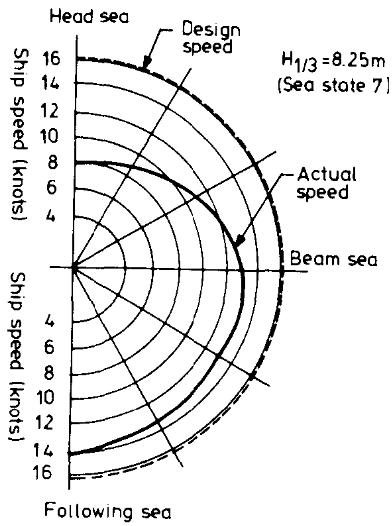



Fig. 1.5. Effect of added ship resistance due to waves and wind (involuntary speed reduction). Ship length = 185 m. ($H_{\frac{1}{3}}$ = significant wave height).

Table 1.2. *General operability limiting criteria for ships* (*NORDFORSK*, 1987)

| | Merchant ships | Naval vessels | Fast small craft |
|---|---|---|---|
| Vertical acceleration at forward perpendicular (RMS-value) | $0.275g$ ($L \leq 100$ m)<br>$0.05g$ ($L \geq 330$ m)[a] | $0.275g$ | $0.65g$ |
| Vertical acceleration at bridge (RMS-value) | $0.15g$ | $0.2g$ | $0.275g$ |
| Lateral acceleration at bridge (RMS-value) | $0.12g$ | $0.1g$ | $0.1g$ |
| Roll (RMS-value) | 6.0 deg | 4.0 deg | 4.0 deg |
| Slamming criteria (Probability) | 0.03 ($L \leq 100$ m)<br>0.01 ($L \geq 300$ m)[b] | 0.03 | 0.03 |
| Deck wetness criteria (Probability) | 0.05 | 0.05 | 0.05 |

[a] The limiting criterion for lengths between 100 and 330 m varies almost linearly between the values $L = 100$ m and 330 m, where $L$ is the length of the ship.
[b] The limiting criterion for lengths between 100 and 300 m varies linearly between the values $L = 100$ m and 300 m.

Table 1.3. *Criteria with regard to accelerations and roll* (*NORDFORSK*, 1987)

| Root mean square criterion | | | |
|---|---|---|---|
| Vertical acceleration | Lateral acceleration | Roll | Description |
| $0.20g$ | $0.10g$ | 6.0° | Light manual work |
| $0.15g$ | $0.07g$ | 4.0° | Heavy manual work |
| $0.10g$ | $0.05g$ | 3.0° | Intellectual work |
| $0.05g$ | $0.04g$ | 2.5° | Transit passengers |
| $0.02g$ | $0.03g$ | 2.0° | Cruise liner |

heave motion criterion is that the heave amplitude should be less than 4 m. It is therefore important to design structures with low heave motion so that it is possible to drill in as high a percentage of the time as possible. Semi-submersibles are examples of structures with very low heave motion in the actual frequency domain. Rolling may also be an important motion mode to evaluate, for example for operation of crane vessels or for transportation of jackets and semi-submersibles on ships and barges. Rolling, pitching and accelerations may represent limiting factors for the operation of process equipment on board a floating production platform.

In the design of mooring systems for offshore structures loads due to current, wind, wave-drift forces and wind- and wave-induced motion are generally of equal importance. There are two important design parameters. One is the breaking strength of the mooring lines. The other is the flexibility of the riser system which means, in practice, for a rigid riser system that the extreme horizontal offsets of the platform relative to the connection point of the riser to the sea floor should be less than say 10% of the water depth.

Wind, current, mean wave drift forces and slowly varying wave drift forces are also important in the design of thrusters and in station keeping of crane vessels, diving vessels, supply ships, offshore loading tankers and pipelaying vessels. Interaction of thrusters with other thrusters, the free-surface and structures may also be important for dynamic positioning systems, towing and marine operations in waves.

Examples of the main objectives of the hydrodynamic analysis of a tension leg platform are, to calculate the vertical dynamic loads on the platform with the purpose of estimating axial forces in the tethers and to calculate the wave elevation in order to evaluate the air gap between the waves and the underside of the platform. The minimum air gap is also an important consideration for other types of platforms.

## HYDRODYNAMIC CLASSIFICATION OF STRUCTURES

Both viscous effects and potential flow effects may be important in determining the wave-induced motions and loads on marine structures. Included in the potential flow is the wave diffraction and radiation around the structure. In order to judge when viscous effects or different types of potential flow effects are important, it is useful to refer to a simple picture like Fig. 1.6. This drawing is based on results for horizontal wave forces on a vertical circular cylinder standing on the sea floor and penetrating the free surface. The incident waves are regular. $H$ is the wave height and $\lambda$ is the wavelength of the incident waves. $D$ is the cylinder diameter. The results are based on the use of Morison's equation (see chapter 7) with a mass coefficient of 2 and a drag coefficient of 1. The linear McCamy & Fuchs (1954) theory has been used in the wave diffraction regime.

Let us try to use the figure for offshore structures. We will consider a regular wave of wave height 30 m and wavelength 300 m. This corresponds to an extreme wave condition. Let us consider wave loads on the caisson of a gravity platform where typical cross-sectional dimensions are 100 m. This implies equivalent $H/D$- and $\lambda/D$-values of 0.3 and 3, respectively. This means that wave diffraction is most important. If we consider the columns of a semi-submersible, a relevant diameter would be approximately 10 m. This implies $\lambda/D = 30$, $H/D = 3$, which means that the hydrodynamic forces are mainly potential flow forces in phase with the undisturbed local fluid acceleration. Wave diffraction and viscous forces are of less significance.

For the legs of a jacket a relevant diameter is approximately 1 m. This implies that viscous forces are most important. By viscous forces we do not mean shear forces, but pressure forces due to separated flow. The examples above are for an extreme wave condition. In an operational wave condition the relative importance of viscous and potential flow effects are different. We should bear in mind that Fig. 1.6 only provides a very rough classification. For instance, resulting forces may be small due to the cancelling out of effects of loads from different parts of the structure.

## ENGINEERING TOOLS

Both numerical calculations, model tests and full-scale trials are used to assess wave-induced motions and loads. From an ideal point of view full-scale tests are desirable but expensive and difficult to perform under controlled conditions. It may also be unrealistic to wait for the extreme weather situations to occur. Model tests are therefore needed. A drawback with model tests is the difficulty of scaling test results to full scale results when viscous hydrodynamic forces matter. The geometrical dimensions and equipment of the model test facilities may also limit the experimentıl possibilities.

Due to the rapid development of computers with large memory capacity and high computational speed, numerical calculations have

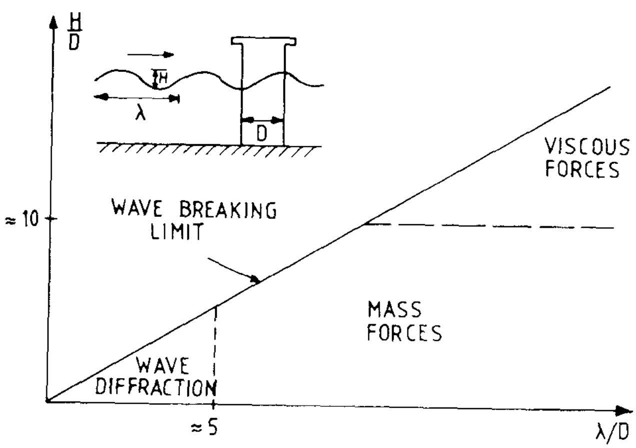



Fig. 1.6. Relative importance of mass, viscous drag and diffraction forces on marine structures.

played an increasingly important role in calculating wave-induced motions and loads on ships and offshore structures. A significant step in the development started about 1970. For offshore structures it was partly connected with the beginning of offshore oil and gas production and exploration in the North Sea. However, it is important to stress that numerical computer programs are also dependent on the development of hydrodynamic theories. More theoretical research is still needed, in particular to increase the knowledge on separated viscous flow and extreme wave effects on ships and offshore structures.

It is unrealistic to expect that computer programs will totally replace model tests in the foreseeable future. The ideal way is to combine model tests and numerical calculations. In some cases computer programs are not reliable. Model tests often give more confidence than computer programs when totally new concepts are tested out.

When computer programs have been validated and the theoretical basis of the computer program has been satisfactorily compared with experimental results, computer programs offer an advantage relative to model tests. Computer programs can often be used in a more efficient way than model tests to evaluate different designs in a large variety of sea conditions. However, sound judgement of results is always important. A basis for this is physical understanding and practical feeling.

One aim of the book is to provide physical understanding to the reader and try to simplify the problems mathematically. In this way one can develop simple tools to evaluate results from model tests, full-scale trials or computer programs.

# 2 SEA ENVIRONMENT

The intention of this chapter is to provide the basic information on waves, wind and current that is needed to evaluate sea loads and motions acting on ships and offshore structures. It is assumed that the reader has a basic knowledge of fluid mechanics and is familiar with the concepts of velocity potential and Bernoulli's equation. A brief survey of the general aspects of free-surface fluid flow problems based on potential theory is given below.

## BASIC ASSUMPTIONS

The sea water is assumed incompressible and inviscid. The fluid motion is irrotational. A velocity potential $\phi$ can be used to describe the fluid velocity vector $\mathbf{V}(x, y, z, t) = (u, v, w)$ at time $t$ at the point $\mathbf{x} = (x, y, z)$ in a Cartesian coordinate system fixed in space. This means that

$$\mathbf{V} = \nabla\phi \equiv \mathbf{i}\frac{\partial\phi}{\partial x} + \mathbf{j}\frac{\partial\phi}{\partial y} + \mathbf{k}\frac{\partial\phi}{\partial z} \tag{2.1}$$

where $\mathbf{i}$, $\mathbf{j}$ and $\mathbf{k}$ are unit vectors along the $x$-, $y$- and $z$-axes, respectively. A velocity potential has no physical meaning itself, but is introduced because it is convenient in the mathematical analysis of irrotational fluid motion. The fluid is irrotational when the vorticity vector

$$\boldsymbol{\omega} = \nabla \times \mathbf{V} \tag{2.2}$$

is zero everywhere in the fluid. Also, since water is incompressible, i.e. $\nabla \cdot \mathbf{V} = 0$, it follows that the velocity potential has to satisfy the Laplace equation

$$\frac{\partial^2\phi}{\partial x^2} + \frac{\partial^2\phi}{\partial y^2} + \frac{\partial^2\phi}{\partial z^2} = 0 \tag{2.3}$$

The complete mathematical problem of finding a velocity potential of irrotational, incompressible fluid motion consists of the solution of the Laplace equation with relevant boundary conditions on the fluid. We will show examples of boundary conditions later.

The pressure $p$ follows from Bernoulli's equation. If we assume the