Loads and Responses for Planing Craft in Waves

Anders Rosén
Aeronautical and Vehicle Engineering
Division of Naval Systems
SE-100 44 Stockholm, Sweden

TRITA-AVE 2004:47
ISSN 1651-7660
ISBN 91-7283-936-8

Akademisk avhandling som med tillstånd av Kungliga Tekniska Högskolan fredagen den 17 december 2004 kl. 10, sal E1, Lindstedtsvägen 3, Stockholm, för teknisk doktorsgrads vinnande framlägges för offentlig granskning av Anders Rosén.

Preface

The work presented in this thesis has been carried out at the Division of Naval Systems at the Department of Aeronautical and Vehicle Engineering at KTH, under the supervision of Jakob Kuttenkeuler. The work was performed with financial support from, and partly in cooperation with, the Swedish Defence Material Administration (FMV). The experiment presented in Paper B and Paper C was performed at the Dynamics Laboratory of Canal de Experiencias Hidrodinamicas de El Pardo (CEHIPAR) in Madrid, and funded by the EU-program Training and Mobility of Researchers – Access to Large Scale Facilities (TMR-ALSF). The colleagues at the Division of Naval Systems – Jakob Kuttenkeuler, Ivan Stenius and Karl Garme – are acknowledged for the unique working environment, which offers a great mix of generosity and demands, ambitions and joy.

Stockholm, November 2004

Anders Rosén

Abstract

Experimental and numerical analysis of loads and responses for planing craft in waves is considered. Extensive experiments have been performed on a planing craft, in full-scale as well as in model scale. The test set-ups and significant results are reviewed. The required resolution in experiments on planing craft in waves, concerning sampling frequencies, filtering and pressure transducer areas, is investigated. The aspects of peak identification in transient signals, fitting of analytical cumulative distribution functions to sampled data, and statistical convergence are treated.

A method for reconstruction of the momentary pressure distribution at hull-water impact, from measurements with a limited number of transducers, is presented. The method is evaluated to full-scale data, and is concluded to be applicable in detailed evaluation of the hydrodynamic load distribution in time-domain simulations. Another suggested area of application is in full-scale design evaluations, where it can improve the traceability, i.e. enable evaluation of the loads along with the responses with more confidence.

The presented model experiment was designed to enable time-domain monitoring of the complete hydromechanic pressure distribution on planing craft in waves. The test set-up is evaluated by comparing vertical forces and pitching moments derived from acceleration measurements, with the corresponding forces derived with the pressure distribution reconstruction method. Clear correlation is found.

An approach for direct calculations of loads, as well as motion and structure response, is presented. Hydrodynamic loads and motion responses are calculated with a non-linear time-domain strip method. Structure responses are calculated by applying momentary distributed pressure loads, formulated from hydrodynamic simulations, on a global finite element model with inertia relief. From the time series output, limiting conditions and extreme responses are determined by means of short term statistics. Promising results are demonstrated in applications, where extreme structure responses derived by the presented approach, are compared with responses to equivalent uniform rule based loads, and measured responses from the full-scale trials. It is concluded that the approach is a useful tool for further research, which could be developed into a rational design method.

Keywords: planing craft, high-speed craft, waves, model tests, full-scale trials, hull-water impact loads, slamming, pressure measurements, pressure distribution reconstruction, experimental analysis, statistical analysis, time-domain, simulations, non-linear strip methods, direct calculations, finite element analysis, design loads, design methods

Dissertation

This doctoral thesis consists of an introduction to the area of research and the following appended papers:

Paper A

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 B

Rosén, A., *Impact Pressure Distribution Reconstruction from Discrete Point Measurements*, International Shipbuilding Progress, Vol.52, No.1, 2005.

Paper C

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

Paper D

Rosén, A., *Direct Calculations of Loads and Responses for Planing Craft in Waves*, To be submitted for publication, 2004.

Division of work between authors

Paper A

Rosén performed the full-scale trials, evaluated the simulations to the trials, and wrote the corresponding sections of the paper. Garme developed and reviewed the simulation method and the evaluation to model tests. The introduction and the summary and conclusions sections were written jointly by the authors.

Paper C

Rosén developed the method for pressure distribution reconstruction, and reviewed and applied it in the paper. Otherwise the paper was written jointly by Rosén and Garme, and the experiment and the analysis was made in co-operation between the authors.

Contents

Background

A ride with a planing craft in rough seas can be a spectacular experience. As illustrated in Fig. 1, high speed in waves can imply large and violent motions. High speed also implies large and complex hydrodynamic impact forces and large inertia loads. The hull must be strong and stiff enough to carry these loads, still light enough to make the craft efficient from point of view of building cost, load capacity, operational cost, powering, environmental influence, etc. The craft must also have seakeeping characteristics which allow for operation in speeds and seas appropriate for its mission, for example from point of view of crew and passenger comfort and habitability.

Fig. 1 Auckland police patrol RIB (courtesy of Industrial Research Ltd).

The last decades has involved a tremendous development in the design, construction and operation of high-speed craft. Faster and larger vessels have been built and new concepts have been introduced. The relatively recent introduction of high-speed ferries, such as *HSC Gotland* in Fig. 2, has on many routes decreased the time required for a journey with as much as a factor 2. The driving forces in this development are of course complex, but can partly be explained by the relatively low oil price and the global economical expansion where time is money. Also the technical development is a driving force in itself. To deal with the large and complex loads in general and special issues related to new craft concepts, the development of high-speed craft has involved extensive research on new materials, material concepts, and methods for structure and hydrodynamic modelling. This has lead to considerable accomplishments. However, the areas of structural design and seakeeping analysis of high-speed craft still involves great challenges.

Fig. 2 *HSC Gotland* (courtesy of Destination Gotland).

An example of a recent and very advanced high-speed craft design is the *Visby* class corvette in Fig. 3. This ship is actually built of plastic. The hull is a sandwich construction, with carbon fibre reinforced vinyl ester laminates separated by a PVC foam core. The material concept was in several aspects found beneficial compared to more conventional materials such as metals and glass fibre reinforced plastics, for example concerning structural weight but also concerning acquisition and maintenance cost. The design is based on massive research in many fields, for example on the new material concept, production techniques, and on methods for structure analysis and optimization. The available methods for seakeeping analysis and design load prediction were found limited, and were complemented by model testing and research, development and application of methods for hydrodynamic simulation, (see for example Lönnö (1998), Hellbratt&Vallbo (1998), Milchert et al (1997)).

Fig. 3 *The Visby corvette* (courtesy of Kockums).

The work in the present thesis is to some extent a continuation of the research on loads and responses performed in connection with the development of *Visby*. As the work has developed, the focus has mainly been on planing craft, i.e. smaller high-speed craft operating in the higher speed regimes. The results are however also relevant for larger craft. The work started with full-scale trials on the planing craft *90E* in Fig. 4, Garme&Rosén (2003)/PaperA. The craft is about 10 metres long, has a displacement of 6.5 tonnes and a maximum speed of +40 knots. The material concept is the same as for *Visby*. The primary purpose of the trials was evaluation of the structural design. The craft was operated in rough seas as hard as possible concerning crew endurance. Structure responses were measured and compared with criteria.

Fig. 4 *Storebro 90E.*

The conclusion from the trials was that the design is generally good, in that the allowable response levels were approached but not exceeded. However, high strain levels were observed in the deck structure in the area of the front corners of the cockpit. This lead to a slight modification of the design, and brought further light on the limitations in the design methods presently available to the planing craft designer. The full-scale trials also enabled collection of reference data, which has been used for evaluation of simulations through the rest of the work. It also gave opportunities to general phenomenological observations, and several questions and problems related to experiments on planing craft in waves where identified.

In connection to the performance of the full-scale trials on *90E*, Rosén and Garme formulated a research program, which implied time-domain monitoring and modelling of loads and responses for planing craft in waves. The aim was increased understanding of the involved mechanisms, and development of methods which could be used in pursuing more optimised planing craft designs. This has resulted in the development of a non-linear simulation method for modelling of hydromechanic loads and rigid body motion responses, (Garme 2004a). Within the present thesis it has resulted in the development of method for reconstruction of the hydrodynamic pressure distribution at hull-water impacts from discrete point measurements, Rosén (2005)/PaperB; design and performance of an experiment on *90E* in model scale addressing the hydromechanic pressure distribution, Rosén&Garme (2004)/PaperC; and development of an approach for direct calculations of hydromechanic and structure inertia loads, as well as motions and structure responses for planing craft in waves, Rosén (2004)/PaperD. The work has also involved further investigation of several of the questions identified during the full-scale trials, for example concerning experimental resolution in terms of sampling frequencies, filtering and pressure transducer areas, and different aspects of statistical analysis, such as identification of peak values in transient signals.

Outside the scope of this thesis, but within the same project frame work at the Division of Naval Systems, the problem of wave measurements during full-scale trials has been further investigated. A portable wave buoy is being developed, which will offer a robust technique to record time series of the wave elevation, from which significant wave height, mean wave period, wave direction and spectra can be determined. A cooperation has been established with IRL (Industrial Research Ltd.) in New Zealand, which for example involves detailed investigation of hydrodynamic loads and fluid-structure interaction for high-speed craft in waves, by means of explicit finite element analysis and experiments with the servo-hydraulic slam test system at IRL. Furthermore, extensive full-scale trials have been performed on *Visby* in Fig. 3. The measurements have so far been used for primary structure evaluation. Further analysis and trials is planned for.

The following five sections gives an introduction to the area of research. The first two sections is a basic introduction to the mechanisms of planing craft and loads and responses in waves. The third section is a critical review of the semi-empirical methods currently available to the planing craft designer. The forth and fifth sections gives a brief introduction to significant experimental and theoretical work on planing craft, and also reviews the contributions of the present thesis. Finally future work is outlined.

Planing Craft

At zero speed a craft is floating according to the physical law referred to as Archimedes' Principle. It states that a body immersed in a fluid is buoyed up by a force that equals the weight of the displaced fluid. As the craft starts to move through the water, the water flow around the hull generates dynamic pressures in addition to the static buoyant pressures. Hereby a wave system, consisting of transversal waves and waves diverging with an angle to the craft heading direction, is formed around the hull as illustrated in Fig. 5.

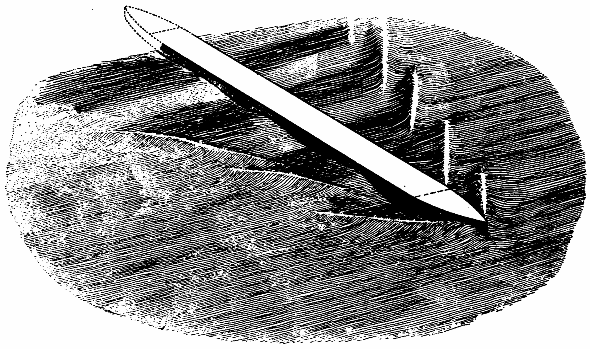

Fig. 5 W. Froude's sketch of craft generated wave system.

Water waves has the peculiarity of a fixed relation, $V_w/\surd L_w$=1.25 $m^{1/2}s^{-1}$, between their speed of propagation V_w and their length L_w. Hence, longer waves travel faster than shorter. The craft generated transversal waves moves with the craft, i.e. $V_w=V$ where V is the craft speed, and consequently has the length $L_w=(V/1.25)^2$ m. As seen, the vessel in Fig. 5 spans approximately four of its own generated transversal waves, i.e. $L/L_w=4$ where L is the craft length. The condition can be described in terms of the craft speed-length ratio, $V/\surd L$=0.625. As speed increase the craft generated waves get longer, and at $V/\surd L$=1.25 waves with lengths equal to the craft waterline length are generated.

The waves are related to pressure variations along the hull, principally lift in the crests and suction in the troughs. At lower speeds, where the craft spans several waves as in Fig. 5, the pressure variations counter measures along the hull length. The draught and trim are the same as at zero speed, and the craft weight is entirely supported by buoyant forces. The power requirements are modest and the hydrodynamic drag is dominated by friction. Craft operating in these speeds are referred to as *displacement* craft, and preferably have hull shapes which are tapered at the stern and curved upwards toward the water line, i.e. has convex sections and buttocks, to minimize flow separation which is another source of drag.

Also the wave generation is a source of drag, since the wave generation implies energy dissipation. Above $V/\surd L$=0.9 the wave making drag becomes considerable. Here the craft spans less than two waves in its own bow wave train. Above $V/\surd L$=1.25 the generated waves are longer than the craft length, and the craft is literally climbing uphill on its own bow wave. Here a rounded hull form results in negative pressures in the after parts of the hull, the craft trims down by the stern, the draught increase, and the wave making resistance becomes a virtual barrier to further speed increase. Passage of this barrier requires a hull shape which avoids the negative pressure developments. The principles of pressure distribution and wave making for different hull shapes is illustrated in Fig. 6.

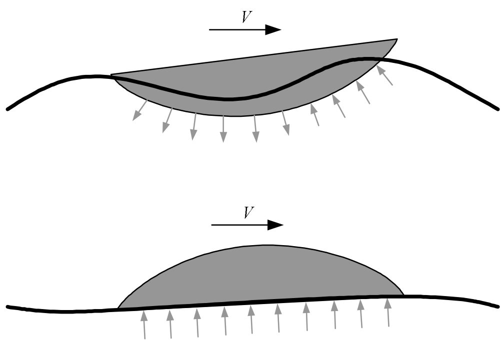

Fig. 6 Principles of pressure distribution and wave generation for different hull shapes.

With flatter buttock lines terminating in a transom stern, the flow is forced to separate cleanly by the transom, and a positive dynamic pressure can develop all over the hull bottom. Increasing speed will here instead decrease the trim and draught. The craft is lifted out of the water and is said to be *planing*. To reach really high speeds, i.e. approximately $V/\sqrt{L}>2.8$, effective flow separation is required, not only at the transom, but also at the sides. This is achieved by hard chines and sometimes also additional spray rails which stimulates the sideways separation. Fig. 7 shows the body plan of *90E* in Fig. 4, which has a typical deep-vee planing hull shape with hard chines and chine flats.

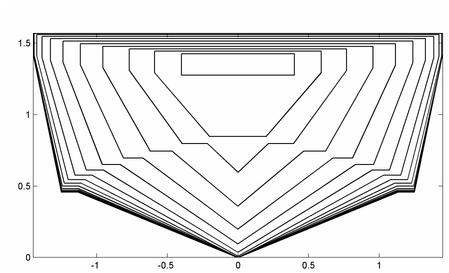

Fig. 7 Body plan of *90E* – typical deep-vee planing hull with hard chines and chine flats.

As the craft is lifted out of the water and starts planing, the wave generation and the related drag is considerably reduced. Fig. 8 is a schematic illustration of the relation between total drag and speed for a planing hull form in comparison to a displacement hull. The speed barrier for the displacement hull is clearly seen as an exponential drag growth. The hump in the planing hull curve is related to the reduction in wave drag at the transition from displacing to planing.

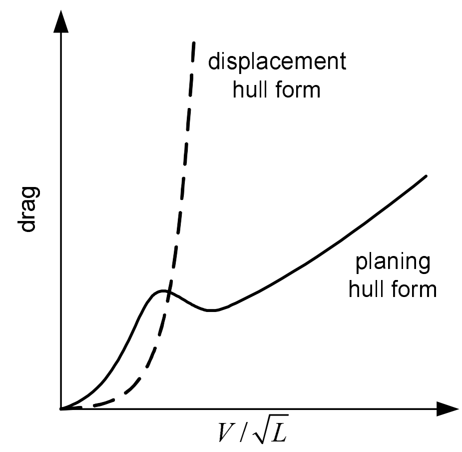

Fig. 8 Relation between total drag and speed for different hull shapes.

For short and wide power boats, such as *90E* in Fig. 4, the transition from displacing to planing is quite significant as illustrated in Fig. 8. However, for more slender hulls like *Visby* in Fig. 3, the transition is more gradual without significant hump. Hence, it is difficult to define an exact speed-length ratio for when the transition occurs. Savitsky (1985) refers to the speed regime $1.25<V/\sqrt{L}<2.8$ as *semi-planing*, because here the hull weight is supported partly by hydrostatic and partly by hydrodynamic forces. For $V/\sqrt{L}>2.8$ the hydrodynamic forces dominates and the craft is *planing*, and for $V/\sqrt{L}<1.25$ the craft is *displacing*. The hydrostatic/hydrodynamic lift fraction as a function of speed is schematically illustrated in Fig. 9.

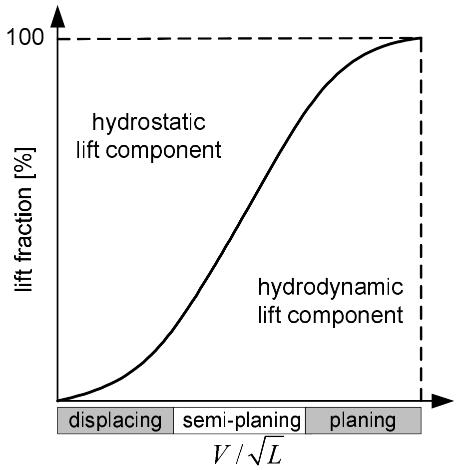

Fig. 9 Hydrostatic/hydrodynamic lift fraction as a function of speed.

Instead of the speed length ratio $V/\sqrt{L}$, different speed regimes are often characterised by the non-dimensional $V/\sqrt{(gL)}$, which is generally referred to as the *Froude number*, where g is the gravitational constant. However, in the planing speed regime, the craft length is not the most relevant measure of the speed-size relationship, since the wetted hull length decrease as the craft is lifted out of the water with increasing speed, and the wave making gets less significant. Here instead the speed beam ratio $V/\sqrt{(gB)}$ is often used, where B is the craft maximum wetted beam. In terms of the speed-beam ratio Savitsky&Brown (1976) characterises the displacing regime as $V/\sqrt{(gB)}<0.5$, semi-planing as $0.5<V/\sqrt{(gB)}<1.5$, and planing as $V/\sqrt{(gB)}>1.5$. Another measure sometimes used is the speed-volume displacement ratio $V/\sqrt{(g\nabla^{1/3})}$, where ∇ is the craft volume displacement in m^3 corresponding to the design waterline. IMO (2000) defines a *high-speed craft* (*HSC*) as a craft with a maximum speed $V_{max}\geq 3.7\nabla^{0.17}$ m/s. Table 1 exemplifies three mono-hull crafts of different sizes, in relation to different speed coefficients and the different speed regimes described above. Craft A is typically a fast ferry like *HSC Gotland* in Fig. 2. As seen in the table it is to be considered as semi-planing according to the definitions above. Craft B is on the border between semi-planing and planing; this could be a passenger craft or a naval craft like *Visby* in Fig. 3. Craft C is similar to the RIB in Fig. 1 and *90E* in Fig. 4 and is fully planing. The IMO definition covers a wide speed range, considering all three as high-speed craft.

Table 1 Mono-hull craft of different sizes in relation to different speed coefficients.

Craft	L [m]	B [m]	∇ [m^3]	V_{kn} [kn]	V [m/s]	$3.7\nabla^{0.17}$	$V/\sqrt{L}$	$V_{kn}/\sqrt{L_{ft}}$	$V/\sqrt{gL}$	$V/\sqrt{gB}$	$V/\sqrt{g\nabla^{1/3}}$
A	110	16	2500	35	18	14	1.7	1.8	0.55	1.4	1.6
B	60	10	600	40	21	9.2	2.7	2.8	0.87	2.1	2.3
C	10	2.5	6.5	40	21	5.1	6.6	7.0	2.1	4.2	4.9

The physical mechanism creating the hydrostatic lift, is the hydrostatic pressure which is acting normal to the hull surface. The hydrostatic pressure increase linearly with the distance below the water surface, i.e. $p_{stat}=\rho zg$, where ρ is the fluid density, g is the gravitational constant, and z is the vertical distance below the water surface. The pressure subjected to a craft at zero speed is schematically illustrated in Fig. 10. The pressure distribution at displacing speed has a similar character, with small additional contributions from dynamic effects.

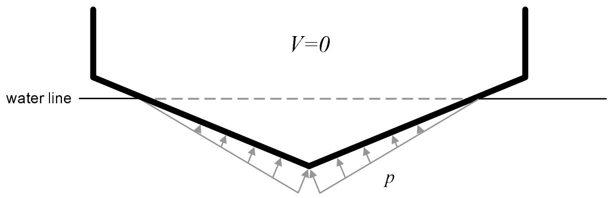

Fig. 10 Hydrostatic pressure distribution at zero speed.

The dynamic pressure on planing craft is related to the relative flow around the hull. It will here be principally described with reference to Fig. 11-Fig. 13. The craft in Fig. 11 is assumed to run in steady state in calm water, i.e. with constant speed V, draught, and trim angle η_5. β is the hull deadrise. CG marks the centre of gravity. α–α marks a virtual cut perpendicular to the keel, at arbitrary longitudinal distance x measured from the transom. The flow velocity component normal to the keel is $U=Vsin(\eta_5)$. As seen, the keel is wetted from a to d, i.e. for $d\leq x\leq a$. The principal characteristics in the *chines-dry* region, i.e. $b<x\leq a$, is illustrated in Fig. 12. As seen, the water surface is deformed and *piles-up* close to the hull. At the *spray-root*, i.e. the intersection between the piled-up water line and the hull, a *spray-jet* is formed. The peak in the hydrodynamic pressure distribution is related to the formation of the jet. Because of the pile-up, the chine is wetted already at $x=b$, well ahead of the intersection between the chine and the undeformed water line. In the *chines-wet* region, $d\leq x\leq b$, the sideways flow separates at the sharp chine, where the hydromechanic pressure adjusts to atmospheric pressure, as pictured in Fig. 13. In the aftmost region, $d\leq x<c$, where the chines are below the undeformed water line, the water line deforms into a hollow. By the transom, the longitudinal flow separates and the hydromechanic pressure adjusts to atmospheric as by the chines.

The dynamic draught and trim corresponds to a total hydromechanic lift equal to the craft weight, and a centre of pressure in vertical line with the centre of gravity. The hydrodynamic pressure is proportional to the normal flow velocity squared, i.e. $p_{dyn}\sim U^2$. The hydrodynamic pressure in the chines-dry region, $b<x\leq a$, is also proportional to the hull deadrise, i.e. $p_{dyn}\sim\beta$, as illustrated in Fig. 14. As seen, the pressure increase with decreasing deadrise. Also the peakiness increase with decreasing deadrise; in the figure the peak pressure is two times the keel pressure for $\beta=20^{\circ}$, and four times for $\beta=10^{\circ}$. From a calm water performance point of view, a small deadrise is preferable, because it decrease the wetted area and thereby also the frictional resistance. However, in waves a larger deadrise is needed to limit the pressures and the corresponding vertical accelerations at wave impacts, as described in the following section. The design of planing craft is therefore always a compromise between calm and rough water performance. This generally results in a *warped* hull, with larger deadrise in the forebody (20-50°) and smaller in the after body (10-20°), as for *90E* in Fig. 7.

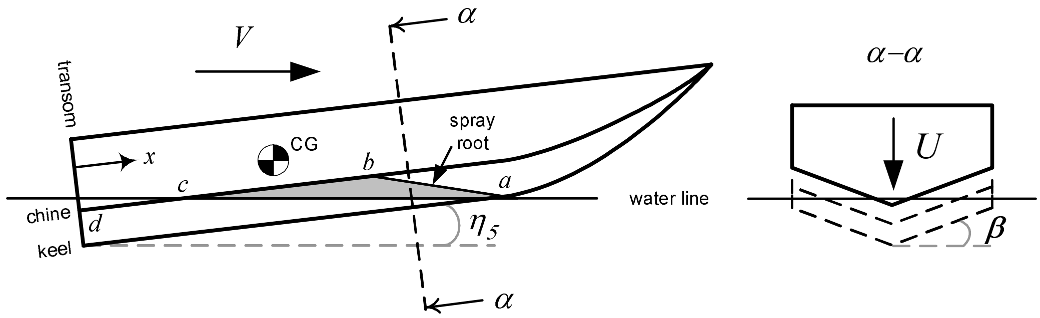

Fig. 11 Profile and frontal view of planing craft at constant speed V.

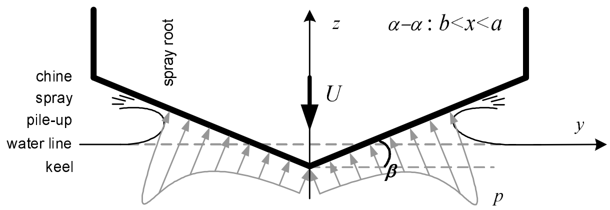

Fig. 12 Chines-dry characteristics.

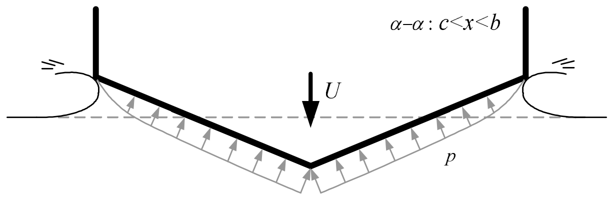

Fig. 13 Chines-wet characteristics.

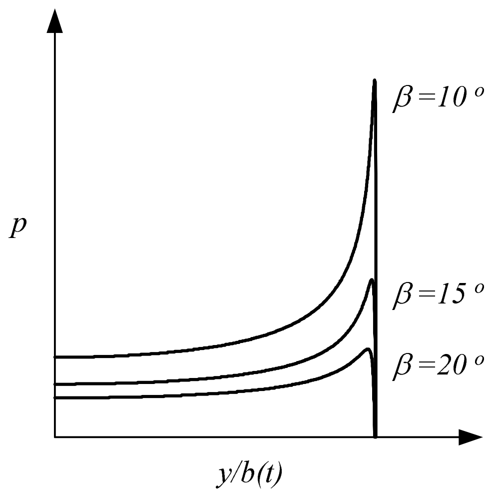

Fig. 14 Pressure distributions for different deadrise for the same constant incident velocity. $b(t)$ is the momentary wetted beam.

Loads and Responses in Waves

In the previous section it was described how the craft relates to the waves generated by its own motion. The craft is of course also affected by wind generated waves. When it is blowing, energy is transferred from the wind to the water through aerodynamic pressure variations, principally pressure forces pushing on the wave crests. The wave motion is an interaction between potential and kinetic energy in the water. For a craft in waves, some of the wave energy is transferred to the craft through hydromechanic pressure variations on the hull surface. Hereby the craft oscillates with the waves. Here some results from the model experiment in Rosén&Garme (2004)/PaperC will be used to describe the loads and responses for planing craft in waves. The model, which is displayed in Fig. 15, is a modified version of *90E* in Fig. 4 in scale 1:10, having a length of one metre and a displacement of 6.5 kg. An example of measured motion response is given in Fig. 16, where the model is heading regular waves with 0.075 m double amplitude and 1.6 s period at a speed of 4.5 m/s.

Fig. 15 The planing craft model used in the experiments in Rosén&Garme (2004)/PaperC.

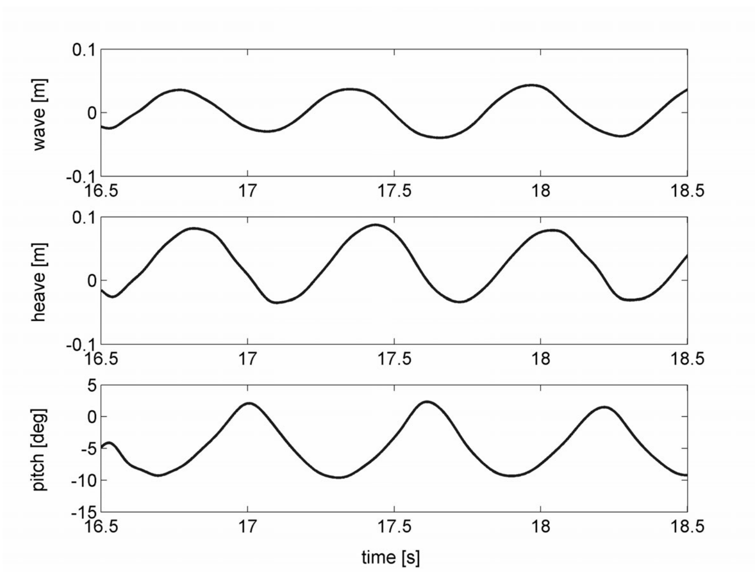

Fig. 16 Heave and pitch response for the model in Fig. 15 heading regular waves at 4.5 m/s.

The hydromechanic pressure variations on planing craft in waves, can be described in a similar way as the pressure at calm water planing in the previous section. At sections with dry chines the pressure is characterised by a peak at the intersection between the hull and wave surfaces. At sections with wetted chines the pressure is more uniform, and the flow separates by the chines and transom, as clearly seen in Fig. 15. The relation between the hydrodynamic pressure, the incident velocity, and the relative geometry is similar as in calm water planing, i.e. p_{dyn}~U^2,β. However, in waves the incident velocity U is the compound effect of craft forward speed, and the oscillating wave velocities and craft rigid body velocities. Similarly, the relative geometry is not only determined by the hull deadrise as in calm water, but is governed also by the wave geometry and the craft motions. The pressure distribution oscillates with the relative oscillation between the hull and the water. The situation is illustrated in Fig. 17 for a sequence where a hull section moves downwards through a wave surface. The pressure distribution is pictured for three time instants, t_1, t_2 and t_3. As illustrated in Fig. 1, planing craft motion can be violent, and the incident velocity when the hull hits back on the wave surface can consequently be very high, resulting in very high pressure levels. Because of the local variations in the involved geometries and velocities, the shape, magnitude and development of the pressure distribution is in reality much more complex than illustrated in Fig. 17.

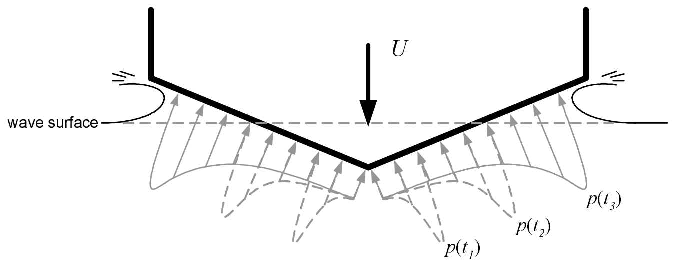

Fig. 17 Pressure distribution schematically pictured for three time instants during hull-wave impact.

Fig. 18 shows measurements with a pressure transducer in the fore part of the planing model for the same sequence as the motion measurements in Fig. 16. The transducer was calibrated at zero speed, and the negative pressure corresponds to atmospheric when this part of the hull is lifted out of the water because of the relative motion between the hull and the wave surface. The sharp rise corresponds to the transducer passage of the pressure peak as the hull hits back through the wave surface, and the sharp drop is when the hull part is again exiting the water.

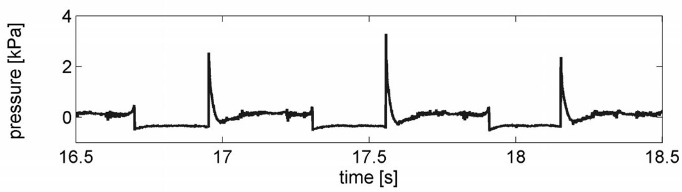

Fig. 18 Pressure measurements in the fore part of the model (simultaneous with Fig. 16).

Fig. 19 shows a typical momentary pressure distribution on a planing craft in head seas, (from Allen&Jones (1978), 1PSI (pound/square-inch)≈7kPa, i.e. 0.7 m static water pillar). The peaked pressure distribution is seen as a high pressure acting over a narrow band of the hull surface. If the hull is moving downwards relative to the wave surface, the high pressure band and the whole pressure distribution will propagate out towards the chines. The pressure is lower in the aftmost one third of the hull because here the chines are wetted. Lower pressure is also seen in the fore part of the hull. Because of the pitch motion and the dynamic interaction with the waves, the incident velocity generally increase towards the stem. As mentioned, the hydrodynamic pressure is proportional to the incident velocity squared. To limit the pressure levels at wave impacts, planing hulls are generally warped with increasing deadrise towards the stem, which according to Fig. 14 decreases the pressure. The hull is subjected to highest pressure in the region where the combined effects of incident velocity and relative geometry is worst.

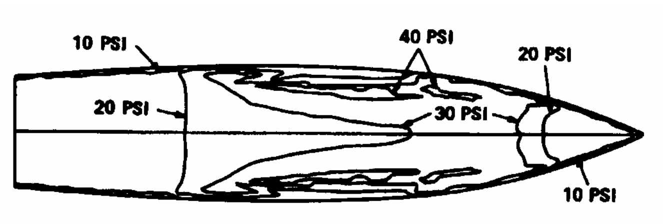

Fig. 19 Example of momentary pressure distribution on a planing craft in head seas, (Allen&Jones 1978).

In displacing speeds, the hydromechanic loads oscillates harmonically with the waves. However, at planing the loading period is related to the period of a pressure pulse propagation across the hull, as illustrated in Fig. 17. This period is generally distinctly shorter than the period of wave encounter, and also shorter that the craft natural period. Hence, the loading of planing craft in waves is to be characterized as transient. The transient loading is clearly illustrated in Fig. 20, which displays acceleration measurements in the fore part of the model for the same sequence as the motion measurements in Fig. 16.

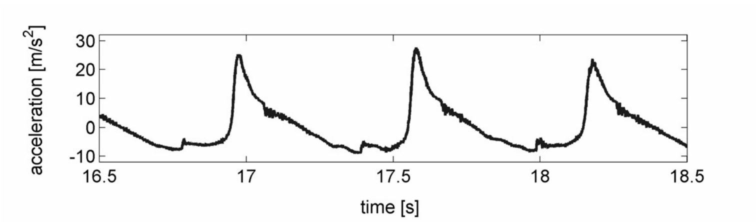

Fig. 20 Vertical acceleration in the fore part of the model (simultaneous with Fig. 16).

The laboratory generated waves in Fig. 16 are smooth and regular. However, because the wind is constantly changing speed and direction, a sea surface is generally characterised by large irregularity and randomness. When analysing waves, this is treated by assuming that the wind speed and direction is relatively constant for periods of a couple of hours. During such period the waves are modelled as a stationary stochastic process. The process can be described in terms of statistical mean values, such as the mean period, and fractional peak mean values as the significant wave height which is the average of the highest one third of the wave heights. The distribution of wave energy on different frequency components can be described in terms of energy spectra. An example of such process is given in Fig. 21. Here an irregular wave system with significant wave height of 0.075 m and mean period 1.056 s, has been created in the wave basin. The loads and responses for a craft in irregular waves are correspondingly irregular and random. Fig. 22 shows measurements of the vertical acceleration for the model when heading the irregular wave system in Fig. 21 at a speed of 4.5 m/s. No wave encounter is like any other, concerning magnitude, shape and propagation of the hydromechanic loads and the resulting responses. Also the responses are treated as stochastic processes and described in terms of statistical measures. The vertical accelerations for planing craft are generally described in terms the average of the highest $1/10^{th}$ or $1/100^{th}$ of the acceleration peak values, which are considered to be representative measures of the crew and passenger experiences and also for describing the loading of the hull structure.

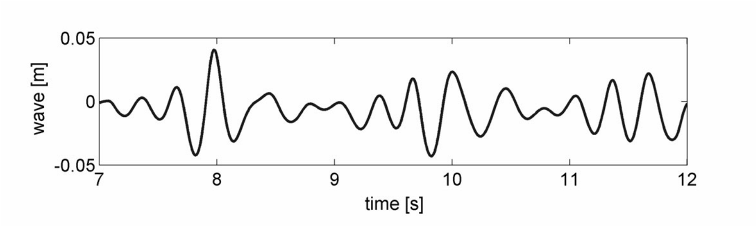

Fig. 21 Irregular wave system generated in the wave basin.

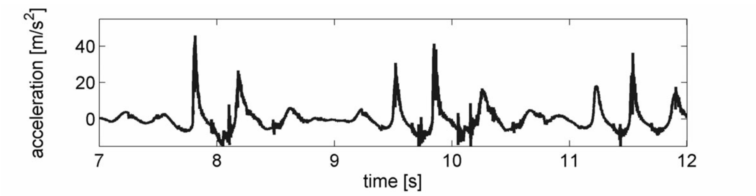

Fig. 22 Vertical acceleration in the fore part of the model at 4.5 m/s in the waves in Fig. 21.

The fundamental function of a craft hull is to keep the water outside, and to define a shape that is appropriate for the craft in question, regarding stability, resistance, seakeeping, manoeuvrability, etc. The fundamental function of the hull structure is to withstand the hydromechanic pressure and preserve the hull shape. A typical hull structure is seen in Fig. 23. This is a Greenland kayak with a primary wood structure covered by skin or fabric. The outside of the bottom is subjected to the hydromechanic pressure. The inside is subjected to atmospheric pressure and additional loads from the kayaker and possible cargo. The hull structure is carrying to the difference between these inner and outer forces, locally where the lateral hydromechanic pressure is carried as membrane stresses in the skin, as well as globally where the hull structure basically works as a beam carrying global bending moments and shear forces.

Fig. 23 Primary wood structure of a Greenland kayak, (Nansen 1891).

These principal functions of the hull structure are independent of craft type or size. Fig. 24 shows a part of the bottom structure of Visby in Fig. 3. This is a high-tech high-speed craft built in carbon fibre sandwich, and as seen there are large similarities with the kayak structure, with longitudinal girders and transverse beams and frames. There are some major differences though. One is in the structure, where the sandwich panels in Visby has a considerable bending and shear stiffness, compared to the skin in the kayak where the pressure loads are completely carried by membrane stresses. The largest differences however are in the applied loads.

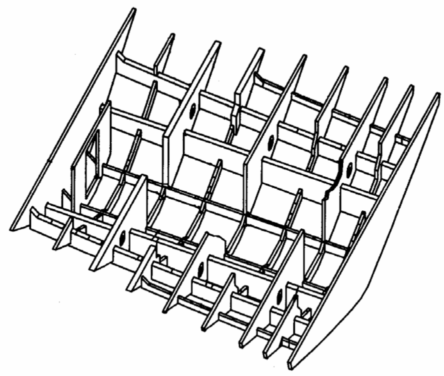

Fig. 24 Primary sandwich beam bottom structure of *Visby* in Fig. 3 (courtesy of Kockums).

The kayak is typically displacing, and the hull structure is subjected to hydromechanic loads with basically the same distributions as the loads at zero speed as illustrated in Fig. 10, which oscillates harmonically with the waves. As described above, the hydrodynamic loads on the high-speed craft are characterized by large gradients, very high magnitudes, and rapid development and propagation across the hull surface. The situation is schematically illustrated in Fig. 25 for a cross section of a hull structure during a wave encounter. BP_i are bottom panels and G_i are longitudinal girders. The distribution of bending moments and shear forces in the structure elements, are governed by the load shapes and the effective boundary constraints. The effective boundary constraints are governed by the structure stiffness, but also by the load shapes. For example the boundaries of a bottom panel can be considered as clamped, if the panel and the adjacent panels are subjected to the same uniform load. However, for a panel subjected to a hydrodynamic impact load, as in Fig. 25, the effective boundary conditions are governed by the stiffness of the surrounding beam structure in combination with the difference in loading between adjacent panels. Further, because of the transient loading and the high acceleration levels for planing craft in waves, structure inertia effects can be considerable. As long as the load frequencies are distinctly lower than the structure eigenfrequencies, the structure response can be considered as quasi-static. This is generally the case for smaller planing craft designed for rough operation, having very stiff structures. However, for larger craft designed for milder operation, the structure eigenfrequencies are lower and might be in the same order of magnitude as the load frequencies. This will result in dynamic effects and hydroelastic interaction between the fluid and the structure. As understood, the hull structure of a planing craft is subjected to a complex and countless variation in load magnitudes and shapes. As for the motion response, the structure responses consequently have to be treated as stochastic processes, and design loads derived by means of statistical extrapolation.

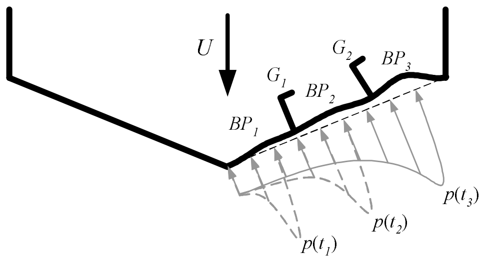

Fig. 25 Schematic illustration of structure response at hull-water impact.

Design Methods

Each craft is built for a particular purpose, e.g. transportation of passengers or cargo, patrol, search and rescue, war, racing, or leisure. The designer is challenged by the task of pursuing optimal performance for the particular purpose, under the constraints of building cost, operational cost, environmental influence, etc. This involves achievement of appropriate performance in calm water as well as in waves, and design of a hull structure that can withstand the highest load the craft will ever be subjected to during its lifetime, but still is as light as possible. Because of the high complexity of planing hydrodynamics, as described in the previous sections, the design of planing craft has basically relied on experience, trial-and-error, and semi-empirical methods, and still do. The prevailing semi-empirical methodology will here be reviewed and discussed.

Two classical methods for analysis of the calm water performance, which are still in frequent use, are presented in Savitsky (1964) and Hadler (1966). Savitsky (1964) synthesised large amounts of experimental data and combined this with theoretical models to formulate a semi-empirical iterative method for prediction of lift, running attitude, resistance and porpoising stability, for planing craft in calm water. Hadler (1966) developed a practical method for prediction of power performance of planing craft, by bringing together research on marine propellers with that on planing.

Based on the model experiments by Fridsma (1969) and (1971), Savitsky&Brown (1976) extended the Savitsky (1964) method to also include formulas for prediction of statistical measures of vertical acceleration and added resistance for planing craft in waves. Hoggard&Jones (1980) made a similar synthesis as Savitsky&Brown (1976), but based on full-scale results. Savitsky has made a great contribution to the understanding and modelling of planing craft. Valuable references in addition to the above mentioned are for example found in Savitsky&Gore (1979) and Savitsky (1985). Savisky&Koelbel (1993) is a comprehensive review of the state of the art in seakeeping analysis and load prediction for planing craft, including descriptions of design features which provide good seakeeping.

A pioneering work on load prediction and structural design was presented by Heller&Jasper (1960). Later contributions have been made, for example by Spencer (1975), Allen&Jones (1978) and Henrickson&Spencer (1982). IMO has specified basic safety requirements for the design and classification of high-speed craft, IMO (2000). From these, rules, requirements and design formulas have been developed by the classification societies, e.g. DNV (1996), ABS (1997), UNITAS (1997) and Lloyd's (1998). The rule formulas are principally based on the above mentioned source works with some improvements for instance by additional input from experience. For example DNV (1996) offers a formula for prediction of the vertical accelerations which very closely resembles the formula by Savitsky&Brown (1976), and a formula for prediction of design loads which is closely related to Allen&Jones (1978).

For larger craft global and local loads are generally treated separately. However, for craft with length $L \leq 50$ metres, minimum strength standards are normally considered to be satisfied from local strength requirements, (DNV 1996). Hence, for smaller craft the focus is mainly on the local loads. The principal idea in the semi-empirical methods for prediction of design loads for planing craft, is to formulate a static uniform design pressure for each of the different structural components (hull panels, beams, girders, etc), which is equivalent to the largest load the component will ever be exposed to during its lifetime, in the sense that it generates equivalent shear forces and bending moments in the component.

The method presented in Allen&Jones (1978) will here be reviewed, as a representative of the state of the art and as a background for a critical discussion. Consider the momentary pressure distribution in Fig. 19, which is taken from Allen&Jones (1978). Fig. 26 is a schematic enlargement of the momentary pressure distribution around the high pressure band on one of the hull sides. Two different areas have been marked, A1 and A2, which could represent the load carrying area of a structural component, e.g. a bottom panel in Fig. 24. As seen, the smaller area A1, is completely covered by the high pressure band. Hence, an equivalent uniform pressure for A1 at this particular instant, must be equal or close to equal to the peak pressure. For the much larger area A2, only a limited part is covered by the high pressure band. The load is instead dominated by the lower pressure levels, and the equivalent uniform pressure for A2 at this particular instant, should be distinctly lower than the peak pressure.

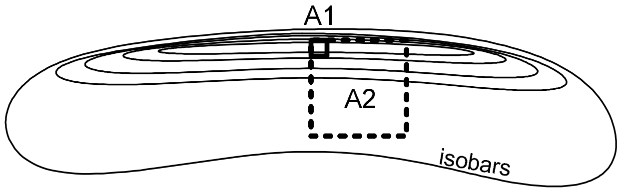

Fig. 26 Schematic enlargement of the momentary pressure distribution in Fig. 19.

As described in the previous section, the hull structure of a planing craft will be subjected to a countless variation in load shapes and magnitudes. However, during extensive semi-empirical simulations and additional full-scale trials on different craft, Allen&Jones noticed that even though there is a large variation in pressure distribution development and magnitudes from wave encounter to wave encounter, and between different conditions in terms of craft speed and sea state, there is a relatively consistent equivalent-pressure/area relationship. By non-dimensionalising the studied areas with a reference area expressed in terms of the craft main particulars, observations on different craft could be formulated into a general pressure-area relationship applicable for arbitrary planing craft. The reference area is assumed to represent the approximate amount of the hull bottom involved in major wave impacts. Hereby the pressure-area relationship could be related to the total vertical load on the hull, which in turn could be expressed in terms of the single variable vertical acceleration at the centre of gravity according to Newton’s second law. Actually, a suggestion to a definition of high-speed craft, is that it is a craft for which vertical acceleration due to wave impact has a significant effect on the loading predictions and structural design, (Fan&Mazonakis 1995). Allen&Jones (1978) refers to the reference area according to Spencer (1975),

$$A_R = \frac{25\Delta_{lt}}{d_{ft}} \tag{1}$$

which is expressed in terms of the craft massdisplacement in long tons Δ_{lt}, and the static draught in feet d_{ft}, and has the dimension of square feet. Transformed to SI-units this yields

$$A_R = 0.7\frac{\Delta}{d} \tag{2}$$

in square metres, where Δ is the craft displacement in metric tonnes and d is the static draught in metres. Allen&Jones (1978) demonstrates that, for typical planing craft geometry, (1) and (2) corresponds to approximately 36% of the idle water plane area, which can be expressed as

$$A_R = 0.3LB \tag{3}$$

where L and B are respectively craft idle waterline length and beam in metres. Koelbel (2001) argues that (3) is more appropriate than the original (1) and (2) because it relates to the craft geometry, and does neither change with displacement nor depend on the static draught. According to Newton's second law, Allen&Jones (1978) expressed the average pressure load on the reference area, P_R, in a condition corresponding to an average vertical acceleration in the centre of gravity a_{CG}, as

$$P_R = \frac{ma_{CG}}{A_R} \tag{4}$$

where m is the craft mass. According to the discussion above, the average pressure load on an infinitesimal area, is equal to the average peak pressure, P_P. Through their observations Allen&Jones (1978) concluded that the average pressure load on the reference area is approximately 14% of the average peak pressure, i.e.

$$P_R = 0.14P_P \tag{5}$$

For areas A_D between infinitesimal and A_R, Allen&Jones (1978) expressed the equivalent uniform pressure as

$$P_D \sim K_D P_P \tag{6}$$

and derived a pressure-area-relation K_D according to Fig. 27. As seen in the figure and in (5)-(6), $K_D \rightarrow 1$ and $P_D \rightarrow P_P$ for small areas, i.e. as $A_D/A_R \rightarrow 0$. Similarly, $K_D \rightarrow 0.14$ and $P_D \rightarrow P_R$ for larger areas, i.e. as $A_D/A_R \rightarrow 1$. Typical design areas, like bottom panel areas and load carrying areas for beams and stiffeners, are generally in the order of magnitude of 1-10% of the reference area. According to Fig. 27 this corresponds to $0.55 > K_D > 0.3$, i.e. equivalent pressures between 55 and 30% of the average peak pressure. (According to Koelbel (1995) K_D can be approximated by $K_D = 0.14(A_D/A_R)^{-0.285}$ for $A_D/A_R > 0.0035$). As described in the previous section, the hull is subjected to highest pressure in the region where the combined effect of relative velocity and relative geometry is worst, see Fig. 19. Allen&Jones (1978) took account of these mechanisms by complementing the equivalent pressure expression in (6), with a longitudinal pressure distribution factor K_L, as displayed in Fig. 28. By combining (4)-(6), Fig. 27 and Fig. 28, Allen&Jones (1978) expressed the equivalent uniform pressure for arbitrary design area as

$$P_D = K_L K_D \frac{ma_{CG}}{0.14A_R} \tag{7}$$

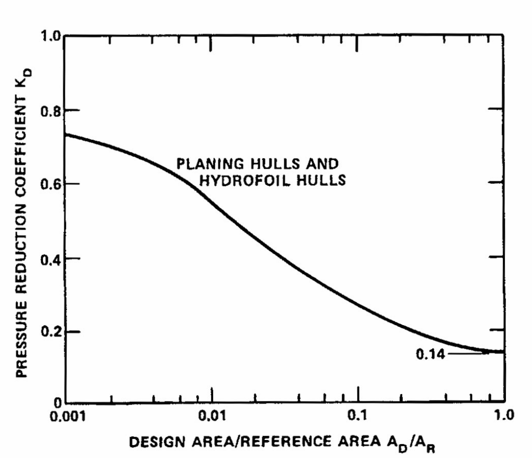

Fig. 27 Pressure reduction coefficient K_D according to Allen&Jones (1978).

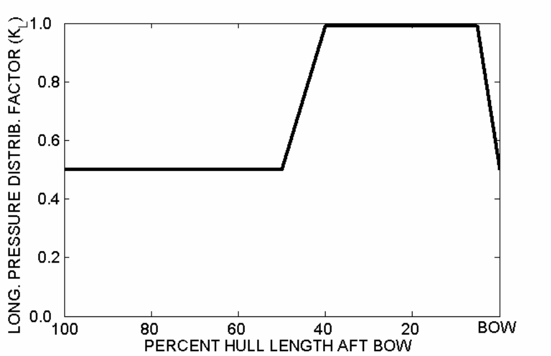

Fig. 28 Longitudinal pressure distribution factor K_L according to Allen&Jones (1978).

With (7) design loads can be derived for arbitrary bottom structure components in arbitrary planing craft, only based on the craft length L, beam B, and mass m, the load carrying area of the structural component in question A_D, and a measure of the vertical acceleration in the centre of gravity, a_{CG}. However, the question is how to determine an appropriate measure of the design acceleration. Allen&Jones (1978) were aware of this and stated that this is a difficult and controversial input. Based on measurements and experience in crew habitability, an acceleration of $5g$ is suggested for small planing craft for rough operation. Koelbel (2001) argues that the hull structure should be designed based on crew habitability and endurance and suggests an acceleration of 6g for small high performance craft, and specifies additional lower levels for larger craft and for craft for milder operation. It is not clear what vertical acceleration measure Allen&Jones (1978) is referring to. Koelbel (2001) however states that the statistical average of the highest $1/10^{th}$ of acceleration peak values should be used in the Allen&Jones (1978) method. DNV (1996) refer to the average of the highest $1/100^{th}$ and, as mentioned, express the design pressure in a similar way as (7). Henrickson&Spencer (1982) completely eliminates the need to predict a design acceleration by deriving a design load expression based on a design sea state defined as H_s=$L/12$, where H_s is the design wave height and L is the craft length.

Allen&Jones (1978) states that the design acceleration optionally could be derived in relation to a design sea state, according to the semi-empirical method presented in Savitsky&Brown (1978). As mentioned, Savitsky&Brown (1978) derived a formula for vertical acceleration prediction by regression analysis on experimental data. The formula yields

$$a_{1/1} = 0.0104\left(\frac{H_s}{B} + 0.084\right)\frac{\tau}{4}\left(\frac{5}{3} - \frac{\beta}{30}\right)\left(\frac{V_{kn}}{\sqrt{L}}\right)^2 \frac{L/B}{C_\Delta} \tag{8}$$

where $a_{1/1}$ [g] is the statistical average of the peak values of the vertical acceleration in the craft centre of gravity in a seastate defined by the significant wave height H_s [ft]. L [ft] is craft length, B [ft] is craft beam, τ [°] is trim, β [°] is the deadrise, V_{kn} [kn] is craft speed, C_Δ [1] is the static beam loading coefficient $C_\Delta = \Delta/(\rho B^3)$, where Δ is craft displacement, and ρ is the water density. The precision of the formula is stated to be ±0.2g, and the range of applicability is given by: $100 \le \Delta_{lt}/(0.01 L_{ft})^3 \le 250$ where Δ_{lt} is the displacement in long tons (=1016 kg); $3 \le L/B \le 5$, $10^o \le \beta \le 30^o$, $3^o \le \tau \le 7^o$, $0.2 \le H_s/B \le 0.7$, $2 \le V_{kn}/\sqrt{L_{ft}} \le 6$. According to Fridsma (1971), the acceleration peak values for planing craft in irregular seas are exponentially distributed. Hereby, the statistical averages on one level, $a_{1/N}$, are related to averages on another level, $a_{1/M}$, as

$$a_{1/N} = a_{1/M}\frac{1 + \ln N}{1 + \ln M} \tag{9}$$

With (9) the acceleration peak average from (8) can be recalculated to the design values, i.e. the fraction averages of the highest $1/10^{th}$ or $1/100^{th}$ peak values. DNV (1996) and the other classification societies, and Hoggard&Jones (1980), offers formulas similar to (8) for prediction of the design vertical acceleration in relation to craft speed and sea state.

According to IMO (2000), high-speed craft should be operated in accordance with a speed/sea state restriction curve, to ensure that the craft is not loaded to hard considering hull structural strength and safety of crew and passengers. The principles of a speed/sea state curve is illustrated in Fig. 29. As seen, the craft is allowed to operate with maximum speed until the significant wave height exceeds H_1, whereas for wave heights above H_2 the craft should stay in harbour. The operational restrictions can be related to the design acceleration for the craft in question, by formulas like (8) or through model tests.

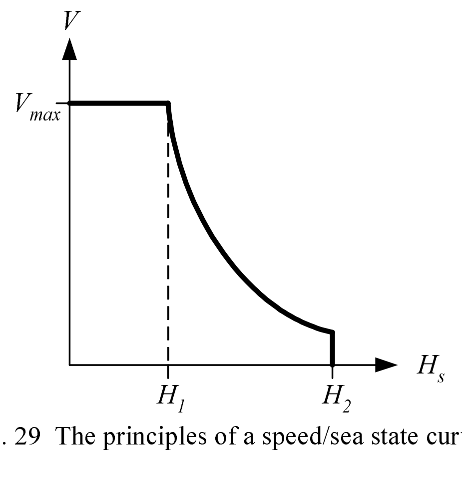

Fig. 29 The principles of a speed/sea state curve.

A critical review of the classification formulas is found in Fan&Miles (1997), where considerable differences in scantlings derived by the different classification rules for the same craft are demonstrated. Fan&Miles (1997) states that these methods are not providing the most appropriate means of pursuing minimum structural weight. Critical reviews of the source works, such as Allen&Jones (1978) and Savitsky&Brown (1976), as well as the classification formulas, are also given in Koelbel (1995), (2000) and (2001). Koelbel describes these methods as amounting to little more than educated guesswork, and points out for example the large variation between the different formulas for prediction of the design acceleration in relation to sea state, and stresses the limitation in the data which they are derived from. Koelbel presents recommendations for removal of some of the inherited uncertainties and improvement of several details in the prevailing design procedure. Another serious limitation in these methods is the uniformity of the design pressure. As described in the previous sections, the hydrodynamic pressure is characterised by large gradients, and it is simply impossible to formulate a uniform design pressure which is equivalent to the real hydrodynamic load, concerning the ratio between bending moments and shear forces in the hull structure. Hereby, direct application of the derived uniform design pressure for example in design of a sandwich panel, will result in different margins to failure in the laminates compared to the core. Furthermore, when a uniform load is applied in structure analysis, the effective boundary conditions, which were described in the previous section, are not taken into account. With uniform design loads it is simply impossible to pursue structures which are optimised in all parts. Hence, the methods available to the planing craft designer, for prediction of seakeeping characteristics as well as design loads, involve large uncertainties and ambiguities. This calls for application of large factors of safety, which in turn limits the possibilities of achieving optimized designs.

Experimental Analysis

The phenomena of planing and hull-water impact has been studied experimentally for nearly a century. Different surveys has addressed different topics and mechanisms, e.g. pressure distribution, wetted surface, dynamic trim and lift, resistance, etc., in calm water planing; motions, transient accelerations, pressure distributions, and structure response, etc., for planing in waves. Experiments have been performed on craft in full-scale as well as model scale. Several experiments has also addressed the 2-dimensional problem of a prismatic body impacting on a water surface, by so called drop tests.

The first analysis of planing and hull-water impact, experimental as well as theoretical, were related to the study of landing of sea planes. Experiments were performed already in the early childhood of aviation, e.g. Baker (1912). Seaplane floats continued to be the main focus through the following decades. Experiments from this period are presented for example in Thompson (1929), a study which involved the development of an early apparatus for measurements of the hydrodynamic pressure. Sottorf (1944) studied the load/resistance relation and the shape of the planing pressure distribution for different planing surfaces. In several studies Smiley examined the pressure distribution on v-shaped planing floats with various trim and deadrise angles, e.g. Smiley (1952). In the 1950ies the focus of research changed towards planing craft. As reviewed in Savitsky (1964), extensive investigations of the complete planing phenomena, theoretical as well as through experiments and synthesization of earlier experiments, were performed at the Davidson Laboratory of the Stevens Institute. In the same laboratory, the most extensive model experiments ever made on planing craft in waves, were performed by Fridsma, on prismatic planing hulls in regular waves, Fridsma (1969), and on the same models heading irregular waves, Fridsma (1971). Impact accelerations and rigid body motions were measured. Allen&Jones (1978) measured pressure, acceleration and structural response on two planing crafts in full-scale, and combined the results with theoretical models to derive the method for prediction of design pressures review in the previous. In Talvia&Wiefelspütt (1991) pressure measurements were performed at offshore trials with a self propelled 1:6 scale model of a 35 metre SAR-vessel. Finch et al (2000) performed sea trials on the 12m RIB in Fig. 1, measuring motions and structure response. Thornhill et al. (2003) examined boundary layer velocities and pressure at various locations on a planing model.

Drop tests, where a test section is dropped from various heights into a water surface, has for example been performed by Chuang (1967) on wedge shaped and flat-bottomed sections. The experiment involved pressure measurements, and for small deadrise angles cushioning was observed, i.e. entrapment of air between the body and the water surfaces, which decreases the magnitude of the hydrodynamic impact. Drop tests with large scale specimens were performed by Hayman et al (1991). This experiment also involved flexible structures and measurement of the elastic structure response. Zhao et al (1996) performed drop tests with the objective to validate theoretical models for the impact pressure distribution. An extensive analysis of 2-dimensional hull-water impact and the impact-planing analogy, is presented in Tveitnes (2001). This study involved drop tests which, in contrast to most other drop tests, were performed with controlled velocity. Drop tests with controlled velocities has also been performed and presented by Battley&Stenius (2003). This study was performed on a flexible sandwich structure, and pressure, impact force and structure responses were measured.

Within the project frame work of the present thesis, full-scale trials have been performed on *90E* in Fig. 4, (Rosén (1998)&(1999), Rosén&Garme (1999), Garme&Rosén (2003)/PaperA). Rigid body motions, and additional vertical transient accelerations where measured along with hydrodynamic pressure in six points, and structure response in the fore part of the hull. The primary purpose of the trials was shake down design evaluation, i.e. to run the craft as hard as the crew could manage and measure the resulting structure response. The trials also enabled general phenomenological studies and collection of reference data, which have been used for evaluation of simulations in Garme&Rosén (2003)/PaperA, Rosén (2005)/PaperB, and Rosén (2004)/PaperD. Also a model experiment has been designed and performed at Canal de Experiencias Hidrodinamicas de El Pardo (CEHIPAR) in Madrid, (Garme&Rosén (2000), Rosén&Garme (2004)/PaperC). The model has here been shown in Fig. 15, and examples of measurement signals have been given in Fig. 16, Fig. 18, Fig. 20 and Fig. 21. The model was a modified version of *90E* in scale 1:10. The model was towed at constant speed in calm water, and head and oblique regular and irregular waves. The hull was equipped with a large number of pressure transducers, and also the craft motions and transient accelerations were measured along with the towing force and the waves. Rosén&Garme have also performed extensive full-scale trials on *Visby* in Fig. 3. The material has so far been used for structure design evaluation. Nothing has yet been published but further analysis and publication is planned for.

This was just a short overview of some of the most referred experimental works, some additional significant studies, and the experiments within the project frame work of the present thesis. Many more experiments have of course been performed and published, some of which can be found in the reference lists of the above mentioned. However, it seems to be a general opinion that there is a lack of experimental data on planing craft. One reason for the lack of data is that experiments generally are very expensive. Further, the full-scale trial situation is very demanding for the measurement equipment as well as for the personnel, and there is always a problem in receiving appropriate sea states and to measure the waves with confidence. Another reason for the lack of data is the described mechanical complexity of the problem of planing craft in waves.

Some of the problems related to experiments on planing craft in waves, are brought forward and discussed in Zseleczky&McKee (1989), for example the noise in rigid body acceleration signals caused by structure vibrations, and the problem of peak identification in transient signals. As described in the previous section the craft responses in irregular waves have to be treated as stochastic processes. Identification of acceleration and structure response peak values is crucial in the statistical analysis. In contrast to harmonic signals, peak identification in transient signals is non-trivial. Peak identification is a significant problem, not only in experimental analysis, but also in the analysis of time-domain simulations. The problem is discussed in Rosén (2004)/PaperD, where the method by Zseleczky&McKee (1989) is applied and evaluated. The method is found relatively stable, still it is concluded that further studies are needed to develop a consistent approach. Rosén (2004)/PaperD further investigates fitting of analytical cumulative distribution functions to sampled data, and evaluates statistical convergence, aspects which are as relevant in the analysis of experiments as for time-domain simulations. In the analysis of the model test in Rosén&Garme (2004)/PaperC the problem of structure noise in acceleration signals is treated by filtration, and the choice of appropriate cut-off levels is discussed. Rosén&Garme (2004)/PaperC also evaluates the experimental resolution, concerning sampling frequencies for the different entities, and pressure transducer areas.

As described, the hydromechanic pressure distributions on planing craft in waves are characterized by large gradients and rapid development and propagation across the hull surface. Measurements with pressure transducers, which is a usual technique, does only give discrete information about the distributions. Hence, to get a somewhat complete picture, either a very large number of transducers, or a method to further process the signals, is needed. The first alternative is expensive and in full-scale applications drilling many holes in the hull is practically impossible. In Rosén (2005)/PaperB a method, for reconstruction of the momentary impact pressure distribution, is presented. The method is based on a set of assumptions and interpolation techniques, by which measurements of the developing pressure distribution in one position of the hull at a particular time instant, can be associated with measurements in other positions at other instants. Hereby, monitoring of the complete pressure distribution in the time-domain, is enabled with a limited number of transducers. The method is evaluated to full-scale data. The approach will be used in the evaluation of the time-domain simulation method which is presented in Garme&Rosén (2003)/PaperA and further developed in Garme (2004a). Other areas of application are for example in experimental derivation of detailed design loads. At full-scale design evaluations the method can be used to improve the traceability, i.e. enable evaluation of the loads along with the responses with more confidence. The method is planned to be used in the further analysis of the *Visby* trials. Time-domain monitoring and analysis of the complete pressure distribution was also the primary purpose of the model experiment presented in Rosén&Garme (2004)/PaperC. The test set-up is evaluated by comparing vertical forces and pitching moments derived from acceleration measurements, with the corresponding forces derived with the pressure distribution reconstruction method. Clear correlation is found. The measurements will be used in further evaluation and development of the non-linear strip method and the direct calculation approach presented in Garme (2004a) and Rosén (2004)/PaperD.

Theoretical Analysis

As mentioned, also the first theoretical works on planing and hull-water impact, were performed in the context of seaplanes. The first approach was to study the 2-dimensional water impact of a wedge, similar to Fig. 17. Pioneering works were made by von Karman (1929) who presented a simple and elegant solution to the impact problem based on conservation of momentum. A few years later Wagner (1932) published a solution based on fluid mechanics, offering a more detailed treatment of the involved physics, for example modelling of the pile-up. The modelling of the problem has been developed through the years, as reviewed for example in Zhao&Faltinsen (1993). Zhao&Faltinsen (1993) presents a non-linear boundary element solution, which later was developed by Zhao et al (1996) to treat arbitrary 2-dimensional sections and flow separation. Methods have been developed for modelling of the hydroelastic interaction between the hydrodynamic pressure and a flexible structure, e.g. Faltinsen (1999). The problem of 2-dimensional section water impact has also been modelled by means of explicit finite element analysis, e.g. Bereznitski (2001), and computational fluid dynamics (CFD), e.g. Tveitnes (2001).

An example of a very advanced CFD-solution, for the complete problem of loads and motion responses for planing craft in waves, is presented in Camponnetto et al (2003). By CFD the 3-dimensional flow can be modelled with very high resolution. However, this is achieved to the cost of computational effort. To reach steady state in regular waves, the approach in Camponnetto et al (2003) required 33 hours CPU time on a high performance PC.

An efficient approach for modelling of planing craft, is to refer to the geometrical similarity between the vertical water impact of a hull section and the projection of the hull running through an imaginary plane. Consider the α-α-plane in Fig. 11 as fixed in space, and the different sections to the right in the figure, as sequential projections as the craft runs through the plane. Hereby, the 3-dimensional fluid mechanical problem can be modelled as a series of 2-dimensional section impact problems, which of course are much easier to solve. Developments of such strip methods for steady planing in calm water, are for example found in Tulin (1957), Zhao et al (1997), and Savander et al (2002).

The *planing-immersing-section analogy*, as it is referred to, has also been generalised to model planing craft in waves. Here the momentary incident velocity U for each section, is modelled as the compound effect of the local water particle velocity in the waves, craft rigid body velocities, craft forward speed, camber and trim. The global forces are achieved by integration of the sectional forces over the hull length. Because of the involved nonlinearities, both in hydrodynamic and hydrostatic terms, the equations of motion have to be solved by iteration in the time domain. The first time-domain strip application for planing craft in waves was presented by Zarnick (1978)&(1979). The Zarnick approach has later been developed by Keuning (1994), Payne (1995), and in the commercially available software POWERSEA, (Akers 2003).

The latest developments of the non-linear strip approach, are found in Garme (2004a). Here the 2-dimensional hydromechanic problem is modelled with a panel method, which treats geometrical complexities like chine flats, and improves the solution in the chines wet phase. The work has also involved development of a technique for correction of the near transom lift, which takes account of the non-2-dimensionallity of the flow in the aft part of the hull. The method has been thoroughly validated, and the simulated motions have shown to compare well with model tests, and full-scale trials as demonstrated in Garme&Rosén

(2003)/PaperA. The development of a method for reconstruction of the wave field around a hull from wave measurements at model tests, (Garme&Hua 1999), has enabled comparison between simulated motions and motions from model tests in the time domain, (Garme 2004a). The model experiment presented in Rosén&Garme (2004)/PaperC, was designed to enable such time-domain evaluation. One of the purposes of the development of the method for pressure distribution reconstruction presented in Rosén (2005)/PaperB, is detailed evaluation of the simulated section forces. Such evaluation is on the list of future work.

As stated by Moan (2003), improved high-speed craft structures require design by direct calculations using first principles of loads as well as strength. Direct calculations of structural responses has reached a high level of development. Advanced finite element methods have been implemented in commercially available software packages, and is nowadays used by rule in the development of larger craft. Advanced methods have also been presented for calculation of global loads for larger high-speed craft in the lower speed regimes, e.g. Wu&Moan (1996). Concerning the local hydromechanic loads there is still need for further research and development. Small craft designers generally do not possess resources to perform direct calculations to any larger extent. However, as stated by Finch et al (2000), meaningful technology improvements could be achieved also in small craft development, by selecting numerical modelling techniques that are at an appropriate level of sophistication relative to the vessel cost. Furthermore, with the development of computers, with rapid improvements in performance and drastic reductions in price, design by direct calculations should become a more realistic alternative also in the small craft industry. This especially comes for high performance craft built in series, like coast guard patrol craft, SAR-vessels, naval craft, etc, but also for leisure and race boats. One example of possible achievements by direct calculations on small high-speed craft is found in Ojeda et al (2004). Here significant stress reductions are demonstrated by laminate scheme tailoring based on global finite element analysis with quasi-static slamming loads according to DNV (1996). The only commercially available software for hydrodynamic analysis on planing craft, i.e. smaller high-speed craft in the higher speed regimes, in waves is POWERSEA (Akers 2003). However, in the present implementation the loads are not available to the user for structure design purposes.

Rosén (2004)/PaperD presents an approach for direct calculations of hydromechanic and structure inertia loads, as well as motions and structure responses for planing craft in waves. Hydrodynamic loads and motion responses are calculated with the Garme (2004a) strip method. Momentary peaked pressure distributions are formulated, from the momentary section forces and the wetted section draughts from the strip calculations, by the introduction of a pressure shape function and scaling technique. This gives a momentary load picture similar to Fig. 19 as input to the structure analysis. The structure analysis is performed with a global finite element model. Boundary conditions are modelled by use of inertia relief, which implies that applied loads are counterbalanced by inertia forces induced by an acceleration field. Hereby, false boundary constraint forces are avoided and structural inertia effects are included. By the peaked load distribution and the accurate modelling of the structure boundaries, the effective boundary conditions in the hull bottom panels are accounted for. The hydrodynamic simulations are performed in arbitrary sea states formulated as stochastic processes. Hereby, also the response time-series outputs are stochastic processes, from which limiting conditions and design values are derived by means of short term statistics. Promising results are demonstrated by applications on *90E* in Fig. 4, where extreme structure responses derived by the presented approach, are compared with responses to equivalent uniform rule based loads, and measured responses from the full-scale trials in Garme&Rosén (2003)/PaperA.

An important aspect of direct calculations and time-domain simulations is applicability, especially in design applications. As mentioned, the strip approach implies simplification of the hydromechanic problem from 3-dimensions to a series of 2-dimensional problems, which decrease the computational effort. The presented direct calculation approach also involves techniques for pre-calculation of hydrodynamic coefficients and structure response, which reduces the computational effort even further. In its present state the approach is a useful tool for further research. It can also be used for evaluation and development of the prevailing semi-empirical design methods. The approach could be developed into a rational design method, which would enable analysis of loads and responses with significantly higher resolution than what is possible with the prevailing semi-empirical methods, and development of more optimized planing craft designs.

Future Work

The presented approach for direct calculations of loads and responses for planing craft in waves will be further evaluated and developed. The simulated section forces will be validated to the presented model tests, by application of the method for pressure distribution reconstruction. The sensitivity in the modelling of the momentary pressure distribution, concerning effective boundary conditions and distributions of bending moments and shear forces in the hull bottom structure, will be further investigated. Preferably should new full-scale trials be performed, to enable detailed evaluation of a complete application of the approach. The approach is primarily developed for application on planing craft, i.e. high-speed craft in the higher speed regimes. The range of applicability concerning larger craft and lower speeds will be evaluated.

Design by direct calculations using first principles of loads and strength, require a different methodology than design by semi-empirical methods. Aspects such as application of permissive in stead of prescriptive criteria, and application of partial safety factors, should be further investigated, to enable full benefit from the direct calculations. Design by direct calculations gives better control and prediction of the loads and responses for a new design. On-board the new design, installation of a system for monitoring of the encountering waves and the resulting loads and responses, gives better control of the operational conditions, which enable safe operation close to the design limits. Further research should be made on on-board monitoring and operational criteria.

The challenge of the future is to design planing craft, which are safer, more weight optimized, cheaper, have better calm and rough water performance, and consume less fuel and have less influence on the environment, than the designs of today. This will be achieved by increased understanding, and improved methods for modelling and monitoring of the involved mechanisms.

References

ABS, *Guide for Building and Classing High Speed Craft*, 1997.

Akers, R., POWERSEA, Ship Motion Associates, 2003.

Allen, R.G., Jones, J.R., *A Simplified Method for Determining Structural Design Limit Pressures on High Performance Marine Vehicles*, AIAA/SNAME Advanced Marine Vehicle Conference, April, 1978.

Baker, G.S., *Some Experiments in Connection with the Design of Floats for Hydro-Aeroplanes*, ARC R&M, No.70, 1912.

Battley, M.A., Stenius, I., *Dynamically Loaded Marine Composite Structures*, 14th International Conference on Composite Materials (ICCM-14), San Diego, USA, 2004.

Bereznitski, A., *Slamming: The Role of Hydroelasticity*, International Shipbuilding Progress, Vol.48, No.4, 2001.

Caponnetto, M., Söding, H., Azcueta, R., *Motion Simulations for Planing Boats in Waves*, Ship Technology Research, Schiffstechnik, Vol.50, No.4, October, 2003.

Chuang, S-L, *Experiments on Slamming of Wedge-Shaped Bodies*, Journal of Ship Research, September, 1967.

DNV, *Design Principles, Design Loads*, Rules for Classification of High Speed and Light Craft, Part 3, Chapter 1, July, 1996.

Faltinsen, O.M., *Water Entry of a Wedge by Hydroelastic Orthotropic Plate Theory*, Journal of Ship Research, Vol.43, pp.180-193, 1999.

Fan, M., Mazonakis, C., *Structural Design Philosophies and Classification Requirements for High Speed Craft*, Proc. 3rd Int. Conf. Fast Sea Transportation (FAST), 1995.

Fan, M., Miles, P., *Structural Design of High Speed Craft – A Comparative Study of Classification Requirements*, Proc. 4th International Conference on Fast Sea Transportation (FAST'97), Australia, 1997.

Finch, G.R., Brown, N.R., Calius, E.P., *Measurements and Predictions of Seakeeping and Structural Response on a 12m Police RIB*, Proc. of the Royal Institute of Naval Architects 5th Int. Conf. on Surveillance, Pilot and Rescue Craft, UK, 2000.

Fridsma, G., *A Systematic Study of the Rough-Water Performance of Planing Boats*, Davidson Laboratory, Stevens Institute, Report 1275, Nov, 1969.

Fridsma, G., *A Systematic Study of the Rough-Water Performance of Planing Boats – Irregular Waves*, Part II, Davidson Laboratory, Stevens Institute, Report SIT-DL-71-1495, March, 1971.

Garme K., Hua J., *A Method to Analyse Seakeeping Model Measurements in Time Domain*, Proc. 9th Int. Offshore and Polar Engineering Conference (ISOPE), France, 1999.

Garme K., Rosén A., *Experimental Pressure Investigation on High-Speed Craft in Waves*, Proc. International Conference on Hydrodynamics of High-Speed Craft, RINA, UK, 2000.

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

Garme, K., *Modelling of Planing Craft in Waves*, PhD Thesis, TRITA-AVE 2004:34, ISBN 91-7283-861-2, Division of Naval Systems, KTH, Stockholm, Sweden, 2004a.

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

Hadler, J.B., *The Prediction of Power Performance on Planing Craft*, Transactions of the Society of Naval Architects and Marine Engineers (SNAME), Vol.74, 1966.

Hayman, B., Haug, T., Valsgård, S., *Response of Fast Craft Hull Structures to Slamming Loads*, Proc. 1st International Conference Fast Sea Transportation (FAST'91), Norway, 1991.

Hayman, B. et al, *Structural Design of High Speed Vessels*, Report of Specialist Committee V.2, Proc. 14th International Ship and Offshore Structures Congress, ISSC, Nagasaki, Japan, 2000.

Hellbratt, S.E., Vallbo, S., *Practical Aspects on Structural Analysis of a Weight Optimized FRP-Sandwich Stealth Corvette*, Sandwich Construction 4, Stockholm, Sweden, 1998.

Heller, S.R., Jasper, N.H., *On the Structural Design of Planing Craft*, RINA, Quarterly Transactions, July, 1960.

Henrickson, W.A., Spencer, J.S., *A Synthesis of Aluminium Crewboat Structural Design*, Marine Technology, Vol.19, No.1, Jan. 1982.

Hoggard, M.M., Jones, J.P., *Examining Pitch, Heave and Accelerations of Planing Craft Operating in a Seaway*, High Speed Surface Craft Symposium, Brighton, England, June, 1980.

IMO, *International Code of Safety for High-Speed Craft*, (2000 HSC Code), 2000.

Karman, Th. von, *The Impact on Seaplane Floats During Landing*, NACA, Technical Note, No.321, 1929.

Keuning, J.A., *The Nonlinear Behaviour of Fast Monohulls in Head Waves*, Thesis, Technische Universiteit Delft, ISBN 90-370-0109-2, 1994.

Koelbel, J.G., *Comments on the Structural Design of High Speed Craft*, Marine Technology, Vol.32, No.2, April, 1995.

Koelbel, J.G., *Structural Design for High Speed Craft – part one*, Professional Boatbuilder, No.67, Oct/Nov, 2000.

Koelbel, J.G., *Structural Design for High Speed Craft – part two*, Professional Boatbuilder, No.68, Dec/Jan, 2001.

Lloyd's Register of Shipping, *Rules and Regulations for the Classification of Special Service Craft*, 1998.

Lönnö, A., *Experiences from Using Carbon Fibre Composites/Sandwich Construction in the Swedish Navy*, Sandwich Construction 4, Stockholm, Sweden, 1998.

Milchert, T., Hua, J., Makinen, K., *Design Slamming Loads for a Semi-Planing Naval Craft Built of Composite Sandwich*, TRITA-FKT, Report 9712, ISSN 1103-470X, KTH, Stockholm, Sweden, 1997.

Moan, T., *Towards Structural Design of High Speed Craft Based on Direct Calculations*, Proc. 7th International Conference on Fast Sea Transportation (FAST'03), Italy, 2003.

Nansen, F., Eskimoliv, 1891.

Ojeda, R. Gangadhara Prusty B., Salas, M., *Finite Element Investigation on the Static Response of a Composite Catamaran Under Slamming Loads*, Ocean Engineering, Vol.31, pp901-929, 2004.

Payne, P.R. *A General Purpose Time Domain Program for High Speed Small Craft*, Royal Institution of Naval Architects, Int. Conf. on Computer Aided Design and Production for Small Craft, (CADAP'95), 1995.

Rosén, A., *Full-Scale Trials Performed on the Combat Craft 90E the Autumn 1997*, Technical report in Swedish, KTH, Stockholm, 1998.

Rosén, A., *Analysis of the Structural Response in the Deck Structure of the Combat Craft 90E*, Technical report in Swedish, KTH, Stockholm, 1999.

Rosén, A., Garme, K., *Slamming Studies on High Speed Planing Craft Through Full-Scale Trials and Simulations*, Proc. 5th International Conference on Fast Sea Transportation (FAST'99), Seattle, USA, 1999.

Rosén, A., Garme, K., *Model Experiment Addressing the Pressure Distribution on Planing Craft in Waves*, International Journal of Small Craft Technology, Vol.146, (RINA Transactions Part B), 2004.

Rosén, A., *Direct Calculations of Loads and Responses for Planing Craft in Waves*, To be submitted for publication, 2004.

Rosén, A., *Impact Pressure Distribution Reconstruction from Discrete Point Measurements*, International Shipbuilding Progress, Vol.52, No.1, 2005.

Savander, B.R., Scorpio, S.M., Taylor, R.K., *Steady Hydrodynamic Analysis of Planing Surfaces*, Journal of Ship Research, Vol.46, No.4, 2002.

Savitsky, D., *Hydrodynamic Design of Planing Hulls*, Marine Technology, Vol.1, No.1, October, 1964.

Savitsky, D., Brown, P.W., *Procedures for Hydrodynamic Evaluation of Planing Hulls in Smooth and Rough Water*, Marine Technology, Vol.13, No.4, Oct, 1976.

Savitsky, D, Gore, J.L., *A Re-Evaluation of the Planing Hull Form*, AIAA/SNAME Advanced Marine Vehicle Conference, 1979.

Savitsky, D., *Planing Craft*, Naval Engineers Journal, February, 1985.

Savitsky, D., Koebel, J. G., *Seakeeping of Hard Chine Planing Hulls*, Technical research bulletin R-42, SNAME, 1993.

Smiley R. F., *Water-Pressure Distribution During Landings of a Prismatic Model Having an Angle of Dead Rise of 22.5° and Beam-Loading Coefficients of 0.48 and 0.97*, NACA TN 2816, 1952.

Sottorf W., *Analysis of Experimental Investigation of the Planing Process on the Surface of Water*, NACA TM 1061, 1944.

Spencer, J.S., *Structural Design of Aluminium Crewboats*, Marine Technology, Vol.12, No.3, July, 1975.

Talvia, J., Wiefelspütt, R., *Offshore Measurements on Large Scale Model and Investigation of Structural Response Aspects of Slamming Loads for High Speed Monohulls*, Proc. 1st International Conference on Fast Sea Transportation (FAST'91), 1991.

Thompson, F. L., *Water-Pressure Distribution on Seaplane Float*, NACA Report, No.290, 1929.

Thornhill E. Oldford, D., Bose, N., *Planing Hull Performance From Model Tests*, International Shipbuilding Progress, Vol.50, No.1&2, 2003.

Tulin, M.P., *The Theory of Slender Surfaces Planing at High Speed*, Schiffstechnik, Vol.4, 1957.

Tveitnes, T., *Application of Added Mass Theory in Planing*, PhD Thesis, University of Glasgow, 2001.

UNITAS (BV, GL, RINA), *Rules for the Construction and Classification of High Speed Craft*, 1997.

Wagner, H., *Über Stoss und Gleitvorgänge an der Oberfläche von Flüssigkeiten*, Zeitschrift für Angewandte Mathematik und Mechanik, Vol.12, 1932.

Wu, M., Moan, T., Linear and Nonlinear Hydroelastic Analysis of High-Speed Vessels, Journal of Ship Research, Vol.40, No.2, 1996.

Zarnick, E.E., *A Nonlinear Mathematical Model of Motions of a Planing Boat in Regular Waves*, David W. Taylor Naval Ship R&D Center, USA, March, 1978.

Zarnick, E.E., *A Nonlinear Mathematical Model of Motions of a Planing Boat in Irregular Waves*, David W. Taylor Naval Ship R&D Center, USA, September, 1979.

Zhao, R., Faltinsen, O., *Water Entry of Two-Dimensional Bodies*, Journal of Fluid Mechanics, Vol.246, pp.593-612, 1993.

Zhao, R., Faltinsen, O., Aarsnes, J., Water Entry of Arbitrary Two-Dimensional Sections with and without Flow Separation, Proc. 21st Symposium on Naval Hydrodynamics, Trondheim, Norway, 1996.

Zhao, R., Faltinsen, O.M., Haslum, H.,A., *A Simplified Nonlinear Analysis of a High-Speed Planing Craft in Calm Water*, Proc. 4th International Conference on Fast Sea Transportation (FAST'97), Australia, 1997.

Zseleczky, J., McKee, G., *Analysis Methods for Evaluating Motions and Accelerations of Planing Boats in Waves*, 22nd American Towing Tank Conference, St. John's, Newfoundland, Aug, 1989.