EXPERIMENTAL TESTING OF SLAMMING PRESSURE ON A RIGID MARINE PANEL

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Abstract

This report presents an experimental investigation of the slamming pressures on a rigid panel using the SSTS (Servo hydraulic Slam Test System) which can be programmed to go with constant velocities, unlike other slamming test facilities based on drop tests. A symmetric wedge was idealised by using one rectangular panel, which was designed and built to have a low bending deflection and a dry natural frequency well above the loading frequency. The pressures were measured using six piezoelectric pressure transducers placed in thirteen different positions on the panel. Pressures were measured along the horizontal direction in order to verify a two-dimensional flow state, which showed that the peak pressures were slightly lower close to the short sides of the panel. However, the flow did not seem to vary considerably around the middle of the panel. The rigidity was examined by measuring deflection of the panel and the whole rig. The main tests were performed with the deadrise angles 10, 20, 30 and 40º and constant impact velocities from 0.5 m/s to 6 m/s. The control system of SSTS managed to keep the velocity constant during the slam event within a tolerance of 10 % up to 5 m/s. The pressures were measured at 5 points along the symmetry line from the keel towards the chine. Load and vertical displacement of the specimen fixture were also measured. The maximum peak and residual pressure measured were 933 respectively 76 kPa, obtained for 10º and 6 m/s. The minimum peak and residual pressure measured were both 6.3 kPa and obtained for 40º and 2 m/s.
1 Introduction

1.1 Background and Objectives
When a hull enters the water surface with a relatively high velocity a pressure pulse arises along the hull structure. This phenomenon is called slamming and occurs when the vessel hits a wave in the front or from below. The latter is for example very common for small planing crafts. In order to improve the understanding of the design loads of the hull there have been many studies which have developed calculation models for slamming pressures. Among all these studies there are not many dealing with the interaction between slamming pressure and structural response.

The aim of this thesis was to examine pressure loads on a rigid structure. One way to study the pressure load is to consider a two-dimensional wedge which is entering the water surface with a controlled vertical velocity. This was experimentally performed with the SSTS (Servo hydraulic Slam test System). Systematic tests were performed with the deadrise angles 10, 20, 30 and 40º and constant velocities, where the latter is unique for SSTS.

Among the measuring equipment were six piezoelectric pressure transducers that were moveable to different positions. Experiments aiming for verifications of the 2D flow state were done by comparing the pressure pulses at various positions on the panel. The rigidity of the panel and the whole rig was also investigated by measuring acceleration and deflection.

The thesis was developed as part of a cooperation between Naval Systems at the Royal Institute of Technology (KTH) in Stockholm, Sweden, and Industrial Research Limited (IRL) in Auckland, New Zealand. SSTS is developed and built at IRL and is used for funded and commercial research.

The experimental results from this work are to be compared with current work on a numerical model of High Speed Craft structure undertaken by Ivan Stenius at Naval Systems, KTH. The model takes fluid-structure interaction into account, based on explicit FE-codes and multi-material Eulerian formulations with penalty contact algorithms.
1.2 SSTS - Servo-hydraulic Slam Test System

Figure 1.1. Left: An overview photograph of SSTS showing the tank and data acquisition. Right: A photograph of the specimen and specimen fixture taken through the inspection window of the tank.

A major ambition with the experimental configuration with SSTS in this work is to create a two-dimensional flow state around a symmetric wedge, i.e. what an infinitely long wedge would generate, during water impact. The symmetric wedge is here idealised by a half wedge. The half wedge is a rectangular panel with an angle to a constraining vertical wall, which can be considered as a mirror in terms of fluid mechanics.

The SSTS (Figure 1.1) is a hydraulic controlled slam test system which can be programmed to run constant and non-constant velocities within a 5% tolerance up to 5 m/s for 10° deadrise angle. SSTS can achieve velocities of more than 10 m/s, however due to the force capacity of the hydraulic system constant velocities can only be achieved at velocities below approximately 6 m/s for 10° with panels of this size. Increasing the deadrise angle leads to lower force and thereby better controlled velocity. See chapter 4.2.1 for more information of velocity control.

The SSTS consists of a cylindrical water tank with a diameter of 3.5 m and a water depth of typically 1.5 m. Figure 1.2 shows the fundamental parts of the test system. The specimen fixture (Figure 1.3), which is attached to the hydraulic ram, slides on vertical rails and hence only moves in one degree of freedom. Two hydraulic accumulators drive the ram and the velocity is controlled by a servo valve. Three vertical panels, two on the sides and one in the back are supposed to constrain the flow to have the 2D behaviour (mentioned above) along the panel. The side panels are supposed to idealise an infinitely long panel. Each side panel is supported by four vertical aluminium bars. The back panel is supported in the
bottom and the top of the tank, but the horizontal movement is mainly restrained by four plastic blocks that are mounted on the fixture and slide on the back side of the back panel. The specimen fixture is mostly made of carbon fibre composite and can be adjusted to the deadrise angles of 0, 10, 20, 30 and 40º. The mass of the specimen fixture including the panel is 75 kg.

A typical test result from a slam is shown in Figure 1.4 where load, displacement and pressure are measured and the velocity is differentiated from the displacement. The slam event is defined as the time range between the keel impact and fully submerged panel and is shown with two vertical lines in the figure. The panel is here considered fully submerged when the chine has reached calm water level.
Figure 1.4. A typical slam showing a larger time range than only the slam event which is in between the vertical black and blue lines. The slam event is defined as the time it takes from the keel reaching the water surface till the panel is fully submerged.
2 Literature Review

This literature review is focused on former slamming tests and does not include full-scale craft tests. The intention is to review slamming tests that are done in similar controlled environment as in the Servo hydraulic Slam Testing System (SSTS) at Industrial Research Limited in Auckland.

One of the first real drop test with wedge-shaped models was done by Chuang 1966 [1]. The tests were performed with one rigid flat bottom model and five rigid wedges with deadrise angles of respectively 1, 3, 6, 10 and 15º. The models had piezoelectric pressure transducers and accelerometers. Video cameras and special electrical probes were used to detect trapped air. It was concluded that the effect of trapped air needs to be taken in account between 0 and 3º. On the basis of the experiments a set of equations for the pressure of a flat bottom impact was derived. The pressure was measured at the keel and away from the keel. The largest pressure was 80 psi away from keel with 7.5 inches drop height and 3º deadrise angle.

In [2] a 0.222-scale model of Apollo command module was used for drop tests. The impact structure had a spherical geometry and was instrumented with piezoelectric accelerometers, strain gauges, displacement transducers and flush-mounted pressure transducers. The magnitude of the pressure peaks are not very reliable because of the fairly slow instrumentation frequency. However, the characteristic pressure pulse with very short rise time and its following mean pressure were significant and could be compared to former full-scale tests.

Wedge-shaped hull sections of sandwich with width 2000 mm and length 1500 mm were used in drop tests for Det Norske Veritas (DNV) by Hayman 1991 [4]. Two models were used, one with stiffened aluminium and one of GRP sandwich with 3 mm skin thickness and 50mm Divinycell H200 PVC foam core. The models had a deadrise angle of 30º but were tilted to different angles down to 5º. The instrumentation consisted of four accelerometers, six pressure transducers and two strain gauges. The maximum pressures for the both models were compared in order to distinguish the influence of the flexibility. However, Hayman says that the differences are largely masked by other variations, e.g. different pressure transducers and sampling rates. Instead the author suggests integration of the structure stresses in order to compute the response load. The response load may then be compared for different models. For the sandwich model failure was obtained as shear in the core close to the chine. A numerical model was also used and showed that a quasi-static analysis reproduced the measured behaviour quite well. The FEA also confirmed that the ratio (maximum core shear stress)/(maximum laminate in-plane shear stress) is larger for slamming loads than for a uniformly distributed pressure. Experimental tests were also performed on a full-scale costal rescue craft with a length of 19.6 m and displacement of 33.5 tonnes. Accelerations and strains were measured in a speed of 18 knots and a significant wave height of 4-5 m. The highest measured acceleration close to cg was 6.2 g which was compared to old and new DnV Rules. Comparing the strain signals at the most severe slamming event to the drop test it corresponded to a drop height of 3 m.

Drop tests comprising flat, wedge, cylindrical and double curved panels were undertaken by Wraith 1998 [6], using a drop test tank. To make the velocity more uniform during impact weights were added to the models. The impact velocity was set by varying the drop height. The deadrise angles of the wedge were selected to 0, 3, 6, 10º to match Chuang [1] and 30º for Hayman [4]. The panels were instrumented with piezoelectric pressure transducers and
accelerometers. Three pressure transducers were mounted in a row on the middle of the wedge from keel to chine and eight along the keel from the middle to the edge. The test results show good agreement with both Chuang and Hayman’s experiments. The acoustic pulse was also considered in the analysis. However, the duration of the acoustic pressure pulse was too short to cause any measurable deformation or deceleration and the magnitude was found to have no significant dependency on the body geometry.

Jensen, 1999, [7] performed drop tests with a sandwich panel. The aim of the study was to develop a sensor system for a vessel of the Norwegian Navy. Their test set up was a horizontal panel mounted on a free falling rig which was dropped into a towing tank with induced waves. The panel had 2 mm GRP skins, 25 mm Divinycell core, had the dimensions 1000*600 mm and was equipped with 16 fibre optical sensors, a range of resistive strain gauges, two pressure transducers and six loadcells. The strain was almost linearly proportional to the drop velocity and the highest strain measured was 2300 microstrain. The study also included natural frequency analysis where the wet frequency was studied for different deadrise angles and velocities. The frequency increases with increasing angle and decreasing with increasing drop velocity. The former is due to decreasing added mass and the latter due to increasing added mass. The angle dependency is more pronounced at lower drop velocities. Only the first normal wet mode was detected, and at velocity 2 m/s and 0º angle the wet natural frequency was about 60 Hz, which is approximately 14 % of the dry frequency obtained from FE analysis.

Garme and Rosén, 2000, [8] performed experimental tests with a high-speed craft model (length 1.05 m and with 0.25 m) in a towing tank with regular and irregular waves. The model was equipped with 20 pressure transducers and 3 accelerometers in order to map the pressure distribution and rigid body motions. The sampling frequency was 2.5 kHz which captured the pressure pulse propagation and the pulse shape very well. In the irregular waves the model was towed with the forward speeds 4.5 and 3.1 m/s and the significant wave heights used were 5, 7.5 and 10 cm. The pressure peaks fitted a Weibull distribution which was used to ensure that the number of wave encounters was enough to perform a statistical analysis. The slamming pressures were in the order of magnitude of 30 kPa. It was also concluded that small model, the high speed and the short duration of the slamming events leads to difficulties making the test design nontrivial.

A comparison between some different computational methods and drop tests were done by Engle [11]. Two wedge-shaped models of 10 and 20º deadrise angle respectively were used in the experiment. Both models were considered rigid and were equipped with five pressure gauges. Maximum peak pressures were recorded to be compared to five numerical methods and theoretical models from Wagner and Chuang.

Manganelli [12] performed model tests, for an Open60’ class yacht using slam patches. Sections of the hull were cut out and replaced by six stiff patches (each 80x80 mm) that were attached to load cells. In that way the average slamming pressure of a section panel were measured. Sampling rates ranging between 2 kHz and 10 kHz were used. The model was exposed to rotational drop tests in calm water and was towed in regular waves instead of irregular waves, motivated by better repeatability. Different boundary conditions and the fact that the whole patch deflected (no bending) suggest higher hydroelastic effects than for full-scale, while the scaling based on Froude's number and constant Cauchy number would lead to lower hydroelastic effects for the model. However, the author refers to Bereznitski [9] who claims that the hydroelasticity is dependent of the ratio between the dry natural frequency and
the loading frequency. The highest equivalent average pressure obtain was around 350 kPa with an impact velocity of 2.75 m/s.

In [10] Stenius performs slamming tests using the Servo-hydraulic Slam Test System at Industrial Research limited in Auckland, New Zealand, with a single curved specimen. The deadrise angle of the specimen increased from 0° at the keel with a constant radius of 650 mm. The pressure was measured at a deadrise angle of 27°. The specimen was also equipped with load cell, strain gauges and accelerometers. The impact velocities presented are 4.5, 3.8, 3.3, 2.1 m/s, where the peak pressure measured for 4.5 m/s was 93 kPa. A new technique to represent a realistic time domain pressure pulse was implemented in a transient dynamic finite element model of the specimen. Previous theoretical prediction methods were also studied and compared to experiments.

Among the above mentioned slamming experiments have all of them more or less been of drop test basis, except the last using SSTS. The SSTS has been further improved since Stenius work, [10], which includes larger force capacity and a sophisticate control system.
3 Test Specimen

As described in chapter 1.2 the symmetric wedge is idealised by a rectangular panel. When comparing the experiments to theoretical analyses the panel will be considered rigid. The definition of rigid is however not trivial and has to do with hydroelasticity. Therefore the plate must be built so that the deflection is minimal and hence the deflection will be the primary parameter when designing the plate. As a consequence of this the bending and shear stiffness must be high. The panel weight must be as low as possible in order to stay within the limits of the SSTS (Servo hydraulic Slam Testing System) capacity. To make the panel both stiff and light it is preferably built as a sandwich.

3.1 Plate Analysis

The sandwich faces were chosen to be epoxy and carbon fibre laminates, where the tensile stiffness is \( E_f \), see Figure 3.1. The tensile stiffness of the core is \( E_c \), but will have small influence on the total bending stiffness because \( E_f \) is much higher. The major part of the normal stresses will be carried by the faces.

![Figure 3.1 The sandwich cross section shows core and face thickness and E-modulus.](image)

The panel is rectangular and simply supported with the sides \( a \) and \( b \), as illustrated in Figure 3.2. Over the panel there is a pressure, \( p \), which is determined by the hydrodynamic pressure from the water impact.

The following analytical solutions of sandwich panels are classical sandwich theory and are taken from Zenkert [5].
The load can be written as

$$q = \sum_{m=1}^{\infty} \sum_{n=1}^{\infty} q_{mn} \sin\left(\frac{m\pi x}{a}\right) \sin\left(\frac{n\pi y}{b}\right)$$

Eq. 3.1

where $q_{mn}$ are the Fourier coefficients that are determined by

$$q_{mn} = \frac{4}{ab} \int_{0}^{a} \int_{0}^{b} p(x, y) \sin\left(\frac{m\pi x}{a}\right) \sin\left(\frac{n\pi y}{b}\right) dx dy$$

Eq. 3.2

for a given load $p(x, y)$. For simplicity the deflection is determined by assuming isotropic faces. The deflection can be divided in two parts, deflection due to bending, $w_b$, and shear, $w_s$.

$$w_b = \sum_{m=1}^{\infty} \sum_{n=1}^{\infty} \frac{q_{mn} (1-\nu^2) \sin\left(\frac{m\pi x}{a}\right) \sin\left(\frac{n\pi y}{b}\right)}{D \left[\left(\frac{m\pi}{a}\right)^2 + \left(\frac{n\pi}{b}\right)^2\right]^2}$$

Eq. 3.3

$$w_s = \sum_{m=1}^{\infty} \sum_{n=1}^{\infty} \frac{q_{mn} \sin\left(\frac{m\pi x}{a}\right) \sin\left(\frac{n\pi y}{b}\right)}{S \left[\left(\frac{m\pi}{a}\right)^2 + \left(\frac{n\pi}{b}\right)^2\right]}$$

Eq. 3.4
\( \nu \) is the Poisson’s ratio for the laminate. \( D \) and \( S \) are bending stiffness and shear stiffness respectively and are determined by assuming thin faces and weak core which means that all the bending is taken by the faces and all the shear by the core,

\[
D = \frac{E_f t_f d^2}{2} \tag{Eq. 3.5}
\]

and

\[
S = \frac{G_c d^2}{t_c} \tag{Eq. 3.6}
\]

where

\[
d = t_c + t_f \tag{Eq. 3.7}
\]

and \( t_c \) is the core thickness, \( t_f \) the face thickness and \( G_c \) the core shear modulus.

### 3.2 Design Load

Several different theoretical models for slamming pressures on a wedge have been developed over the years. One theoretical method that is easy to formulate and is both time and space dependent was found by Wagner. According to Stenius [10] Wagner’s method is one of the most over estimating methods. The equations by Wagner, Eq. 3.8-10, are taken from chapter 9 in [3] and is partly found from the velocity potential for a flat infinitely long plate moving with a velocity \( V \) perpendicular to its surface and with an expanding width \( 2c(t) \). Combining this with the Bernoulli equation the pressure distribution is identified as

\[
p(x) = \rho V \frac{c(t)}{\sqrt{c(t)^2 - x^2}} \frac{dc}{dt} \tag{Eq. 3.8}
\]

where

\[
c(t) = \frac{\pi}{2} \frac{V t}{\tan \beta} \tag{Eq. 3.9}
\]

for a wedge illustrated in Figure 3.3.
Figure 3.3. A 2-D wedge falling down into the water showing the pile up water.

Eq. 3.8 is only valid for $x < c(t)$. The maximum pressure at constant impact velocity $V$ according to Wagner is

$$p_{\text{max}} = \frac{1}{2} \rho \left( \frac{dc}{dt} \right)^2$$  \hspace{1cm} \text{Eq. 3.10}$$

and is located close to $c(t)$. The pressure peak in Figure 3.4 (which is infinite according to Eq. 3.8) was reduced to $p_{\text{max}}$ when applying it to the panel and looks as in Figure 3.5.

Figure 3.4. A plot of Eq. 3.8 showing the spatial pressure distribution across the panel. Figure 3.5. The pressure distribution with the peak reduced to stagnation pressure. The dashed line represents a more realistic decay after the peak.

### 3.3 Panel Rigidity

When predicting the impact pressure with Wagner’s solution the panel is considered rigid. In reality the panel will deflect and thereby affect the pressure, the effect is known as hydroelasticity. However, in this study the experimentally measured pressures are preferred to
be close to what a rigid panel would give. To find out if the rigid assumption is reliable the significance of the hydroelastic effect must be examined. The first approach below uses Bereznitski [9] who studied beams subjected to slamming loads and suggests that only the relation between dry natural frequency and load period may be used. A method to estimate the reduction of pressure due to panel deflection is also presented.

What determines the pressure load on the panel are the deadrise angle and the relative velocity between the water and the panel. The latter is not only the impact velocity of the whole panel but also how quickly the panel is deflecting, which depends on the natural frequency and the loading frequency. Hence, an aim of this design is to build a panel with higher natural frequency than the loading frequency.

3.3.1 Dynamic Effects

In [9] Bereznitski states that the ratio between the duration, \( T_{dur} \), of the impact and the first period of the natural vibration of the dry structure, \( T_{period} \), is the key factor for defining when hydroelasticity is important.

\[
\text{ratio} = \frac{T_{dur}}{T_{period}}
\]

Bereznitski concludes that if the ratio is larger than 2, hydroelasticity does not play a significant role. The natural frequencies for a sandwich plate with isotropic faces, taken from [5], can be expressed as

\[
\omega_{mn} = \pi \left[ \left( \frac{mb}{a} \right)^2 + n^2 \right]^{1/2} \left[ \frac{D}{\rho^* b^4 (1-v^2)} \right] \left[ 1 + \pi^2 \theta \left( \frac{mb}{a} \right)^2 + n^2 \right]
\]

where

\[
\theta = \frac{D}{S (1-v^2) b^5},
\]

\[
\rho^* = 2 \rho_f + \rho_e
\]

\( a \) and \( b \) the sides of the panel and \( m \) and \( n \) the vibration modes.

The period is then

\[
T_{period} = \frac{1}{f_{mn}} = \frac{2\pi}{\omega_{mn}}.
\]
The impact duration is here assumed to be the time it takes from the keel entering the water surface to the instant when the pressure peak has moved to the chine. The vertical distance from the keel of the panel to the water surface as illustrated in Figure 3.3 is

\[ \eta_b = c(t) \tan \beta = \{ \text{due to Eq. 3.9} \} = \frac{\pi V t}{2}. \]

When \( c(t) \) has reached the chine this distance is

\[ \eta_b = b \sin \beta. \]

So the impact duration is

\[ T_{\text{dur}} = \frac{2b \sin \beta}{\pi V}, \]

and the ratio to be considered is found as

\[ \text{ratio} = \frac{T_{\text{dur}}}{T_{\text{period}}} = \frac{b \sin \beta \epsilon_{\omega_1}}{\pi^2 V}. \]

### 3.3.2 Estimation of Pressure Reduction

When a panel is subjected to a load it has to deflect, even a panel which is considered rigid will deflect some amount. But what is not that trivial is how the deflection interacts with the load. A change in deflection will theoretically give a change in slamming pressure and vice versa. The below analysis is an attempt to see how much a certain deflection would affect the pressure, i.e. without concerning about what has caused the deflection. This may lead to a conservative quantification of the effect by hydroelasticity.

It is known from Bernoullis equation that for an incompressible fluid the stagnation pressure is

\[ p_0 = (z - z_0) \rho g + \frac{1}{2} \rho V^2 \]

and if the gravitational part is neglected this becomes

\[ p_0 = \frac{1}{2} \rho V^2. \]

Hence, the pressure is proportional to the velocity squared, or

\[ p \sim V^2. \]
Now let \( V \) be the vertical impact velocity in the slam event. When the panel is deflecting the panels surface will experience a slightly smaller velocity. If the panel’s local velocity, or speed of deflection, is \( V_p \) the new velocity relative to the calm water surface is \( (V - V_p) \). Using the relation of Eq. 3.21 the change in pressure may be expressed as

\[
\text{reduction} = \frac{p_2 - p_1}{p_1} = \frac{p_2}{p_1} - 1 = \frac{(V - V_p)^2}{V^2} - 1
\]

Eq. 3.22

If \( V_p \) is chosen as a function of time it can be visualised what the reduction is for every time instant. For example if the deflection is assumed to have a sine shape as

\[
w(t) = \frac{w_{max}}{2} \left( 1 - \cos \left( \frac{2\pi}{T_{dur}} \right) \right)
\]

Eq. 3.23

the deflection velocity can be found from derivation.

\[
V_p(t) = \frac{dw}{dt} = \frac{w_{max}}{T_{dur}} \pi \sin \left( \frac{2\pi}{T_{dur}} \right)
\]

Eq. 3.24

Eq. 3.23 and Eq. 3.24 are plotted in Figure 3.6 and Eq. 3.22 in Figure 3.7.

![Figure 3.6](image)

**Figure 3.6.** Equation (3) and (4) with \( w_{max} = 3 \text{ mm} \), \( V = 6 \text{ m/s} \), \( T_{dur} = 20 \text{ ms} \).
From this example with only a deflection of 3 mm the pressure is significantly reduced with almost 16%. However this is the maximum and occurs only at one certain position.

3.4 Final Design

A Matlab programme was developed to calculate analytical equations of the deflection of the sandwich panel (Eq. 3.3 and Eq. 3.4), assuming isotropic faces. The load applied was the pressure distribution by Wagner, Eq. 3.8 and Eq. 3.9. In order to also have a solution to an orthotropic sandwich panel a finite element analysis (FEA) was performed using the FEA programme NISA. The load applied to the finite element model was an equivalent uniform pressure.

The materials of the sandwich panel were chosen to be carbon fibre and epoxy for the laminate and balsa as core. The reason to choose balsa instead of foam was the higher shear modulus of Balsa. The laminate data is presented in Table 3.1 to Table 3.3.

<table>
<thead>
<tr>
<th>Lamina type</th>
<th>CNC500, 0/90°, 60/40% CDB400, -45/45°</th>
</tr>
</thead>
<tbody>
<tr>
<td>Weight [kg/m²]</td>
<td>500</td>
</tr>
<tr>
<td></td>
<td>400</td>
</tr>
<tr>
<td>Tensile modulus, $E_x$ [GPa]</td>
<td>73</td>
</tr>
<tr>
<td></td>
<td>20</td>
</tr>
<tr>
<td>Tensile modulus, $E_y$ [GPa]</td>
<td>49</td>
</tr>
<tr>
<td></td>
<td>20</td>
</tr>
<tr>
<td>In-plane shear modulus, $G_{xy}$ [GPa]</td>
<td>6</td>
</tr>
<tr>
<td></td>
<td>29</td>
</tr>
<tr>
<td>Poisson ratio, $\nu_{xy}$</td>
<td>0.04</td>
</tr>
<tr>
<td></td>
<td>0.7</td>
</tr>
<tr>
<td>Density [kg/m³]</td>
<td>1500</td>
</tr>
<tr>
<td></td>
<td>1500</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Core</th>
<th>Balsa, Baltek CK100</th>
</tr>
</thead>
<tbody>
<tr>
<td>Thickness, $t_c$ [mm]</td>
<td>50</td>
</tr>
<tr>
<td>Density, $\rho_c$ [kg/m³]</td>
<td>145</td>
</tr>
<tr>
<td>Shear modulus, $G_c$ [MPa]</td>
<td>159</td>
</tr>
</tbody>
</table>
The entire sandwich laminate may also be described as:
[CNC5003/CDB400/CNC5002/CK100/ CNC5003/CDB400/CNC5003]

Table 3.3 Face laminate data.

<table>
<thead>
<tr>
<th>Face Carbon/Epoxy</th>
<th>Thickness, $t_f$ [mm]</th>
<th>$E_x$ [GPa]</th>
<th>$E_y$ [GPa]</th>
<th>$G_{xy}$ [GPa]</th>
<th>$v_{xy}$</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>3</td>
<td>65.6</td>
<td>44.0</td>
<td>6.74</td>
<td>0.13</td>
</tr>
</tbody>
</table>

The SSTS can only be set to angles of $\beta = 0, 10, 20, 30, 40^\circ$, although $0^\circ$ is not intended to be used here. Hence, the most severe deadrise angle is $10^\circ$ to design the panel. Previous tests have demonstrated that for the size of this panel a nominally constant velocity of up to 6 m/s is achievable for $10^\circ$. Because of the time dependence in Eq. 3.8 the design pressure for the panel is chosen for a time instant when the deflection, $w$, has its maximum, which was found at $t=9$ ms. The resulting prediction of panel properties are presented in Table 3.4 where it can be seen that the dominant part of the deflection is due to shear.

Table 3.4. Predicted panel response.

<table>
<thead>
<tr>
<th>Quantity</th>
<th>Analytic</th>
<th>FEA</th>
</tr>
</thead>
<tbody>
<tr>
<td>Load</td>
<td>Wagner, $\beta = 10^\circ$, $V = 6$ m/s</td>
<td>210 kPa uniform pressure</td>
</tr>
<tr>
<td>$w$ [mm]</td>
<td>1.87</td>
<td>1.33</td>
</tr>
<tr>
<td>$w_b$ [mm]</td>
<td>0.42</td>
<td>-</td>
</tr>
<tr>
<td>$w_c$ [mm]</td>
<td>1.45</td>
<td>-</td>
</tr>
<tr>
<td>$f_{11}$ [Hz], Dry natural frequency</td>
<td>624</td>
<td>598</td>
</tr>
<tr>
<td>Ratio (Eq. 3.18)</td>
<td>5.7</td>
<td>5.9</td>
</tr>
<tr>
<td>Reduction (Eq. 3.22) [%]</td>
<td>9.9</td>
<td>7.1</td>
</tr>
</tbody>
</table>

Ratio is larger than 2 from both the analytical and the FEA solution which implies that there is no need to consider hydroelasticity according to Bereznitski. Reduction is less than 10% which may be considered as a general engineering tolerance.

3.5 Construction

The manufacturing of the panel was done by using vacuum resin infusion which is based on covering the whole laminate with a plastic bag sealed to the mould and applying vacuum to one side and resin inlet on the other side. The lower pressure on the inside forces the resin through the laminate. One advantage of this method is the very controlled stacking procedure. All the laminae including the core are placed in their right positions before they are impregnated with the resin. The method also implies a minimum of resin/fibre volume fraction and a low content of voids. A step by step description of the procedure is presented in Figure 3.8.
In addition to the set of laminae chosen in the final design, strips of thin carbon fibre cloth were added where the major part of the pressure transducers were to be. Also an extra lamina of the same cloth was added as the outermost laminate layer. The reason for adding the cloth was better surface properties than for the rest of the layers (carbon fibre weave). What could happen otherwise with the weave if the surface is machined or notched is that fibres can be ripped off of the whole length of the panel as a result of water pressure during slamming. The strips are located along the line of p1 to p5 and the line of p11, p8 and p9 shown in Figure 4.1. The strips are of 100 mm width.

1. Two sheets of balsa core were pierced with 1 mm holes to improve the impregnation.
2. The carbon laminae, including the strips, were stacked.
3. One layer of carbon fibre cloth was added between the balsa sheets.
4. All fibre layers in place on top of a glass surface.
5. Peel ply (white) and porous carrier (green).
6. Vacuum bag wrapped around everything and sealed to the table.
7. Vacuum applied at top and glass plate turned to vertical position.
8. More than half the panel impregnated by epoxy resin.
9. Finished sandwich panel cut into right dimensions. Peel ply already peeled off.

Figure 3.8. Manufacturing procedure step by step.
In the slamming experiments the specimen is preferred to correspond to a symmetric wedge as in Figure 3.3, which was achieved by restraining the flow with a vertical panel close to the keel edge of the panel. In order to shorten the distance between the panel edge and the back panel, the panel was designed to have an angle at the keel edge. The angle can be seen in Figure 3.9 and was chosen to be 35° to the in-plane line. The largest deadrise angle for the tests was 40°, so the reason for not choosing the angle at the keel edge of the panel to 40° was that the upper surface of the panel would have too little supporting area to the specimen fixture. Additionally plastic strips were attached to the keel edge (see Figure 3.9) in order to reduce friction with the back panel.

Figure 3.9. The panel with keel strips on the edge to reduce friction with the back panel.
4 Experimental Details

4.1 Instrumentation

The pressure transducers are to measure the pressure distribution along the symmetry line of the panel from the keel to the chine. But to be able to verify the flow as two-dimensional (i.e. no variation in the horizontal direction) some transducer fixtures were also placed along the horizontal direction. Three extra fixtures, 6 to 8, were placed in the same vertical level as number 3 as in Figure 4.1. Number 9 to 12 were to compare the pressure front to the 1 to 5 row. Number 13 is to verify symmetry.

![Figure 4.1. A principal sketch of the panel showing positions of pressure measurement points, where a = 60, b = 107.5, c = 120 and d = 200 mm.](image)

The panel was prepared with 13 holes for the pressure transducers, where 10 of them were equipped with the flush mounting pressure transducer fixtures shown in Figure 4.2. The rest of the holes were to fit the moveable fixture as in Figure 4.3. Drawings of the fixtures can be seen in appendix B. All empty holes were plugged. Six dynamic piezoelectric pressure transducers were used. The sensors diaphragms have a diameter of 5.5 mm. Each one has a sensitivity of 1.45 mV/kPa and a maximum pressure of 68.9 MPa. See appendix C for further details. All the transducers were calibrated from the manufacturer. In between the specimen fixture and the ram a load cell was mounted (see Figure 1.2). A displacement transducer was also mounted on the specimen fixture in order to measure the vertical position. To measure the deflection of the panel one strain gauge and one displacement transducer were mounted at the middle of the panel for some tests. Also two accelerometers were used, one was permanently attached on the load cell and was used as feedback to the control system. The accelerometer was also connected to the PC-card to be compared to the numerically differentiated acceleration from the position transducer. The second accelerometer was used at different positions: on the main vertical beam of the rig to measure horizontal accelerations and on the specimen fixture in order to measure vertical acceleration at another distance from the one on the load cell.

The data acquisition was done by a 16 bit PC-card from National Instruments and is sampled with the Labview software at a sampling rate of 20 kHz per channel.
Table 4.1. Instrument details.

<table>
<thead>
<tr>
<th>Instrument</th>
<th>Position</th>
<th>Model</th>
</tr>
</thead>
<tbody>
<tr>
<td>Pressure transducer, 6 units</td>
<td>p1-13</td>
<td>PCB 113 A26</td>
</tr>
<tr>
<td>Load cell</td>
<td>Between ram and specimen</td>
<td>Precision Transducers LPC 10000</td>
</tr>
<tr>
<td>Displacement transducer, LVDT type</td>
<td>Engaged to lower ram edge</td>
<td>Vishay REC-139L</td>
</tr>
<tr>
<td>Panel displacement transducer, LVDT type</td>
<td>Middle of panel close to p3</td>
<td>Schaevitz 1000 HCD</td>
</tr>
<tr>
<td>Accelerometer 1</td>
<td>On top of load cell</td>
<td>PCB 321 A02</td>
</tr>
<tr>
<td>Accelerometer 2</td>
<td>Vert. beam close to water.</td>
<td>PCB 308 B</td>
</tr>
<tr>
<td>Strain gauge</td>
<td>Middle of panel</td>
<td>EA-13-12SAC-350</td>
</tr>
<tr>
<td>DAQ-card</td>
<td></td>
<td>National Instruments, PCI-6013</td>
</tr>
</tbody>
</table>

4.2 Testing The System: Evaluation and Improvements

The aim of the experimental testing in this project was to map the slamming pressures of a rigid marine panel for a number of different impact velocities and deadrise angles. The intention was mainly to measure the pressures with constant impact velocity, but also some tests with accelerating and decelerating impacts.

Before the main tests were performed some tests were done in order to investigate the characteristics of the test set up, such as repeatability, two-dimensional flow state and specimen rigidity.

4.2.1 Velocity Control

The velocity was controlled to be as constant as possible during the slamming event. The system managed to be controlled very well for low velocity, i.e. no extra valve drive was required. In the controller’s software, MACS which is developed by MOOG, the system can...
be manually programmed to give the hydraulic valve extra drive at specific times. The procedure of tuning the velocity profile was to run a slam and pick up the time from the velocity curve where the velocity was starting to decrease, and then change the settings in MACS. Appendix D shows screen shots of the controller software. An average of 15 loops were carried out for each condition of velocity and deadrise angle. The tolerance of the velocity was within 10%, although the major part was within 5%.

4.2.2 Importance of Flush Mounted Sensors

In the first set of tests a negative pressure dip was visible just before the pressure spike from some of the sensors, as can be seen in Figure 4.4. It was examined whether the sensors were properly clamped in its fixtures or not, which they were. Because of the fact that this dip did not show up for every pressure sensor, there were probably differences in how the pressure transducers were fitted in the fixtures. One visible difference was that all the pressure transducers were not entirely flush, with a variation within 0.5 mm. The sensors that were not flush were p4 and p10. The explanation could be that the spray, which is in the in-plane direction, has a very high velocity and a bump in its way would make it locally change direction and give a lower pressure on top of the bump (in this case the sensor).

![Figure 4.4. An early test with 2 m/s and 10° showing negative pressures for some of the transducers.](image)

After adjusting the depth of the pressure transducers and running the same velocity and deadrise angle the negative pressure dips were less pronounced. Also seen, that if the sensors instead were slightly recessed a small positive bump showed up just before the spike.

4.2.3 Sampling Frequency

The maximum sampling frequency for the data acquisition card was 200 kHz which is divided by the number of sampling channels to get the actual sampling rate. Since the number of channels was 8, 25 kHz was the limit, but because of the recommendation not to use the maximum rate 20 kHz was chosen as the limit sampling frequency. To find the required sampling rate for the slam testing a number of runs were made with different velocities and sampling rates. The shortest rise time of the pressure is expected at 10° and with the highest velocity 6 m/s, thus the most difficult condition to capture the pressure peak. The different
sampling rates used in the comparison were 10, 15 and 20 kHz and three tests were done for each frequency. By zooming in at the pressure peak it was found that the peak value was not captured for 20 kHz and that the pressure spike was rather chopped off which can be seen at the pt1 DAQ curve in Figure 4.5.

![Figure 4.5. The pressure peaks from the DAQ card with 20 kHz and the Signal Analyzer plus the interpolated DAQ signal.](image)

A well known formulation concerning signal analysis is the Nyquist theorem which explains the aliasing effect. The aliasing effect is what you see for instance when looking at the wheels of a car on television. At a certain speed the wheels seem to change rotation direction which is explained by the frequency the camera samples the picture. The Nyquist theorem says that if the frequencies of all the frequency components of the system to be measured are less than half the sampling frequency (which is called the Nyquist frequency $f_N$) all other points in between the sampling points are also known. If this condition is not satisfied all the frequency components above $f_N$ will be mirrored to the left side of $f_N$ in the measured data and add on the other components and the measured signal would not resemble the real signal. In order to investigate the frequency spectrum for the slam tests a signal analyzer with a sampling frequency of 262 kHz was connected to one of the pressure transducers. The pressure signal was sampled by both the signal analyzer and the data acquisition card (DAQ) during the same test. The voltage signal in respect to time as well as to frequency was saved on the signal analyzer. The spectrum is shown in Figure 4.6, where it can be seen that the amplitude can be assumed as zero somewhere before 10000 Hz. Hence, instead of increasing the sampling frequency, interpolation can be done to find the other points. In Figure 4.5 both the pulse recorded by signal analyzer and by the DAQ card are plotted, as well as the interpolated DAQ signal. The interpolation was done using the Matlab-function INTERP, which with a symmetric filter allows the original data to pass through unchanged and interpolates between so that the mean square error between the sampled points and their ideal values is minimized.
Figure 4.6. The frequency spectrum from the signal analyzer showing Nyquist frequency which is the half sampling frequency.

4.2.4 2D Verifications

If a straight horizontal line on the panel is considered, the pressure pulse would arrive at exactly the same time instant on every point on that line and have exactly the same shape and amplitude in ideal two-dimensional conditions, although that is hardly achievable in reality because of different contributing factors. In order to ensure that these factors do not affect the results in a considerable amount some tests were done to investigate the pressure distribution over the panel. The pressure transducers were placed in three different combinations, Figure 4.7, Figure 4.12 and Figure 4.13.

The first combination comprised the pressure points p8, p7, p6, p3 and p13 and was aimed to search for differences in pressure along the horizontal middle line and where the flow state possibly changed from two to three-dimensional. Figure 4.8 represents one of three tests with the deadrise angle of 10° and a velocity of 6 m/s, showing the value of the peak pressures. The pressures seem to be slightly lower on the sides (i.e. p8 and p13) than in the middle, which is representative for the other tests too. Although no such systematic difference is seen between p6 and p3, which suggests a less 2D flow state at the sides. In appendix A the pressure graphs in the time domain for the three tests are zoomed in around the peaks of p8, p7, p6, p3 and p13. In test 1 and 3 the peaks appear in the same order, while in test 2 all the peaks are more gathered and p3 is later than the others instead of first. The divergence in test 2 could be caused by a non flat water surface during the test. Having the distance between p3 and p4, $b$, it is possible to calculate the propagation velocity of the pressure pulse by reading the time difference, $t_{p4} - t_{p3}$, between respectively peaks:
\[ V_{\text{pulse}} = \frac{b}{t_{p4} - t_{p3}} \]  

Eq. 4.1

Eq. 4.1 gives a propagation velocity of 53.5 m/s, which may be multiplied to the largest time difference of p8, p7, p6, p3 and p13 in order to also have the largest spatial difference:

\[ d_{\text{max}} = V_{\text{pulse}} \Delta t_{\text{max}} \]  

Eq. 4.2

This distance \( d_{\text{max}} \) is thus 53.75\( \times \)0.3 mm \( \approx \) 16 mm. This difference is between p3 and p13 which are \( 2d = 400 \) mm apart. Some differences between these two are expected because p13 is closer to the boundaries. 16 mm along the panel corresponds to a vertical distance of 2.8 mm and could be caused by minor disturbances of the water surface. In Figure 4.9 the shape of the pressure front is estimated using Eq. 4.2.

![Figure 4.8. Peak pressures measured along the horizontal middle line at 6 m/s and 10°.](image1)

![Figure 4.9. Estimated shape of the pressure front along the horizontal middle line.](image2)

Also for the least severe testing condition, i.e. at 2 m/s and 40°, the pressures along the middle horizontal line arise nearly at the same time instant according to Figure 4.10, although the pressure signal is fairly blurry and oscillating which is further discussed in chapter 4.2.5. In appendix A the pressure graphs from two tests for this condition are zoomed in around p8, p7, p6, p3, p13. The largest time difference is approximately \( \Delta t_{\text{max}} = 4 \) ms, and the time it takes for the pulse to travel from p3 to p4 is 47 ms which implies according to Eq. 4.1 that the pulse is propagating in 2.28 m/s and that the larger spatial distance between the pressure peaks along the horizontal middle line is 9 mm according to Eq. 4.2.
Figure 4.10. Pressures measured along the horizontal middle line at 2 m/s and 40º.

Figure 4.11. Peak pressures measured along the horizontal middle line at 2 m/s and 40º.

Figure 4.12. Two different test set up for 2D verification.

Figure 4.13. Pressure pulses measured for both test set-ups in Figure 4.12, at 10º and 6 m/s.

Figure 4.14. Pressure pulses measured for both test set-ups in Figure 4.12, at 10º and 6 m/s.

In Figure 4.14 only the magnitudes of the pressure peaks are shown for p11, p12, p1, p9, p10 and p5. It is clear that the pressure tends to increase from the side towards the middle, i.e. from p11 to p1 and from p9 to p5. Although, the magnitudes from the keel and the chine should not be compared because the velocity was decreased significantly as the upper transducers were submerged.
Also for 40° and 2 m/s the peak pressures occur very close in time for both test set up, which can be seen in Figure 4.15.

The magnitudes of the peaks from the pressure points close to the keel and to the chine are shown in Figure 4.16. Also here the pressures are slightly lower at the sides than at the middle. However, by comparing p1 to p11 and p5 to p9, the points in the middle, p1 and p5, are not necessarily the highest.
From the 2D-verification tests at least two conclusions can be drawn. The pressures at the sides are lower than in middle, which suggests that the boundaries are affecting significantly. But the pressure points next to symmetry line are not necessarily lower than for the points on the symmetry line. Also the pressure spikes for the pressure points that are positioned at the same horizontal line appear very close in time. For the most sever condition, 10º and 6 m/s, Figure 4.9 shows that the largest spatial difference of the peak occurrence along the horizontal middle line is 16 mm, which may be considered as a reasonable straight pulse front.

4.2.5 Rigid Verifications

As a major topic for this project to measure the slamming pressures for a rigid wedge the rigidity of the whole system was experimentally investigated by measuring the panel deflection and accelerations of the whole rig. The sensors used for panel deflection are explained in chapter 4.1. The panel displacement transducer was only used for the 10º deadrise angle because it did not fit in the specimen fixture for the other angles. However, the laminate strain was measured in the panel’s x-direction for both 10 and 40º. The rigidity of the rig was examined by measuring the horizontal acceleration at the main vertical beam (Figure 1.2), which the specimen fixture is attached to. An attempt was also made to measure the vertical acceleration at two positions of the specimen fixture to discern a rotation of the specimen fixture.

Panel Deflection

The panel displacement was measured for the most severe condition, 6 m/s and 10º, and is shown in Figure 4.17. The maximum measured deflection was 3.05 mm. Figure 4.17 also shows the velocity of deflection, $V_p$, at the middle of the panel which is calculated by differentiating the displacement signal. Applying $V_p$ to Eq. 3.22 gives an estimated pressure reduction seen in Figure 4.18 which suggests a maximum reduction of 12.3 %. In the estimation of pressure reduction the impact velocity was set constant 6 m/s. In reality the impact velocity is non constant which also changes the pressure.

![Figure 4.17. The panel displacement and the velocity of deflection at the middle of the panel for 6 m/s and 10º.](image)
Vibration In The Rig

In the 2D verifying tests an oscillation of the pressure signal was obvious at the condition 2 m/s and 40° which can be seen in Figure 4.15. In Figure 4.19 p3 is zoomed in around the largest oscillations. The largest oscillations in the figure vary approximately from 167 to 180 Hz.

In the time range of Figure 4.19 all the other pressure signals oscillate in the same amount, which can be seen in Figure 4.15, even the signal from p1 which is straight below the supporting edge of the specimen fixture. If the oscillations were caused by vibration of the panel, p1 should oscillate with a lower amplitude than p3 due to larger deflection at the middle of the panel. The natural frequency of the panel was measured to 1136 Hz which is much higher than the oscillations in the pressures.

Vertical accelerations were measured at two points of the specimen fixture, on top of the loadcell and on the front beam. To excite vibrations in the specimen fixture the rig was run at 1 m/s and was then abruptly stopped before water impact. The acceleration signals were transformed to the frequency domain using the FFT (Fast Fourier Transform) function in
Matlab. The frequency spectrum from the accelerometer on the loadcell is shown in Figure 4.20. The first two spikes are identified as the natural frequencies of the hydraulic ram and are at 23 and 44 Hz respectively. The third spike, at 178 Hz, corresponds very well to the frequency which the pressures are oscillating at in Figure 4.19, 167 to 180 Hz. The spectrum from the accelerometer on the front beam was less pronounced for higher frequencies but did also comprise the spike at 178 Hz. Efforts were made to avoid exciting this natural frequency by adjusting the precision in the rails that the specimen fixture is guided by, but without success. The oscillations of the pressure signals seem to always occur at the same elevation of the specimen fixture but do not affect the measurements of peak pressure for all sensors. The residual pressure may be filtered with a band stop filter in order to eliminate the oscillations mathematically.

![Figure 4.20. Frequency spectrum of the acceleration on the loadcell.](image)

An accelerometer was engaged on the main vertical beam of the rig in order to measure horizontal acceleration of the whole rig. The acceleration signal was piecewise integrated to find the velocity and the displacement, shown in Figure 4.21. The maximum horizontal velocity and displacement were 0.48 m/s and 5.22 mm respectively. This velocity transformed perpendicular to the panel surface is 0.083 m/s, which with Eq. 3.22 would give a pressure reduction of 2.7%.

For the least severe condition, 2 m/s and 40º, the horizontal acceleration was very small and would probably not correspond to the horizontal acceleration of the specimen fixture.

![Figure 4.21. Velocity and displacement over time.](image)
Figure 4.21. Left: The horizontal acceleration low pass filtered at 400 Hz of the vertical main beam of the rig. The condition was 6 m/s and 10°. Right: Velocity and displacement numerically integrated from the horizontal acceleration.

The oscillations of the test system discussed above appear to be a structural response of the moving test fixture and/or the test frame. The instant where the oscillations occurred was after the slamming event for 10° deadrise angle but started to appear within the event as the deadrise angle was increased. For 40° deadrise angle the oscillations occurred at the middle of the event, which can be seen in chapter 5.4.

4.3 The Load

4.3.1 Correction of Load Due To Acceleration

The forces acting on the rig can be divided into five components, ram force $L$, gravitation $mg$, force from the hydrodynamic pressure $P$, the reaction force $R$ from the linear guide tracks and the friction $F$ and are shown in Figure 4.22. The force equilibrium of the vertical components can be defined as in equation Eq. 4.3,

$$ \downarrow: \quad ma = L + mg - P \cos \beta - F $$

Eq. 4.3

where $a$ is the acceleration, $m$ the mass of the panel and the specimen fixture and $\beta$ the deadrise angle. The load cell is calibrated to zero when the rig statically hangs in the ram and therefore $mg$ can be excluded, which gives

$$ L = ma + P \cos \beta + F $$

Eq. 4.4

So when presenting the load in order to show the hydrodynamic force acting on the panel the inertia force, $ma$, must be subtracted, i.e.

$$ \text{Corrected load} = L - ma $$

Eq. 4.5

One should be slightly careful of relying on the corrected load as well because of the unknown amount of friction contribution by the linear guide tracks. However, the corrected
load is a good indicator of the order of magnitude and may also be compared to the pressure result integrated over the area of the panel.

### 4.3.2 Load Due To Hydrodynamic Pressure

One way to obtain the load caused by hydrodynamic pressure on the panel is to integrate the measured pressure over the area of the panel, assuming a two-dimensional flow state over the panel. For the main tests, the pressure was measured at five points. Numerical integration of just five points would give a too low resolution. Therefore a new pressure function was created by interpolating between the measuring points. The interpolation method chosen was piecewise cubic Hermite interpolation and was performed with the Matlab-function INTERP1. The time instant in the slam event when the interpolation was done was chosen when a specific pressure transducer had reached its maximum, i.e. peak pressure. Because of the oscillations, explained in chapter 4.2.5 the specific pressure point chosen varied between different deadrise angles.

The integration was done by summing all the elements in the interpolated pressure vector, \( p \), and multiplying by the distance between the transducers, \( \Delta x \), and the length of the panel, \( a \). Hence, the vertical force is

\[
Force = \cos \beta \sum_{i=1}^{n} p_i \Delta x a
\]

Eq. 4.6

where \( n \) is the length of the vector \( p \) and \( p_i \) is the element \( i \) of \( p \). \( \beta \) is the deadrise angle.

Figure 4.23 shows the interpolated pressure distribution from one test with 10º and 2 m/s.

![Figure 4.23. The interpolated pressure in the spatial domain for the test 10deg_2ms_081204_3 which is further analyzed in chapter 5.1.](image)

The force from Eq. 4.6 is used in the results in chapter 5, except for 40º deadrise angle, in order to compare the measured force from the load cell.
4.4 Slam Test Programme

Main Tests
The main tests were run with the deadrise angles of 10, 20, 30, and 40° and with velocities shown in Table 4.2, where the factor 3 in the third column is the number of repetitions for each condition.

Table 4.2. The test runs with constant velocity profile.

<table>
<thead>
<tr>
<th>Deadrise angle, [°]</th>
<th>Velocity, [m/s]</th>
<th>Number of runs</th>
</tr>
</thead>
<tbody>
<tr>
<td>10</td>
<td>0.5, 1, 2, 3, 4, 5, 6</td>
<td>7*3 = 21</td>
</tr>
<tr>
<td>20</td>
<td>2, 3, 4, 5, 6</td>
<td>5*3 = 15</td>
</tr>
<tr>
<td>30</td>
<td>2, 4, 5, 6</td>
<td>4*3 = 12</td>
</tr>
<tr>
<td>40</td>
<td>2, 4, 6</td>
<td>3*3 = 9</td>
</tr>
</tbody>
</table>

Thus, a total number 57 runs were performed for the main tests. In addition a various number of runs were done in order to tune the velocity profile described in chapter 1.2.

Tests With Shallow Tank
A few tests were performed with 100 mm lower water level at 10° deadrise angle. The velocities used were 4 and 5 m/s. The tests intended to find differences in slamming pressures due to more shallow water. No obvious changes were obtained in the limited number of tests made. A more extensive set of tests is required in order to understand how much the boundaries in the tank are affecting the slamming pressures. Hence, further discussion of water depth is excluded in this report.

Tests With Accelerating and Decelerating Impacts
In addition to the runs with constant velocities, tests were done with accelerating and decelerating impacts.

The accelerating runs were performed by starting with a very low velocity, 0.5 m/s, then as the keel reaches the water start to accelerate up to 5 m/s as the chine reaches the calm water surface.

The decelerating runs were performed by accelerating the specimen to 5 m/s before the water surface and then start to decelerate just as the keel hits the water surface and keeps decelerating till the specimen stops with the chine in the calm water level.
5 Results

The results of the slamming experiments in this chapter are presented in a time window of the whole acquisitioned data in order to only show the slamming event defined as the time period from the keel reaching the water surface to the chine reaching the calm water surface. The original raw voltage data file created by Labview was calibrated and processed (for example velocity was calculated from the displacement) in Excel where a text file of the processed data was created. The processed data file was then used by a Matlab programme for further processing, such as plotting pressures and load.

The residual pressure presented below is defined as the more steady value which the pressure signal is converging to after the peak.

5.1 10º Deadrise Angle

In Table 5.1 all the peak pressures are presented for the 10º tests. The residual pressures are read from the data as the pressure from p1 in the time instant that p5 has reached the peak pressure. This residual pressure value from p1 is an average taken over a range of 200 sampled points.

Table 5.1. Maximum and residual pressures from the slamming experiments with 10º deadrise angle. The tests marked with bold letters are graphically presented below.

<table>
<thead>
<tr>
<th>Test name</th>
<th>Velocity [m/s]</th>
<th>p1 max [kPa]</th>
<th>p2 max [kPa]</th>
<th>p3 max [kPa]</th>
<th>p4 max [kPa]</th>
<th>p5 max [kPa]</th>
<th>Residual pressure [kPa]</th>
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</thead>
<tbody>
<tr>
<td>10deg_05ms_141204_1</td>
<td>0.5</td>
<td>6.5</td>
<td>9.1</td>
<td>10.3</td>
<td>10</td>
<td>9.8</td>
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<tr>
<td>10deg_05ms_141204_2</td>
<td>6.7</td>
<td>8.3</td>
<td>8.7</td>
<td>9.0</td>
<td>9.2</td>
<td>9.6</td>
<td></td>
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<tr>
<td>10deg_05ms_141204_3</td>
<td>7.4</td>
<td>8.3</td>
<td>8.5</td>
<td>9.1</td>
<td>8.6</td>
<td></td>
<td></td>
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<tr>
<td>10deg_1ms_071204_1</td>
<td>125.4</td>
<td>22.7</td>
<td>30.7</td>
<td>33.9</td>
<td>32.7</td>
<td></td>
<td></td>
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<tr>
<td>10deg_1ms_071204_2</td>
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<td>19.6</td>
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<td>29.8</td>
<td>35.2</td>
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<td>25.6</td>
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<td>94.0</td>
<td>151.0</td>
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<td>125.8</td>
<td>126.4</td>
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<td>97.9</td>
<td>122.0</td>
<td>125.7</td>
<td>116.4</td>
<td></td>
<td></td>
</tr>
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<td>10deg_3ms_081204_1</td>
<td>3</td>
<td>205.8</td>
<td>178.1</td>
<td>230.7</td>
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<td>249.3</td>
<td>233.5</td>
<td>34.0</td>
<td></td>
</tr>
<tr>
<td>10deg_3ms_081204_3</td>
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<td>215.1</td>
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<td>10deg_4ms_081204_1</td>
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<td>353.8</td>
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<td>477.3</td>
<td>473.2</td>
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</tr>
<tr>
<td>10deg_4ms_081204_2</td>
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<td>446.1</td>
<td>444.7</td>
<td>458.1</td>
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<td>57.0</td>
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</tr>
<tr>
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<td>478.3</td>
<td>475.2</td>
<td>521.2</td>
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</tr>
<tr>
<td>10deg_5ms_081204_1</td>
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<td>523.6</td>
<td>646.7</td>
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<td>10deg_6ms_141204_2</td>
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</tr>
</tbody>
</table>

10º deadrise angle was the most difficult condition to tune for an acceptable velocity profile partly because 10º gives the largest forces, but also because the event is shorter and is
therefore more difficult to control. The velocity profiles for 10° that corresponds to the particular tests presented are shown below every pressure and load graph.

**Test 10deg_05ms_141204_1**
Figure 5.1 shows the pressure where small variation of peak pressures between p2 to p5 can be seen. The lower peak value of p1 does not seem to correspond to lower velocity according to the velocity profile. The well pronounced periodic variations of the load seen in Figure 5.2 is due the system tuning itself for a constant velocity. Good agreement between the load cell signal and the corrected load due to acceleration, calculated with Eq. 4.5.

The load calculated from the interpolated spatial pressure, Eq. 4.6, is corresponding well with the measured load. However, the magnitudes are very low because of the low velocity and are therefore difficult to compare.

![Figure 5.1](image1.png)  
*Figure 5.1. Above, the pressures measured at p1-5 for 10° and 0.5 m/s and below the corresponding velocity profile.*

![Figure 5.2](image2.png)  
*Figure 5.2. Above, the load from the load cell both with and without correction of acceleration. The load calculated from pressures, Eq. 4.6, is also shown. Below the corresponding velocity profile.*

**Test 10deg_1ms_071204_2**
Figure 5.3 shows the pressures measured for 1 m/s. Variations of the peak pressure can be seen, although the residual pressures seem similar for all the pressure curves. The values of the peak and residual pressures are found in Table 5.1. The peaks tend to increase from the keel towards the chine, except for p2 which has the lowest peak and seem to correspond to lower velocity according to the velocity profile. Very good load agreement seen in Figure 5.4.
Figure 5.3. Above, the pressures measured at p1-5 for 10° and 1 m/s and below the corresponding velocity profile.

Figure 5.4. Above, the load from the load cell both with and without correction of acceleration. The load calculated from pressures, Eq. 4.6, is also shown. Below the corresponding velocity profile.

Test 10deg_2ms_081204_3

Figure 5.5 shows lower peak pressures for p1 and p2. p1 is lower than p2 even though the velocity is higher for the instant of maximum p1.

The load calculated from the pressures agrees very well with the measured load.

Figure 5.5. Above, the pressures measured at p1-5 for 10° and 2 m/s and below, the corresponding velocity profile.

Figure 5.6. Above, the load from the load cell both with and without correction of acceleration. The load calculated from pressures, Eq. 4.6, is also shown. Below the corresponding velocity profile.
Test 10deg_3ms_081204_1
In this test p2 has the lowest peak which seems to correspond to lower velocity at that time shown in the velocity profile (see Figure 5.7).

The calculated load in Figure 5.8 is higher than the measured load and is also higher in proportion compared to the test with 2 m/s.

![Figure 5.7](image1.png)  
**Figure 5.7.** Above, the pressures measured at p1-5 for 10° and 3 m/s and below, the corresponding velocity profile.

![Figure 5.8](image2.png)  
**Figure 5.8.** Above, the load from the load cell both with and without correction of acceleration. The load calculated from pressures, Eq. 4.6, is also shown. Below the corresponding velocity profile.

Test 10deg_4ms_081204_2
Also here a lower peak pressure for p1 is seen in Figure 5.9. The gap between the measured and calculated load seen in Figure 5.10 has increased even more in proportion compared to the tests with lower velocity. A smooth dip for the corrected load starting at the time 90 ms can be seen, which indicates a significant deceleration which the velocity profile confirms as the velocity is decreasing.
Figure 5.9. Above, the pressures measured at p1-5 for 10º and 4 m/s and below, the corresponding velocity profile.

Figure 5.10. Above, the load from the load cell both with and without correction of acceleration. The load calculated from pressures, Eq. 4.6, is also shown. Below the corresponding velocity profile.

Test 10deg_5ms_081204_2
Same tendency as for 4 m/s is seen in both Figure 5.11 and Figure 5.12 with lower peak for p1, larger calculated load and lower corrected load. As shown in the velocity profile, the system did not manage to keep the velocity close to 5 m/s throughout the slam event.

Figure 5.11. Above, the pressures measured at p1-5 for 10º and 5 m/s and below, the corresponding velocity profile.

Figure 5.12. Above, the load from the load cell both with and without correction of acceleration. The load calculated from pressures, Eq. 4.6, is also shown. Below the corresponding velocity profile.
For the 6 m/s test the velocity was decreasing from approximately 6.1 to 4.7 m/s which has a considerably effect on the hydrodynamic pressure due to relation 3.14 and should be considered when comparing the pressure peaks in Figure 5.13. Despite the drop in velocity the peaks of p1 and p5 are almost of the same magnitude.

The calculated load in Figure 5.14 is more than twice as high as the corrected measured load. The corrected load is significantly lower than the measured.

![Figure 5.13. Above, the pressures measured at p1-5 for 10° and 6m/s and below, the corresponding velocity profile.](image1)

![Figure 5.14. Above, the load from the load cell both with and without correction of acceleration. The load calculated from pressures, Eq. 4.6, is also shown. Below the corresponding velocity profile.](image2)

The load calculated from the measured pressures are significantly higher than the measured load for the velocities over 2 m/s. A possible explanation could be that the interpolation method used for the spatial pressure distribution may not fit well enough when the peak pressure is much higher than the residual pressure, which is the case for 10° deadrise angle. Also the fact that the pressures are lower closer to the sides (see chapter 4.2.4) could be more pronounced for higher velocities.

### 5.2 20° Deadrise Angle

The residual pressure is now read from p1 as p4 reaches the peak pressure. This because the oscillations discussed in chapter 4.2.5 start to appear for 20°. The calculated load is also calculated at the instant of peak pressure at p4.
Table 5.2. Maximum and residual pressures from the slamming experiments with 20° deadrise angle. The tests marked with bold letters are graphically presented below.

<table>
<thead>
<tr>
<th>Test no.</th>
<th>Velocity</th>
<th>p1 max</th>
<th>p2 max</th>
<th>p3 max</th>
<th>p4 max</th>
<th>p5 max</th>
<th>Residual pressure</th>
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<td>29.7</td>
<td>31.1</td>
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<td>30.5</td>
<td>10.9</td>
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<td>249.6</td>
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<td>268.7</td>
<td>260.6</td>
<td>227.3</td>
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<td></td>
</tr>
</tbody>
</table>

The velocity profile was not significantly easier to tune for 20° than for 10°, although the system managed to hold the velocity longer throughout the event. The velocity profiles are shown below every pressure and load graph.

20deg_2ms_061204_2

For this test the peaks in Figure 5.15 are fairly even throughout the event and may be explained by the even velocity profile which stays within a tolerance of 5 % for all peaks. However, for such low velocity the pressure is affected less. Also the loads show good correlation in Figure 5.16.
Figure 5.15. Above, the pressures measured at p1-5 for 20º and 2 m/s and below, the corresponding velocity profile.

Figure 5.16. Above, the load from the load cell both with and without correction of acceleration. The load calculated from pressures, Eq. 4.6, is also shown. Below, the corresponding velocity profile.

20deg_3ms_061204_1
Also for 3 m/s there was a small variation of the pressure peaks, shown in Figure 5.17. Like the 2 m/s tests there was good correlation of the loads shown in Figure 5.18.

Figure 5.17. Above, the pressures measured at p1-5 for 20º and 3 m/s and below, the corresponding velocity profile.

Figure 5.18. Above, the load from the load cell both with and without correction of acceleration. The load calculated from pressures, Eq. 4.6, is also shown. Below, the corresponding velocity profile.
For 4 m/s p1 has the highest peak and considerably higher than p2 which may correspond to the lower velocity at the peak of p2 seen in the velocity profile (see Figure 5.19). Although, a consistency of the correlation between magnitude of peak pressure and velocity is harder to discern than for the tests presented above. For example, the peak of p3 is slightly higher than for p4 even though the velocity is lower for p3.

Figure 5.19. Above, the pressures measured at p1-5 for 20º and 4 m/s and below, the corresponding velocity profile.

Figure 5.20. Above, the load from the load cell both with and without correction of acceleration. The load calculated from pressures, Eq. 4.6, is also shown. Below, the corresponding velocity profile.
A better correlation between the velocities and peak pressures is seen in Figure 5.21 than for 4 m/s.

The loads in Figure 5.22 are well correlated but with a slightly higher calculated load.

Figure 5.21. Above, the pressures measured at p1-5 for 20º and 5 m/s and below, the corresponding velocity profile.

Figure 5.22. Above, the load from the load cell both with and without correction of acceleration. The load calculated from pressures, Eq. 4.6, is also shown. Below, the corresponding velocity profile.

Figure 5.23 shows a fairly small variation of peak pressures given that a change in velocity has larger influence on the pressure for such high velocity as 6 m/s. The peak of p5 is slightly lower which also corresponds to a lower velocity at that time instant.

The corrected load and the calculated load are more separated from the measured load than for 5 m/s, which is reasonable considering the higher velocity.
Figure 5.23. Above, the pressures measured at p1-5 for 20º and 6 m/s and below, the corresponding velocity profile.

Figure 5.24. Above, the load from the load cell both with and without correction of acceleration. The load calculated from pressures, Eq. 4.6, is also shown. Below, the corresponding velocity profile.

Very good agreement between the calculated load and measured load is obtained for 20º except for 6 m/s, which in similarity with 10º may be caused by the higher peak/residual ratio.

5.3 30º Deadrise Angle

The oscillations discussed in chapter 4.2.5 appear more significantly for 30º than for 20º in terms of how large part of the event is affected. Hence, the residual pressure is read when the pressure peak has reached p3.

Table 5.3. Maximum and residual pressures from the slamming experiments with 30º deadrise angle. The tests marked with bold letters are graphically presented below.

<table>
<thead>
<tr>
<th>Test no.</th>
<th>Velocity</th>
<th>p1 max</th>
<th>p2 max</th>
<th>p3 max</th>
<th>p4 max</th>
<th>p5 max</th>
<th>Residual pressure</th>
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<td>[m/s]</td>
<td>[kPa]</td>
<td>[kPa]</td>
<td>[kPa]</td>
<td>[kPa]</td>
<td>[kPa]</td>
<td>[kPa]</td>
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<td>97.4</td>
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<td>100.6</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>
30deg_2ms_021204_3
For the 30° deadrise angle a significant difference of the peak/residual ratio is obtained compared to the lower angles. For 2 m/s the peak pressure is little pronounced but still visible, according to Figure 5.25. The magnitudes of the peaks are increasing from p1 to p5, even though the velocity is fairly constant throughout the event.

Figure 5.26 shows good correlation between the measured and corrected load but a slightly lower calculated load. The measured load is higher than the calculated which could be due to the friction in the tracks.

30deg_4ms_021204_1
For 4 m/s the peak of p1 is larger than the others and also narrower in shape. The width of the spikes seem to increase from p1 to p5 (see Figure 5.27). The pressures are clearly affected by the oscillation for all the pressure curves after p3. The peak of p1 is higher than the rest even though the velocity is only 0.1 m/s lower for p2.
Figure 5.27. Above, the pressures measured at p1-5 for 30º and 4 m/s and below, the corresponding velocity profile.

Figure 5.28. Above, the load from the load cell both with and without correction of acceleration. The load calculated from pressures, Eq. 4.6, is also shown. Below, the corresponding velocity profile.

30deg_5ms_031204_1

Same behaviour of p1 is obtained in Figure 5.29 as for 4 m/s. Figure 5.30 shows an excellent correspondence between the measured and the calculated pressure profiles.

Figure 5.29. Above, the pressures measured at p1-5 for 30º and 5 m/s and below, the corresponding velocity profile.

Figure 5.30. Above, the load from the load cell both with and without correction of acceleration. The load calculated from pressures, Eq. 4.6, is also shown. Below, the corresponding velocity profile.
Also for 6 m/s p1 is significantly higher than the rest. The load calculated from the measured pressures agrees very well with the measured load for 5 and 6 m/s.

What seems general for the 30º pressure graphs is that the peak tends to become wider as the pressure pulse travels towards the chine. Also, the rise time of the pressure spikes seem to become longer in the same manner.

5.4 40º Deadrise Angle
For 40º deadrise angle it is not possible to reasonably calculate the load from the measured pressures because of the major part of oscillations and to only interpolate between the two points p1 and p2 would not be possible. The residual pressure is also difficult to distinguish, even for p1 and p2 which are not significantly affected by the oscillations. Hence, the author suggests to be aware of uncertainties of the presented data concerning peak and residual pressure for p3, p4 and p5.
Table 5.4. Maximum and residual pressures from the slamming experiments with 40° deadrise angle. The tests marked with bold letters are graphically presented below.

<table>
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<tr>
<th>Test no.</th>
<th>Velocity</th>
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<th>p2 max</th>
<th>p3 max</th>
<th>p4 max</th>
<th>p5 max</th>
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<tr>
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<td>58.1</td>
<td>46.0</td>
<td>51.3</td>
<td>46.9</td>
<td>57.4</td>
<td>37.5</td>
<td></td>
</tr>
<tr>
<td>40deg_6ms_021204_3</td>
<td>58.2</td>
<td>45.3</td>
<td>49.7</td>
<td>46.0</td>
<td>57.8</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

The constant velocity profiles were much easier to maintain for 40° than for the other angles.

Figure 5.33 to Figure 5.38 shows the pressure and the load signals for the 40° tests. In similarity with the 30° tests the peak of p1 is higher than p2 for 4 and 6 m/s.

40deg_2ms_021204_2

Figure 5.33. The pressures measured at p1-5 for 40° and 2 m/s.

Figure 5.34. Load cell signal both with and without correction of acceleration. The load calculated from pressures, Eq. 4.6, when p4 is maximum is also shown.
Figure 5.35. The pressures measured at p1-5 for 40° and 4 m/s.

Figure 5.36. Load cell signal both with and without correction of acceleration. The load calculated from pressures, Eq. 4.6, when p4 is maximum is also shown.

Figure 5.37. The pressures measured at p1-5 for 40° and 6 m/s.

Figure 5.38. Load cell signal both with and without correction of acceleration. The load calculated from pressures, Eq. 4.6, when p4 is maximum is also shown.
Like the 30º deadrise angle the spikes tends to become wider and the rise time longer as the pressure pulse travels across the panel. Also, the peak pressure for p1 is significantly higher than for p2.

5.5 Pulse Width versus Sensor Diameter
For deadrise angles of 20º and above an increase of rise time and pulse width, from the keel towards the chine, was visible. Hence, the width of the pulse in the space domain is also increasing. Due to the finite width of the diaphragm of the pressure sensors the increasing pulse width would also lead to an increase of measured peak value. If the pressure spike is very narrow, the sensor is only experiencing a mean value, i.e. if the pressure is not even over the diaphragm. This would for instance explain equal or higher pressure for p5 than p4 despite lower velocity at p5.

One way to estimate the significance of the effect is to consider the rise time for the most severe case 6 m/s and 10º, which in the space domain corresponds to the x-distance of the dashed line in Figure 3.5. At p3 the rise time is approximately 0.15 ms. Using that in Eq. 4.2, where the pulse velocity is calculated with Eq. 4.1 the distance from the peak to zero is 7 mm, which is relatively close to the 5.5 mm diaphragm diameter of the sensor. However, this distance is not equal to the width at the peak, though in the same order.

5.6 Summary of Peak Pressures
In order to illustrate how the peak pressures depend on deadrise angle, the mean values of the results presented in chapter 5.1 to 5.4 are plotted in Figure 5.39. When calculating the mean values different pressure transducers were used for different conditions in order to exclude errors caused by oscillations and too high or too low velocity.
Figure 5.39. Peak pressures plotted against deadrise angle for 3 different velocities.
5.7 Accelerating and Decelerating Impacts

When considering a boat or ship the impact velocity may be considered constant over a short distance of the hull. However, over a longer distance the velocity is far from constant. The impact can either be accelerating or decelerating. The following two tests measures the slamming pressure and velocity when the rig is accelerating from 0 to 5 m/s respectively decelerating from 5 to 0 m/s, see Figure 5.40 and Figure 5.41. In the velocity graphs however, it can be seen that the velocity varies within a smaller range than from 0 to 5 m/s because the spatial distance from p1 to p5 is smaller than the distance from the keel to the chine edge. For example in the first test when p1 has reached the water surface, the rig has already accelerated to 1.8 m/s.

Figure 5.40. Pressure and velocity signal from the accelerating test.
Figure 5.41. Pressure and velocity signal from the decelerating test.
6 Conclusions

From the systematic slamming tests with different deadrise angle and velocities, peak pressure, residual pressure and load was successfully measured with constant impact velocity for most conditions of velocity and deadrise angle.

The two-dimensional flow assumption is verified and found valid between the symmetry line to the line of p10, p6 and p12. At the sides, i.e. p9, p8, p11 and p13, the maximum pressure was slightly lower. The rigidity of the panel was accepted according to [9]. When the panel deflection was measured for the most severe condition 10° and 6 m/s the pressure reduction due to panel deflection was estimated to approximately 12 % which may be considered as relatively high. However, the major part of the main tests has much less pressure loading which also decreases the effect of panel deflection. The oscillations that occur in the pressure graphs are considerably affecting the result for 30° and 40° deadrise angle. However, the peak pressure was measured unaffected by p1 and p2.

The control system of SSTS did manage to keep the velocity constant within a tolerance of 10 % and a major part within 5 %. The differences of pressure peak values in each test did correspond to the instantaneous velocity for the major part of all tests. Hence given the instantaneous velocity the pressure result can be compared to a theoretical model by using the same velocity as obtained in the experiments.

The load that was calculated by interpolating between the pressure points at a certain time instant and integrated along the panel width did coincide with the measured load for moderate velocities. From 3 m/s for 10° and from 5 m/s for 20° the calculated load started to diverge upwards from the measured load. The reasons are probably the simplicity in the used interpolation method in combination with the non 2D flow close to the panel borders.

The distance from peak to zero pressure in the space domain was calculated to the same order as the diaphragm diameter of the sensor, which indicates that the peak pressure can have been slightly averaged over the diaphragm for some tests. This explains the relation between increasing rise time and peak pressure from the keel towards the chine for many tests.

Future work after this thesis would mainly be to make similar tests with an elastic panel and measure panel displacement and strain for more combinations of deadrise angle and velocity. But also to improve the understanding of the behaviour of the SSTS such as investigate the effects by the tank boundaries and localise the vibrations described in this report.
7 Acknowledgements

This thesis work was carried out at Industrial Research Limited in Auckland, New Zealand under supervision of Dr Mark Battley whom I wish to acknowledge for making this work at IRL possible. Mr Ivan Stenius and Dr Anders Rosén at the department of Naval systems at The Royal Institute of Technology in Stockholm, made the project possible. Ivan Stenius continuously supervised from the opposite side of the globe. The author wants to address his appreciation to the group "Smart materials and structures" for excellent working atmosphere, especially to Peter Beck for invaluable help with the slam rig, to Tim Wyatt for his flexibility in the workshop and to Eric Wester for good help with signal analysis. Gratitude's to High Modulus in Auckland for sharing their facilities and knowledge when manufacturing the sandwich panel.
8 References


Appendix A: Graphs of 2D Verifications

The following three graphs represent three tests with a velocity of 6 m/s and a deadrise angle of 10º and are zoomed in on the pressure pulses measured along the horizontal middle line.

Pressure pulses measured along the horizontal middle line with $V=6$ m/s and $\beta=10^\circ$. Test 1.
Pressure pulses measured along the horizontal middle line with $V=6$ m/s and $\beta=10^\circ$. Test 2.

Pressure pulses measured along the horizontal middle line with $V=6$ m/s and $\beta=10^\circ$. Test 3.
Pressure pulses measured along the horizontal middle line with $V=2\,\text{m/s}$ and $\beta=40^\circ$. Test 1.

Pressure pulses measured along the horizontal middle line with $V=2\,\text{m/s}$ and $\beta=40^\circ$. Test 2.
Appendix B: Drawings of pressure transducer fixtures

The drawing of the permanent fixture. 10 pieces were manufactured.
A cad-model of the moveable fixture and the pressure transducer in the middle.
### Appendix C: Pressure transducer details

**ICP® PRESSURE SENSOR SPECIFICATIONS**

<table>
<thead>
<tr>
<th>Performance</th>
<th>ENGLISH</th>
<th>SI</th>
</tr>
</thead>
<tbody>
<tr>
<td>Measurement Range (for ±5V output)</td>
<td>500 psd</td>
<td>3450 kPa</td>
</tr>
<tr>
<td>Useful Overrange (for ±10V output)</td>
<td>1000 psd</td>
<td>6895 kPa</td>
</tr>
<tr>
<td>Sensitivity (±0.5 mV/pS)</td>
<td>10 mV/s</td>
<td>0.069 kPa</td>
</tr>
<tr>
<td>Maximum Pressure</td>
<td>10 kpsi</td>
<td>6895 kPa</td>
</tr>
<tr>
<td>Resolution</td>
<td>10 ps</td>
<td>0.069 kPa</td>
</tr>
<tr>
<td>Resonant Frequency</td>
<td>≥500 kHz</td>
<td>≥500 kHz</td>
</tr>
<tr>
<td>Rise Time</td>
<td>≤1.0 μ sec</td>
<td>≤1.0 μ sec</td>
</tr>
<tr>
<td>Low Frequency Response (±5%)</td>
<td>0.01 Hz</td>
<td>0.01 Hz</td>
</tr>
<tr>
<td>Non-Linearity</td>
<td>≤1.0 % FS</td>
<td>≤1.0 % FS</td>
</tr>
</tbody>
</table>

**Environmental**

| Acceleration Sensitivity | ≤0.002 psd/g | ≤0.0014 kPa/ms² |
| Temperature Range (Operating) | -100 to +278 °F | -73 to +135 °C |
| Temperature Coefficient of Sensitivity | ±0.1 5%/°F | ±0.18 °F/C |
| Maximum Flash Temperature | 3000 °F | 1649 °C |
| Maximum Vibration           | 2000 g pk     | 19014 m/s² pk  |
| Maximum Shock               | 2000 g pk     | 19014 m/s² pk  |

**Electrical**

| Output Polarity (Positive Pressure) | Positive | Positive |
| Discharge Time Constant (at room temp) | ≤50 sec | ≤50 sec |
| Excitation Voltage | 20 to 30 VDC | 20 to 30 VDC |
| Constant Current Excitation | 2 to 20 mA | 2 to 20 mA |
| Output Impedance | <100 ohms | <100 ohms |
| Output Bias Voltage | 8 to 14 VDC | 8 to 14 VDC |

**Physical**

| Sensing Geometry | Compression | Compression |
| Sensing Element | Quartz | Quartz |
| Housing Material | 174 Stainless Steel | 174 Stainless Steel |
| Diaphragm | Invar | Invar |
| Sealing | Welded Hermetic | Welded Hermetic |
| Electrical Connector | 10-32 Coaxial Jack | 10-32 Coaxial Jack |
| Weight (with clamp nut) | 0.2 oz | 6.0 gm |

**Optional Versions** (Optional versions have identical specifications and accessories as listed for standard model except where noted below. More than one option maybe used.)

- E - Enamel coating
- H - Hermetic Seal
- M - Metric Mount
- N - Negative Output Polarity
- S - Stainless Steel Diaphragm

### Notes

1. For ±10 volt output, minimum 24 VDC supply voltage required. Negative 10 volt output may be limited by output bias.
2. Zero-based, least-squares, straight line method.
3. See PCB Declaration of Conformance PS023 for details.
4. For sensor mounted in threaded adapter, see adapter installation drawing for supplied accessories.
5. Used with optional mounting adaptor.

### Supplied Accessories

- 960A03 Clamp nut, 5/16-24-2A thd, 1/4" hex, stainless steel
- 965A02 Seal ring, sensor flush mount, 0.245" OD x 0.218" ID x 0.016" flk, brass
- 965A06 Steel sleeve sensor recess mount 0.245" OD x 0.221" ID x 0.240" flk 17-7 (1)

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**PCB PIEZOTRONICS**

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Appendix D: Screen Pictures of MACS

The upper picture shows the main window where for example the position of the ram is displayed and can be changed manually with very low speed if the system is set to manual mode. The lower picture shows the command window where percentage and timing of extra valve drive for feed forward are set. MACS is a product of MOOG.