Modeling of Planing Craft in Waves

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Preface

The work presented in this thesis was completed at the division of Naval Systems, Department of Aeronautical and Vehicle Engineering at the Royal Institute of Technology (KTH) under supervision of Dr. Jakob Kuttenkeuler and financed by the Swedish Defense Material Administration (FMV). The studies began at the division of Naval Architecture, lead by Professor Olle Rutgersson, where Dr. Jianbo Hua introduced me to the subject. The division of Naval Architecture was closed in December 2001, and supported by FMV, Professor Jan Bäcklund, Dr. Jakob Kuttenkeuler and Professor Dan Zenkert the work continued at the division of Naval Systems.

The experiments included in the thesis were performed at the former KTH towing tank and funded by the *Gösta Lundeqvists fond för skeppsteknisk forskning*, and at the Dynamics Laboratory of the *Canal de Experiencias Hidrodinámicas de El Pardo* (CEHIPAR), Madrid funded by the EU-program Training and Mobility of Researchers –Access to Large Scale Facilities (TMR-ALSF). The tests at CEHIPAR were performed together with Anders Rosén who has supported me all through this struggle.

I also want to address my appreciation to Jakob Kuttenkeuler for his honest and straightforward supervision and encouraging leadership and to Ivan Stenius for his valuable comments on my work.

Stockholm, September 2004

Karl Garme

Abstract

Simulation of the planing hull in waves has been addressed during the last 25 years and basically been approached by strip methods. This work follows that tradition and describes a time-domain strip model for simulation of the planing hull in waves. The actual fluid mechanical problem is simplified through the strip approach. The load distribution acting on the hull is approximated by determining the section load at a number of hull sections, strips. The section-wise 2-dimensional calculations are expressed in terms of added mass coefficients and used in the formulations of both inertia and excitation forces in the equations of motions. The modeling approach starts from the hypothetic assumption that the transient conditions can be modeled based on those section-wise calculations. The equation of motion is solved in the time-domain. The equation is up-dated at each time step and every iteration step with respect to the momentary distribution of section draught and relative incident velocity between the hull and water and catches the characteristic non-linear behavior of the planing craft in waves.

The model follows the principles of the pioneering work of E. E. Zarnick differing on model structure and in details such as the modeling of the lift in the transom area. A major part of the work is concerned with experiments and evaluation of simulations with respect to performed model tests and to published experiment data. Simulations of model tests have been performed and comparisons have been made between measured and simulated time series. The link between simulation and experiment is a wave model which is based on a wave height measurement signal. It is developed and evaluated in the thesis.

The conclusions are in favor of the 2-dimensional approach to modeling the conditions for the planing hull in waves and among further studies is evaluation of simulated loads and motions to full-scale trial measurement data.

Keywords: Planing craft, time-domain simulation, design loads, transient motion, non-linear motion, impact acceleration, waves, 2-dimensional strip method, pressure measurement.

Dissertation

This thesis consists of a brief introduction to the area of research and the following appended papers:

Paper A

Garme K. & Hua J., *A Method to Analyse Seakeeping Model Measurements in Time Domain*, 9th Int. Offshore and Polar Engineering Conference ISOPE'99, 1999.

Paper B

Garme K. & Rosén A., *Time-Domain Simulations and Full-Scale Trials on Planing Craft in Waves*, International Shipbuilding. Progress., Vol. 50, no 3, 2003.

Paper C

Rosén A. & Garme K., *Model Experiment Addressing the Impact Pressure Distribution on Planing Craft in Waves*, International Journal of Small Craft Technology, Vol. 146, 2004.

Paper D

Garme K., *Improved Time Domain Simulation of Planing Hulls in Waves by Correction of the Near-Transom Lift*, Submitted for publication in June 2004.

Division of work between authors

Paper A

Garme developed the method and wrote the *Wave equation determination*, *Comparison of calculated and measured waves* and *Summary and conclusions* sections. Garme computed the wave model input to the ship motion simulations performed by Hua who wrote the sections *Introduction*, *Comparison of calculated and measured ship motions* and *Applications*.

Paper B

Garme wrote the sections *The simulation model* and *Validation of simulations* except the subsection *Full-scale trials* which together with the section *Discussion on full-scale trial results* were written by Rosén. The *Introduction* and *Summary and conclusions* sections were jointly written by Garme and Rosén.

Paper C

The method to reconstruct and integrate pressure transducer signals is developed by Rosén who also reviewed and used it in this paper. Otherwise the paper is written jointly by Garme and Rosén and the tests and analysis are made in co-operation between the authors.

Contents

Introduction

The superior position of ships for transportation of heavy and voluminous cargo is partly due to the low energy demand compared to other means of transport. This of course, presupposes that the forward speed is low, typically 10-25 knots, thus moderate 20-50 km/h, basically depending on ship size. The hull resistance in water depends strongly on the forward speed and for common cargo ships the resistance increases so dramatically above a certain speed that it is practically impossible and economically insane to increase the speed any further. At that threshold, the hump speed, the aft settles, the trim increases and the ship sails uphill. The wave resistance, i.e. the energy transferred to the wave systems around the ship, causes this speed limit. Nevertheless, water borne craft can exceed the hump speed and for some applications, typically, defense, police, ambulance and Search and rescue (SAR) units it is worth the effort. In addition there is economical potential in fast sea passenger transport that can compete with or complement the air and road transports. The leisure boat industry should also be mentioned to whose costumers the high speed is a value in itself. To pass the threshold of the displacement ship and reaching speeds at 25-50 knots or for exceptional designs as fast as say +75 knots, the wave resistance has to be overcome by hull shapes for which dynamic lift develops as the speed increases, lifting the hull and minimizing the submerged volume. The most common concept is the planing hull but also the hydrofoil is a tried-and-true solution. The latter is a more complicated and less robust design and has generally worse low speed characteristics than the planing hull which limits its use. At high speed in waves the hydrodynamic conditions vary tremendously and the v-shaped planing hull is a good compromise to satisfactory resistance and seakeeping characteristics during those circumstances and furthermore, it is simple and robust technique. Taken all in all, this has made the planing hull concept the incomparably most common high-speed craft.

High-performance planing vessels, e.g. patrol boats, SAR vessels, naval craft and race boats, are structurally designed to withstand the impact loads occurring as the ship hits the water surface. The loads depend on forward speed, sea conditions and the ship motional response due to the first two. The high loads cause large acceleration of the hull, crew and equipment. Figure 1 shows the measured acceleration at the centre of gravity for a 1:10 scale model of a 10 meter, 6.5 tones planing craft. During the sequence the ship leaves the water, falls free for a moment before hitting the water surface violently. The acceleration rise time and duration is in the order of milliseconds, important characteristics both for the instrumentation of the experiment and for the actual impact on the ship and its crew. The hydrodynamic pressure causing this acceleration response of almost 10 g constitutes a peaked propagating distribution difficult to predict.

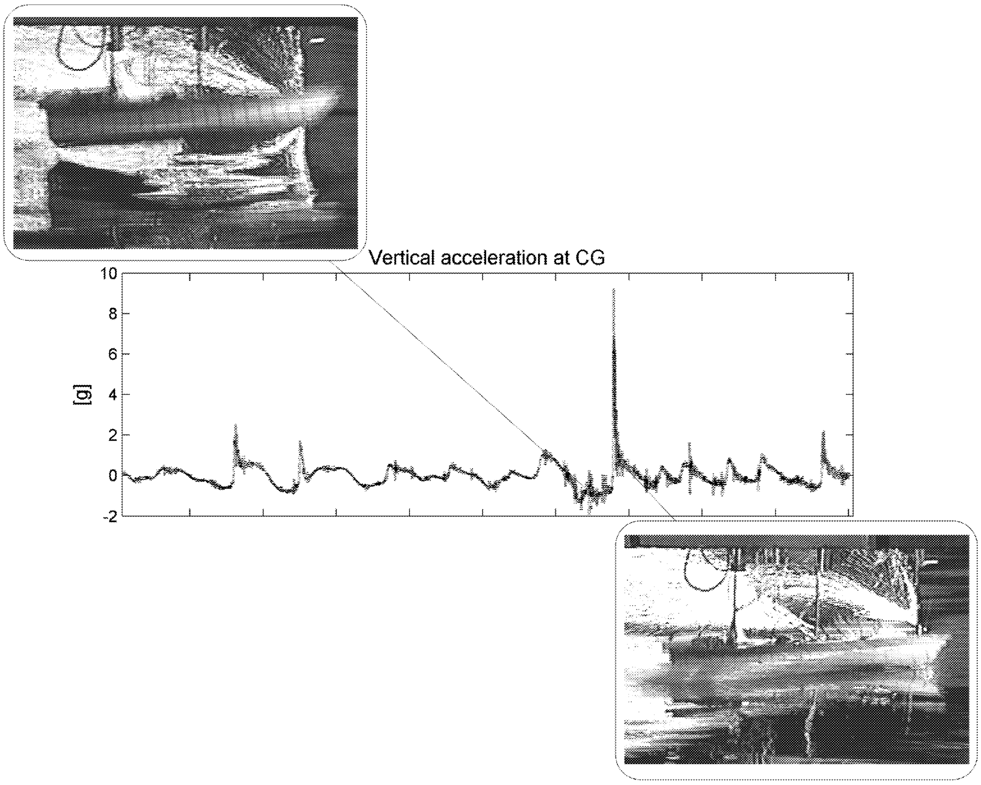

Figure 1. Model test measurement signal from an accelerometer located at the model centre of gravity (CG).

A ship is structurally designed, arranged and equipped for certain operational conditions, that is, to be safely run at specified speeds and sea conditions with the crew having acceptable working conditions. The ship should also be built as light as possible to minimize the required power and maximize the load capacity and range. With modern structure analysis tools, i.e. finite element methods (FEM), the hull structural design can be investigated in order to identify weaknesses, and to attend to them, early in the process of construction. Despite this, the outcome of such analysis and the opportunities for optimization are limited due to the difficulties to accurately estimate design loads. Numerical modeling of the planing craft in the seaway could constitute a solid basis for choosing design loads and could characterize the motional behavior of the vessel during those conditions. Moreover, the seakeeping and dynamic stability, i.e. the risk of porpoising could be surveyed and propulsion and ride control systems could be investigated.

The question is to what extent, and effort, the planing craft in waves can be modeled. Assuming that the hull geometry is known, the problem is fluid mechanical with a free water surface partially penetrated by the moving hull. The wave motion and the vessel speed influence the ship motions, and the boundaries between the hull and water as well as between the wave surface and the air, continuously changes. Those moving boundaries are necessary to formulate the fluid mechanical problem but dependent on its solution which makes the problem more complex. Limiting the investigation to hydrodynamic pressure and ship motions the hydromechanics can be simplified to inviscid, incompressible and irrotational potential flow. That step from the Navier-Stokes and the continuity equations to the Bernoulli and Laplace equations is a large simplification. During the last decade efforts have been made to solve the flow around a planing hull in calm water as a 3-dimensional potential flow problem, for instance by Lai & Troesch (1995) presenting a vortex-lattice method later used by Wagner & Andersson (2003) to model a similar situation but increasing the hull complexity with sprayrails. There are also examples when viscosity is accounted for. Capponetto et al. (2003) review results from calculations made with commercial CFD code solving the Reynolds Average Navier Stokes equations (RANS). The program was applied to the planing craft in calm water and in regular waves. The authors concluded that the steady state solution for the planing hull heading regular waves, was achieved after about 33 hours and that approaching planing in irregular waves was out of scope. In comparison, the steady state motion of the craft in regular waves was reached after about 8 minutes on a comparable computer with the simplified code assuming inviscid and 2-dimensional flow used in Paper D, a computational time that definitely could be shortened several times by more efficient coding. This indicates the advantages if the problem can be addressed with a 2-dimensional approach.

2-dimensional approach

In difference to calm water and regular waves, the irregular sea condition will never lend the planing hull steady state conditions. Thus, to formulate and solve the 3-dimensional fluid mechanical problem has not been judged as a realistic approach. Instead the problem could be simplified to a sum of 2-dimensional ones, in line with the successful strip method for motion analysis of displacement ships. The 2-dimensional approach to model the planing hull dates back to the 1920s and 1930s and pioneering studies on loads on seaplane floats. The most well known are the papers by von Karman (1929) and Wagner (1932). The slender hull and the high speed at landing resulted in a basically transverse impact flow that justified the assumption of 2-dimensional potential flow. Moreover, the geometrical similarity was observed between a falling hull cross section and the projection of the hull running straight through an imaginary vertical plane. This is often referred to as the *planing immersing section analogy*, see Figure 2.

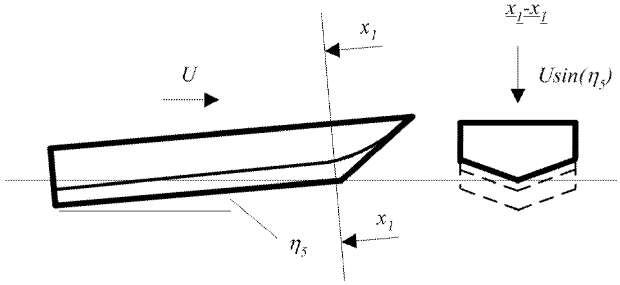

Figure 2. Sketch to illustrate the *planing immersing section analogy*. U denotes forward speed and η_5 the pitch angle.

The analogy indicates that the stationary 3-dimensional problem can be transformed to a 2-dimensional time dependent problem for which each instant is linked to a position along the hull. The sectional load calculation that is the foundation of the 2-dimensional approach has almost exclusively been addressed to the impact force developed as the section penetrates an initially calm water surface with constant velocity until the water has reached the section chine. Considerably less effort have been spent on the chines wet and exit problems, i.e. the reverse situation when the section is lifted out of the water. Among the few works on the latter is Tveitnes (2001) who has studied the chines dry, wet and exit phases experimentally and analytically. The large impact force occurs during the chines dry phase which of course is the reason for the focus on this situation but if the 2-dimensional section shall be used to model a ship in waves the chines wet, and to some extent also the exit case must be considered. A crucial matter to the load is when the chine is wetted. As the section moves down through the water with large velocity, the water is forced upwards at the section side and outwards from the section. The local surface deformation is often referred to as pile-up and will determine when the chine is wetted. When the ship is modeled the pile-up level influences the running attitude.

Models for planing craft in waves

Modeling of the planing craft in waves by a 2-dimensional strip approach began with the study on the coupled heave and pitch instability in calm water, called porpoising, Martin (1978a). The same linear equations of motions were used to model the heave and pitch response to regular waves by Martin (1978b). The linear model showed promising results on motions despite the fact that the rigid body motions or accelerations of a planing hull in waves are generally not linear with the wave amplitude, a conclusion of the important experiment series of Fridsma (1969 & 1971). Although, heave and pitch motion could turn out relatively harmonic also in severe wave conditions which is shown for instance from the full-scale data of Garme & Rosén (2003) –paper B. Figure 3 shows measurement signals on heave motion and vertical acceleration at the centre of gravity where it is clearly seen that the impact acceleration peaks of short duration hardly influence the heave motion.

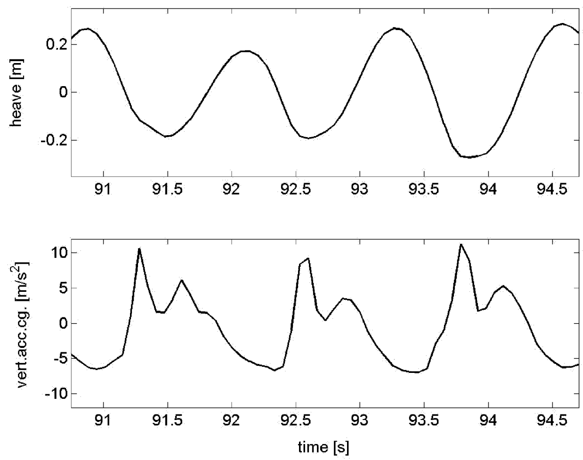

Figure 3. Examples of measurement signals of heave and vertical acceleration at the centre of gravity from full-scale trials on the Swedish Navy combat craft 90E.

Martin (1978b) concludes that the linear frequency domain model could not reproduce accelerations accurate enough and that this problem could be solved by non-linear equations of motion. Martin suggests time-domain analysis which is presented the same year by

Zarnick (1978). The Zarnick model becomes the basis of most following simulation models. It takes about a decade before implementation and developments of the Zarnick model are published, maybe because the computer power becomes available and to a fair prize first at about 1990. The simulation codes that develop are FASTSHIP, Keuning (1994), BOAT3D, Payne (1995) and POWERSEA Akers (1999) of which the latter is the only commercially available program. Despite modifications and developments the basis of all three is the same as for the Zarnick (1978) model which also goes for the simulation model in Garme & Rosén (2003) –paper B and in Paper D. Keuning (1994) adds a semi-empirical tool for estimating the calm water running attitude in order to determine correction terms to the dynamic simulation procedure. POWERSEA has included the Zarnick model in a Windows interface and added a hull geometry editor and semi-empirical tools as the Savitsky model for power prediction and comparative calm water analysis and also a semi-empirical module for including the effect of appendices into the analysis, Ship Motion Associates (2003).

The simulation model presented and used in Garme & Rosén (2003) –paper B and in Paper D, takes into account, similar to POWERSEA, the effect of sections with a surface normal component directed out of the hull cross section plane. Also the model structure is different from the classical Zarnick formulation with a pre-calculation scheme for hydrostatic and hydrodynamic coefficients and in that the complete load distribution is determined before integration to the rigid body equations of motion. The corrections of the near transom lift presented in Paper D is also a modification with respect to the Zarnick model. The aim has been to determine the load distribution on the planing hull in waves together with the rigid body motions and accelerations, and a great effort has been addressed to evaluation of the simulation results. This has been made in steps leading finally to simulation of model tests and comparison between calculated and measured time series. This is performed in Paper D, where the link between experiments and simulations is the wave model determined from the wave height meter signal by means of the method of Garme & Hua (1999) –paper A. In the following sections the 2-dimensional modeling are discussed focusing on evaluation and applications.

Strip approach to planing

The slender 3-dimensional hull in steady state planing on a calm water surface can, following the *planing-immersing section analogy*, be recognized as a 2-dimensional hull cross section moving down through the water surface, see Figure 2. Every instant during the section's way through the surface, represents a position along the hull and the 2-dimensional time dependent situation could be interpreted as a sweep downstream along the hull. If the hull is slender and prismatic, and the speed is high, the geometrical analogy will be reasonable but even if it would be geometrically exact the flow especially at transom could not be described satisfactory by a 2-dimensional analysis. If the hull is not prismatic, which of course always is the case in the bow area, or if the ship runs in waves implying that the section does not move with constant speed down through the water surface, and maybe not at all downwards, the situation departs further from the ideal assumptions of 2-dimensionality and slenderness. The limits for slenderness need to be defined which is often done in reference to Ogilvie (1967) through a ratio typically between the wetted hull length and the draught at transom. The slenderness parameter is coupled to forward speed through the Froude number defining the validity range of the slenderness assumption. In waves the shape of the submerged volume will vary considerably and the underwater body is definitely neither slender at all times nor will the relative velocity between the water and section be large at all instances.

Ogilvie (1969) states that slender body theory implies a lack of communication upstream, that the load closest to the transom can not be modeled but that such local weaknesses do not affect the possibilities to determine the loads further upstream. In this thesis the modeling hypothesis is that the loads acting on the hull can be modeled from the local conditions although it is obvious that the precision will vary and that local or momentary compensations must be used in areas were the model is known to be weak, for instance in the transom area. The simulation model used and described in Garme & Rosén (2003) –paper B and in Paper D is formulated from the following hypothesis:

-the analogy of a planing hull in calm water and the water entry of a 2-dimensional section, can be generalized to approximate the hydrodynamic loads on a slender planing hull at high-speed during transient conditions in waves.

It is a large step from the stationary conditions in calm water to the irregular situation in the fully developed seaway. The transient events demanding for time-domain simulation comprise the dimensioning conditions being the main target for the modeling efforts.

Section force formulation

It is computationally pleasant to approximate the 3-dimensional fluid mechanic problem with the sum of independent 2-dimensional ones and the analysis of the planing craft in waves is made tremendously much easier if approached by such 2-dimensional strip method. If the hydrodynamics is assumed to follow potential theory, i.e. inviscid, irrotational and incompressible flow, the dynamic section load, df_3, can be expressed by the dominating term in the Bernoulli equation, developed and eventually expressed by the product between the incident velocity of the flow, u_3, and a function that integrated over the wetted area is called added mass, a_3, according to

$$df_3 = -\rho \int_S \frac{d\phi}{dt} n_3 ds = -\rho \frac{d}{dt}\left(u_3 \int_S \phi_3 n_3 ds \right) = -\rho \frac{d}{dt}\left(u_3 a_{33}\right) \tag{1}$$

where n_3 is the hull surface normal vector component in direction x_3 (see Figure 4) and ϕ the potential function expressible as, (Newman 1977),

$$\phi = u_3 \phi_3 \tag{2}$$

where u_3 is the incident velocity in direction x_3.

The load acting on the section, equation (1), is thus expressed as the time rate of change of the incident velocity times the added mass, which could be interpreted as the time rate of change of fluid momentum in accordance with Newton second law. The added mass is only depending on the section geometry and can be pre-calculated. Thus, the difficult fluid mechanic problem does not have to be solved repeatedly at each time step during the simulation. The local draught and incident velocity is determined momentarily during simulation as the section loads are calculated and the equations of motions become nonlinear through its time and motion dependent coefficients. By separation of the geometry dependence and the incident velocity the 2-dimensional sectional load calculations can be combined to any load situation by the momentary draught and the incident velocity distributions. If the simulation made use of a 3-dimensional added mass it must be determined at every time step during the simulation and a pre-calculation similar to the 2-dimensional model is impossible since it would demand for an infinitely large number of coefficients to represent all possible positions of the wave surface relative to the hull.

In fact the added mass is frequency dependent but approaches asymptotically a value for large frequencies. The assumption of high speed, which also is the link between impact and planing, justifies the use of the high frequency added mass in modeling of the planing craft. This results consequently in limitation at too low forward speed and adds uncertainty in waves where the incident velocity fluctuates largely. The incident velocity distribution along the hull is made up from velocity components of the wave motion, the forward speed and the pitch and heave motions. In waves, situations occur that not necessarily are consistent with the *planing-immersing section analogy* or in violation with the assumptions for the load calculation. A possible situation is low or even negative incident velocity but as long as the ship has a bow-up trim the velocity of incidence is normally positive. In classical expressions the impact load is proportional to the square of the impact velocity which might lead to the mistake that lift is determined also as a section has negative incident velocity. For the simulation model in Garme & Rosén (2003) –paper B and Paper D negative velocity of incidence results in suction of the same magnitude as the lift would have been at the reverse flow direction. An alternative to this is to prescribe zero force if the velocity becomes negative which is applied by Chiu & Fujino (1989). The Tveitnes (2001) study of the exit problem showed that suction does occur but to a lower magnitude than the lift in the reverse situation.

The 2-dimensional impact is illustrated by Figure 4 that shows the principal shape of the pressure distribution and the local deformation of the water surface. The v-shaped hull creates a high pressure so rapidly that the displaced water moves upwards at the section sides. The water surface rise close to the section, so called pile-up, and the water sprays up the hull sides. The local surface deformation results in a larger wetted part of the section than would have been the case if the surface had remained flat, which is important to account for when the pressure distribution on the hull section is determined.

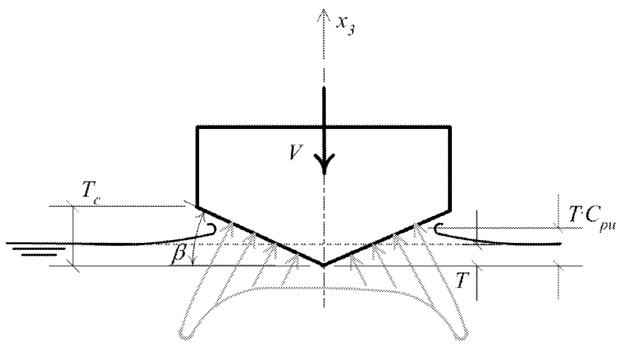

Figure 4. Illustration of the surface deformation, pile-up, and pressure distribution on a v-shaped section with deadrise β penetrating a calm water surface with velocity V.

The pressure peaks at the spray roots become more pronounced and more rapidly propagating the flatter the bottom, i.e. the smaller the deadrise. When the section is immersed to the level when the chines are wetted the pressure peaks disappear and the section load decreases clearly, see Figure 5.

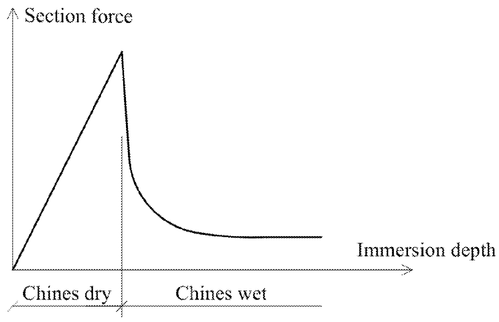

Figure 5. The dynamic vertical force principally sketch for a v-shaped section with vertical sides above the hard chines. The section penetrates the water surface with constant downward velocity.

Section force calculation

The 2-dimensional potential flow impact problem is a difficult task to model. The work of Wagner (1932) still contains the most used and referred analytical results. Wagner approximated the flow around the falling wedge to the known solution of a plate in a uniform flow. By letting the plate expand as a function of time the increasing wetted surface of the immersing wedge was modeled. The approach is often referred to as the *expanding plate theory* and is a formulation for infinitely small deadrise although Wagner suggests and exemplifies applications for deadrise in a range applicable to the planing hull. The local surface deformation according to Wagner becomes $C_{pu}=\pi/2$ using the notation of Figure 4 regardless of the deadrise angle. In fact the pile-up is deadrise dependent and the $\pi/2$ suggested by Wagner is usually considered as an upper limit. In the simulation model of Garme & Rosén (2003) –paper B and Paper D where the pile-up is not explicitly determined the following expression is used:

$$C_{pu} = \frac{\pi}{2} - \beta\left(1 - \frac{2}{\pi}\right) \tag{3}$$

This pile-up level was also used by Keuning (1994) and Payne (1995) but origin from Pierson. The expression (3) is reasonably close to the numerical results presented by Zhao & Faltinsen (1993).

The methods to solve the potential flow problem of the immersing section have developed since the classical work of Wagner (1932). The aim has been to solve the problem for real hull section shapes comprising: large deadrise, non-constant deadrise, chines and sprayrails. A great effort has been made on theoretical studies developing the expanding plate theory and in the last decades on numerical approaches to the problem. In addition, the problem has been addressed by, semi-empirical and experimental studies. Ferdinande (1966), for instance, models the wedge by an expanding lozenge in a uniform flow to which a fluid mechanical solution is found by means of conformal mapping technique. The procedure catches the deadrise dependence of the pile-up. Also Watanabe (1986) starts out from the solution of Wagner (1932) but refines the solution in the spray area. The Wagner solution is used outside the spray area and the two solutions are matched. Stavoy & Chuang (1976) make use of the Wagner theory expressing section impact loads on prismatic hulls in waves interpreting the constant immersing velocity as the momentary relative velocity between water and hull. In the experimental study of Chuang (1967) the maximum pressure is investigated by means of drop tests and compared to the Wagner expressions. The deadrise was in the range of 0° to 15°. At the smallest angles 0° and 1°, entrapped air between the section and the water surface showed to influence the impact pressure. Constant velocity impact tests have been performed by Tveitnes (2001) with the objective to measure the impact force on wedge shaped sections with hard chines in the range of 5°-45° of deadrise. Those referred experiments are in line with the common opinion that the Wagener theory constitutes an upper limit for the pressure, load and pile-up levels.

An important step towards the more complex and ship-like section geometries was taken during the last ten years by development of numerical methods adopted to determine the pressure distribution on 2-dimensional sections with chines and non-constant deadrise. The boundary element methods (BEM) introduced by Zhao & Faltinsen (1996) and Savander et al. (2002) are impressive examples. With the impact problem described by Eulers equations Arai et al. (1994) approached the solution by a volume of fluid (VOF) method showing promising results. Arai et al. stressed the ability to model the transient surface deformation and the numerical stability which might be the reason for the choice of a VOF-method. The more common approach is to assume potential flow and in that case a BEM would result in less elements and simpler numerical analysis. Savander et al. (2002) apply their method to the case of steady state calm water planing but there are no examples, to the author's knowledge, where such high-resolution method is adopted to simulation models for the planing hull in waves. Garme & Rosén (2003) –paper B reviews an attempt to determine the sectional added mass of wedges with hard chines. The approach, inspired by Tulin & Hsu (1986), was to divide the potential problem in two parts resulting in one source and one vortex singularity distribution to be determined in order to satisfy the boundary conditions. The vortex distribution was intended to model the transverse flow in the pile-up area as the section immerges. The source and vortex distributions were solved by a panel method. Section loads in level with published results were determined and used in paper B. Later uncertainties on the implementation have come up and it was understood that added mass coefficients for the simple geometries could be determined from the source distribution alone producing equally reasonable section loads. This insight simplified the simulations and was used in Paper D. Thus, doubts remain concerning the added mass calculation and it is at this moment not validated for calculations on actual hull section shapes.

As mentioned, the advanced 2-dimensional methods to calculate the added mass have not been used for simulation purposes. On the other hand has basically only the prismatic v-shaped hull been studied with the methods following Zarnick (1978) and accordingly it has been reasonable to make use of approximate analytical solutions of the added mass. Those expressions, i.e. Wagner solutions or similar, are valid to wedges but used for v-shaped sections they are only valid as long as the chines are dry. The section force after chines wetting has instead been expressed by a semi-empirical term, originally from Shuford (1958), denoted *cross flow drag*. Figure 6 shows the section force on a v-shaped section with hard chines at constant impact velocity. The panel method solution expresses a dynamic load also for chines wet similar to the semi-empirical expression from Tveitnes (2001). The analytical expression on the other hand is only valid for chines dry. The simulation models using such expression, e.g. Zarnick (1978), Keuning (1994) and Akers (1999) adds the *cross flow drag*, principally to model the more or less constant chines wet load illustrated by the panel method and semi-empirical results in Figure 6.

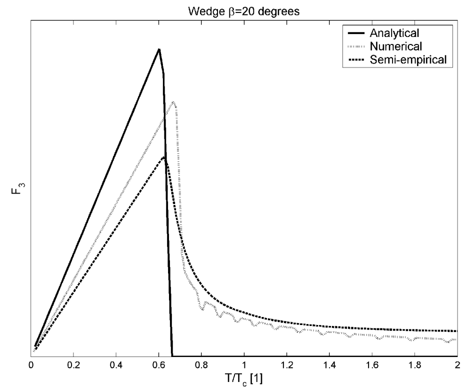

Figure 6. Non-dimensional dynamic lift force, F_3, on a wedge shaped section with hard chines as a function of penetration draught, T, non-dimensionalzed by the distance between the chine and section apex, T_c,. The analytical expression is the same as used by Zarnick (1978) basically the Wagner (1932) result with a pile up factor of $\pi/2$. The numerical result is from the panel method used in Paper D with a pile-up level according to expression (3) and the curve denoted semi-empirical is based on Tveitnes (2001) assuming a pile-up factor of $\pi/2$. The difference in the choice of pile-up level influences the force magnitude as well as the penetration draught of chines wetting.

The fact that sections, especially in the bow, with a non-zero surface normal component alongships contribute to lift and of course to resistance due to the alongships flow is modeled in Garme & Rosén (2003) –paper B and Paper D as coupling terms of the added mass. Those added mass terms are determined by assuming that the relation between the source strength of the distribution determined as the section is exposed to in-plane flow is the same also if the flow is directed normal to the cross-section plane. Thus the added mass coefficients are scaled with respect to the section surface normal component in the direction of the incoming flow. The simulation program POWERSEA also accounts for the deviation from the strictly prismatic hull and lift and resistance is compensated with respect to keel camber and deadrise variation, Ship Motion Associates (2003).

The 2-dimensional approach to model the planing hull presupposes of course an accurate method to determine the section pressure distribution or the lift force. If this is achieved it is still not as simple as to sum the section force results. The running attitude, especially the trim, is sensitive to the load in the bow and transom areas where the uncertainties to the assumed 2-dimensionallity are largest. A conclusion by Garme & Rosén –paper B is that the weakness of the modeling approach is mainly an overestimation of the lift in the transom area disadvantageously influencing the trim. A consequence of a poorly modeled running attitude is that the geometrical hull-water intersection, and thereby the determined impact force as well as the rest of the load distribution, becomes incorrect. Thus, adequate methods to determine the impact force is one necessary feature to simulate the design loads and accelerations but the wrong mean running attitude implies that the modeled situations are geometrically irrelevant. The trim angle has traditionally been corrected, addressing the dry transom stern, by a reduction of the hydrostatic lift and moment, Zarnick (1978 & 1979), Akers (1999), Keuning (1994) and Payne (1994). Paper D suggests a reduction of the aftmost part of the 2-dimensional load distribution prescribing zero lift at the transom stern. The shape of the corrected load distribution at calm water planing is illustrated by Figure 7, where the load increases from the bow and abaft until the chine is wetted. The load drops to a lower level and increases slightly with the increased displacement and from a distance afore the transom it decreases again to zero at transom.

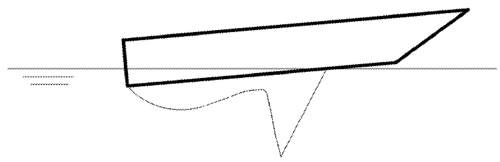

Figure 7. Principal sketch of the section load distribution for a planing craft.

The pile-up level influences the section load magnitude and the instance of chines wetting. While modeling the planing hull the pile-up level influences the alongships position of the chines wetting which due to the large difference in load magnitude for dry and wet chines, influences the running attitude. The alongships position of chines wetting is also important with respect to porpoising and the accuracy is relevant for the ability of the model to predict that coupled heave and pitch instability.

Experiments and simulation validation

Model tests are regularly used in ship design. The most common investigations are resistance measurement on towed or self propelled models and surveys of seakeeping characteristics. For quite some years great effort has been put in numerical analysis of the flow around the displacement hull targeting hull resistance and the visionary term *numerical towing tank* is frequently used. For this problem viscosity must be accounted for and the numerical task is very demanding and still not solved to accuracy competitive to the model test, although numerical results are valuable in design for comparing hull shapes and thereby for choosing lines. For seakeeping analysis the viscosity is subordinate, the fluid mechanic problem easier to solve and in the future it can be expected that accurate numerical predictions will partly overtake the roll of model testing. The obvious advantage of simulation tools is that studies can be varied to an extent that is both practically and economically unrealistic to achieve experimentally. For the development of the numerical tools on the other hand, the importance of the test facilities can not be emphasized too strongly. In this case when the planing hull in waves is studied from a simplified 2-dimensional approach there is an obvious need for experiments both to validate the simulation model, its subroutines, and for principal studies showing the behavior of the physical reality. Experiments will possibly also be used to collecting data for derivation of empirically based corrections of the computational model.

A valuable source of information is published experiment data from tests where characteristic parameters have been varied systematically and the semi-empirical methods based on such experiment series. Fridsma (1969 & 1971) performed studies on planing prismatic v-shaped hulls where the deadrise, length, centre of gravity and mass were varied apart from forward speed and wave characteristics. The tests were performed in calm water, regular and irregular waves. The Fridsma (1969) results have been used as reference data in several studies over the years: Martin (1978b), Zarnick (1978 & 1979), Chiu & Fujino (1989), Keuning (1994), Akers (1999), Garme & Rosén (2003) –paper B and Paper D. In the work of Savitsky & Brown (1976) the Fridsma (1971) experiment series in irregular waves constitute the main basis to the semi-empirical procedures for added resistance in waves and design acceleration. The design guidelines of Savitsky & Brown (1976) is a continuation of the, to the small craft naval architect classical, work of Savitsky (1964) where a computational procedure on running attitude, power requirements and porpoising stability is presented. The computational scheme is built up by semi-empirical expressions based on measurement data collected from experiments on planing prismatic craft performed from the late 1940s and the 1950s at the Davidson Laboratory at the Stevens Institute of Technology where also Fridsma made his experiments around 1970. A systematic series on non-prismatic planing hull shapes is presented by Clement & Blount (1963). The series is designated TMB Series-62, and were performed at the David Taylor Model Basin. The parent model hull was chosen on the basis of experimental experience aiming for a hull of low resistance, good rough water performance and steering qualities. Another four models were built based on the parent hull with the significant difference on the length-to-beam ratio. The models were run in calm water at various speeds, load and LCG and the running attitude, resistance and porpoising characteristics were investigated. Those published data and empirically based methods for planing hull performance are reliable sources that have been used by many researchers during the years and constitute an important reference while evaluating a simulation model.

Experimental evaluation of the simulation model

Method evaluation comprises consideration of many aspects and details. This results in a demand for information that hardly can be satisfied from published data alone. For instance does the Fridsma experiment series show the double amplitude of heave and pitch responses in regular waves, adequate and valuable information but concealing the mean running attitude and whether the oscillations were harmonic or not. Therefore, experimental studies designed to evaluate parts of the computational model or details not reviewed in the public data have been performed. In Garme & Rosén (2003) –paper B and in Paper D comparisons between simulation results and published reference data is complemented by data from those new experiments. The laboratory provides the possibility to refine the planing-in-waves event. Figure 8 sketches the test set up reviewed in Garme & Rosén (2003) –paper B.

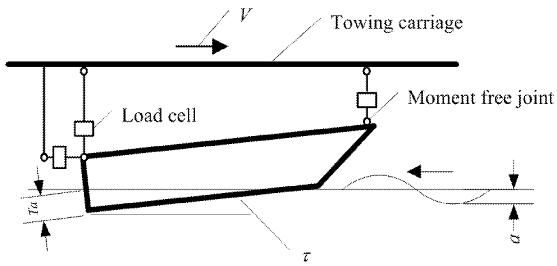

Figure 8. Captive test set-up. V is the forward speed, T_a is the draught at transom, τ the trim angle and a the amplitude of the incident regular wave.

The experiment was performed at the KTH towing tank and lift and towing force were measured. Tests were made in calm water and regular heading waves at a constant trim angle. The draught and forward speed was varied. A principal advantage of such simplified experiment is that parts of a compound calculation can be checked. If the experiment can be performed to different degrees of complexity focusing on different objectives it is a powerful tool to identifying weaknesses or errors of the mathematical model. In this case the captive model made the ship motions vanish from the calculation when the experiment was simulated and the other terms of the load determination routine could be evaluated. The calm water runs showed on overestimated lift and bow down moment while the tests in regular waves confirmed the prediction of the fluctuating load to both level and shape. Based on the captive test results, results from the Fridsma (1969) tests, predictions following the Savitsky (1964) method and on simulation results conclusions could be drawn on how the hull motion influenced the simulation.

The process of evaluation continued with data collected from the set-up of Figure 9.

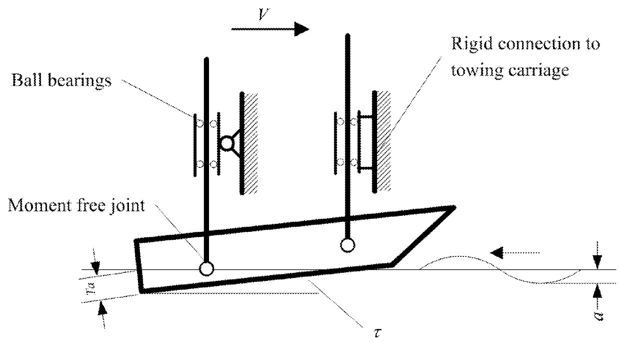

Figure 9. Test set-up for a towed planing model. The model is free to motion in heave, pitch and roll. V is the forward speed, T_a is the draught at transom, τ the trim angle and a the amplitude of the incident regular wave.

The tests were performed at the CEHIPAR ship dynamics laboratory in Madrid. The model was towed in a straight path at constant forward speed, free to move in heave, pitch and roll. The set-up serves well-defined geometric conditions for the hull and wave surface intersection which was demanded for in order to compare the measured and simulated time series. The excitation force calculation is the basis of the simulation and the load distribution to shape and magnitude is important to model correctly. The rigid body acceleration is a major design parameter and a quantity with coupling towards structural design as well as crew and passenger comfort. Aiming for a detailed picture of the excitation, preferably the pressure distribution, and the response, the hydrodynamic pressure was measured together with rigid body acceleration and motion. The vertical acceleration demands for much larger sampling frequency than the motions. This, together with practical difficulties to integrate the high frequency acceleration signals, motivates the common choice to measure both motions and accelerations. The pressure transducer and accelerometer instrumentation is illustrated in Figure 10.

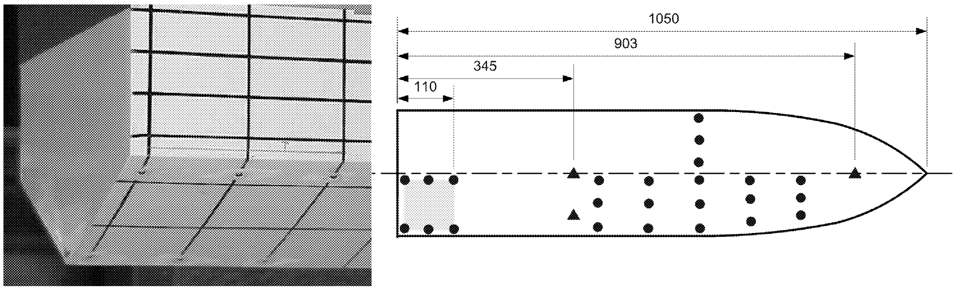

Figure 10. To the right, instrumentation overview indicating pressure transducer (●) and accelerometer location (▲). The shadowed area marks the transducers shown in the picture to the left.

The correspondence between impact loads and acceleration, between excitation and response, was evaluated by Rosén & Garme (2004) –paper C and the test set-up was confirmed to catch those quantities. The pressure distribution at occasions of impact was reconstructed from the discrete pressure transducer signals and integrated to force which was compared to the inertia force determined from the accelerometer signals. To cope with the large gradients and rapid changes characteristic for the impact pressure distribution, the method of Rosén (2004) was used to convert the transducer signals to force. This exemplifies that among the difficulties meeting the experimentalist are the often demanding task of interpretation and analysis of the measurement signals.

When evaluating the simulation model the measured pressure in the aft was integrated to section loads simply by an integration of a polynomial fitted to the signals at each station, a procedure considered adequate presupposing that the sections were submerged to a wet chines level. The study confirmed previous conclusions on the load distribution and showed on the load decrease towards the transom stern. This is discussed in Paper D and a correction of the calculated 2-dimensional load distribution in the near transom area is introduced that increases the agreement with the results of Fridsma (1969). After implementation of the near-transom lift correction, simulation of the model experiment of Figure 9 was performed. Comparisons between measured and calculated time series were possible thanks to the well-defined, and measured, wave and model hull motions. Figure 11 reviews the measured and simulated acceleration signal of a test sequence and snapshots of the model airborne and, a moment later, impacting the water surface.

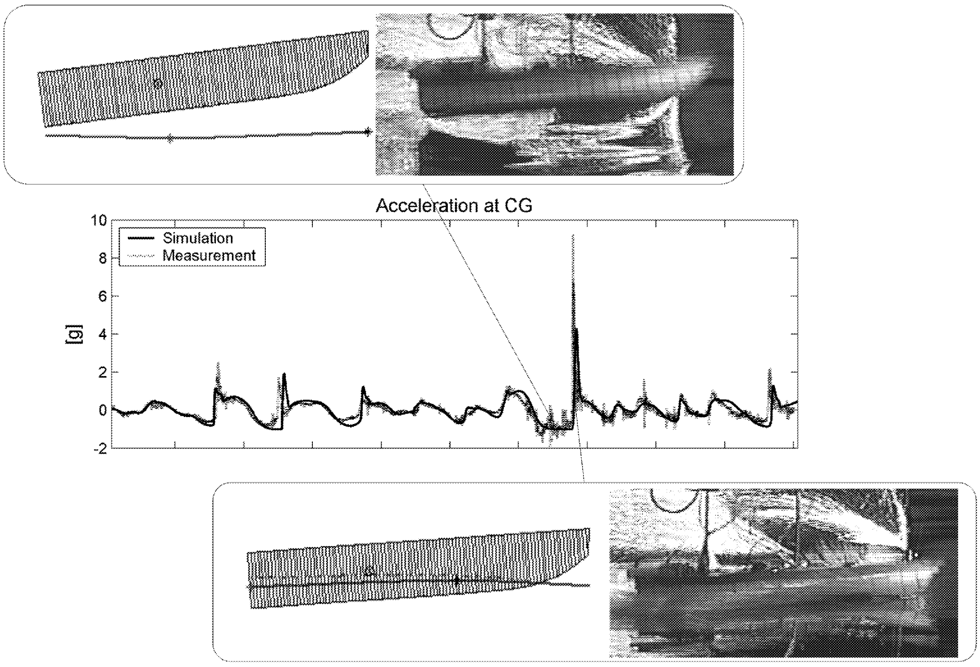

Figure 11. Illustration of time-domain comparison of a simulated model test sequence.

The link between the simulation and the experiment is the wave model. The experiment was performed in linear regular and irregular waves and the wave height was measured at a point diagonally in front of the model where the incident waves were uninfluenced by hull generated waves. The wave signal is a function of time and of the global measurement position at every sample. Due to the dispersive characteristic of the waves the surface shape changes momentarily and the relation between a point measurement and the wave profile along the model hull is not just geometrical but depending on the mechanical characteristics of the water surface. The wave model for the simulation of the experiment sequence was created from the wave height meter signal by means of the method of Garme & Hua (1999) – paper A. The method gives a linear model of the waves as a function of time and location and the wave surface and motion along the model hull are thereby known. The method is reliable and useful especially for tests in irregular waves where evaluation otherwise are limited to a statistical analysis.

The evaluation in the time domain strengthens the confidence in the ability of the strip approach to generate technically useful results. But the refined evaluation also opens for questions on details, for instance how the transition to atmospheric pressure at the dry transom look like during the variety of geometrically different situations that occur as the planing craft runs through rough waves. At the sequence shown in Figure 11, the aft end is first to hit the water surface at re-entry and impact load develops all over the wetted length. Figure 12 shows the calculated load distribution from the first water contact until the acceleration starts to decline. At this specific event when the chines are dry also at the transom stern the modeling of the load transition to atmospheric pressure is significant for the total lift and moment. It is conceivable that the near transom load is principally different for the chines dry and chines wet situations. If the pressure adopts to atmospheric for a shorter distance than assumed at the simulation illustrated in Figure 11 and Figure 12, the simulated acceleration and pitch velocity would have been slightly larger which would have increased the agreement at the very moment of impact. On the other hand the sequence before the jump is satisfactory modeled with the chosen correction. For further studies, detailed model measurements of the near transom pressure distribution are necessary to get a clear picture of the conditions and thereby enable improvement of the load distribution modeling.

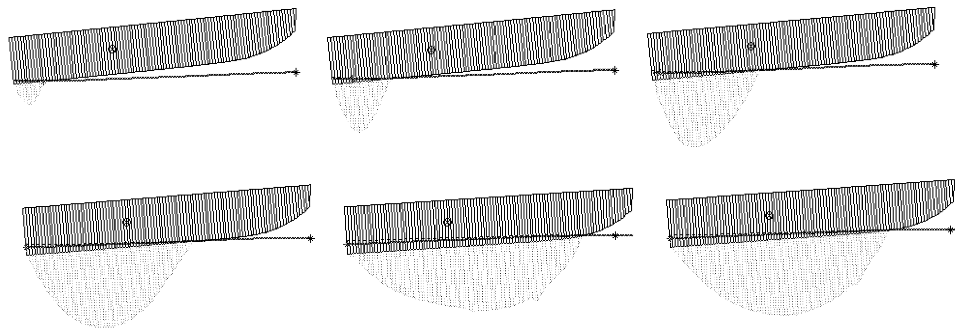

Figure 12. The simulated load distribution stepped during the impact event illustrated in Figure 11. It is approximately 0.006 s between the pictures.

The simplified and well-defined situations necessary for method development and evaluation can be achieved in the ship model laboratory. Still, it should be kept in mind that the actual aim is to predict a reasonable correct picture of the planing vessel at real conditions and simulations must be evaluated in comparison to full-scale results. Full-scale trials and the following data analysis are connected with difficulties. With possible motion in all degrees of freedom, and as speed, wave direction, course and wind conditions can not be kept constant, uncertainties are unavoidable. In Garme & Rosén (2003) –paper B, a statistical comparative analysis of simulations and full-scale trials was performed which is the only reasonable approach when the sea state is not known in detail. The wave condition is probably the most delicate circumstance to keep track of while testing at sea. The seaway usually described in terms of a variance spectrum, is often estimated by the eye in terms of a significant wave height or at best measured by means of a stationary wave buoy in or close to the test area. It would be of great value to all evaluation of full-scale measurements if the incident wave surface could be measured by an onboard based system. There are examples where radar technique is used but for the high speed craft in rough sea with spray and breaking waves the success is not obvious. Anyway, field measurement of waves is a subject to further studies. Eventually it should be mentioned that full-scale test could also throw light on possible scale effects not present in the simulation model.

Application

The simulation models have been developed for the planing craft in head seas. They are principally also valid in following seas but the important issues in that case are addressed to stability out of the centerline plane for instance broaching that is dependent on roll and yaw, degrees of freedom which are not modeled. Head seas, on the other hand, constitute the condition for the most severe relative motions and velocities and represent design conditions with respect to local and global hull loads and vertical acceleration. Garme & Rosén (2003) – paper B show on the basis of full-scale pressure measurements that the local load could be even larger in bow seas as the relative deadrise occasionally becomes very small resulting in a large impact pressure on one side of the hull. In the light of such result the design loads of the local structure might be referred to bow sea condition rather than head seas. Still, the limitation of modeled degrees of freedom does not leave the simulation models without practical applicability. A long term aim has been to simulate loads relevant for structural design which demands for accurate routines to determine the load distribution. In order to determine also the local design loads the load distribution needs to be resolved into pressure. The simulated load distribution appears to be determined reasonably accurate but additional verification especially addressing impact loads in rough seas remain. Simulation results on the rigid body acceleration can be used to determine design acceleration for structural design, setting equipment specifications, and predicting the working conditions aboard.

Simulations indicate new possibilities but lack a methodological frame. In Akers (1999) the ship is simulated in a random seaway and the peaks of the mean pressure time history, determined from the simulated loads, are treated statistically determining the most probable largest mean pressure for a certain number of wave encounters. This statistical average is connected to the maximum and the design pressure by means of verified approximations. Rosén & Garme (2001) argues similarly addressing direct calculations. The sketched scenario is to perform simulations with time-domain realizations of a design sea state and from the knowledge of the relationship between the transverse pressure distribution and the simulated section load deliver the instantaneous pressure distribution as load input to a Finite Element Analysis investigating the critical events. In both cases the question of how to choose the design sea state is left open and also how the simulation matrix should be defined. Troesch & Hicks (1992) approaches those questions and discusses a methodology to analyze the planing craft and incorporating the use of a planing hull simulator. They propose a stepwise refined analysis where the initial steps lead to a limited number and range of variables to be analyzed by simulations. They exemplify that critical excitation frequencies and LCG positions indicating porpoising instability can be identified from linear and simplified non-linear analysis of the non-linear dynamic system. Those analyses are less time consuming than simulations and will serve the overview of a large number of parameters and their influence on the dynamics which could be used to set a simulation matrix. Clearly, a simulation tool is a complement to the variety of existing design tools and needs support from experience, rules of thumb and more or less advanced approximations. Insight on the influence of various parameters on power requirement and seakeeping performance is given by Savitsky & Koelbel (1993) describing the planing hull dynamics. The relations between running attitude, resistance, LCG and the risk of porpoising have to be investigated at an early stage of design and are typically approached by the methods of Savitsky (1964), Savitsky & Brown (1976) and Mercier & Savitsky (1973). A rational working process is exemplified by Savitsky et al. (1972) reviewing a pre-design for a patrol craft including model experiments to verify the design. Imagine the next design cycle to refining the outlined design and the analysis would be extended with simulations followed by structural design. Obviously there is a need for a methodology of the high speed craft design in order to make efficient use of simulation tools. Partly, it is a question of the principles of design and what actually is the demanded vessel performance. A related subject is the choice of design acceleration. Koelbel (1995) brings up the interesting thought that the acceleration is limited by the motivation and endurance of the crew. That statement leads to the question on the correlation between design loads and the loads that the vessel actually is exposed to during operation. This could be surveyed by long-term measurements to picture the operational conditions for different types of vessels and their variety of activities. Monitoring the loads during operation would open for more precise design conditions and a more optimized hull structure.

Developments and further studies

The most central step is perhaps the methodology development discussed in the previous section. The simulation model is at a stage where it should be evaluated for design load generation but the principles on design sea state and what simulations to perform and how the results should be interpreted have to be decided for. Simulations can also be used to predict running attitude, motion and acceleration in head seas which besides seakeeping characteristics picture the operability. An important step is to implement a more detailed routine to determine the sectional added mass. The previously mentioned methods by Zhao & Faltinsen (1996) and Savander et al. (2002) are designed to section shapes of non-constant deadrise and with chines. Implementing such method is definitely demanding but would make the simulation method more general with respect to hull shape and is assumed to increase the accuracy and to increase the possibility and reliability of detailed studies. The increased numerical complexity would appear in the pre-calculation and thus not effect the simulation times. Although, for more advanced hull shapes the total number of sections might have to be increased and together with a shortened time step the processor load would increase. As a reference it could be mentioned that the simulations of the Fridsma hulls in Paper D were made with 120 sections for the 45” hulls (80 section at the 36” prismatic part and 40 sections for the 9” bow) and a 0.0015 s time step which was estimated refined enough for the speeds and encounter frequencies of those simulations.

Another aim is of course to model motions out of the centre line plane which would open for modeling oblique waves and maneuvering. A leap of development as such, would presume modification of the pre-calculation scheme and of the routine for added mass calculation in order to deal with asymmetric sections. Asymmetric impact has been studied numerically by Iafrati (2000) and the method Zhao & Faltinsen (1996) would also be applicable. Steady asymmetric planing has been addressed for instance by Xu & Troesch (1999) but asymmetric planing in waves is definitely an area lacking both experimental and theoretical investigations. Developing the simulation model in order to increase the degrees of freedom is of course not limited to the sectional added mass calculation but would imply several more or less radical modifications. A less grandiose development would be an implementation of a ride-control model for the vertical motions. A routine to model how trim control flaps, interceptors or T-foils interact with motions and influence the global hull loads in order to minimize motions would be useful. Another step within reach is the implementation of a simple trust force model.

In continuation of Paper D is a quantitatively precise experimental investigation of the pressure distribution in order to study the pressure decrease towards the chines and transom in detail, the load fluctuation in waves and the differences with respect to whether the planing surface with a transom stern has dry or wet chines. Further should a systematic comparison of simulations of the measurements presented in Rosén & Garme (2004) –paper C be performed making the most of this available data. On the track of direct calculations will simulated loads be used as Finite Element Analysis input and the structural response evaluated with respect to the full-scale data of the experiment presented in Garme & Rosén (2003) –paper B and to data from planed new full-scale trials.

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