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SEA TESTS OF THE USCGC UNIMAK
PART 3 - PRESSURES, STRAINS, AND DEFLECTIONS OF THE BOTTOM
PLATING INCIDENT TO SLAMMING

by

Joshua E. Greenspon

RESEARCH AND DEVELOPMENT REPORT

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TABLE OF CONTENTS

ABSTRACT ........................................................................................................................................... 1
INTRODUCTION ..................................................................................................................................... 1
TEST INSTALLATION .......................................................................................................................... 2
PRESENTATION AND ANALYSIS OF DATA.......................................................................................... 4
   Pressures, Strains, and Deflections Experienced by Bow Plating .................................................. 4
   Computed Strains ................................................................................................................................. 8
   Comparison of Computed and Measured Strains ............................................................................. 10
   Comparison of Stresses Estimated for Several Ships ....................................................................... 10
   Strength Estimates for Typical Bottom Plates of Selected Ships .................................................... 12
DISCUSSION OF RESULTS .................................................................................................................. 12
CONCLUSIONS ..................................................................................................................................... 13
ACKNOWLEDGMENTS .......................................................................................................................... 13
APPENDIX A - THEORY ....................................................................................................................... 14
   Strain at any Point in Either of the Coordinate Directions ............................................................... 14
APPENDIX B - COMPARISON OF COMPUTED AND MEASURED TIME VARIATION OF PLATE STRAIN .................................................................................................................. 17
REFERENCES ....................................................................................................................................... 19
LIST OF ILLUSTRATIONS

Figure 1 - Location of Gages and Shell Expansion, Frame Spacing 24 Inches .......... 2
Figure 2 - Photographs of Test Setup ........................................................................ 3
Figure 3 - Representative Slamming Records ............................................................. 5
Figure 4 - Pressure Distribution on Bottom (Oscillogram No. 3965-B)
   For One of the Recorded Slams ........................................................................... 9
Figure 5 - Sketch of UNIMAK Test Plate ................................................................. 9
Figure 6 - Sketch of Plate .......................................................................................... 14
Figure 7 - Triangular Pulse ......................................................................................... 18
Figure 8 - Theoretical Experimental Time History Comparison ................................ 19

LIST OF TABLES

Table 1 - Instruments Used during the Test ................................................................. 4
Table 2 - Peak Pressures and Strains Incident to Slamming ........................................ 8
Table 3 - Comparison Between Measured and Calculated Strains .............................. 11
Table 4 - Theoretical Maximum Stresses in Typical Bottom Plating ............................ 11
Table 5 - Pressures at Various Stages of Deformation Based on Theory ..................... 12
USCGC UNIMAK
ABSTRACT

This report presents and discusses some of the test data taken in the early part of 1955 during rough-water sea trials of a Coast Guard cutter on weather patrol duty. The data includes the impact pressures incident to slamming as well as the corresponding strains and deflections of the forward bottom plating. These measurements are compared with results obtained from theoretical considerations, and strength estimates are made for typical bottom plates of selected ships.

INTRODUCTION

Examination of structural failures in the hulls of ships operating in rough seas indicated that more information should be obtained concerning the magnitudes of the loads acting on the bow structure. Accordingly, in 1947, the Bureau of Ships requested that the Taylor Model Basin obtain information on the forces acting on the bows of ships. During the spring of 1951, the U.S. Coast Guard cutter CASCO was instrumented for the measurement of strains and motions at sea. The installation included a diaphragm pressure gage, especially developed for this purpose, installed in the bottom to measure impact pressures incident to slamming. In these tests the response of the pressure gage was approximately linear up to 100 psi, and the recording system was linear up to about 60 cps. However, only low-speed records were taken; the peak slamming pressures damaged the gage and thus were not recorded. When the pressure gage was removed, it was found that the diaphragm had a permanent set, and it was estimated that a pressure exceeding 250 psi had caused this set. On the basis of these tests it was decided that further studies should be conducted in order to determine the magnitudes and time durations of the slamming pressures.

In the fall of 1954, the David Taylor Model Basin requested the U.S. Coast Guard to make another cutter available for further testing, and the USCGC UNIMAK, a sister ship to the USCGC CASCO, was assigned for this purpose. The UNIMAK operated at two weather stations in the North Atlantic, Stations Delta and Baker. The best records were obtained during the second trip at Station Baker. The ship operated in very heavy seas, and the Captain allowed the ship to undergo severe slamming so that records could be taken.

This report gives some of the results of the measurements taken on the UNIMAK* during the second trip at Station Baker. The measurements discussed here are the pressures incident to slamming as well as the corresponding strains and deflections in the forward bottom plating. Several examples of these measured quantities are compared with the responses predicted by use of theoretical considerations. Estimated representative pressures are used to calculate

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References are listed on page 19.

*This report is one of a series of reports on these tests. Reports 976 and 977 contain a general outline of tests and test results, statistical distribution patterns of hull motions and stresses, and wave observations.
the approximate stresses that would be expected in typical panels of several types of ships when subjected to these pressures.

TEST INSTALLATION

The portion of the instrumentation which was designed to study the effects of slamming included eight pressure gages, three strain gages to measure bending strain in the plate, three strain gages to measure tensile strain in the middle surface of the plate, one strain gage to measure strain in the keel, one deflection gage to measure deflection at the center of the test panel, and an accelerometer to measure the linear acceleration of the ship's bow during slamming. The locations of these gages are shown in Figure 1, and some of the photographs taken

<table>
<thead>
<tr>
<th>Gage</th>
<th>Pressure Gages Location</th>
<th>Gage</th>
<th>Strain Gages Location</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>3 in. Fwd of Fr 21, 8 1/4 in. Stbd of C</td>
<td>A</td>
<td>Longitudinal Tensile Strain, Center of Panel Fr 22-23, 8 1/4 in. Stbd C</td>
</tr>
<tr>
<td>2</td>
<td>3 in. Fwd of Fr 23, 8 1/4 in. Stbd of C</td>
<td>B</td>
<td>Longitudinal Bending Strains, Center of Panel Fr 22-23, 8 1/4 in. Stbd C</td>
</tr>
<tr>
<td>3</td>
<td>7 5/8 in. Aft of Fr 23, 8 1/4 in. Port of C</td>
<td>C</td>
<td>Longitudinal Tensile Strain, 15.9 in. Aft Fr 23, 8 1/4 in. Stbd C</td>
</tr>
<tr>
<td>4</td>
<td>7 5/8 in. Aft of Fr 23, 8 1/4 in. Stbd of C</td>
<td>D</td>
<td>Longitudinal Bending Strain, 15.9 in. Aft Fr 23, 8 1/4 in. Stbd C</td>
</tr>
<tr>
<td>5</td>
<td>7 5/8 in. Aft of Fr 23, 3 in. Stbd of 6 in. Longitudinal</td>
<td>E</td>
<td>Longitudinal Tensile Strain, 22.1 in. Aft Fr 23, 8 1/4 in. Stbd C</td>
</tr>
<tr>
<td>6</td>
<td>7 5/8 in. Aft of Fr 23, 3 in. Stbd of Longitudinal 1</td>
<td>F</td>
<td>Longitudinal Bending Strain, 22.1 in. Aft Fr 23, 8 1/4 in. Stbd C</td>
</tr>
<tr>
<td>7</td>
<td>3 in. Aft of Fr 24, 8 1/4 in. Stbd of C</td>
<td>G</td>
<td>Longitudinal Hull Strain 15 in. Aft Fr 23 on C Keel</td>
</tr>
<tr>
<td>8</td>
<td>11 3/4 in. Aft of Fr 28, 8 1/4 in. Stbd of C</td>
<td>I</td>
<td>Deflection Gage on Center of Test Panel</td>
</tr>
<tr>
<td>9</td>
<td>5G, Accelerometer 2 in. Fwd of Fr 24 on C Keel</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

Figure 1 - Location of Gages and Shell Expansion, Frame Spacing 24 Inches

*The 8 1/4 in. Starboard centerline is the location of the center of the panel next to the keel. All dimensions were taken from the edge of the stiffeners.
of the test setup are given in Figure 2. It should be noted that the test plate was extra heavy, installed because of a history of bottom damage.

Table 1 gives the type and range of instruments used in the tests, together with the frequency responses of each circuit.

The output of the sensing elements listed in Table 1 was amplified by Consolidated Engineering Company System D carrier amplifiers and recorded on a Consolidated Engineering Company 18-channel oscillograph equipped with Type 7-228 galvanometers.
TABLE 1
Instruments Used during the Test

<table>
<thead>
<tr>
<th>Gage</th>
<th>Quantity Measured</th>
<th>Sensing Element</th>
<th>Linear Range</th>
<th>Linear Frequency Response of System</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Pressure</td>
<td>Differential Transformer</td>
<td>0-1000 psi</td>
<td>0-500</td>
</tr>
<tr>
<td>2</td>
<td>Pressure</td>
<td>Differential Transformer</td>
<td>0-800 psi</td>
<td>0-500</td>
</tr>
<tr>
<td>3</td>
<td>Pressure</td>
<td>Differential Transformer</td>
<td>0-800 psi</td>
<td>0-500</td>
</tr>
<tr>
<td>4</td>
<td>Pressure</td>
<td>Differential Transformer</td>
<td>0-1000 psi</td>
<td>0-500</td>
</tr>
<tr>
<td>5</td>
<td>Pressure</td>
<td>Differential Transformer</td>
<td>0-800 psi</td>
<td>0-500</td>
</tr>
<tr>
<td>6</td>
<td>Pressure</td>
<td>Differential Transformer</td>
<td>0-800 psi</td>
<td>0-500</td>
</tr>
<tr>
<td>7</td>
<td>Pressure</td>
<td>Differential Transformer</td>
<td>0-800 psi</td>
<td>0-500</td>
</tr>
<tr>
<td>8</td>
<td>Pressure</td>
<td>Differential Transformer</td>
<td>0-800 psi</td>
<td>0-500</td>
</tr>
<tr>
<td>9</td>
<td>Acceleration</td>
<td>Unbounded Strain Gages</td>
<td>0-5 g</td>
<td>0-100 (approximately)</td>
</tr>
</tbody>
</table>

A Tensile Strain SR-4 Gage 2
B Bending Strain SR-4 Gage 2
C Tensile Strain SR-4 Gage 2
D Bending Strain SR-4 Gage 2
E Tensile Strain SR-4 Gage 2
F Bending Strain SR-4 Gage 2
G Strain in Keel SR-4 Gage 2
H Central Deflection Differential Transformer ±0.5 in.

PRESENTATION AND ANALYSIS OF DATA
PRESSURES, STRAINS, AND DEFLECTIONS EXPERIENCED BY BOW PLATING

Figure 3 contains several typical records obtained during the test.
Table 2 lists the maximum slamming pressures obtained on Gages 2, 3, 4, and 5 and the peak bending strains obtained at Station F for a number of selected oscillograms, where each oscillogram corresponds to a given slam. These gages experienced the largest pressures and
Estimated fundamental period of plate obtained by hitting plate and allowing it to vibrate freely.*

Heave Acceleration 1/2 g

Roll Angle

Pitch Angle

Roll Angle

Heave Acceleration

Roll Angle

Heave Acceleration

Pitch Angle

Pitch Angle

Figure 3a - Record 4064

Figure 3b - Record 4098

Figure 3 - Representative Slamming Records

See Figure 1 and Table 1 for Instrument Identification. The repetitive signal on Gages B, D, and F is hash.

*This period was measured when the ship was in the water.
strains during the tests. Often the pressure was not uniform over the plate since the pressure pulse moved across the plate; consequently, the peak pressures did not occur at the same time. Distributions of pressure over the forward bottom for one of the slams recorded during the tests are given in Figure 4. The curves demonstrate the rather localized nature of the high pressure. Curves of such a type should be useful to the designer in future designs of the bottom framing in the vicinity of slamming. In general, the times of rise and durations of the impact pressures acting at the gage were estimated to be greater than the fundamental period of the panel of plating, so that the pressure may be considered to act statically for purposes of computing the corresponding maximum stresses and deflections. This is partially illustrated by the absence of high frequencies in the strain records. It was noted that the maximum pressures were usually experienced by the pressure gages near the keel. During any given slam the gage near the turn of the bilge (Gage 6) never indicated a greater maximum pressure than the gage near the keel.

Unfortunately the deflection gage was damaged after only a few deflection measurements were taken and the few deflections recorded were small (of the order of 0.01 in., as can be seen from Figure 3).

At the time of slamming, large accelerations of short duration occurred in the bow. These accelerations can be seen on the oscillograph records in Figure 3 (Gage 9). Part of acceleration is due to hull bending and part to rigid body motion.
TABLE 2

Peak Pressures and Strains Incident to Slamming

<table>
<thead>
<tr>
<th>Record No.*</th>
<th>Peak Pressure, psi</th>
<th></th>
<th></th>
<th>Peak Bending Strains†</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Gage 2</td>
<td>Gage 3</td>
<td>Gage 4</td>
<td>Gage 5</td>
</tr>
<tr>
<td>4064</td>
<td>144</td>
<td>201</td>
<td>99</td>
<td>150</td>
</tr>
<tr>
<td>4098</td>
<td>79</td>
<td>136</td>
<td>127</td>
<td>122</td>
</tr>
<tr>
<td>4039</td>
<td>29</td>
<td>144</td>
<td>70</td>
<td>100</td>
</tr>
<tr>
<td>4049</td>
<td>94</td>
<td>52</td>
<td>95</td>
<td>53</td>
</tr>
<tr>
<td>4104B</td>
<td>88</td>
<td>52</td>
<td>52</td>
<td>68</td>
</tr>
<tr>
<td>4034</td>
<td>71</td>
<td>34</td>
<td>29</td>
<td>31</td>
</tr>
<tr>
<td>4096</td>
<td>51</td>
<td>50</td>
<td>29</td>
<td>37</td>
</tr>
<tr>
<td>4076</td>
<td>102</td>
<td>47</td>
<td>61</td>
<td>82</td>
</tr>
<tr>
<td>3965B</td>
<td>71</td>
<td>59</td>
<td>87</td>
<td>152</td>
</tr>
<tr>
<td>4095</td>
<td>38</td>
<td>38</td>
<td>33</td>
<td>46</td>
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<td>4080</td>
<td>47</td>
<td>38</td>
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<td>37</td>
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<td>4075</td>
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<td>36</td>
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<td>4067</td>
<td>148</td>
<td>51</td>
<td>113</td>
<td>142</td>
</tr>
<tr>
<td>4057</td>
<td>48</td>
<td>112</td>
<td>51</td>
<td>42</td>
</tr>
<tr>
<td>4102</td>
<td>27</td>
<td>49</td>
<td>31</td>
<td>35</td>
</tr>
<tr>
<td>4101</td>
<td>59</td>
<td>61</td>
<td>48</td>
<td>51</td>
</tr>
<tr>
<td>4033</td>
<td>32</td>
<td>33</td>
<td>31</td>
<td>41</td>
</tr>
<tr>
<td>4062</td>
<td>72</td>
<td>34</td>
<td>42</td>
<td>51</td>
</tr>
<tr>
<td>4083</td>
<td>261**</td>
<td>268**</td>
<td></td>
<td>295**</td>
</tr>
<tr>
<td>4092</td>
<td></td>
<td>183**</td>
<td></td>
<td>184**</td>
</tr>
<tr>
<td>4077</td>
<td></td>
<td>48</td>
<td>69</td>
<td>63</td>
</tr>
</tbody>
</table>

*Some of these records are illustrated in Figure 3.

**The pressure lines on these oscillograph records were not well-defined so that these values are only approximate.

†Peak values do not necessarily occur simultaneously.

COMPUTED STRAINS

The theoretical formula that is utilized here for the computation of the strain at Station F is derived from the theory given in Reference 3. The assumptions made in calculating the theoretical strain due to a given pressure are as follows:

1. The plate undergoes small deflections (less than about 0.7 of its thickness).
2. The plate has all edges clamped.
3. Two alternative assumptions were used to compute the pressure.
   a. The pressure is uniformly distributed and is statically applied to the panel. This uniform pressure is taken to be the average of the pressures on Gages 2, 3, 4, and 5 at any given time.
b. The uniform pressure is taken equal to the average of the peak pressures on the four gages statically applied to the panel.

Under either of these assumptions the formula for the approximate bending strain $\epsilon_F$ at Station F is as follows:* 

$$\epsilon_F = 0.095 \frac{P_0 a^2}{E h^2} \tag{1}$$

*See Appendix A for derivation of this formula.
where $P_0$ is the average pressure,

- $a$ is the width of the panel, (use 10.53 in. as width of plate),
- $h$ is the thickness of the panel, (7/16 in.) and
- $E$ is the modulus of elasticity of the panel material.

**COMPARISON OF COMPUTED AND MEASURED STRAINS**

Table 3 shows a comparison between measured and theoretical strains computed by Equation [1], using Assumptions 3a and 3b. Some of the causes for discrepancies could be as follows:

1. Inaccuracy in assuming that the average pressure is uniformly distributed over the panel.
2. Inaccuracy of the theory which assumes that the modes of a clamped plate can be represented in terms of beam functions.
3. Inaccuracies in limiting the number of terms in the approximations to the mode shapes.
4. Inaccuracy in assuming that the edges of the plate were clamped and in neglecting motion of the supports.
5. Inaccuracy in using four pressures to calculate the average pressure, in view of the fact that three of these pressures were not taken directly on the test panel.
6. Inaccuracies in the instrumentation, the calibrations, and the reading of the data.

A comparison of computed and measured time variation of the strain at Station F is given in Appendix B. Good agreement between the calculated and measured values is indicated.

**COMPARISON OF STRESSES ESTIMATED FOR SEVERAL SHIPS**

If it is assumed that the small-deflection theory of plates holds and that the pressure is applied statically,* then the maximum stress in the plate can be readily computed.** Table 4 lists maximum stresses that might be expected in typical bottom plating of various ships subjected to slamming, assuming that the pressures are of the same order of magnitude as those measured on the UNIMAK.†

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*On the basis of the measurements and comparisons, these two assumptions seem to be reasonable.

**The approximate formulas for the maximum stresses are tabulated for small deflections in Table 1 of Reference 3 or page 228 of Reference 4. The formulas in Reference 4 were used to calculate the stresses in Table 4 since they are undoubtedly more accurate.

†This assumption can only be verified by further field tests or model tests. However, it can be noted that for ships with finer lines, such as destroyers or cruisers, lower pressures than the ones measured in these tests may be expected; see Reference 5.
### TABLE 3
Comparison Between Measured and Calculated Strains

<table>
<thead>
<tr>
<th>Record No.</th>
<th>Average Pressure at Time of Maximum Strain (psi)</th>
<th>Maximum Measured Strain (μ in/in.)</th>
<th>Calculated Strain Using Average Pressure at Time of Maximum Strain (μ in/in.)</th>
<th>Measured Strain (R&lt;sub&gt;2&lt;/sub&gt;)</th>
<th>Average of Peak Pressures Using Average Pressure on Gages 2, 3, 4, 5 (see Table 2) (psi)</th>
<th>Calculated Strain Using Average Peak Pressure (μ in/in.)</th>
<th>Measured Strain (R&lt;sub&gt;2&lt;/sub&gt;)</th>
</tr>
</thead>
<tbody>
<tr>
<td>4098</td>
<td>86</td>
<td>200</td>
<td>157</td>
<td>1.27</td>
<td>116</td>
<td>211</td>
<td>0.95</td>
</tr>
<tr>
<td>4039</td>
<td>36</td>
<td>89</td>
<td>66</td>
<td>1.35</td>
<td>86</td>
<td>157</td>
<td>0.57</td>
</tr>
<tr>
<td>4049</td>
<td>33</td>
<td>86</td>
<td>60</td>
<td>1.43</td>
<td>74</td>
<td>135</td>
<td>0.64</td>
</tr>
<tr>
<td>4104 B</td>
<td>56</td>
<td>111</td>
<td>102</td>
<td>1.09</td>
<td>65</td>
<td>110</td>
<td>0.94</td>
</tr>
<tr>
<td>3965 B</td>
<td>82</td>
<td>171</td>
<td>150</td>
<td>1.14</td>
<td>92</td>
<td>167</td>
<td>1.02</td>
</tr>
<tr>
<td>4076</td>
<td>57</td>
<td>154</td>
<td>104</td>
<td>1.48</td>
<td>73</td>
<td>133</td>
<td>1.16</td>
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<tr>
<td>4096</td>
<td>28</td>
<td>93</td>
<td>51</td>
<td>1.62</td>
<td>42</td>
<td>77</td>
<td>1.21</td>
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<td>4034</td>
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<td>90</td>
<td>53</td>
<td>1.70</td>
<td>41</td>
<td>75</td>
<td>1.20</td>
</tr>
<tr>
<td>4064</td>
<td>91</td>
<td>168</td>
<td>166</td>
<td>1.01</td>
<td>148</td>
<td>269</td>
<td>0.62</td>
</tr>
<tr>
<td>4095</td>
<td>31</td>
<td>139</td>
<td>57</td>
<td>2.44</td>
<td>39</td>
<td>71</td>
<td>1.36</td>
</tr>
</tbody>
</table>

Arithmetic Mean = 1.41

### TABLE 4
Theoretical Maximum Stresses in Typical Bottom Plating

<table>
<thead>
<tr>
<th>Ship</th>
<th>Panel Size (Approximate) in.</th>
<th>Material</th>
<th>Approximate Maximum Stress for 100-psi Pressure* psi</th>
</tr>
</thead>
<tbody>
<tr>
<td>Coast Guard Cutter</td>
<td>10½ x 24 x 0.44 (reinforced plate)</td>
<td>Mild Steel</td>
<td>29,000</td>
</tr>
<tr>
<td>Destroyer A</td>
<td>24 x 26 x 0.44</td>
<td>High Tensile Steel</td>
<td>100,000**</td>
</tr>
<tr>
<td>Destroyer B</td>
<td>21 x 24 x 0.37</td>
<td>Mild Steel</td>
<td>120,000**</td>
</tr>
<tr>
<td>Cruiser</td>
<td>46 x 41 x 0.50</td>
<td>Mild Steel</td>
<td>220,000**</td>
</tr>
<tr>
<td>Liberty Ship</td>
<td>27 x 36 x 0.70</td>
<td>Mild Steel</td>
<td>64,000**</td>
</tr>
<tr>
<td>Cargo Vessel</td>
<td>27 x 32 x 0.81</td>
<td>Mild Steel</td>
<td>42,000**</td>
</tr>
</tbody>
</table>

*It is assumed that the elastic limit has not been exceeded and that the pressure is uniformly distributed.

**These very high values indicate that the plate has probably already yielded. The actual stresses can then no longer be computed by the elastic theory.
TABLE 5
Pressures at Various Stages of Deformation Based on Theory

The following values were used in the computations: yield stress of mild steel, 30,000 psi; yield stress of high tensile steel, 50,000 psi; ultimate stress for mild steel, 60,000 psi; ultimate stress for high tensile steel, 85,000 psi; Young's modulus, $30 \times 10^6$ psi; and Poisson's ratio, 0.3.

<table>
<thead>
<tr>
<th>Ship</th>
<th>Panel Size in.</th>
<th>Material</th>
<th>Pressure at Yield (Hovgaard's Curve A) psi</th>
<th>Pressure at Which Permanent Set is 20 Percent of Deflection (Hovgaard's Curve B) psi</th>
<th>Allowable Pressure According to Clarkson's Criterion psi</th>
<th>Conservative Estimate for Ultimate Pressure Load psi</th>
</tr>
</thead>
<tbody>
<tr>
<td>Coast Guard Cutter</td>
<td>103 x 24 x 0.44 (reinforced plate)</td>
<td>Mild Steel</td>
<td>96</td>
<td></td>
<td>200</td>
<td>1900</td>
</tr>
<tr>
<td>Destroyer A</td>
<td>24 x 26 x 0.44</td>
<td>High Tensile Steel</td>
<td>38**</td>
<td>115**</td>
<td>220</td>
<td>1950</td>
</tr>
<tr>
<td>Destroyer B</td>
<td>21 x 24 x 0.37</td>
<td>Mild Steel</td>
<td>20</td>
<td>63</td>
<td>102</td>
<td>1250</td>
</tr>
<tr>
<td>Cruiser</td>
<td>46 x 41 x 0.50</td>
<td>Mild Steel</td>
<td>30</td>
<td>35</td>
<td>70</td>
<td>860</td>
</tr>
<tr>
<td>Liberty Ship</td>
<td>27 x 36 x 0.70</td>
<td>Mild Steel</td>
<td>38</td>
<td>104</td>
<td>150</td>
<td>1760</td>
</tr>
<tr>
<td>Cargo Vessel</td>
<td>27 x 32 x 0.81</td>
<td>Mild Steel</td>
<td>39</td>
<td>117</td>
<td>180</td>
<td>2000</td>
</tr>
</tbody>
</table>

*Based on the information contained in Reference 8, the actual ultimate load will be somewhat greater than this.

**These values were obtained by multiplying the values obtained from Hovgaard's curves by 50,000 psi.

STRENGTH ESTIMATES FOR TYPICAL BOTTOM PLATES OF SELECTED SHIPS

Experience has shown that the forward bottom plating in many ships has either been dished in or has failed completely. It is therefore important to be able to predict what load the plate can withstand before onset of large permanent sets or failure. Table 5 gives some information on allowable pressures for the panels considered in Table 4.

DISCUSSION OF RESULTS

Examination of Table 4 indicates that high stresses can be developed in ship plating due to pressures incident to slamming of the order of magnitude measured on Coast Guard cutters. However, panels of ship bottom plating can withstand much higher pressures than those experienced by the UNIMAK before undergoing large plastic deformations or failure, as can be seen from Table 5.

Theory predicted that, for the UNIMAK test panel, pressures of the order of 100 psi would give stresses of the order of the yield stress at the middle of the long edge of the panel, while the same pressure would give only around 150 to 200 $\mu$in/in. at the point where the strain was measured. The tests verify this.

The times of rise and durations of the pressures were estimated to be greater than the fundamental period of the test panel so that the load could be approximated as static for the computations of strains. In these theoretical calculations the results of assuming that the
pressure is uniformly distributed and statically applied gives reasonably good comparison between theory and experiment, in view of the simplifying assumptions that were used in calculating the theoretical strain.

Since only one ship was tested, it is not possible to generalize the results to other types of ships; however, one outstanding factor can be emphasized – the impact pressures that can be expected at the bottom plating of the bow during slamming are of the order of several hundred pounds per square inch.

**CONCLUSIONS**

1. Pressures acting on the ship's bottom forward, incident to slamming, are of the order of one hundred to several hundred pounds per square inch. The times of rise and durations of this pressure are sufficiently long, at least in the case of the UNIMAK, so that the pressure may be considered to act statically for the purpose of simplifying the computation of the maximum resulting stresses and deflections.

2. The distribution of the pressure induced by slamming is such that only a relatively limited area of bottom plating is usually exposed to the large pressures at any one time. This is illustrated by the fact that relatively low pressures were measured on Gages 6 and 8 (see Figures 1 and 3) when high slamming pressures occurred on Gages 2, 3, 4, and 5. Thus high local loads and stresses may be induced without necessarily inducing large hull girder stresses. The maximum measured slamming pressures were always greater near the keel (Gages 2, 3, 4) than near the turn of the bilge (Gage 6), during any given slam.

3. Bottom panels in the area subject to slamming could be designed to experience stresses within the elastic limit for most slam-induced pressures or they could be designed for a given set.

4. For present bottom designs it is expected that plastic deformation incident to slamming occurs often; however, no perceptible permanent set occurred during the UNIMAK tests.

**ACKNOWLEDGMENTS**

The UNIMAK sea trials were conducted under the supervision of Mr. J.T. Birmingham of the Vibrations Division at the David Taylor Model Basin. The untiring and enthusiastic efforts of Mr. W.E. Smith of the Instrumentation Division did much to make these trials a success. Mr. Mills Dean, III also of the Instrumentation Division, was responsible for the difficult strain gage installations. The overall program of tests and analysis was conducted under the direction of Mr. N.H. Jasper.

The data on which this report is based could not have been obtained without the splendid cooperation received from the U.S. Coast Guard, especially its Naval Engineering Division, the Coast Guard Yard in Baltimore, Maryland, and the Captain, CDR Maynard F. Young, of the USCGC UNIMAK.
APPENDIX A

THEORY

A brief derivation of the formula for the strain at any point in a panel is given here. A more complete explanation of the theory is given in Reference 3.

STRAIN AT ANY POINT IN EITHER OF THE COORDINATE DIRECTIONS

Figure 6 - Sketch of Plate

To derive an expression for the static strain at any point the expressions for the stresses in the coordinate directions will be considered as a starting point. These stresses are

\[ \sigma_x = \frac{6 M_x}{h^2}, \quad \sigma_y = \frac{6 M_y}{h^2} \]  \[2\]

where \( M_x \) and \( M_y \) are bending moments per unit length and \( h \) is the thickness of the panel. The expressions for the bending moments are (see Reference 3, pages 16 and 17).

\[
M_x = \sum_{i=1}^{\infty} \sum_{j=1}^{\infty} D (X_i'' Y_j + \nu X_i Y_j'') \frac{P_0 a^2}{\rho h} \int_0^b \int_0^b (X_i Y_j)^2 dx dy \]  \[3\]

\[
M_y = \sum_{i=1}^{\infty} \sum_{j=1}^{\infty} D (X_i'' Y_j + \nu X_i'' Y_j) \frac{P_0 a^2}{\rho h} \int_0^b \int_0^b (X_i Y_j)^2 dx dy \]  \[4\]

where \( D = \frac{E h^3}{12 (1-\nu^2)} \), the plate modulus or plate stiffness,

\[
p_{ij} = \frac{K_{ij}}{a^2} \sqrt{\frac{D}{\rho h}} \]  \[5\]

the natural frequency of the \( ij^{th} \) mode (in a vacuum),

\( \rho h \) is the mass per unit area of the plate material,
\( b \) is the length of the plate,
\( a \) is the width of the plate,
\( P_0 \) is the magnitude of the uniformly distributed pressure,
\( \nu \) is Poisson's ratio,
\( E \) is the modulus of elasticity, and 
\( h \) is the plate thickness.

Also
\[
X_i'' = \frac{d^2 X_i}{dx^2}
\]
where \( X_i \) is a function of \( X \) alone and is the mode function for a clamped-clamped beam, and
\[
Y_j'' = \frac{d^2 Y_j}{dy^2}
\]
where \( Y_j \) is a function of \( y \) alone and is also the mode function for a clamped-clamped beam.

Values of \( X_i, Y_j, X_i'', Y_j'', \int_0^a X_i \, dx, \int_0^b Y_j \, dy, \int_0^a X_i^2 \, dx, \) and \( \int_0^b Y_j^2 \, dy \) are tabulated in References 3, 9, and 10.

The expressions for the strains in terms of stresses are as follows:
\[
\varepsilon_x = \frac{1}{E} [\sigma_x - \nu \sigma_y]
\]
\[
\varepsilon_y = \frac{1}{E} [\sigma_y - \nu \sigma_x]
\]
Hence
\[
\varepsilon_y = \frac{1}{E} \left[ \frac{6 M_j}{h^2} - \frac{6 M_x}{h^2} \right] = \frac{6}{E h^2} [M_y - \nu M_x]
\]
Substituting in the above equations the values for \( M_x \) and \( M_y \) given in [3] and [4], respectively, the strain in the \( y \)-direction for the \( ij \)th term is
\[
(\varepsilon_y)_{ij} = \frac{6(1-\nu^2)}{E h^2} X_i Y_j'' \frac{P_0}{K_{ij}^2} \frac{a^4}{\int_0^a \int_0^b (X_i Y_j)^2 \, dx \, dy}
\]
where \( K_{ij} \) is tabulated in Table 1 of Reference 3 for the first four symmetrical modes of a clamped plate. So
\[
(\varepsilon_y)_{ij} = \frac{6(1-\nu^2)}{E h^2} X_i \frac{\beta_i^2}{b^2} \left( \frac{b^2}{\beta_j} Y_j'' \right) \frac{P_0}{K_{ij}^2} \frac{a^4}{\int_0^a \int_0^b (X_i Y_j)^2 \, dx \, dy}
\]
or
\[
(\varepsilon_y)_{ij} = \frac{P_0 a^2}{E h^2} \left[ 6(1-\nu^2) \left( X_i \frac{b^2}{\beta_j^2} Y_j'' \right) \frac{\beta_j^2}{\beta_i^2} \frac{b^2}{a^2} \frac{\int_0^a \int_0^b (X_i Y_j)^2 \, dx \, dy}{\int_0^a \int_0^b X_i^2 Y_j^2 \, dx \, dy} \right]
\]
*This value is tabulated for different modes in References 3 and 9.*
and

\[ \varepsilon_y = \sum_i \sum_j (\varepsilon_y)_{ij} \]  \hspace{1cm} [7]

where \( \beta_i, \beta_j \) are constants for a given mode; see Reference 3. The calculations for the mode corresponding to \( i = 1, j = 1 \) are given as an example. From Table 1 of Reference 3 (corresponding to \( \frac{b}{a} = 2.23 \))

\[ K_{ij} = K_{11} = 24.10 \]

From Table 5 of Reference 3

\[ \int_0^a X_i \, dx = \int_0^a X_1 \, dx = 0.8309 \, a \]
\[ \int_0^a X_i^2 \, dx = \int_0^a X_1^2 \, dx = a \]
\[ \int_0^b Y_j \, dy = \int_0^b Y_1 \, dy = 0.8309 \, b \]
\[ \int_0^b Y_j^2 \, dy = \int_0^b Y_1^2 \, dy = b \]

\[ X_i = [X_1] \text{ evaluated at } x = \frac{a}{2} = 1.5882 \]

and from Table 1 of Reference 8

\[ \frac{\beta_j^2}{\beta^2} Y_j'' = \frac{\beta_j^2}{\beta_1^2} Y_1'' = [\phi_1]^* \text{ evaluated at } \gamma = 0.06b = 1.4428 \]

The values for the term in the brackets are as follows for \( X = 0.5 \, a, \ y = 0.06 \, b^{**} \)

<table>
<thead>
<tr>
<th>Value for term in brackets</th>
<th>Value for term in brackets</th>
</tr>
</thead>
<tbody>
<tr>
<td>( i = 1 ) ( j = 1 )</td>
<td>( i = 1 ) ( j = 1 )</td>
</tr>
<tr>
<td>( i = 1 ) ( j = 3 )</td>
<td>( i = 3 ) ( j = 3 )</td>
</tr>
<tr>
<td>( i = 1 ) ( j = 5 )</td>
<td>( i = 3 ) ( j = 3 )</td>
</tr>
</tbody>
</table>

Hence an approximate expression for the strain at Gage F is

\[ \varepsilon_y = 0.095 \frac{P_0 \, a^2}{E \, h^2} \]

*According to the notation of Reference 8.

**This is the location of Gage F.
APPENDIX B

COMPARISON OF COMPUTED AND MEASURED TIME VARIATION OF PLATE STRAIN

In the body of the report the maximum strain only was computed theoretically and compared with the measured strain. In this section the time history of the strain obtained in a selected slam is compared with the time history computed from theoretical considerations. The assumptions used in computing the theoretical time history are as follows:

1. The elastic small deflection theory which takes into account bending stresses only can be used.
2. The plate has all edges clamped.
3. The average pressure is uniformly distributed over the panel. This average pressure is calculated at any instant of time by taking the average of the pressures on Gages 2, 3, 4, and 5 at that time.

From Reference 3 it can be seen that the strain at Gage F due to a uniformly distributed load can be written as follows:

\[
\varepsilon_F = \sum_m (\varepsilon_{\text{static}})_m \times (R_m) \quad \text{(8)}
\]

where

\[
(R_m) = \frac{1}{p_m} \int_0^t F(\tau) \sin p_m (t - \tau) \, d\tau
\]

From Equations [7] and [8] the strain at Gage F can be written as:

\[
\varepsilon_F = \frac{P_0 a^2}{E h^2} \left[ 0.0690 \int_0^t F(\tau) \sin p_{11} (t - \tau) \, d\tau 
+ 0.0280 \int_0^t F(\tau) \sin p_{13} (t - \tau) \, d\tau 
+ 0.0001 \int_0^t F(\tau) \sin p_{15} (t - \tau) \, d\tau 
- 0.0010 \int_0^t F(\tau) \sin p_{31} (t - \tau) \, d\tau 
- 0.0009 \int_0^t F(\tau) \sin p_{33} (t - \tau) \, d\tau \right]
\]
From the experimental data obtained during the tests it is believed that the slamming pulse can be approximated by an asymmetrical triangle as follows:

\[ F(t) = 2p_0 \frac{t}{\alpha t_0} \quad 0 < t < \alpha t_0 \]

\[ = 2p_0 \frac{t_0-t}{(1-\alpha)t_0} \quad \alpha t_0 \leq t \leq t_0 \]

![Figure 7 - Triangular Pulse](image)

From Reference 11 the response factor

\[ R_m = p_m \int_0^t F(\tau) \sin p_m (t-\tau) \, d\tau \]

can be obtained. The response is as follows:

\[ R_m = \frac{2}{\alpha p_m t_0} (p_m t - \sin p_m t) \quad \text{For } 0 < t < \alpha t_0 \]

\[ R_m = \frac{2}{(1-\alpha) p_m t_0} [\alpha (p_m t_0 - p_m t) - (1-\alpha) \sin p_m t + \sin (p_m t - \alpha p_m t_0)] \]

\quad \text{for } \alpha t_0 < t < t_0 \]

\[ R_m = \frac{2}{\alpha (1-\alpha) p_m t_0} [\sin (p_m t - \alpha p_m t_0) - (1-\alpha) \sin p_m t - \sin (p_m t - p_m t_0)] \]

\quad \text{for } t > t_0 \]

\[ \alpha \neq 0, \quad \alpha \neq 1 \]

The strain-time curve has been calculated for Record 4104 B (see Figure 3). The theoretical strain-time curve for the strain in Gage F is compared with the measured strain-time curve in Figure 8, assuming that the average pressure-time curve can be approximated by an asymmetrical triangular pulse (as given in Figure 7) with the following constants:

\[ 2p_0 = 60 \text{ psi} \]

\[ \alpha = 0.286 \]

\[ t_0 = 0.0658 \text{ sec.} \]

Comparison between measured and computed strain indicates rather good agreement.
REFERENCES


