ABSTRACT

This paper presents the recent developments at ABS to revise the requirements for slamming impact loads on high speed naval craft. According to the ABS Guide for Building and Classing High Speed Naval Craft (HSNC 2007), slamming impact load is one of the most critical factors for the scantling design of hull structures. As modern naval craft design requires higher ship speed with increasing ship size, ABS is continuously putting efforts to refine prescriptive rules, analysis procedures and numerical tools. Recently, ABS investigated large high speed naval craft designs and proposed new requirements for slamming impact loads.

This paper is mainly focused on the refinement of prescriptive rules for bottom slamming design pressure on mono-hulls and wetdeck slamming design pressure on the cross-structure of multi-hulls. Extensive numerical simulations were carried out using the nonlinear time domain seakeeping program LAMP. Vertical acceleration, impact forces and slamming pressures were calculated and compared with available model test data and design practices.

This paper also presents ABS’s on-going efforts for the development and validation of computational fluid dynamics (CFD) code as an alternative numerical tool to analyze the extremely violent nonlinear free-surface flows such as sloshing, slamming and green water impact problem. Some of the most recent CFD simulation results are presented including the wetdeck slamming of a catamaran using the level-set Finite-Analytic Navier-Stokes (FANS) code.

INTRODUCTION

As the new generation of high speed naval craft becomes larger and faster, slamming impact loads on these vessels are a critical design concern. Currently available rules for slamming design pressure were mostly developed for small planing hulls based on experimental and theoretical work undertaken in the 60’s and 70’s [2, 3].

To revise the current design criteria specified in the ABS Guide for Building and Classing High Speed Naval Craft (HSNC) [1], ABS recently carried out extensive numerical analysis for the new designs of high speed naval craft. As testing vessels, high speed naval craft of large semi-planing mono-hull, small planing mono-hull, displacement mono-hull, and wave-piercing catamaran are considered, and the state-of-the-art nonlinear seakeeping program LAMP is used for numerical simulation.

1. Design Conditions

1.1. Design sea states and ship speeds

For the determination of design loads on high speed naval craft, design sea states are to be defined by significant wave height, wave modal period and ship speed. Table 1 shows an example of the sea states and significant wave heights typically being used for the design of naval craft operating in the North Atlantic. Vm denotes the maximum design speed.

<table>
<thead>
<tr>
<th>Table 1: Sea states in North Atlantic</th>
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</thead>
<tbody>
<tr>
<td>Sea State</td>
</tr>
<tr>
<td>-----------</td>
</tr>
<tr>
<td>2</td>
</tr>
<tr>
<td>3</td>
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<tr>
<td>4</td>
</tr>
<tr>
<td>5</td>
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<tr>
<td>6</td>
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<tr>
<td>7</td>
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<tr>
<td>8</td>
</tr>
</tbody>
</table>
Table 2 shows the design sea states and ship speeds defined in ABS HSNC guides (3-2-2/Table 1). Note that the significant wave height of survival condition is not to be taken less than L/12. \(v_m\) denotes the maximum speed for the craft in the design condition, and the ship speed of 10 knots in survival condition is to be verified by the Naval Administration.

<table>
<thead>
<tr>
<th>Table 2: Design sea states and ship speed</th>
</tr>
</thead>
<tbody>
<tr>
<td>Operational Condition</td>
</tr>
<tr>
<td>(h_{1/3})</td>
</tr>
<tr>
<td>Naval Craft</td>
</tr>
<tr>
<td>Coastal Naval Craft</td>
</tr>
<tr>
<td>Riverine Naval Craft</td>
</tr>
</tbody>
</table>

1.2. Tested vessels
This study is mainly focused on the slamming impact loads on high speed naval craft. In this study, two semi-planing mono-hulls and one displacement mono-hull and one planing mono-hull are considered for bottom slamming design pressure. Also a wave-piercing high speed catamaran is considered for wetdeck slamming design pressure.

\[
n_{cg} = N_2 \left[ \frac{12h_{1/3}}{B_w} + 1.0 \right] \tau \left( 50 - \beta_{cg} \right) \frac{V^2 B_w^2}{\Delta} \tag{1}
\]

where
- \(n_{cg}\) average of the 1/100 highest vertical acceleration at LCG in g’s
- \(N_2 = 0.0078\)
- \(h_{1/3}\) 1/3 highest significant wave height, in m, as given in Table 2
- \(B_w\) maximum waterline beam, in m
- \(\beta_{cg}\) deadrise angle at LCG, in degrees, not to be taken less than 10° nor more than 30°
- \(V\) design speed in operation and survival conditions, in knots, as given in Table 2
- \(\Delta\) displacement in kg
- \(\tau\) running trim angle at \(V\), in degrees

Note that the vertical acceleration in equation (1) is a function of running trim angle of the hull. At the early stages of design, however, the running trim angle is not known a priori. Furthermore, the running trim angle of the vessel in a seaway varies in time and is not much relevant to the vertical acceleration of the vessel operating in severe sea states.

Based on the numerical study presented in Section 3, the empirical formula for vertical acceleration is revised as follows.

\[
n_{cg} = 35N_2C_V \left[ \frac{12h_{1/3}}{B_w} + 1.0 \right] \tau \left( 50 - \beta_{cg} \right) \frac{V^2 B_w^2}{\Delta} \tag{2}
\]

where
- \(n_{cg}\) average of the 1/100 highest vertical acceleration at LCG in g’s
- \(C_V = 0.657\sqrt{L - 2.5}\), not to be taken less than 1.5 nor more than 5

The vertical acceleration at any section of the hull along the ship length may be expressed as below.

\[
n_{xx} = n_{cg} K_V \tag{3}
\]

where
- \(n_{xx}\) average of the 1/100 highest vertical acceleration at any section, in g’s
- \(K_V\) vertical acceleration distribution factor

A revision of the vertical acceleration distribution factor \(K_V\) is proposed, as given in Fig. 2. The factor has been significantly increased for survival condition, based on the numerical simulation results presented in Section 3.
2.2. Bottom Slamming Pressure

In the early 1960s, for small planing craft, Heller and Jasper suggested a slamming pressure formula in a very concise form [3]. Based on the numerical analysis in this study, the current slamming design pressure given in HSNC has been validated first and then a revised slamming design pressure is proposed as below.

For semi-planing hull, the slamming design pressure can be expressed as:

$$p_{bx} = \frac{N L_{w}}{B_{w}}\left[1 + n_{w}\right]\left[\frac{70 - \beta_{bs}}{70 - \beta_{vg}}\right]F_{L}$$  \hspace{1cm} (4)

where

- \(p_{bx}\): bottom design pressure at any section, in kN/m²
- \(N\): 0.01
- \(\Delta\): displacement in kg
- \(L_{w}\): craft length on the waterline, in m
- \(B_{w}\): maximum waterline beam, in m
- \(\beta_{bs}\): deadrise angle at any section, in degrees, not to be taken less than 10° nor more than 30°
- \(F_{L}\): longitudinal pressure distribution factor

In the proposed formula, longitudinal pressure distribution factor \(F_{L}\) is introduced to consider 3D flow effects near the bow and stern area, as shown in Fig. 3.

For semi-planing hull, the slamming design pressure may be simplified as:

$$p_{bx} = \frac{N L_{w}}{B_{w}}\left[1 + n_{vg}\right]F_{V}$$  \hspace{1cm} (5)

where

- \(F_{V}\): vertical acceleration distribution factor, defined in HSNC 3-2-2/Figure 8

3. BOTTOM SLAMMING SIMULATION FOR MONO-HULLS

3.1. LAMP method

The Large Motion Amplitude Program (LAMP) is a 3D panel method developed by SAIC for the time-domain simulation of nonlinear ship motions and wave loads in extreme wave conditions. LAMP development started as a DARPA project in 1988 and has been supported by the US Navy, USCG, SAIC/MIT and ABS [12].

3.2 MONO-1 hull

MONO-1 is a large high speed semi-planing naval craft, recently built to ABS Class in accordance with the ABS Guides for Building and Classing High Speed Naval Craft (HSNC). The overall length of the craft is more than 110m and loading conditions are as given in Table 3.

<table>
<thead>
<tr>
<th>Loading Condition</th>
<th>Displacement (tons)</th>
<th>Speed (knots)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Full Load Departure</td>
<td>3100</td>
<td>38</td>
</tr>
<tr>
<td>Full Load Arrival</td>
<td>2900</td>
<td>41</td>
</tr>
<tr>
<td>Full Load Minimum Operation</td>
<td>2800</td>
<td>42</td>
</tr>
<tr>
<td>Full Load Survival</td>
<td>3100</td>
<td>10</td>
</tr>
</tbody>
</table>

Fig. 4 shows the LAMP nonlinear geometry model of MONO-1 that includes the hull geometry above the mean waterline. Nonlinear hydrostatic restoring and Froude-Krylov forces acting on the instantaneous wetted hull surface are calculated over the nonlinear geometry model.

Fig. 5 shows the LAMP linear hydro panel model of MONO-1. A large number of quadrilateral or triangular panels are distributed on the hull surface below the mean waterline as well as on the truncated free surface. This
model is used to calculate radiation and diffraction forces due to linear wave-body interaction.

Once the geometry models were prepared, nonlinear time-domain seakeeping analysis was carried out using LAMP. Fig. 6 shows the time history of vertical acceleration calculated at x=90m from AP.

As given in Eq. (2), the slamming design pressure is expressed in terms of the average of 1/100 highest vertical acceleration in a design sea state under consideration. Once the time series of vertical acceleration are obtained from seakeeping analysis in time domain, peak analysis is to be performed to estimate the 1/100 highest vertical acceleration.

Fig 7 shows the 1/100 highest vertical acceleration of MONO-1 at operational condition: full load departure with ship speed V=38knots and significant wave height Hs=4m. Compared with model test measurements (green dot) and LAMP simulation results (pink square), the proposed vertical acceleration (red line with triangle) has been significantly improved.

Fig. 8 shows the 1/100 highest vertical acceleration of MONO-1 at survival condition: full load departure with V=10knots and Hs=9m. The proposed vertical acceleration shows much better agreement with LAMP results.

Fig. 9 shows the bottom slamming design pressure distribution along the ship length of MONO-1 at operational condition. The main difference between the current and proposed design pressure is coming from vertical acceleration, which is known as the most critical design factor for high speed craft. The figure presents the actual design pressures that were actually used for the scantling check of bottom plates. The actual design pressures were determined by the vertical acceleration at LCG directly measured from model tests. The proposed slamming pressure in this study is very close to the actual design pressure of MONO-1.
pressure has been significantly increased by the proposed formula in better agreement with actual design pressure.

Numerical calculation of slamming impact pressure using the 2D boundary element method can be a very challenging and time-consuming task. Instead, direct calculation of vertical impact forces on each sectional cuts using wedge approximation is considered in this study, which is a more efficient and reliable method. As a design practice, the slamming pressure may be estimated from sectional impact force as follows:

\[ p = \frac{W}{B/6} \]  \hspace{1cm} (6)

where

- \( p \) slamming pressure, in N/m\(^2\)
- \( w \) sectional impact force, in N/m
- \( B \) maximum beam, in m

Fig. 11 shows the time history of sectional impact forces simulated by LAMP/LMPOUND, calculated at three representative sections \( x = 20 \)m, 50m and 90m from AP at operational condition. The current and proposed rule values are also compared in the figure.

Note that the relative ship motion and acceleration is expected high at \( x = 90 \)m from AP, but the impact force was actually low because of the high deadrise angle of that section.

In general, the proposed impact forces have been significantly increased because of the increased vertical acceleration.

Peak analysis is required for the representation of statistical properties of the response in time series, either calculated by numerical simulation or measured by model tests. In this study, only a highest peak is counted between zero-crossings. No intermediate peaks are counted as independent events. To eliminate the noise in time series, only those peaks exceeding a threshold value are counted. In this study, 10% of the average of the 1/100 highest peak is used as threshold value. To avoid transient response, the first 1/5 of time series is ignored in the peak counting.

Those counted peaks are to be represented by relevant probability distribution functions. In this study, two-parameter Weibull distribution is considered as follows:

\[ Q(x > x) = \exp\left(-\frac{x^b}{\theta^b}\right) \]  \hspace{1cm} (7)

Fig. 12 shows the peak analysis results of vertical impact forces using Weibull distribution. Note that the response at a exceedance probability level of 0.001 is a typical target value that corresponds to the most probable short-term extreme value out of 1000 encountered wave cycles.
3.3. MONO-2 hull

MONO-2 is a medium-size high speed semi-planing naval craft recently built to ABS Class in accordance with the ABS Guides for Building and Classing High Speed Naval Craft (HSNC). The overall length of the craft is about 60m and loading conditions are as given in Table 4.

<table>
<thead>
<tr>
<th>Loading Condition</th>
<th>Displacement (tons)</th>
<th>Speed (knots)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Full Load Departure</td>
<td>750</td>
<td>35</td>
</tr>
<tr>
<td>Full Load Survival</td>
<td>750</td>
<td>10</td>
</tr>
</tbody>
</table>

Fig. 12: Peak analysis of vertical impact force at operational condition

Fig. 13 shows the LAMP geometry model of MONO-2 that includes the hull geometry above the mean waterline.

Fig. 13: LAMP geometry model

Fig. 14 shows the vertical acceleration of MONO-2 in operational condition. Compared with LAMP results, the proposed vertical acceleration has been increased in the stern area. Fig. 15 shows the vertical acceleration in survival condition. It can be seen that the proposed vertical acceleration has been significantly increased.

Fig. 14: Vertical acceleration at operational condition: full load at $V=35$ knots and $H_s=4$m

Fig. 15: Vertical acceleration at survival condition: full load at $V=10$ knots and $H_s=6$m

Fig. 16 shows the slamming design pressure of MONO-2 at operational condition. Note that the proposed design pressure has been significantly increased in the bow and stern areas.

Fig. 16: Slaming design pressure of MONO-2 at operational condition.

Table 4: Loading conditions of MONO-2
3.4. MONO-3 hull

MONO-3 is a displacement-type naval craft recently built to ABS Class. The overall length of the craft is about 100m and loading conditions are as given in Table 5.

Table 5: Loading conditions of MONO-3

<table>
<thead>
<tr>
<th>Loading Condition</th>
<th>Displacement (tons)</th>
<th>Speed (knots)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Full Load Departure</td>
<td>2200</td>
<td>25</td>
</tr>
<tr>
<td>Full Load Survival</td>
<td>2200</td>
<td>10</td>
</tr>
</tbody>
</table>

Fig. 17 shows the LAMP geometry model of MONO-3 that includes the hull geometry above the mean waterline.

Fig. 17: LAMP geometry model

Fig. 18 shows the vertical acceleration of MONO-3 in operational condition. Fig. 19 shows the vertical acceleration in survival condition. Note that MONO-3 is a displacement vessel and has smaller vertical acceleration than the semi-planing vessels of MONO-1 and MONO-2. The proposed vertical acceleration shows good agreement with LAMP simulation results.

Fig. 18: Vertical acceleration at operational condition

Fig. 19: Vertical acceleration at survival condition

3.5. MONO-4 hull

MONO-4 is a typical high speed planing naval craft. The overall length of the craft is about 25m and loading conditions are as given in Table 6.

Table 6: Loading conditions of MONO-4

<table>
<thead>
<tr>
<th>Loading Condition</th>
<th>Displacement (tons)</th>
<th>Speed (knots)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Full Load Departure</td>
<td>75</td>
<td>30</td>
</tr>
<tr>
<td>Full Load Survival</td>
<td>75</td>
<td>10</td>
</tr>
</tbody>
</table>
Fig. 22 shows the vertical acceleration of MONO-4 in operational condition. Note that MONO-4 is a small planing hull and higher vertical acceleration is expected, but the proposed vertical acceleration is slightly reduced compared to the current vertical acceleration which is believed to be overpredicted. Fig. 23 shows the vertical acceleration in survival condition. The proposed vertical acceleration has been significantly increased.

Fig. 24 shows the slamming design pressure on planing hull of MONO-4 at operational condition, calculated by eq. (4). Note that the proposed design pressure has been reduced due to the reduction of vertical acceleration in operational condition, as given in Fig.22.

Fig. 25 shows the slamming design pressure on planing hull of MONO-4 at survival condition.

4. WETDECK SLAMMING DESIGN PRESSURE FOR MULTI-HULLS

The underside of wet deck or cross structure of multi-hulls is likely subject to significant slamming impact loads in severe sea states. In this study, the current wetdeck slamming design pressure given in HSNC is investigated for a latest high speed catamaran.

Numerical simulation of wetdeck slamming impact loads can be a very challenging and time-consuming task. Recently, a very efficient wetdeck slamming module has been implemented into the LAMP system based on a 2D longitudinal cut model [10].

LAMP wetdeck module is used for the prediction of wetdeck slamming pressure of a catamaran operating in design operational condition and survival condition. According to the LAMP simulation results, a revised wetdeck slamming design pressure is proposed as follows:

\[ p_{\text{wd}} = 30 F_I V V_I (1 - 0.5 h_u / h_{1/3}) \]  

(8)

where

- \( F_I \) wet deck pressure distribution factor as given in Fig 26
- \( V \) design speed in operation and survival conditions, in knots, as given in Table 2
- \( V_I \) relative impact velocity as given below

\[ V_I = \frac{4 h_{1/3}}{\sqrt{L}} + 1 \text{ (m / s)} \]

- \( h_u \) vertical distance, in m, from waterline to underside of wetdeck
- \( h_{1/3} \) significant wave height, in m, as given in Table 2
In the comparison to the slamming model test data for CAT-1, it was found that additional pitch damping is needed for an accurate LAMP simulation of the pitch motion in the most severe slamming conditions. In order to match with measured pitch motion, the supplementary pitch damping model is considered in LAMP simulations as follows:

$$ME_2 = -v_5 \cdot KL_5 - v_5^2 \cdot \frac{v_5}{v_5^*} \cdot KQ_5$$

where

- $v_5$: pitch velocity
- $KL_5$: linear pitch moment coefficient
- $KQ_5$: quadratic pitch moment coefficient

Using this added pitch damping model with coefficient tuned based on the pitch response near resonance, a very good agreement between the predicted and measured ship motions was achieved.

Fig. 28 shows the sensor locations of pressure patches used in the slamming model test of CAT-1 performed by DTMB [11].

Fig. 29 shows a typical wetdeck slamming impact pressure at four sensor locations of CAT-1 in a model test condition with regular waves. The wetdeck impact pressure consists of the first sharp peak with short duration due to slamming impact followed by a second round peak with longer duration due to nonlinear hydrostatic and Froude-Krylov forces.

Fig. 30 shows the time history of wetdeck slamming impact pressure of CAT-1 calculated at a pressure location $P_1$, as shown in Fig. 28. The vessel is operating in operational condition with ship speed $V=40$ knots and significant wave height $H_s=4m$. A significant increase of wetdeck slamming pressure is proposed based on LAMP simulation results.

Fig. 31 shows the peak analysis results of wetdeck slamming pressure at $P_1$ using Weibul distribution, comparing with current and proposed wetdeck slamming design pressure. Note that the exceedance probability level of 0.001 corresponds to the typical most probable short-term extreme value.
Fig. 32 shows the time history of wetdeck slamming impact pressure at the pressure location $P_1$ in survival condition with ship speed $V=10$ knots and significant wave height $H_s=9$ m. Note that, in this study of the CAT-1 vessel, the wetdeck slamming pressure of survival condition is found to be more severe than that of operational condition, which is likely due to the higher chance of underdeck wetness in the severe sea state of $H_s=9$ m.

Fig. 33 shows the peak analysis results of wetdeck slamming pressure at $P_1$ in survival condition.

In 2006, full-scale trials measurements of bow accelerations of CAT-1 have been made in severe sea states. Further study will be carried out for the validation of LAMP wetdeck simulations using full-scale measured data.

5.3. CFD Simulations

ABS is putting significant efforts for the development and validation of computational fluid dynamics (CFD) code as an alternative numerical tool to deal with extremely violent nonlinear wave flows such as sloshing, slamming and green water impact problem.

The level-set Finite-Analytic Navier-Stokes (FANS) code is a CFD two-phase flow solver with multi-block overset-grid scheme, developed for highly nonlinear wave flows around ships and offshore structures [7, 8].

Recently, the level-set FANS method has been successfully validated for the prediction of sloshing impact pressure on tank boundary by comparing with sloshing model test data [9]. Fig. 34 shows an example of FANS simulated flow inside a LNG tank at a low filling condition with a prescribed transverse motion. Currently, ABS is working with Texas A&M to implement a compressible gas model into FANS code in order to more accurately predict sloshing and slamming impact pressure considering air cushion effect on trapped air. ABS is also building up a high performance computing system using PC cluster for CFD simulation.
FANS code was also used for the wetdeck slamming simulation of the catamaran CAT-1. Fig. 35 shows the overset moving grid system of FANS geometry model with 25 blocks, 2.2 million nodes on half domain (y>0). The simulations were performed using 16 processors on a Linux cluster.

Figures 36-38 show an example of a wetdeck slamming event of CAT-1 in regular waves. The catamaran is towed at a constant forward speed with Froude number Fr=0.3. It is allowed to heave and pitch freely in waves as shown in Figure 36. The incident wave length to ship length ratio (λ/L) is 1.0 and the wave amplitude to ship length ratio is H/L = 0.04. The heave displacement and pitch angle of the catamaran were obtained by solving the following two degree-of-freedom motion equations using the 4th-order Runge-Kutta method:

\[
\begin{align*}
    m\ddot{z} &= F_z \\
    I_{yy}\ddot{\theta} &= M_y; \quad I_{yy} = mr^2
\end{align*}
\]

It is seen from the free surface patterns in Fig. 36 and the wave elevation contours in Fig. 37 that the level-set FANS method has successfully predicted highly nonlinear waves in front of the bow and along the CAT-1 hulls. Due to the large amplitude heave and pitch motions, wet deck slamming (denoted by the blue color on the lower deck in Fig. 38) was observed when the wave height between the catamaran hulls exceeded the lower deck clearance. A detailed examination of the animation movie indicates that the ship bow was lifted upward by the incident wave with the bulbous bow partially emerged out of the water prior to the slamming impact. Wet deck slamming was observed around the wave peak region when the bow plunges into the water under combined heave and pitch motions.
CONCLUSIONS

In this paper, slamming impact loads have been studied for the high speed naval craft recently built to ABS class. Extensive nonlinear seakeeping analyses were carried out to calculate vertical accelerations and sectional impact forces along the ship length. Based on LAMP simulation results, validated by model test measurements and design practice, new requirements for bottom slamming design pressure on mono-hulls and wetdeck slamming design pressure on multi-hulls were proposed.

This paper also presents ABS’s on-going efforts for the development and validation of CFD methods to predict fully nonlinear 3D impact loads. The level-set FANS method successfully demonstrates the nonlinear free surface capturing capability for the benchmarking cases of sloshing and wetdeck slamming analyses.

To minimize computational time, we are considering the coupling of LAMP and FANS codes. FANS can be used for nonlinear viscous flow near the ship while LAMP is used for linear potential flow away from the ship. This coupled analysis method will be challenging, but also very promising for the simulations of slamming and green water impact loads, sloshing impact loads coupled with ship motion and damaged stability of vessels in waves.

ACKNOWLEDGMENTS

Since the 1988 DARPA project, the LAMP system has been developed for the advanced ship motion simulation under the sponsorships of the US Navy, the US Coast Guard, and the American Bureau of Shipping. The authors would like to thank Dr. L. Patrick Purcell of ONR for his continuous supports for LAMP development.

REFERENCES

1. ABS, Rules for Building and Classing High Speed Naval Craft, 2007


