Numerical modelling of structure responses for high-speed planing craft in waves

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ABSTRACT

The paper presents an approach for time-domain simulation of structure responses, along with hydromechanic and structure inertia loads and motions responses, for high-speed planing craft in waves. Hydromechanic loads and motion responses are calculated with a non-linear time-domain strip method. A pressure shape function is introduced which enables formulation of detailed slamming pressure distributions sequences from the section forces in the strip method simulations. Structure responses are calculated quasi-dynamically by applying the momentary distributed pressure loads on a global finite element representation of the hull structure with use of inertia relief. From the time series output extreme responses are determined by means of short-term statistics. Promising results are demonstrated in applications on a high-speed planing craft, where extreme values of simulated structure responses are compared with responses to uniform design pressures from classification rules and measured responses from full-scale trials. The approach is concluded to be a useful tool for further research which has potential to form the basis for establishment of a computationally efficient simulation-based design methodology. A corresponding experimental modelling approach is presented in a parallel paper.

1. Introduction

High-speed planing craft structures are normally designed based on semi-empirical methods developed in the 1970s, such as Savitsky and Brown (1976) and Allen and Jones (1978) as implemented in scantling codes such as DNVGL (2015) and ISO (2019). The transient, non-uniform, stochastic slamming pressures that are ruling the structure design of such craft, are in these methods modelled as static and uniform. Hereby, standard handbook type formulas can be used to calculate deflection and stresses in the beams and plates constituting the hull structure. This approach is good in that it is well established, straightforward, and requires limited efforts. Since high-speed planing craft are dynamically supported their functionality, fuel consumption and related cost and environmental impact, are strongly related to their mass, and large benefits can be achieved by reducing the structure mass (e.g. Stenius et al., 2011; Garme et al., 2014). Optimizing the structure mass and creating good balance between performance, environmental footprint, and safety, by careful and well informed consideration of the installed power, the crew, passenger and systems conditions and tolerances, and the structure load carrying capacity, however requires a level of detail and accuracy in the load and response predictions that the prevailing semi-empirical design approach does not provide. The prevailing semi-empirical methods further embodies substantial uncertainties as demonstrated and discussed for example in Koelbel (1995), Rosén et al., (2007), Grimsley et al., (2010), McCue (2012), Bowles and Soja (2014), Razola et al., (2014a), Razola et al., (2016), Rosén et al., (2017).

An obvious potential for improvement would be to use direct calculation methods. Due to the high speed, the high level of non-linearity, the transient nature of the loads and responses, and the stochastic nature of the waves and responses, the situation of a high-speed planing craft in waves is however very challenging to model, numerically as well as experimentally.

In its full complexity the slamming problem involves hydroelastic interaction between the fluid and structure. Substantial efforts have been made in recent years on modelling the hydroelastic problem of panels and simple structures impacting on a calm water surface using various techniques, e.g. coupled plate/beam theory and Wagner (1932) theory techniques (e.g. Den Besten and Huijsmans, 2009; Stenius et al., 2011), coupled boundary element methods and finite element methods.
BEM–FEM (Qin and Batra, 2009), fluid dynamics codes coupled with finite element methods, CFD-FEM (Piro and Maki, 2013; Camilleri, 2017), smoothed particle hydrodynamics codes, SPH (e.g. Brizzolara et al., 2008), smoothed particle hydrodynamics coupled with finite element codes, SPH-FEM (e.g. Campbell and Patel, 2010), and explicit arbitrary Lagrange–Euler finite element methods, ALE-FEM (e.g. Stenius et al., 2011; Wang and Soares, 2018). Some attempts have also been made to take hydroelastic effects into consideration in the modelling of high-speed craft in waves (e.g. Volpi et al., 2017; Drimer et al., 2017).

Various modelling techniques have been explored for simulating planing craft rigid body motions in waves, spanning from non-linear strip methods (e.g. Zarnick, 1978, 1979; Keuning, 1994; Payne, 1995; Barry et al., 2002; Garme; Rosén, 2003; Garme, 2005); through more advanced 2D + T methods as for example presented in Sun and Faltinsen (2011) and Dashtimanesh et al. (2019) which can also account for oblique impacts; 3D potential flow boundary element methods that allow for simulation of six degrees of freedom as for example presented in O’Reilly et al. (2018); to very advanced CFD codes e.g. as presented in Caponnetto et al. (2003), Wang et al. (2012), Fu et al. (2014), Aktas et al. (2017), Masumi and Nikseresht (2019), and Judge et al. (2020). The non-linear strip methods have obvious advantages regarding their computational efficiency but rely on substantial simplifications of the fluid mechanic problem. The most advanced CFD methods provide very high level of detail, however to the cost of very large computational efforts.

The structure mechanics problem has been modelled simplistically by applying global design loads from classification rules on finite-element representations of the craft structure, for example as presented in Ojeda et al. (2004). Hou et al. (2019) presents a method for structure dynamic analysis of high-speed planing craft where pointwise acceleration data collected from sea trials are enforced as excitation in a global structure finite element analysis. Rosén (2004) models structure responses for a planing craft in waves by applying hydrodynamic loads derived with a non-linear strip method on a global finite element representation of the hull structure. A similar approach is used in Tuitman and Hoogendoorn (2013) to model the fatigue life of high-speed aluminium craft. Phillips et al. (2004) present an approach where a simplified hydrodynamic solver is used to calculate the momentary slamming pressure distribution which is then applied quasi-dynamically on a finite-element representation of the hull. Extensive efforts with the aim of advancing state-of-the-art regarding high-speed craft structural design has also been made in a project executed by CDI Marine (e.g. Gupta, 2009), where a methodology for route/mission dependent determination of design loads was presented. However, the study focuses on the external loads while details such as the local structure loads due to slamming are not presented in the public reports.

Different options for modelling structure responses for high-speed planing craft in waves could be categorized as schematically illustrated in Fig. 1. The prevailing semi-empirical modelling approach is at the lower end of the accuracy and computational effort scales. Hydroelastic modelling approaches could provide very detailed representation of the involved physics but require computational efforts that are completely unfeasible for design purposes. In between there could be various hybrid approaches that rely on more simplistic descriptions of the fluid-structure coupling and combines more or less advanced hydrodynamic modelling approaches with finite element modelling of the hull structure. Except from the semi-empirical approach, none of these approaches are however very well explored and direct calculation approaches feasible for designing high-speed planing craft with respect to slamming are still waiting to get established.

This paper presents an approach for time-domain simulation of structure responses for high-speed planing craft in waves that has been developed with careful consideration of the trade-off between accuracy and computational effort to make it feasible for design purposes. The approach is demonstrated through application on a particular high-speed planing craft, evaluated in comparison to the prevailing semi-empirical modelling approaches and full-scale trials, and crucial modelling assumptions are highlighted and discussed. A parallel study presented in Begovic et al. (2020) explores the prospect of corresponding experimental modelling of the local structure responses.

2. Modelling Approach

The time-domain simulation approach presented is outlined in Fig. 2. Input is defined in terms of craft geometry, mass, mass moment of inertia, structure layout, scantlings, speed, and sea state properties. Output is given, either as time series or as statistical measures of craft motions, accelerations, hydromechanic loads and structure responses. The different analysis steps are described in the following sub-sections.

2.1. Hydromechanic simulations

Hydromechanic loads and craft motions are simulated using the non-linear time-domain strip method presented and verified in Garme and Rosén (2003) and Garme (2005). Some of the key assumptions in this method are that the fluid is considered ideal and that there is no hydroelastic interaction between the fluid and the structure so that the hull can be modelled as rigid. Following in the tradition of Zarnick...
(1978, 1979), the modelling technique is a generalization of the planing-immersing-section analogy where the 3-dimensional problem of a planing craft in waves is modelled as a series of 2-dimensional impact problems. Planing and slamming forces are accounted for in terms of the rate of change of the added mass. Sectional added masses are calculated based on potential theory using a 2D panel method, which enables modelling of section geometry complexities such as chine flats. 3D effects by the transom are accounted for as described in Garme (2005). A pre-calculation technique is applied where the sectional added masses are pre-calculated and stored for a number of draughts. Then, during the simulation, the momentary added masses and rate of change of added masses are calculated by interpolation between the pre-calculated draughts. This pre-calculation technique significantly reduces the computational effort. The draught resolution is determined based on a convergence check of the vertical section forces. The longitudinal and temporal resolutions are determined based on convergence checks of the craft’s running attitude in calm water and responses in waves.

Simulations are performed at constant speed in regular or irregular waves formulated in terms of a linear wave potential function. Simulations in irregular waves are performed using a Monte-Carlo scheme where the velocity potential is formulated as a superposition of regular components with random phases and with amplitudes derived from an appropriate wave spectrum. Results are acquired in terms of time series of rigid body motions, velocities and accelerations, section draughts, and hydromechanic section forces. In its present state the method is limited to the vertical plane degrees of freedom, i.e. heave and pitch, in long crested head seas.

In the strip calculations the hydromechanic loads are modelled in terms of section forces. To enable detailed structure mechanic simulations, a detailed description of the momentary hydromechanic pressure distribution is however needed. By assuming that the major part of the hydromechanic load is related to planing and slamming, and by introducing a non-dimensional pressure distribution shape function \( C_p(x,y,t) \), the pressure distribution on the hull bottom is formulated by scaling the
shape function with the simulated section forces,
\[ p(x, y, t) = k(x, t)C_p(x, y, t) \]  \hspace{1cm} (1)

where \( t \) is time, \( x \) is the longitudinal space coordinate, \( y \) is the transversal coordinate with \( y = 0 \) in the craft centre line. \( k(x, t) \) is a scale factor that is calculated as
\[ k(x, t) = \frac{f(x, t)}{2 \int_{0}^{b(t)} C_p(x, y, t)dy} \]  \hspace{1cm} (2)

where the section forces, \( f \), and the wet section half beams, \( b \), are determined from the strip calculations. The pressure scaling technique Eq. (1) and (2) allows for application of arbitrary pressure shape functions. Here a simple shape function is formulated based on Wagner (1932) where the pressure distribution on a wedge with a deadrise \( \beta \) impacting a water surface with a constant impact velocity \( V \) is formulated as
\[ p(y, t) = \frac{1}{2} \rho V^2 \frac{\pi}{\tan \beta} \frac{b(t)}{\sqrt{b(t)^2 - y^2}} \forall |y| \leq b(t) \]  \hspace{1cm} (3)

where \( \rho \) is the water density. In its simplest form the Wagner theory implies incorrect modeling of the boundary conditions at \( |y| = b(t) \), resulting in \( p(y, t) \rightarrow \infty \) as \( |y| \rightarrow b(t) \). The maximum pressure in the vicinity of \( |y| = b(t) \) is however according to Wagner (1932) limited by
\[ p_{\text{max}} = \frac{1}{2} \rho V^2 \frac{\pi^2}{4 \tan^2 \beta} \]  \hspace{1cm} (4)

By combining and non-dimensionalizing Eq. (3) and (4) a pressure distribution shape function for deep-V hull sections is here formulated as
\[ C_p(x, y, t) = \min \left\{ \frac{b(x, t)}{\sqrt{b(x, t)^2 - y^2}} \left( \frac{\pi}{4 \tan \beta(x, y)} \right) \mid b(x, t) \leq \frac{B(x)}{2}, \ |y| \in [0, b(x, t)] \right\} \]  \hspace{1cm} (5)

\[ C_p(x, y, t) = 0 \quad \forall \ |y| \geq b(x, t) \]

where \( \beta(x, y) \) is the local deadrise and \( B(x) \) is the local chine beam. In Fig. 3 examples of the shape function Eq. (5) are given for different constant-deadrise sections.

Craft geometries such as chine flats, where the angle is zero, results in the shape function Eq. (5) breaking down. Further, at small dead-rise angles, less than \( \beta(x, y) \approx 3^\circ \), the impact is expected to be cushioned by entrapped air. To handle these two effects the shape function is supplemented with
\[ C_p(x, y, t) = \frac{\pi}{4 \tan 3^\circ} \quad \forall \ b(x, t) \leq \frac{B(x)}{2}, \ |y| \in [0, b(x, t)], \ \beta(x, y) \leq 3^\circ \]  \hspace{1cm} (6)

For a wedge of finite beam \( B \), the pressure distribution Eq. (3) is representative only for \( b(x, t) < B/2 \). When the complete beam is wetted the flow separates at \( y = B/2 \) and the pressure distribution looses its peaked character, becomes more uniform, and adjusts to atmospheric pressure at the separation point. To account for this effect the pressure shape function is supplemented further with
\[ C_p(x, y, t) = 1 \quad \forall \ b(x, t) > \frac{B(x)}{2}, \ |y| \in \left[ 0, \frac{B(x)}{2} \right] \]  \hspace{1cm} (7)

2.2. Structure mechanic simulations

For the structure mechanic simulations, it is assumed that there is no hydroelastic interaction between the fluid and the structure and that the structure responses can be considered as quasi-static and linear. The simulations can hereby be performed by applying the simulated hydrodynamic pressure distribution sequences, Eq. (1), quasi-statically on a finite element representation of the hull structure. By using a global model and defining the boundary conditions by the use of inertia relief, which implies that applied loads are counterbalanced by inertia forces induced by an acceleration field, false constraints are avoided and structural inertia effects are included. The solution can hereby be considered to be quasi-dynamic. The hydromechanic pressure could be applied time-step by time-step in the finite element model. However, the computational effort can be significantly reduced by instead applying a pre-calculation technique similar as in the hydromechanic simulations.
The pre-calculations are performed by formulating a constant pressure load case \( p_c \) for each of the hull bottom elements, i.e. \( e \in [1..N_e] \) where \( N_e \) is the total number of hull bottom elements. For each single-element load case, the FE-equation system is solved and the resulting responses \( \varepsilon_{ae} \) (e.g. stresses, strains or nodal displacements) for a selected number of nodes of interest, i.e. \( n \in [1..N_n] \) where \( N_n \) is the total number of selected nodes, are stored in a data base. This pre-calculation is performed once and for all nodes for the structure in question. Then, during simulation in any speed and sea state, the simulated pressure distribution Eq. (1) is discretized in accordance with the finite element mesh, and the resulting pressure loads \( p_e(t) \) for each of the individual hull bottom elements, are multiplied with the pre-calculated responses to give the structural response time series in the selected nodes as

\[
\varepsilon_n(t) = \frac{\varepsilon_{ae}}{p_c} p_e(t)
\]

(8)

2.3. Post-processing

In the following section the application of the here presented simulation approach will be demonstrated and evaluated for a high-speed planing craft in irregular waves. Results will be presented in terms of peak value statistics where the response peak values are identification by using the method presented in Zslesecky and McKee (1989). Vertical acceleration in the craft centre of gravity will be presented in terms of the statistical average of the highest 1/10th of the peak values,

\[
m_{1/10} = \frac{1}{N_{sim}/10} \sum_{i=1}^{N_{sim}} X_i
\]

(9)

where \( X_i \in [1..N_{sim}] \), is the sorted population of peaks values from a particular simulation and \( N_{sim} \) is the total number of peaks. The structure responses will be presented in terms of statistical extreme values, which are determined by fitting the simulated peak values to analytical distribution functions and extrapolating to appropriate extreme levels. Slamming related vertical acceleration and pressure peak processes for high-speed planing craft can be assumed to follow a two-parameter Weibull distribution,

\[
F(x) = 1 - e^{-(x/a)^c}
\]

(10)

where \( a \) and \( c \) are scale and shape parameters (e.g. Kaplan, 1991; Wang and Moan, 2004; Razola et al., 2016; Begovic et al., 2016). By taking logs of Eq. (10) twice,

\[
\ln(-\ln(1-F)) = c \ln x - c \ln a
\]

(11)

and expressing the simulated short-term cumulative distribution as

\[
F(x) = \frac{i - 1/2}{N_{sim}}
\]

(12)

the coefficients \( a \) and \( c \) can be determined by least square approximation. As shown in Kaplan (1991) and Garme and Rosén (2003), high-speed craft response peak values are distributed differently on different levels, typically approximately Rayleigh distributed (\( c \approx 2 \)) on the lower levels corresponding to linear wave loads and exponentially distributed (\( c \approx 1 \)) on the higher levels corresponding to non-linear slaming loads. Since it is the slamming loads that are of interest the coefficients \( a \) and \( c \) are determined for the tail of the distribution. In this paper the Weibull-fitting is performed on the highest 5\% of the response peak values (if not stated otherwise). An example of such fitting is given in Fig. 4.

According to Ochi (1973) the extreme value \( X_{N,a} \) among \( N \) peak values that is exceeded with a certain small probability \( a \), is related to the corresponding short term peak distribution \( F(x_{N,a}) \) as

\[
F(x_{N,a}) = 1 - \frac{a}{N}
\]

(13)

By combining Eq. (10) and (13) and using the Weibull coefficients derived from the fitting, extreme values can be calculated as

\[
X_{N,a} = a \left( \ln \left( \frac{N}{a} \right) \right)^{1/c}
\]

(14)

The structure responses will be presented in terms of the extreme value with 1\% probability of exceedance during 3 h operation, i.e. Eq. (14) with \( a = 0.01 \) and

\[
N = T \frac{N_{sim}}{T_{sim}}
\]

(15)

where \( T_{sim} \) is the simulation time, \( N_{sim} \) is the number of peak values during \( T_{sim} \) and \( T = 3 \) h = 10,800 s. This is the design measure stipulated for direct calculations by DNVGL (2019).

3. Application & evaluation

The capabilities of the simulation approach are here demonstrated and evaluated by application on the high-speed planing craft Storebro 90E, which for example has been in service for the Swedish Navy, Coast guard, and Marine police. The craft general arrangement and structure layout are shown in Fig. 5. It is about 10 m long, has a displacement of 6.5 tonnes and a maximum speed of +40 knots. The hull has deep-V sections and wide chine flats. The structure is a sandwich construction with carbon-fibre/vinyl-ester laminates and foam core. The internal structure consists of transverse bulkheads at 3.10, 6.53 and 8.80 m measured from the transom (marked with full-lines in the structure layout), a couple of floors and one longitudinal girder on each side (marked with dashed lines). The shaded panel in Fig. 5 is the particular focus in this study. As indicated in Fig. 5 this panel is 2.27 m long and tapered, its beam is 0.68 m and the local deadrise is 25 deg at the aft end and 0.3 m and 35 deg respectively at the forward end. Simulations are performed at 35 knots in various realizations of the ITTC78 wave spectrum. For each condition the simulation time is chosen to give at least 1000 wave encounters.

Fig. 6 gives an example of simulated vertical acceleration in the craft centre of gravity at 35 knots in an irregular sea state with 1 m significant wave height and 3.5 s zero crossing period. As seen, the method captures the transient nature of planing craft in waves, with very large craft motion indicated by periods of ~1 g acceleration when the craft is completely out of the water. Fig. 7 summarize the statistical average of the highest 1/10th of vertical acceleration peak values from 15 simulations at 35 knots in different sea states. As seen the highest accelerations are reached for a wave mean period of 3.5 s. The craft was designed with a design acceleration corresponding to 4.4 g. As seen in Fig. 7, 4.4 g is reached when the craft operates at 35 knots in 0.75 m wave height and 3.5 s wave period.

For the structure modelling the FEA-package ANSYS is used. The sandwich panels are modelled using SHELL91, which is a shell element with equivalent properties defined by multiple layers with sandwich logic where the core is assumed to carry the transverse shear and the laminates are assumed to carry the in-plane forces and bending moments. The intersections between sides and decks, which are built as single skin, are modelled with SHELL93. The beams and girder webs are modelled with SHELL91 and the beam flanges are modelled using LINK8. Boundary conditions are modelled using inertia relief, hereby avoiding false boundary constraint forces and including structural inertia effects. Only structure members that are contributing to the hull strength are modelled. Other components are however included as masses, where the masses of some of the major components such as the engine are well defined in both magnitude and location, whereas masses and locations of other components such as insulation and furniture are approximated, in total summing up to the 6.5 tones craft displacement. Fig. 8 gives examples of simulated deformations at two successive time instants during a wave encounter, together with the corresponding...
pressure distributions and vertical accelerations at CG. The simulated pressure is here applied directly on the FE model. The rest of the presented results are however calculated using the pre-calculation approach according to Eq. (8).

Fig. 9 gives examples of time series of simulated inner laminate strains in the transversal direction at four different transversal positions at \( x = 7.267 \text{ m} \) in the studied panel during one particular wave encounter. For this particular wave encounter it can be seen that the largest strains occur towards the upper panel edge \((y = 0.88 \text{ & } 1.01)\). As seen, the considered positions in the inner laminate are here mainly loaded in tension. The compression seen for \( t < 122.625 \text{ s} \) is a result of the loading and deformation of the panel on the other side of the girder which is loaded before the studied panel.

Fig. 10 gives examples of simulated transversal strains in the inner laminate and core shear along a section at \( x = 7.267 \text{ m} \) at one particular instant during one particular wave encounter. As seen, the largest laminate strains here occur by the keel line \((y = 0)\) whereas the largest core shear occur by the keel line \((y = 0)\) and by the longitudinal girder \((y = 0.49)\).

Detailed validation of the simulation approach will be needed and the parallel study presented in Begovic et al. (2020) explores the prospect of creating a model experiment setup that could enable such validation in the future. The experimental modelling however involves equally large challenges as the numerical modelling and further
development is needed before even attempting to go through with detailed quantitative validation. In the present paper the evaluation of the simulation approach is therefore limited to qualitative comparison between simulations, design limits and full-scale trials.

Fig. 11 summarizes laminate strain and core shear along the transversal section at $x = 7.267$ m in the studied panel, both derived from the simulations (marked with “o”) and by using uniform design pressure according to DNVGL (2015) (marked with “x” and “*”). The simulations have been performed at 35 knots in a sea state with 0.75 m significant wave height and 3.5 s mean zero crossing period, which as mentioned in connection with Fig. 7 should be in parity with the design condition for the craft. The simulations are presented in terms of extreme values (1% probability of being exceeded during 3 h). As observed in Fig. 9, the studied points are subjected to compression (negative) as well as tension (positive) during one impact cycle. Fig. 11 therefore presents the simulations in terms of both compression/negative extreme values (lower curves) and tension/positive extreme values (upper curves). As seen, the largest simulated laminate compressive strain $-0.1\%$ is found at the lower edge of the panel (at the girder at $y = 0.49$ m), whereas the largest simulated laminate tensile strain $0.13\%$ is found close to the mid span of
the panel (at \( y = 0.88 \) m). The largest core shear is found at the panel edges with the maximum absolute value 1.75\% found at the upper panel edge by the chine.

The craft was designed using a maximum allowable strain in the laminates of 0.2\% in compression and 0.42\% in tension and a maximum allowable core shear strain of 1.4\%. Comparing with the results in Fig. 11 it can be noted that the simulated laminate strains are far below the allowable. The low utilization of the laminates can be explained by the laminates thicknesses having been ruled, not by slamming, but to account for rough shore landings. The largest simulated core shear 1.75\% (absolute value) exceeds the allowable core shear 1.4\% that was ruling the design. It should however be noticed that the production related local variations in geometry, core and laminate thicknesses, etc, in the built-up chine areas, are not modelled in detail by the here used shell elements.

Values marked with “\( \square \)" in Fig. 11 are responses resulting from a uniform pressure 39 kPa applied between the girder and the hull side in the finite element representation of the studied structure, whereas values marked with “\( \times \)” corresponds to a uniform pressure 45 kPa applied between the girder and chine flat and 105 kPa applied on the chine flat. These pressures have been derived according to DNVGL (2015) with a design acceleration of 4.4 g corresponding to the statistical average of the highest 1/10th of peak values. As seen the maximum compressive strain and shear strain resulting from these rule loads are in parity with the corresponding maximum simulated strains.

Simulations have also been performed at 35 knots in a significant wave height of 0.50 m and a wave zero crossing mean period of 2.5 s. This corresponds to conditions in which full-scale trials have been performed where laminate strain and core shear were measured in several positions (Garme and Rosen, 2003). Fig. 12 shows extreme values of
lamine strains and core shear along the transversal section \(x = 7.267\) m in the studied panel, both derived from the simulations (marked with “o”) and from the trials (marked with “x”). The comparison between the simulations and the measurements of course involves large uncertainties. To begin with the full-scale experimental set up was not made specifically for simulation evaluation purposes but for evaluating the hull structure design. The positions of the transducers and the thickness of the laminates involves uncertainties. Further, as observed in Rosén (2005), the shear gauge at \(y = 0.62\) is probably malfunctioning. Yet another large uncertainty is in the observation and representation of the waves, where the sea state at the trials was determined through an approach where the vessel was used as a wave buoy at zero speed and

Fig. 10. Example of simulated structure responses at one particular instant in one particular wave encounter. (90° at \(H_s = 1\) m, \(T_z = 3.5\) s, \(V = 35\) kn, \(x = 7.3\) m, \(0.0 \leq y \leq 1.2\) m).

Fig. 11. Values marked with “o” are compression/negative extreme values (lower curves) and tension/positive extreme values (upper curves) derived from simulations at \(V = 35\) kn, \(H_s = 0.75\), \(T_z = 3.5\); “□” are responses resulting from a uniform pressure 39 kPa applied between the girder and the hull side, whereas “x” corresponds to a uniform pressure 45 kPa applied between the girder and chine flat and 105 kPa applied on the chine flat.
then represented in the simulations by a two-parameter ITTC78-spectra through a Monte-Carlo scheme. Still clear qualitative agreement can be observed, with good correlation between simulated and measured laminate strains at $y = 1.01$, and acceptable correlation in the mid span of the panel, whereas the correlation is rather poor at $y = 0.62$. The maximum simulated shear is close to the maximum measured shear, but in a slightly different position.

4. Discussion

The presented results are encouraging, showing that the method predicts vertical accelerations and structure responses in consistency with responses measured in full-scale, design limits for the studied craft and the prevailing semi-empirical design methods. A simulation comprising 1000 wave encounters can be performed in just a few minutes on a standard PC, making the approach feasible for design purposes where very extensive simulations are needed for capturing the craft operational profile and limiting states. The computational efficiency is however relying on a number of rather drastic assumptions and idealizations. Some of these assumptions and idealizations, and some other modelling issues, will here be discussed in more detail.

4.1. Fluid-structure interaction

Key assumptions in the presented modelling approach are that there is no hydroelastic interaction between the fluid and the structure and that the structure responses can be considered quasi-static. Hereby the hydromechanic problem can be solved for a rigid hull and the derived pressure distribution at each time step can be applied quasi-statically in the structure model. This significantly reduces the computational efforts compared to if a fully coupled solution would have been used. Globally these assumptions are certainly valid for small high-performance planning craft with very stiff structure as the one studied in this paper. Regarding local structure such as hull bottom panels, Faltinsen (1999) states that local hydroelastic effects only need to be considered for deadrise angles less than 10 deg, which is much lower than for the craft studied here. According to Berezinitski (2001) hydroelastic effects can be considered limited if the loading frequency is smaller than half the dry natural frequency of the considered structure component. Typical dry panel frequencies for 90E are in the order of 200 Hz (determined by modal analysis in ANSYS), whereas typical load frequencies at 35 knots in a sea state with a significant wave height of 0.75 m and a wave mean period of 3.5 s are around 30 Hz (determined from the simulations). The non-hydroelastic assumption is hence confirmed also in relation to Berezinitski’s criteria. Stenius et al., 2011 formulates a similar measure for characterizing hydroelastic effects, taking into account inertia related as well as kinematic hydroelastic effects. Also the Stenius’ criterion confirms the non-hydroelastic assumption for the craft studied here. For application of the presented simulation approach on other craft, particularly on larger craft with more flexible panels, the non-hydroelastic assumption must be evaluated further, for example by following the procedures in Stenius et al., 2011.

The non-linear strip method has been thoroughly evaluated and simulated motions and loads have shown to compare well with model experiments as well as full-scale trials (Garme and Rosén, 2003; Garme, 2005). It has also been successfully applied in design projects and various research studies (e.g. Garme et al., 2014; Razola et al., 2014a; Razola et al., 2016). A key feature in the approach presented here is the detailed modelling of the slamming pressure distribution by introducing a pressure shape function scaled with the hydrodynamic section forces from the strip calculations. In this paper a simple pressure shape function derived based on Wagner (1932) theory with some adjustments to account for chine flats and chine separation, has been used. Detailed evaluation of this pressure modelling technique and alternative pressure shape functions is performed in Razola et al. (2014b). As mentioned, a first step towards quantitative validation of the simulated structure responses is taken in the parallel study presented in Begovic et al. (2020).

4.2. Post-processing

The simulation results have in this paper been presented in terms of short-term peak value statistics. In the present study the peak
identification method presented in Zseleczky and McKee (1989) has been used. This method relies on a user-specified threshold, which defines the magnitude required for a peak to be considered as a peak. In the present paper, a threshold of 150% of the considered signals’ standard deviation has been used. Fig. 13 is a brief investigation of the sensitivity in the choice of threshold, where the threshold is formulated in terms of multiples of the signal standard deviation. As seen, the different thresholds result in significantly different numbers of identified peaks, and the number of identified peaks also varies between the different positions in the panel for the same threshold. However, as seen, the corresponding extreme values are remarkably stable. As described, extreme values are calculated by fitting a Weibull distribution to the highest 5% of the simulated peak distributions. Fig. 14 is a brief investigation of the sensitivity in this fitting, where fitting of the highest 4, 6, and 8% of peak values are compared. As seen the Weibull parameters and the corresponding extreme values are rather stable. Another issue is how long simulation times and how many peak values that are required for convergence of the statistical measures. According to Savitsky and Koelbel (1993), a test record on planing craft should contain at least 75 wave encounters in order to obtain statistically acceptable samples whereas according to ITTC Recommended Procedures and Guidelines (2017) the minimum number of encounters to be tested is 50, testing more than 100 is standard and more than 200 is considered excellent practice. Fig. 15 is a brief investigation of the convergence of simulated acceleration quantile averages. The results indicate that 75 wave encounters are all too little and that rather 250 peak values are required for convergence of the average of the highest 1/10th of peak values and even more for the average of the highest 1/100th. Similarly, the extreme values of the structural responses are found to require up to 500 peak values for convergence. As mentioned, simulation times resulting in at least 1000 wave encounters are used in the present study. Various issues related to peak value identification and statistical analysis of high-speed planing craft responses are thoroughly explored in Razola et al. (2016).

5. Conclusions

An approach for time-domain simulation of structure responses for high-speed planing craft in waves is presented and demonstrated on a particular high-speed planing craft. The following conclusions are made:

- The presented approach captures the characteristic phenomena for planing craft in waves such as the nonlinear hydrodynamics, the large craft motions and transient accelerations, the peaked propagating pressure distribution and the related structure responses including inertia effects. Hereby the approach enables analysis of loads and responses related to slamming with significantly higher resolution than what is possible with the prevailing semi-empirical design methods.
- The relatively simplistic non-linear strip method, and the pre-calculation of sectional added masses and structure responses, makes the approach highly computationally efficient. If effectively implemented, real time simulations should be achievable. This enables long simulations times feasible for statistical analysis in design applications. The approaches for short term statistical analysis applied in the paper are found to be consistent.
- Calculated extreme response values show consistent agreement with measured responses from full-scale trials and responses to uniform design pressures from classification rules. The here performed comparison is however limited and involves uncertainties and more thorough validation is needed.
- The presented approach is an important step towards establishing a direct calculation approach for structure design of high-speed planing craft. The approach has also been found useful in research, for example in evaluation and further development of the prevailing semi-empirical design methods.

Future work should for example consider investigation of appropriate levels of detail in the finite element structure calculations, evaluation of the range of applicability of made assumptions, and systematic experimental validation. High-speed planing craft in waves is however equally challenging to model experimentally as numerically. A first step towards enabling more thorough experimental validation of the here presented numerical approach is taken in the parallel study presented in Begovic et al. (2020) that explores a novel setup for experimental
modelling of local slamming related structure responses for a high-speed planing craft in waves.

Declaration of competing interest

The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.

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References