CHALLENGES IN HYDRODYNAMICS OF SHIPS AND OCEAN STRUCTURES

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ABSTRACT

Violent fluid motions, high speed marine vehicles and Computational Fluid Dynamics (CFD) are selected as main topics. Violent fluid motions deal with green water on deck, sloshing and slamming. Slamming involves many physical effects. When analyzing slamming, one must always have the structural reaction in mind. This necessitates that hydroelastic effects are considered. Many hydrodynamic phenomena matter for the three main categories of high-speed vessels, i.e., vessels supported by the hull, foils and air cushions. Dynamic instabilities, cavitation and ventilation are limiting factors for their performance. The coupling with automatic control is discussed. A brief overview of the many different CFD methods is given and advantages and disadvantages are discussed.

Keywords:

Green water on deck, sloshing, slamming, hydroelasticity, high-speed marine vehicles, CFD.

1. INTRODUCTION

There is a broad area of marine structures needed for ocean transport, exploitation of subsea hydrocarbons and wave energy, sea food production as well as for marine infrastructure. Challenging hydrodynamic aspects are focused on. However, hydrodynamics must be linked to other disciplines such as structural mechanics and automatic control.

There is a general tendency that specialists in marine hydrodynamics work on separate disciplines such as resistance, propulsion, manoeuvring and seakeeping without always combining the knowledge. For instance, the presence of sea waves can clearly influence the manoeuvring ability of a ship. This is of practical concern in, for example, replenishment operations between ships at sea. An important aspect is to properly account for the mean wave forces and moments in the mathematical manoeuvring model. Figure 1 shows results for the 161m long *Mariner* in a starboard turning manoeuvres. The rudder angle is 20.9degrees and the approach forward speed is 15.4 knots. The period, height and direction of the incident waves are 7s, 3m and 150 degrees, respectively. The figure demonstrates clearly that the presence of waves can have a large effect on the turning ability of a ship, with a noticeable wave-dependent loss in speed

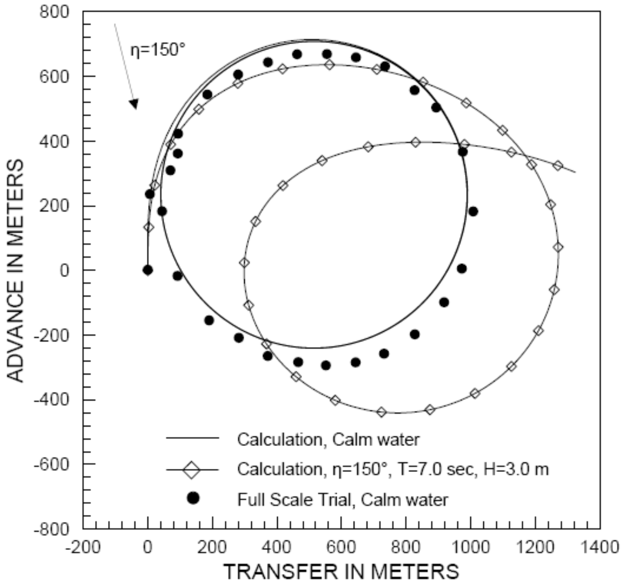

Figure 1: Turning circle manoeuvre of *Mariner* in calm water and waves, respectively (Skejic, unpublished).

A ship is often hydrodynamically optimized in calm water conditions. Because good seakeeping behaviour is an important feature of a high-speed vessel, optimization in calm water conditions may lead to unwanted behaviour in a seaway. Both wave resistance and wave radiation damping are due to the ship’s ability to generate waves. Because low wave resistance may imply low wave radiation damping in heave and pitch, the result can, for instance, be unwanted large resonant vertical motions of a catamaran with slender side-hulls and no automatic control.

Another example is the recent designs of passenger cruise vessels with very shallow local draught and nearly horizontal surfaces in the aft part of the ship. This was the result of hydrodynamic optimization studies in calm water. One does not need to be a hydrodynamicist to understand that this caused slamming (water impact) problems. Aft bodies with shallow draught should also be of concern for directional stability and for ventilation of water jet inlets in waves. Hydrodynamic optimization studies must therefore consider resistance, propulsion, manoeuvring and seakeeping. There are obviously also constraints of nonhydrodynamic character. For instance, minimizing ship motions may lead to higher global structural loads.

Several of our examples in the main text deal with ship applications and we will in particular focus on violent fluid motions and hydrodynamics of high-speed vessels. The future increased role of computer simulations relative to experiments is also addressed.

Examples on marine structures that are left out of the detailed discussions are fish farms and fishing nets. Increased knowledge about the flow through fishing nets in current and waves with due consideration of the net deformation is important, for instance, for the design of mooring systems of fish farms. Vortex induced vibrations is a challenging problem, for instance, for the design of pipelines. The towing of flexible long seismic cables is difficult to theoretically predict under dynamic conditions due to limited knowledge about flow separation at small angles of attack.

2. VIOLENT FLUID MOTION

Examples on fluid motions that will be discussed are green water on deck, sloshing and slamming. Capsizing of damaged ship with water ingress /egress in waves has similarities both with sloshing in tanks and green water on deck. Sloshing in moon pools is another example that could be mentioned. An important issue is to design efficient damping devices, such as perforated walls in the pool. A similar resonance phenomenon occurs between two adjacent ships in waves, or for a ship alongside a terminal. Conventional engineering tools based on linear panel methods cannot accurately describe the flow.

2.1 Green water on deck

There is extensive work worldwide in applying Computational Fluid Dynamics (CFD) to green water on deck. However, most work is related to two-dimensional flow. An attempt has been made by Greco et al. (2007) to classify how the different green water phenomena occur as a function of wave parameters in head sea conditions for stationary ships with blunt bows, based on experimental and theoretical studies of a restrained two-dimensional body. A typical application is to a Floating Production Storage and Offloading (FPSO) vessel. Interaction between the incident waves and the hull plays an important role. One type of green water is the so called dam-breaking phenomenon, where a vertical wall of water is generated at the edge of the deck due to the relative vertical motion between the ship and the waves. The subsequent motion of the water resembles the breaking of a dam, water flooding at high speed (15-20 m/s) along the deck. Its impact against deck structures and equipment can cause serious damage.

A second scenario is water hitting the deck as a plunging breaker. Actually, the plunging breaker may hit a deck house in the forward part of the ship. A third case is the "hammer fist" effect of water on deck, using an analogy to karate. A large mass of water rises above the deck and collapses heavily over a substantial area of the ship during a "hammer fist" type of water on deck. Figure 2 shows how these different phenomena occur as a function of the incident wave steepness and the W_w/W_b-ratio. Here W_w is the maximum vertical velocity of the incident waves and W_b is the maximum vertical fluid velocity at the ship's bow.

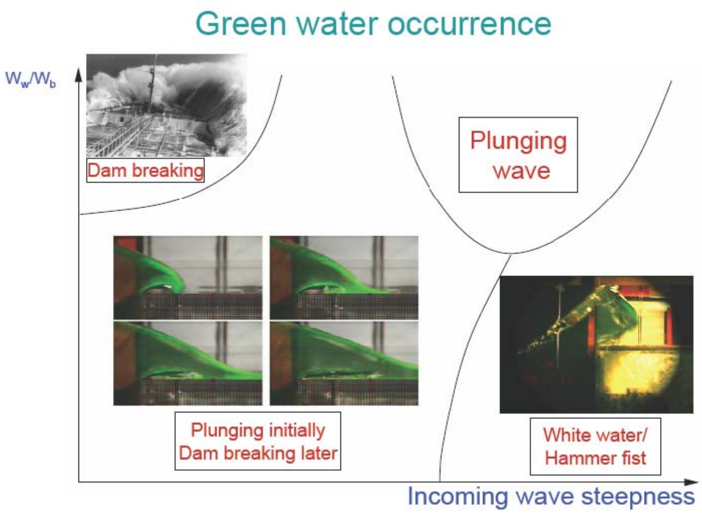

Fig. 2: The effect of green water on deck as a function of the incident wave steepness and W_w/W_b-ratio. W_w is the maximum vertical incoming wave velocity and W_b is the maximum relative fluid velocity at the bow.

Wave-body interaction plays a similar decisive role in bow stem slamming as in the green-water situation.

2.2 Sloshing

Sloshing in a partially filled container is a violent *resonant* free surface flow with strongly nonlinear behaviour. Sloshing must be considered for almost any moving vehicle or structure containing a fluid with a free surface.

A partially filled ship tank will experience violent fluid motion when the ship motions contain energy in the vicinity of the highest natural period for the liquid motion inside the tank. Impact between the liquid and the tank roof is then likely to occur for larger filling ratios. The consequence is wave breaking, spray and mixing of gas and liquid. Actually, extreme cases with gas bubbles everywhere in the liquid have been experimentally observed.

Because sloshing is a typical resonance phenomenon, it is not necessarily the most extreme ship motions or external wave loads that cause the most severe sloshing. This implies that external wave induced loads can in many practical cases be described by linear theory. However, nonlinearities must be accounted for in the tank liquid motions. Because it is the highest sloshing period (natural period) that is of prime interest, vertical tank excitation is of secondary importance. Lateral and angular tank motions cause the largest liquid response in the frequency range of interest. An increased tank size increases the highest natural period of the liquid flow. As a consequence, higher sea states and larger ship motions excite sloshing around resonance. The less internal structures obstructing the flow in the tank are present, the more severe sloshing is.

There is a variety of ship tank shapes. This includes rectangular, prismatic, tapered and spherical tanks as well as horizontal cylindrical tanks. The fluid may be oil, liquefied gas, water or heavy density cargoes like molasses and caustic soda. Ideally one should be able to predict two phase flow due to strong mixing of gas with the liquid. However, it is hard enough to predict one phase flow.

Sloshing has always been an important design criterion for oil tankers, even though partial filling is rare in actual operation. Environmental concerns have led to requirements about double hull tankers. Ship owners try to avoid internal structures in cargo tanks for cleaning reasons. The resulting wide and smooth oil tanks increase the probability of severe sloshing. Sloshing is also of concern for FPSO units and shuttle tankers. The severity of sloshing is connected to possible filling height restrictions for oil tankers, gas carriers, shuttle tankers and FPSO units. Often, operators require no restrictions on filling heights to achieve loading flexibility. Because ballast exchange is required outside the port for a bulk carrier, there are possibilities for sloshing damages. Particularly, the hatch cover is vulnerable. Partial fillings in LNG carriers are a consequence of gas boil-off during operation. Sloshing in tanks has received increased attention due to the design of new types of prismatic LNG tanks. The interior tank surface is relatively smooth.

The hydrodynamic loading inside a tank can be classified either as impact loads or ‘dynamic', non-impulsive, loads. In this context dynamic loads mean loads that have dominant time variations on the time scale of the sloshing period, while impact loads may last only from 10^{-3} to 10^{-2} seconds. It is the slamming pressures and resulting stresses in the membrane structure that are of the main concern in the design of prismatic LNG tanks. Sloshing loads are of significance for both fatigue and ultimate strength. Local structural response due to liquid impact is an important response variable. Loads on possible internal structures must be considered. Some internal structures, as a horizontal stringer on the wall or web-frame at the tank roof, may be in and out of the liquid so that impact loads as well as dynamic loads may matter. Hydroelastic effects are sometimes of importance for impact loads. Total dynamic loads on the tank are of interest in order to estimate tank support reactions as for instance for a spherical LNG tank and possible global interaction with ship dynamics.

The ship motions excite sloshing, which in return affects the ship motions. Ships equipped with anti-rolling tanks utilize this effect. The sloshing induced roll moment on the vessel will cause roll damping by properly choosing the highest natural sloshing period close to the roll natural period. FPSO units sometimes have several partially filled tanks during operation. The wave induced motions and loads on these ships will then be influenced by the dynamic motion of the liquid in the tanks. Because ship motions can strongly affect the mean and slowly-varying wave drift forces and moments, sloshing may also matter in a station-keeping analysis.

The tank shape, the level of filling and the characteristics of the tank motion, for example amplitude and frequency content, make up the principal parameters that determine the nature of the free surface flow. The relative importance of the different parameters depends on the characteristics of the flow, i.e. the response. There is a clear difference between sloshing in a shallow liquid condition and higher filling level conditions. For small ratios between liquid depth and tank length and an excitation frequency around resonance, a hydraulic jump or bore, which travels back and forth in the tank, is formed. When the steep front of the bore hits the tank wall, an impact occurs and a thin vertical jet shoots upwards. When the liquid depth is non-shallow and the liquid motion is two-dimensional, the free surface motion resembles a standing wave. Swirling or rotational flow is a special feature of three-dimensional flow, for instance, in a spherical tank, a vertical circular tank or a square-based tank.

When the interior tank surface is smooth and there are no internal structures as stringers obstructing the flow, the viscous damping of the resonant fluid motions is small as long as wave breaking does not occur. The damping for a smooth tank can be very small at finite depth, i.e. it takes a very long time for transience to die. The damping increases in general with decreasing liquid depth. Large amplification of the liquid motion occurs in a resonant condition. This leads to important nonlinear liquid behaviour inside the tank associated with nonlinear transfer of energy between different modes of liquid motion. *Secondary resonances* may matter. This means that higher harmonic oscillations caused by nonlinearities excite resonance periods lower than the primary resonance period(s). Secondary resonance increases in importance with decreasing liquid depth and increasing excitation amplitude.

It is popular to use Computational Fluid Dynamics (CFD) to model sloshing. We will address this approach in a more general way later in a separate section. However, the analytically based multimodal modal time domain method presented by for instance Faltinsen & Timokha (2001) and Faltinsen et al. (2005) is more suitable to understand the many different flow configurations that can occur during sloshing. The assumptions are potential flow, no overturning waves and infinite tank roof height. The tank wall surface must be vertical in the free surface zone. The method is extremely fast from a computational point of view. The method has been derived for two-dimensional flow in rectangular tanks and for three-dimensional flow in prismatic tanks with a rectangular base. The effect of small chamfer at the tank bottom has been studied in a two-dimensional case. The method has been extensively validated by experimental results for wave elevation, lateral force and roll moment. The 2D studies have examined how the flow changes physical character by going from finite through intermediate to shallow liquid depth. The excitation amplitude is also an important parameter. The physical dissipation increases with decreasing depth.

The steady-state solutions may have several branches. Some of the branches may be unstable. We cannot have steady-state solutions along an unstable branch. Jumps can happen between different branches of solutions.

The multimodal method has been applied to three-dimensional flow with a square-based tank. There are three types of possible dominant steady-state wave response for longitudinal harmonic excitation, namely, ‘planar’ (two-dimensional), ‘swirling’ (rotary motions) and so-called ‘squares’-like three-dimensional steady-state waves formed by a combination of diagonal wave patterns. Even with longitudinal excitation along a tank wall, 3D waves will occur due to non-linear interaction in a nearly square-based tank.

By adopting a stability analysis scheme, one can calculate effective frequency domains for the different wave behaviour and find critical depths where either the frequency domains of stable regimes or their wave response may change dramatically. Frequency domains with no steady-state solutions, i.e. “chaos” exist. This has been experimentally confirmed and gives important guidance for CFD calculations.

Both prismatic and spherical tanks are commonly used on LNG carriers. The most important load on a spherical tank is the sloshing-induced hydrodynamic force, the predominant component of which is the lateral force. Swirling occurs readily so there is in general a force component perpendicular to the forced oscillation direction.

2.3 Slamming

Slamming is of concern in many marine applications. Slamming on ships is often categorized as bottom, bow-flare, bow-stem and wetdeck slamming. Wetdeck slamming has similarities with other nearly horizontal parts of a ship such as large overhanging sterns. Green-water impact on deck structures and bow-stem slamming are of concern for FPSO units. Slamming is important in the design of a ship tank. There are similarities between slamming on ships and offshore platforms. Breaking waves can impact on a ship hull or the columns of a platform. Run-up along the columns can cause local damage of the platform deck. A platform is normally designed with an air gap to avoid global water impact. However, slamming may happen due to unanticipated large waves (Figure 3) or due to the subsidence of the sea floor for bottom-mounted platforms. Bottom slamming should be considered on shallow-draft barge-type Very Large Floating Structures (VLFS) proposed as floating airports in the coastal zone. Examples on more special types of water impacts are slamming on air bags of Surface Effect Ships (SES), drop of mines, accidental drops of pipes from platforms and analysis of free-fall life boats. The deceleration, slamming loads and hydrodynamic loads on the top cover of the life boat may matter. This issue has lately got increased attention for life-boats on platforms in the North Sea. A too low maximum operating sea state can have serious economic consequences due to stop in oil and gas production.

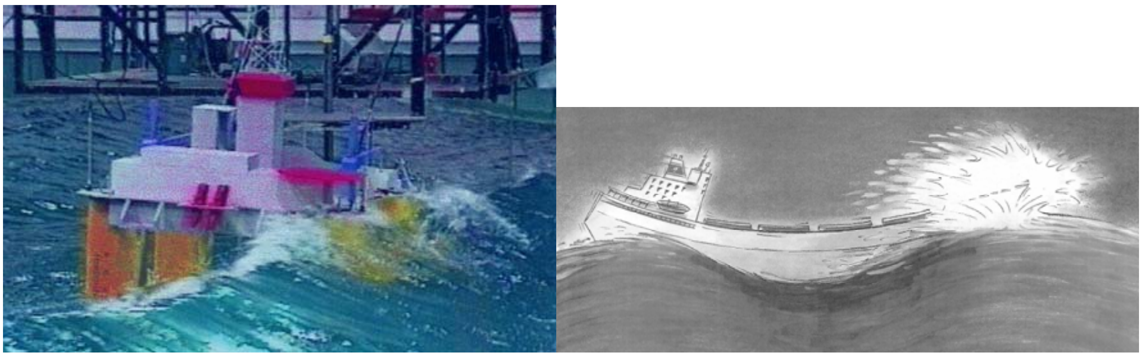

Figure 3: Wetdeck slamming on a semi-submersible platform and artist's impression of bow slamming causing global elastic vibrations (whipping) of the ship hull.

Slamming in a ship tank is associated with violent liquid motion and many possible impact situations have to be considered. For instance, large filling ratios can cause important slamming loads. Examples are high curvature free surface impact and impact with an oscillating gas cavity. Another possible scenario is a sudden flip-through of the free surface at the tank wall. A liquid wedge with a high velocity will as a consequence impact on the tank roof. A chamfered tank roof is likely to reduce the severity of slamming.

Steep waves impacting on a vertical tank wall represents an important consideration for shallow and intermediate liquid depths of a tank. An example is hydraulic jumps that can be formed at resonant conditions for shallow liquid conditions.

It is common to analyze slamming in a tank by assuming a two-dimensional flow. However, swirling may cause important impact against the tank roof corners for non-small liquid depths in a nearly square-based tank.

Both local and global slamming effects must be considered. This is illustrated in the case of bow slamming on a ship in Figure 3 where the resulting global elastic vibrations (whipping) of the ship are exaggerated. The local slamming analysis is typically done by first calculating the ship motions without impact and then consider the impact with given conditions for the impacting hull. However, this procedure does not account for the mutual interaction between slamming and the global ship behaviour and needs to be improved in the future.

Many physical effects may have to be considered such as gas cushions, liquid compressibility and hydroelasticity. When analyzing slamming, one must always have the structural reaction in mind. An important consideration is the time scale of a particular hydrodynamic effect relative to wet natural periods for structural modes contributing significantly to large structural stresses. If the time scale of a hydrodynamic effect is very small relative to important structural natural periods, the hydrodynamic effect can be neglected. When the hydrodynamic loads occur on a time scale of important structural periods, hydroelasticity must be considered. This implies that the fluid (liquid, gas) flow must be solved simultaneously with the dynamic elastic structural reaction.

Local external slamming effects on ship structures of steel and aluminum are first discussed. Drop tests with horizontal plates with correctly scaled elastic properties of steel and aluminum on calm water as well as varying wave conditions and impact conditions have shown recorded maximum pressures with very large variations for a given drop speed. A maximum pressure of about 80 bar was recorded with a drop speed of 6 m/s. However, the recorded strains were not sensitive at all and showed a time variations dominated by the lowest beam mode. The maximum strain occurred approximately one quarter of the highest natural period after the impact. The effect of the impact was to cause a force impulse to the plate. The very high slamming pressures are unimportant, i.e. they cannot be used as a static loading and there is no correlation between pressures and strains. Numerical simulations of water entry of wedge- formed ship cross-sections with stiffened plating have shown the importance of the ratio between the loading (wetting) time and the highest natural period for the structure. If these results are translated into realistic ship structural dimensions and relative vertical motions between the ship and the waves, a rough guidance is that local hydroelasticity should be considered when the angle between the impacting free surface and the hull surface is less than 5 degrees.

We then consider global slamming effects and choose wet deck slamming on a catamaran in head sea as an example. Because of the very different time scales of local and global structural vibrations, we can consider the local structure of the wetdeck as rigid in the global analysis. Heave, pitch and two-node longitudinal bending are important modes of the ship. Because the duration of the impact and the water exit (decrease of wetted surface) is smaller than one quarter of the natural period of heave and pitch, the force impulse becomes important. This enables us to simplify the analysis. Both water entry and exit must be considered. Nonlinear hydrostatic and Froude-Kriloff loads matter. Studies have shown that the Wagner method is unnecessary. Further, only the added mass force is needed during water exit.

Slamming in ship tanks will now be focused on. The discussion has also relevance for slamming in other applications. It is common in tank design to do model tests for sloshing induced slamming effects by means of forced oscillation tests. However, the scaling of the model test results represents a challenge due to the many physical effects that may matter.

Because sloshing is associated with gravity waves, we must require the Froude number is the same in model and full scale. Further, the wave induced ship motions that excite sloshing, is also Froude scaled. If harmonically forced oscillation of the tank with frequency σ is considered, Froude scaling implies that $\sigma\sqrt{L/g}$ must be the same in model and full scale. Here L is a characteristic tank dimension such as the tank breadth. A conventional model test approach does only consider the effect of Froude scaling. However, other scaling parameters of possible importance are summarized below.

If *hydroelasticity* matters during impact, we must ensure that the relevant natural frequencies for the elastic structural vibrations are Froude scaled. For instance, let us consider a steel tank. The bending stiffness matters for the natural elastic frequencies of importance. Further, the length of the elastic plate must be geometrically similar in model and full scale.

The scaling of the natural frequency may be achieved by having different bending stiffness *EI* in model and full scale. The bending stiffness may be properly scaled by considering a different material and/or changing properly the thickness of the material. Because the main interest is to find the slamming induced structural stresses, care must also be shown in scaling structural stresses. More structural modes may be needed for membrane structures than for steel structures. Some of the important structural modes for membrane structures may have relatively lower natural periods than for steel structures.

If slamming is associated with the formation of gas pockets, the Euler number must be the same in model and full scale. The *Euler number* is defined as $Eu = p_a / \rho U^2$. Here p_a is the ullage pressure, i.e., pressure in the ambient gas of the tank. The consequence of both Froude and Euler number scaling is that the ratio between p_a in model and full scale is equal to the ratio between the length scale *L* in model and full scale.

A gas pocket has a natural frequency associated with the compressibility of the gas and a generalized added mass due to the liquid oscillations caused by the gas cavity oscillations. If only Froude scaling is used, it has been shown that Froude scaling is conservative. Further, even though the gas cavity oscillations may be linear in model scale they may be strongly nonlinear in full scale with a time history that can cause larger hydroelastic effects.

Because LNG is boiling, the cavitation number is a factor. The *cavitation number* is defined as $\sigma = (p_a - p_v)/(0.5\rho U^2)$ where p_v is the vapor pressure. There is strongly limited knowledge about the effect of boiling on the slamming loads in an LNG tank.

The *Cauchy number* $C_a = \rho U^2 / E_v$ characterizes the effect of compressibility on the flow. Here E_v is the bulk modulus for elasticity. The effect of liquid compressibility is less important in the case of a liquid with no gas bubbles and may be disregarded for a steel tank. This is a matter of what are the important wet structural natural periods. However, the speed of sound in a liquid can be substantially lower in the case of a mixture of gas and liquid. The consequence is an increased characteristic time scale for the effect of liquid compressibility. Because a mixture of gas and liquid can happen during violent sloshing and because LNG is boiling, we cannot out rule the effect of compressibility on slamming loads. The mixture of LNG and gas is not homogeneous. It is only the upper layer of LNG that is boiling. Because the sloshing and the impact affect the pressure distribution in the fluid and the pressure determines if cavitation occurs, the amount of bubbles is time dependent. The void fraction is therefore a function of time and space. This is an area requiring future research. Surface tension and viscosity are believed to be less important effects during slamming.

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3. HIGH-SPEED MARINE VEHICLES

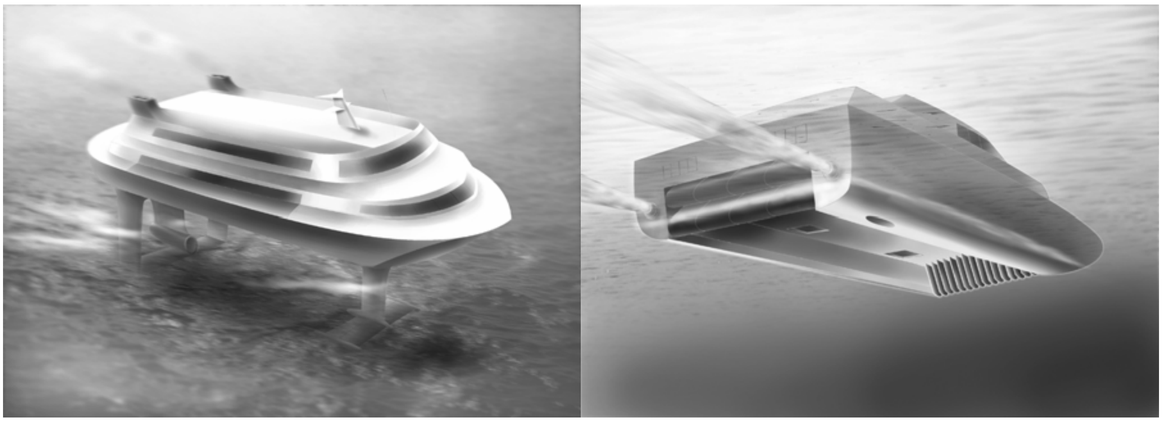


Figure 4: Examples on high-speed marine vehicles supported by foils or an air cushion.

The focus is on the three main categories of high-speed marine vehicles supported by the submerged hull, foils or air cushion. It is important for high-speed marine vehicles to consider all hydrodynamic aspects in design. Considered items have to be limited in this presentation. A more comprehensive discussion of physical effects is given in the book by Faltinsen (2005).

3.1 Surface Effect Ship (SES)

An air cushion is enclosed between the two side hulls and by flexible rubber seals in the bow and aft end of a Surface Effect Ship (SES) (Figure 4). The air cushion raises the vessel, depresses the water surface between the two hulls and lowers the zero-speed metacentric height. However, static stability does not represent a problem at zero speed. Further, the air cushion causes a wave resistance. Generally speaking, the resistance and needed power of a SES is clearly lower than a similarly sized catamaran at corresponding speeds. A danger is leakage from the air cushion in higher sea states with the consequence that the submergence of the vessel increases and the vessel speed drops. Other problem areas for an SES are wear of skirts and as well as power peaks and wear and tear of propulsion/machinery systems caused by ventilation and cavitation due to the low submergence of the water jet inlet. The latter effect is most probable to happen in a seaway or during a turning manoeuvre. Wetdeck slamming and bow diving may occur. Further, berthing of an SES at high wind speeds may be difficult.

Cobblestone oscillations cause unpleasant vertical accelerations in small sea states and are the result of resonant compressible flow effects in the air cushion. It is called cobblestone oscillations to make the resemblance to driving a car on roughly laid cobblestones. Compressibility effects of the air in the cushion are essential. The oscillations are excited because the waves change the enclosed air cushion volume. It is the vertical vessel accelerations that are of concern. When cobblestone oscillations are not excited, i.e. for higher sea states, an SES will in general have good seakeeping behaviour relative to a similarly sized catamaran.

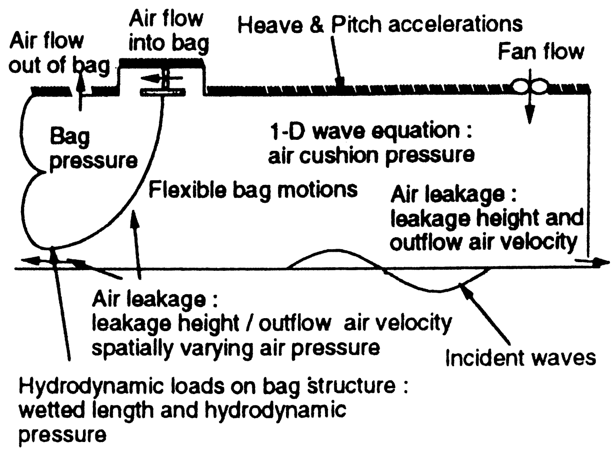


Figure 5: Physical effects influencing cobblestone oscillations of an SES *(Ulstein, 1995).*

Figure 5 gives an overview of the physical effects that matter in describing the cobblestone oscillations (Ulstein, 1995). The 1-D wave equation referred to in the figure means that spatially varying one-dimensional standing acoustic waves and spatially uniform dynamic air cushion pressures are studied. Representative values for a 30 m long SES of the spatially uniform pressure resonance frequency and the one node longitudinal acoustic resonance frequency are 2Hz and 5Hz, respectively. There are effects of the dynamic pressure in the air bag and the fact that the water waves impact on the bag and elastic vibrations of the bag. The vibrations of the bag are like a wave maker for the acoustic wave motions in the air cushion. The figure also mentions spatially varying air pressure in the vicinity of the air bag. Because this occurs on a length scale that is short relative to the important acoustic wavelength, it can be analyzed by assuming incompressible fluid. Due to continuity of fluid mass, the escaping air flow under the air bag must have a mean velocity that is dependent on the local height between the air bag and the water surface. Because high velocity implies small pressure, the escaping air flow causes a suction force on the air bag. This influences the mean escape area of the air from the air cushion. The leakage and inflow to the air cushion influences the damping level of the cobblestone oscillations.

In reality one would use an automatic control system to damp out some of the "cobblestone" effect. This is done by controlling the air flow out from the cushion in such a way that it effectively acts as a damping on the system. In order to do that properly one needs a simplified but rational mathematical method that accounts for the dynamic pressure variations in the air cushion in combination with the global heave and pitch accelerations of the vessel. Sørensen (1993) used a louver system consisting of two vent valves at the deck in the front of the air cushion. The opening and closing of the vent valves control the air flow from the air cushion so that one gets a damping effect on the system. The placement of the louver system is essential. For instance, if the louver system is placed at midships, it will have a negligible effect on the acoustic resonance mentioned above. The reason is simply that the acoustic pressure component has a node, i.e. no amplitude, at midships, while it has its maximum amplitude at the ends of the cushion.

Both the Euler and Froude numbers matter for cobblestone oscillations. A depressurized tank with a wave maker generating high quality waves with small wave lengths is needed. Simplified numerical methods accounting for automatic control are commonly used instead of model tests.

3.2 Hydrofoil vessels

. A hydrofoil vessel with a fully-submerged foil system is illustrated in Figure 4. It has in foilborne conditions in general good seakeeping characteristics, create small wash and have small speed loss due to incident waves. Foils are normally designed for subcavitating conditions. However, the possibility of cavitation is then an important issue. Our discussion assumes subcavitating foils.

The designer tries to maximize the foil’s lift-to-drag ratio and the speed for cavitation inception. Further, the weight of the strut-foil system must be minimized with due consideration of structural strength.

Relevant structural loads are slamming, hull bending moments in foilborne condition and bending of the forward foil and strut during recovery from a forward foil broach.

Slamming on the side hulls of a hydrofoil catamaran is not considered as a problem. The reason is large deadrise angles. Because a monohull hydrofoil vessel uses typically a planing hull with relatively small deadrise angles, slamming loads matter. If a hydrofoil catamaran is hullborne in bad weather, wetdeck slamming must be considered. The possibility of grounding and hitting of objects like logs against the strut-foil system must also be evaluated.

Flutter of foils and struts could cause catastrophic failure, but this has never occurred yet. The classical flutter scenario is dynamic instability of combined bending-torsion vibrations of a strut-foil system. The mass ratio, i.e. the ratio of a typical density of structural material to the density of the fluid is an important parameter. This difference creates a clear advantage for a hydrofoil vessel relative to an aircraft when it comes to flutter.

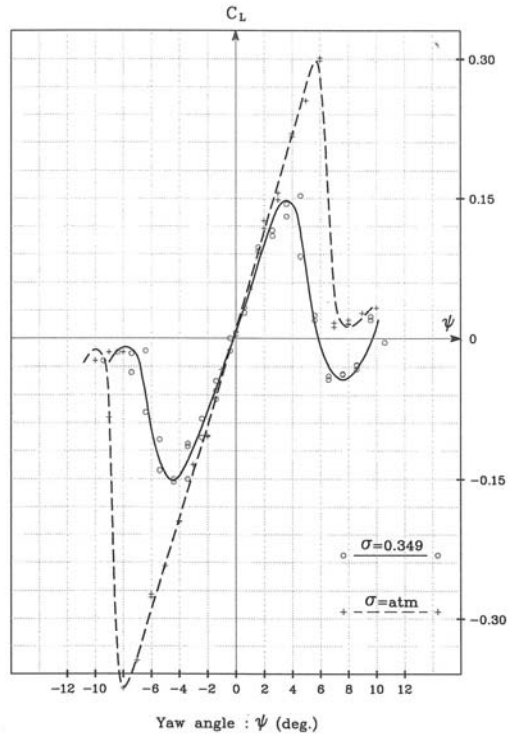
A surface-piercing hydrofoil system in foilborne condition stabilizes the vessel in heave, roll and pitch. This can be understood by means of a quasi-steady analysis. Consider for instance that the heave motion increases. Here heave is positive upwards. This causes the reduction of the wetted foil area. Because the lift is proportional to the wetted area, the lift due to the foils decreases. The weight of the vessel balances the lift in the equilibrium position. The increased heave implies that the vessel weight will force the vessel downwards. Another way of saying this is that there is a restoring force in heave bringing the vessel back to the equilibrium position. Similar static analysis can be made for heel and trim

A hydrofoil vessel with a fully-submerged foil system needs automatic control to stabilize the vessel in heave, pitch and roll. Platforming and contouring modes are used in connection with an active control system. The contouring mode is used in longer waves to minimize relative vertical motion between the vessel and the waves and to avoid ventilation and broaching of the foils. The platform mode is used to minimize vertical accelerations of the vessel in relatively short waves.

The Reynolds number scaling of foil tests and how to ensure turbulent boundary layers during model tests is an important issue. One effective way is to use Hama strips. The basis of a Hama strip is a tape. A saw-tooth shape is made on the upstream edge by means of a scissor.

Cavitation on foils designed for subcavitating conditions limits the vessel speed to the order of 50 knots. Cavitation appears when the pressure in the water is equal to the vapour pressure, i.e. close to zero. The pressure distribution along the foil should be relatively flat in order to minimize the possibility of cavitation, i.e. there must not be pronounced local pressure minima (suction peaks). The consequence of cavitation is that the material of the foil can be quickly destroyed and the lift capabilities of the foils can be significantly reduced. Another consequence is that the drag on the foils increases. Because cavitation is accompanied by noise generation, one can hear on board the vessel when there is the possibility of damage due to cavitation. If the vessel speed should be increased substantially beyond 50 knots, supercavitating foils must be used to avoid cavitation damage. They are characterized by much lower lift-to-drag ratios and lift coefficients than subcavitating foils.

The importance of cavitation and ventilation for hydrofoil vessels will be illustrated by model test results for the side force coefficient of a strut used as a rudder on a hydrofoil catamaran. Ventilation means that air enters from the atmosphere to low pressure areas on the strut. It occurs typically at angles of attack ψ higher than 4° to 6° for a strut on a hydrofoil vessel in foilborne condition. However, this depends on the Froude number. Ventilation will cause a significant drop in the force. Hysteresis is associated with ventilation. This means that ψ may have to be lowered significantly to restore non-ventilated conditions. The detailed physical understanding of ventilation is limited.



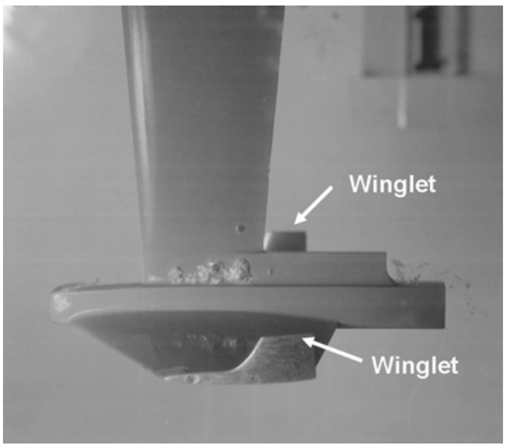


Figure 6: Model test results of the side force coefficient C_L of a front strut-foil system used as rudder on a hydrofoil catamaran (Minsaas, unpublished).

Figure 6 shows model test results for a front strut-foil system used as a rudder on a hydrofoil catamaran. The model tests were done in the depressurized circulating free surface tank at the Technical University of Berlin. The side force coefficient C_L is presented as a function of the yaw angle ψ both at atmospheric air condition and at the cavitation number

$\sigma = 0.349$ corresponding to the full-scale condition. The pitch angle is 2.7° and the submergence Froude number Fn_h is 5.96 corresponding to a full-scale speed of 50 knots. The drop in the absolute value of C_L with either increasing positive ψ-values or decreasing negative ψ-values is an indication of the start of ventilation. This occurs, for instance, at $\psi \sim 4°$ when $\sigma = 0.349$ and at $\psi = 6°$ at atmospheric air condition.

A consequence of the ventilation at the strut is that air penetrates to the foil and causes a significant drop in the lift force on the foil. This drop occurred at ψ approximately equal to $\pm 6°$. The lift force on the ventilated foil was the order of 50% of the non-ventilated lift.

3.3 Semi-displacement and planing vessels.

Semi-displacement vessels without damping systems may have unsatisfactory seakeeping behaviour. One reason is that the excitation loads in heave and pitch at resonance condition in head sea increases with forward speed. Increased heave and pitch damping of semi-displacement vessels is commonly achieved by using T-foils in the bow (Figure 7). Because ventilation and cavitation may limit the operability in higher sea states, wrong conclusions about seaworthiness can result from conventional towing tank tests. While the wave-induced response of displacement vessels can normally be adequately described by linear theory, this is not true for planing vessels. A linear 2.5D (2D+t) theory based on potential flow theory has become popular in describing wave resistance and wave induced motions and loads on semi-displacement vessels.

Figure 7: T-foil placed below the keel and used to damp vertical wave-induced ship motions at high speed (SEASTATE).

The steady wave pattern (wash) associated with fast semi-displacement vessels are of concern in coastal and inland waters. The wash may cause nearby small boats to capsize or ground or large moored ships to move and mooring lines to break. The waves may cause erosion or even collapse of a bank. When the waves approach a beach, the amplitudes increase and the waves break. This may happen when the ship is out of sight, surprise swimmers, and represent a risk factor.

An example of the measured wave elevation in the near aft of a trimaran in deep water is shown in Figure 8. Free surface nonlinearities due to flow separation at the transom stern and breaking waves matter. Numerical methods have clear limitations in describing the wave field.

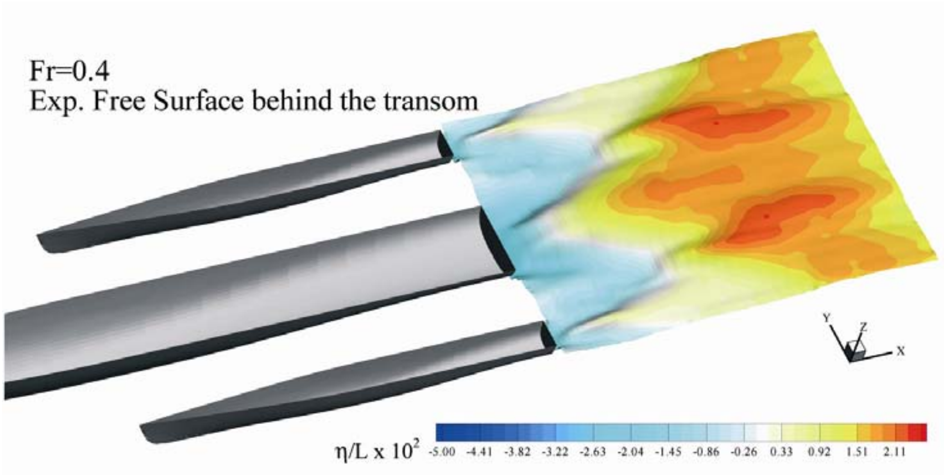


Figure 8: Model tests on high-speed trimaran in calm water. Wave elevation measurements at Froude number 0.4, and colour code for the ratio η / L between the wave elevation and the ship length (INSEAN).

The water depth can have an important influence on the wave pattern when the water depth-to-ship length ratio *h/L* is small. An important parameter is then the depth Froude number $Fn_h = U / \sqrt{hg}$ where U is the ship speed. $Fn_h = 1$ corresponds to the critical speed. Large changes occur at the critical depth Froude number. The ship waves are very different for subcritical, critical and supercritical speeds in shallow water. The wave pattern at subcritical speed consists of transverse and divergent waves while only divergent waves exist for supercritical speed. The supercritical waves have a very small decay with distance which makes them of particular concern from a wash point of view. The angle (Kelvin angle) between the boundary of the wave system and the ship course is $19°28'$ in deep water and when $Fn_h < 0.5 - 0.6$. A rapid increase in the Kelvin angle occurs for $Fn_h > 0.9$, and the angle is $90°$ for $Fn_h = 1$. False wall effects can therefore occur during shallow water model tests around the critical Froude number. If the vessel is in the vicinity of critical speed and in a channel, large waves can build up, be unsteady relative to the ship and travel faster than the ship. Russel described in 1865 that one day, the happiest of his life, something unexpected happened during his towing tests. A large solitary wave occurred ahead of the ship. He could not follow it by foot and got on the horseback and followed it for more than a mile.

Dynamic instability has increased importance with increasing Froude number. One reason is the increased importance of hydrodynamic pressure relative to hydrostatic pressure with increasing forward speed. There is a broad variety of dynamic stability problems. The loss of steady restoring moment in heel with forward speed can cause a sudden list of a round bilge monohull to one side. This can at high speed be followed by a violent yaw to one side. The consequence can be capsizing. This “calm water broaching” is the main reason round-bilge hulls should not operate beyond a Froude number of 1.2 (Lavis, 1980). The loss of steady restoring heel moment with speed should be accounted for in the design by having sufficiently high metacentric height at zero speed.

When a high-speed catamaran in following waves has a speed close to the phase speed of the incident waves, the catamaran can come in a position relative to the waves so that the fore part of the vessel dives into a wave crest. If there is not sufficient buoyancy in the fore part of the vessel, a critical situation may occur.

Examples of dynamic instabilities for planing vessels are “corkscrew” pitch-yaw-roll oscillations, chine walking and porpoising. Porpoising is dynamic instabilities in the vertical plane and is commonly seen. Cavitation and ventilation in the aft body can cause dynamic instabilities.

The Froude number dependence and roll, sinkage and trim matter for manoeuvring of semi-displacement vessels. Strong nonlinearities may occur for a trimaran. For instance, steady heel may cause sign change in steady yaw moment for relatively small heel angles. Numerical methods have difficulties in describing those effects.

Global wave loads matter in the design of vessels longer than 50m. Important global loads for catamarans are transverse vertical bending moment (often called split moment), vertical shear force and pitch connecting moment, as illustrated in Figure 9. Torsional moments, vertical shear forces and vertical bending moments at transverse cross-sections are also of concern, as for monohulls. Both continuous wave loading and slamming will cause global loads. When wetdeck slamming-induced global loads are analyzed, the vessel must be considered elastic. The response is called whipping. When we consider the effect of continuous wave action, the catamaran is often considered as rigid in the determination of the global loads. However, it should be ensured in a linear analysis that the natural frequencies of the global elastic modes are higher than the encounter frequencies of practical interest. When the global elastic modes are accounted for during continuous wave loading, the phenomenon is called springing. Nonlinear effects may also cause springing.

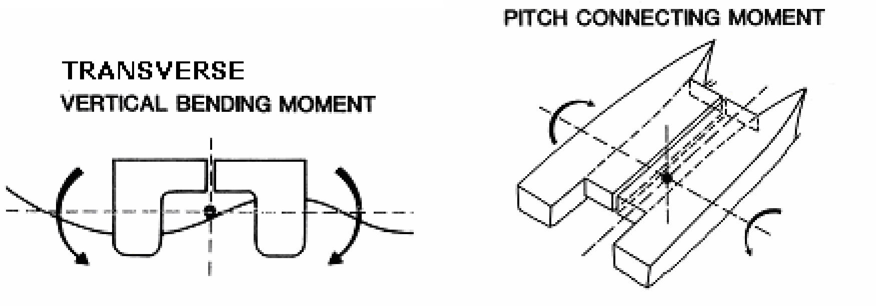


Figure 9: Examples on global wave loads on a catamaran.

The largest global wave loads in a longitudinal cut of a catamaran occur in oblique sea conditions. The time window for testing in existing model tank facilities in oblique sea can be limited due to the high speed and the fact that transients must die out. Experimental error sources associated with non-constant wave conditions in a basin should also be considered.

4. COMPUTATIONAL FLUID DYNAMICS

It has become popular to use Computational Fluid Dynamics (CFD) to solve fluid flow. There is a broad variety of numerical methods. An overview is given in Figure 10. By CFD we mean that the fundamental governing equations, with initial conditions and nonlinear boundary conditions based on either potential flow, inviscid or viscid fluids are solved. For instance, we do not mean linear strip theories for ship motions.

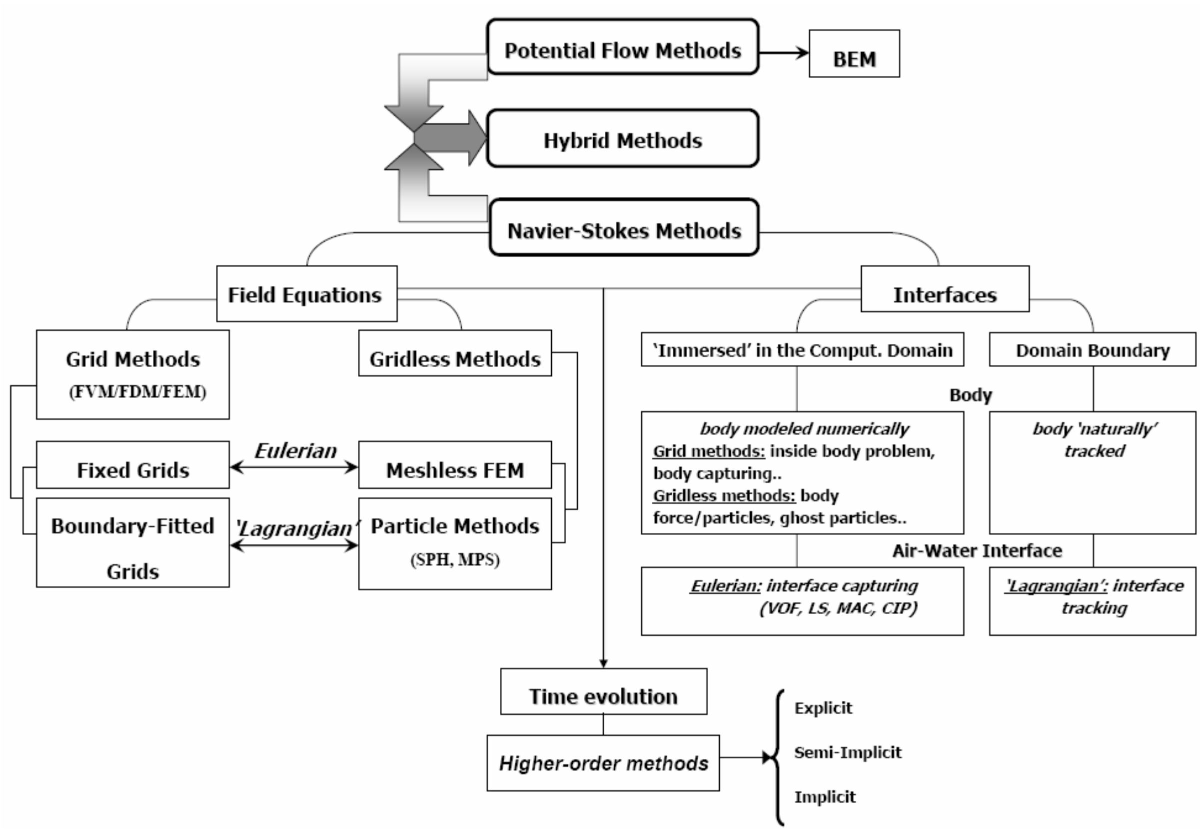


Figure 10: Overview of numerical methods in fluid dynamics with emphasis on possible solution strategies within Navier-Stokes solvers (Greco, unpublished).

We may ask: What are then the advantages and disadvantages of using CFD? Advantages are that complex structures and general excitation may in principle be considered. A CFD method may provide good flow visualization including details such as the vorticity distribution. Flow separation around structures can be simulated. It seems generally accepted that CFD codes have difficulties in predicting impact loads when the angle between the impacting free surface and the body surface is small. Actually hydroelasticity may then play an important role. Most computations do not consider this fact and assume a rigid structure.

Another disadvantage is that the CFD methods are time consuming which makes statistical estimates of response variables in a stochastic sea difficult. Some methods may not be robust enough. For instance, a Boundary Element Method breaks down when an overturning wave hits the underlying free surface. Numerical problems may also arise with a BEM at the intersection between the free surface and the body boundary. When a BEM works, it is in general a fast and accurate method relative to other CFD methods. Nonlinear effects associated with water entry of two-dimensional bodies can in many cases be adequately described by potential flow theory and the Boundary Element Method.

CFD methods based on the Navier-Stokes equations are often very robust. Movies based on CFD may look convincing without always representing the reality. Care must be shown that the solutions are true physical solutions. If sufficient care is not shown, some of the methods may numerically lose or generate liquid mass on a long time scale. This is of particular concern in sloshing simulations. Because the highest natural period of the liquid motion is strongly dependent on the liquid mass, this can result in an unphysical numerical simulation. Verification and validation are a must. By verification is meant that the solutions are consistent with the governing equations, initial and boundary conditions that have been used. By validation is meant comparisons with model tests and full scale trials. Experiments are, of course, not free of errors. So it is important that the experiments are accompanied by an error analysis. Verification can be done by temporal and spatial convergence tests and by controlling that conservation of global mass, momentum and energy are satisfied. Comparisons should be made with analytical methods such as the linear potential flow theory.

Because commercial CFD codes are generic of nature and special physical features are specific for the different application fields, one cannot necessarily trust documentation of verification and validation in other applications than those one is interested in.

If a step should be made in improving seakeeping calculations, we have to rely on Computational Fluid Dynamics (CFD) and solve the Navier-Stokes equations. A graph was made by the author in 1988. Major previous achievements in computations of wave induced motions and loads were listed as a function of available computer power and year. A prognosis of the computer power in year 2000 was made and it was asked if it would be common to solve the 3D Navier Stokes equations in waves for ships and ocean structures. Even though the available computer power was underestimated, we are still not at the stage where solving the Navier Stokes equations in waves is standard practice.

A domain decomposition method is suggested for strongly nonlinear flow at the ship. The violent flow near the ship including possible green water on the deck and water impact is handled by a CFD solver based on Navier-Stokes equation with two-phase flow while the more moderate free surface motion at the rest of the sea is solved by a Boundary Element Method. The following example assumes the near-field solver to be a Finite Difference Method combined with a Level Set-method to describe the interface between water and air. Computational time is roughly estimated by Greco (unpublished) to be 16 hours for one wave period with a PC Pentium IV 2.8 GHz. If 1080 oscillation periods is assumed in a 3 hour storm, this means a computational time of approximately two years. Further, many realizations of each sea state as well as many sea conditions are needed in the design analysis. Computational speed may not be an obstacle with the combined use of supercomputers, future improvements in numerical methods and alternative design procedures by selecting time windows with extreme conditions. However, we have still not solved all physical problems needed for design such as ventilation, air cushions and proper turbulence modelling.

5. CONCLUSIONS

There is an increased focus on Computational Fluid Dynamics (CFD). A broad variety of numerical methods exist and it is difficult to favour one particular method to solve all marine hydrodynamic problems. A straightforward application of CFD in solving the behaviour of ship and ocean structures in severe weather is not conceivable in the near future if proper response statistics should be derived. CFD for high-speed vessels is, in general, not ready for design applications.

Simplified numerical methods based on rational physical approximations should be further developed. An example is on describing cobblestone oscillations of an SES.

A balance between analytical methods, CFD, model tests and sea tests is recommended. Further, links between the different disciplines of hydrodynamics as well as other topics such as structural mechanics and automatic control have been stressed.

Research on violent fluid motions such as slamming, green water on deck, sloshing in tanks, damaged ship with ingress/egress of water are important future activities. Knowledge about the effect of boiling on slamming loads in a LNG tank is needed. A physical understanding of what flow parameters matter for response variables of importance in design and operation is essential. The time scale of a particular physical effect in order for the structure to react, for instance in terms of maximum stresses, is a basis for deciding what physical effects matter.

Dynamic instability, ventilation and cavitation are limiting factors for operation of high-speed vessels. There is limited knowledge about ventilation. A depressurized towing tank with wave maker and 6DOF forced oscillator is desirable. Large horizontal tank dimensions may be needed for seakeeping and manoeuvring tests of high-speed marine vessels.

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