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NAVY DEPARTMENT
DAVID TAYLOR MODEL BASIN

STRAINS AND MOTIONS OF USS ESSEX (CVA 9)
DURING STORMS NEAR CAPE HORN

by

Norman H. Jasper, Dr. Eng.
and
John T. Birmingham

STRUCTURAL MECHANICS LABORATORY
RESEARCH AND DEVELOPMENT REPORT

August 1958
Report 1216

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1. Strains, ship motions, and the associated sea conditions measured during an eastward passage of USS ESSEX (CVA 9) around Cape Horn in July 1957 are reported in enclosure (1). The measurements were requested by reference (a) in an effort to determine the reasons for failure of the hangar deck of TICONDEROGA (CVA 14) during a westward passage around the Horn in heavy weather.

2. The results of the measurements on ESSEX, together with the damage sustained by TICONDEROGA, indicate that the buckling strength of portions of the hangar deck is inadequate to withstand the large vibratory hull bending stresses caused by hydrodynamic forces associated with immersion of the bow flare.

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STRAINS AND MOTIONS OF USS ESSEX (CVA 9)
DURING STORMS NEAR CAPE HORN

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Norman H. Jasper, Dr. Eng.
and
John T. Birmingham

August 1958

Report 1216
NS 731-040
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ABSTRACT

Strains, ship motions, and wave heights were measured during a passage of USS ESSEX (CVA 9) around Cape Horn in very rough seas. The results of the measurements on ESSEX, together with the damage sustained by TICONDEROGA (CVA 14) during a similar passage, indicate that the buckling strength of portions of the hangar deck (uppermost strength deck) is inadequate.

The measurements show that hull stresses associated with intermittent whipping in the first flexural mode were much larger than hull-bending stresses induced at the frequency of wave encounter. The analysis indicates that the whipping motions are primarily caused by hydrodynamic forces associated with the immersion of the bow flare. The report recommends strengthening of the main-deck longitudinals and consideration of whipping stresses in ship design.

INTRODUCTION

Strains, ship motions, and the associated sea condition were measured during an eastward passage of USS ESSEX (CVA 9) around Cape Horn in July 1957. The measurements were made at the request of the Bureau of Ships in an effort to determine the reasons for failure of the hangar deck of TICONDEROGA during a westward passage around the Horn in heavy weather.

The principal areas in which TICONDEROGA had been damaged were between Frames 63 and 74 forward and Frames 96 and 115 in the midships area. The damage was greatest near the centerline of the deck.

The area between Frames 100 and 111 was selected for study on ESSEX. In addition, strains were measured on the port longitudinal at and on both sides of the expansion joint at Frame 62 in order to determine if the expansion joint causes appreciable changes in hull-girder stresses.

In this report, measurements obtained on ESSEX are presented, analyzed, and discussed in relation to the hull-girder stresses and motions, local buckling strength of the hangar deck, prediction of extreme values, and ship operation. An analysis of similitude relationships is made to estimate the variation of stresses experienced in rough seas with the size of the ship.

1References are listed on page 30.
Gage Identification

<table>
<thead>
<tr>
<th>Gage</th>
<th>Location</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>On Web, Longitudinal, 1¼ in. Below Deck</td>
</tr>
<tr>
<td>3</td>
<td>Underside of Main Deck</td>
</tr>
<tr>
<td>5</td>
<td>On Web, Longitudinal, 1¼ in. Below Deck</td>
</tr>
<tr>
<td>7</td>
<td>Underside of Main Deck</td>
</tr>
<tr>
<td>9</td>
<td>Underside of Main Deck</td>
</tr>
<tr>
<td>11</td>
<td>Underside of Main Deck</td>
</tr>
<tr>
<td>13</td>
<td>On Flange of Longitudinal</td>
</tr>
<tr>
<td>15</td>
<td>On Flange of Longitudinal</td>
</tr>
<tr>
<td>17</td>
<td>On Flange of Longitudinal</td>
</tr>
<tr>
<td>19</td>
<td>On Flange of Longitudinal</td>
</tr>
<tr>
<td>21</td>
<td>On Flange of Longitudinal</td>
</tr>
<tr>
<td>20</td>
<td>Centerline of Third Deck, 4 in. forward of Frame 102</td>
</tr>
<tr>
<td>23</td>
<td>At Center of Column</td>
</tr>
</tbody>
</table>

Figure 1 – Location of Strain Gages
INSTRUMENT INSTALLATION

Figure 1 shows the location of the strain gages on ESSEX. The first port longitudinal outboard of centerline was selected for special attention. Gages 1, 3, 5, 7, 9, 11, and 20 were intended to measure the stress* due to hull-girder flexure and to be free of local stresses, whereas Gages 13, 15, 17, 19, and 21 were to respond to the combined axial and local bending stresses. The strain gages were of the SR-4 type and were temperature-compensated. A wave meter, developed by the British Institute of Oceanography, was used to measure wave height. This meter was installed amidships, at a point 15 ft below the load waterline.

Figure 2 – Strain Gages at Station 7
Similar installations were made at Stations 3, 9, and 11.

*Whenever a stress is mentioned in this report, it refers to a stress computed from measured strains.
Roll angle, pitch angle, heave acceleration, transverse acceleration, and wave height* were also measured. Records were taken on the TMB automatic statistical recorder, on a Sanborn oscillograph, and on a Consolidated oscillograph. At least one oscillograph was recording whenever severe seas were encountered. Photographs of the gage and instrument installations are shown in Figures 2 through 4.

*The numerical wave height given in this report is the average height of the 10-percent highest waves (trough to crest).
Figure 4 – Athwartship and Vertical Acceleration and Pitch-Angle Measuring Stations
TEST RESULTS AND DISCUSSION

HULL STRESSES AND MOTIONS

Figure 5a shows oscillograms corresponding to the most severe whipping stresses recorded with the Consolidated oscillograph, and 5b and 5c the oscillograms obtained at the same time by the other two recorders. The records of Figures 5 and 6 are similar to many others obtained. Oscillograms corresponding to the maximum amplitudes of the quantities recorded on the automatic statistical recorder and Sanborn oscillograph are reproduced in Figure 7. These peak values are listed in Table 1 for ready reference. The following maximum peak-to-peak variations of ship motion were observed throughout the trip:

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Vertical acceleration</td>
<td>0.48 g</td>
</tr>
<tr>
<td>Athwartship acceleration (at Frame 144, 57.7 ft above keel)</td>
<td>0.10 g</td>
</tr>
<tr>
<td>Roll angle</td>
<td>13 deg</td>
</tr>
<tr>
<td>Pitch angle</td>
<td>9.5 deg</td>
</tr>
<tr>
<td>Wave height</td>
<td>46 ft</td>
</tr>
</tbody>
</table>

Figures 5 and 6 clearly show that, at intervals, the ship whipped in the fundamental mode of hull vibration (52 cpm), and that the associated stresses were much larger than the ordinary, more slowly varying, wave-induced stresses on which the whipping stresses are superimposed. It is also evident that these higher-frequency stresses die out very slowly,* indicating that many cycles of relatively large stress variations will be experienced by the ship during its service life.

The largest hull-girder stress variation, free of local stresses but including whipping stresses, measured amidships was 23,000 psi (Gage 3) peak-to-peak and corresponds to about 13,000-psi compression and 10,000-psi tension relative to the still-water level. The largest hull stress due to whipping in the two-noded vertical mode of hull vibration was 17,500 psi peak-to-peak (Gage 3). **

The corresponding distributions of bending moment, shear force, and amplitude of whipping vibrations calculated by the methods of vibration analysis are shown in Figure 8. Measured and calculated whipping motions amidships agree well, indicating that the values shown in Figure 8 may be considered substantially correct.

The whipping stresses are due to suddenly applied forces at the bow. If this force lasts longer than half the natural period of the fundamental hull vibration, the maximum sag stress will be equal to the ordinary wave-induced stress plus the peak-to-peak variation in whipping stress. If the force is applied as an impulse, then the maximum sag stress will be

---

*The logarithmic decrement is about 0.037.

**The wave passing the ship at this instant (wave hollow amidships, Figure 5a) had a computed length of 821 ft, an apparent length of 1028 ft, and a height of 26 ft.
TABLE 1

Maximum Measured Variations

The direction from which the waves came was about 30 deg off the bow, except where noted.

<table>
<thead>
<tr>
<th>Station</th>
<th>Quantity Measured</th>
<th>Consolidated Oscillograph</th>
<th>TNB Automatic Statistical Recorder</th>
<th>Sanborn Oscillograph</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Date/Time (Z)</td>
<td>Date/Time (Z)</td>
<td>Date/Time (Z)</td>
<td>Date/Time (Z)</td>
</tr>
<tr>
<td>1</td>
<td>11 Jul 0114Z</td>
<td>14 Jul 1427Z</td>
<td>14 Jul 1427Z</td>
<td>12 Jul 1427Z</td>
</tr>
<tr>
<td>3</td>
<td>18,000'</td>
<td>22,000'</td>
<td>18,000'</td>
<td>20,000'</td>
</tr>
<tr>
<td>5</td>
<td>16,000'</td>
<td>12 Jul* 1427Z</td>
<td>12 Jul* 1427Z</td>
<td>19,000'</td>
</tr>
<tr>
<td>7</td>
<td>15,000'</td>
<td>12 Jul* 1427Z</td>
<td>12 Jul* 1427Z</td>
<td>18,000'</td>
</tr>
<tr>
<td>9</td>
<td>5,000'</td>
<td></td>
<td>12 Jul* 1427Z</td>
<td>20,000'</td>
</tr>
<tr>
<td>11</td>
<td>17,500'</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>13</td>
<td>17,000'</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>15</td>
<td>7,500'</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>17</td>
<td>12 Jul* 1427Z</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>19</td>
<td>12 Jul* 1427Z</td>
<td>7,500'</td>
<td></td>
<td></td>
</tr>
<tr>
<td>21</td>
<td>12 Jul* 1427Z</td>
<td>7800'</td>
<td></td>
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<td>22</td>
<td>12 Jul* 1427Z</td>
<td>7800'</td>
<td></td>
<td></td>
</tr>
<tr>
<td>25</td>
<td>12 Jul* 1535Z</td>
<td>12 Jul* 1427Z</td>
<td>12 Jul* 1427Z</td>
<td>19,000'</td>
</tr>
<tr>
<td>27</td>
<td>0.31'</td>
<td>0.38'</td>
<td>0.38'</td>
<td></td>
</tr>
<tr>
<td>29</td>
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<td>46</td>
</tr>
<tr>
<td>31</td>
<td>2</td>
<td>8</td>
<td>8</td>
<td>13</td>
</tr>
<tr>
<td>33</td>
<td>7.3</td>
<td>9.5</td>
<td>9.5</td>
<td></td>
</tr>
</tbody>
</table>

*During these runs the ship was heading into the waves.

1 See Figure 1 for location of strain gages.

2 The wave height given here is the average of the highest 10 percent waves.

3 The symbol W denotes that the measured variation corresponded to the hull-whipping motion.

The symbol T denotes that the measured variation is the total peak-to-peak variation.

4 The value 28,000 psi is a fictitious stress, corresponding to the measured strain, which includes some plastic strain.
**Figure 5a — Record from Consolidated Oscillograph**

<table>
<thead>
<tr>
<th>Station</th>
<th>Quantity Measured</th>
<th>Sensitivity</th>
<th>Polarity</th>
</tr>
</thead>
</table>
| 31      | Roll-Angle        | 20 deg/in.  | Stable Down |}
| 33, 29  | Pitch Angle       | 13 deg/in.  | Bow Up   |}
|         | Wave Height       | 20 ft/in.   | Crest    |}
| 27      | Heave Accel.      | 1.0 g/in.   | Down     |}
| 25      | Horiz. Accel.     | 0.5 g/in.   | To Port  |}
| 23      | Stress            | 40 kpsi/in. | Compression |}
| 21      | Stress            | 40 kpsi/in. | Compression |}
| 20      | Stress            | 10 kpsi/in. | Compression |}
| 19      | Stress            | 10 kpsi/in. | Compression |}
| 17      | Stress            | 10 kpsi/in. | Compression |}
| 15      | Stress            | 10 kpsi/in. | Compression |}
| 13      | Stress            | 10 kpsi/in. | Compression |}
| 11      | Stress            | 10 kpsi/in. | Compression |}
| 9       | Stress            | 10 kpsi/in. | Compression |}
| 7       | Stress            | 9 kpsi/in.  | Compression |}
| 5       | Not Operating     |             |          |}
| 3       | Stress            | 10 kpsi/in. | Compression |}
| 1       | Stress            | 10 kpsi/in. | Compression |}

Computations for this wave give: Apparent length 1028 ft; Real length = 821 ft; Height 26 ft.
Figure 5b — Record from Sanborn Oscillograph

Figure 5c — Record from TMB Automatic Statistical Recorder

Figure 5 — Oscillograms Showing Severe Whipping

The identification of oscillogram traces is as follows:
Consolidated oscillograph: By Gage Number Only.
Sanborn oscillograph: 3S1 Meaning Gage 3,
Sanborn Oscillograph
Channel 1.
TMB Automatic Recorder: 19A3 Meaning Gage 19,
Automatic Recorder,
Channel 3.

All oscillograms clearly show the very intense whipping motion that occurred just prior to 0114 Z.

The environmental and operating conditions at the time of these measurements were as follows:

Time: 0114 Z (Greenwich Time), 11 Jul 1957
Waves: Coming from 240 deg

Location: 36 deg S; 74 deg W
Wind: 8 knots from 315 deg

Course: 203 deg
Wave Height, (H1/10): 26 ft

Speed: 16 knots
Direction of Waves Relative to Ship: 37 deg
Figure 6a — Oscillogram Taken at 0025 Z on 12 July 1957

The environmental and operating conditions at the time of these measurements were:

Location: 42 deg S; 76 deg W
Course: 193 deg
Speed: 12 knots
Waves: Coming from 220 deg
Wind: 31 knots from 225 deg
Wave Height: 26 ft
Direction of Waves Relative to Ship: 027 deg
Figure 6b – Oscillogram Taken at 0104 Z on 11 July 1957

The environmental and operating conditions at the time of these measurements were:

Location: 36 deg S; 74 deg W
Course: 203 deg
Speed: 16 knots
Waves: Coming from 235 deg
Wind: 12 knots from 315 deg
Wave Height: 26 ft
Direction of Waves Relative to Ship: 032 deg

Figure 6 – Oscillograms from Consolidated Oscillograph Showing Moderate Whipping
Figure 7 - Oscillograms of Most Severe Conditions Recorded on Sanborn Oscillograph and TMB Automatic Statistical Recorder

Figure 7a - Maximum Wave Height
Figure 7b - Maximum Heave Acceleration
Figure 7c – Maximum Roll Angle
Figure 7d — Maximum Pitch Angle
Figure 7e – Maximum Stress at Station 19
Figure 7f - Maximum Stress at Stations 7, 11, and 17
Figure 8 — Calculated Deflection, Shear, and Bending Moment for USS ESSEX Due to Whipping

Gage 3 was 54.69 ft above the baseline. The neutral axis is about 30 ft above the baseline. The section modulus applicable to Gage 3 is 158,000 in.²-ft.

The curves correspond to the maximum measured vibratory strains. These peak-to-peak variations could be caused by an impulse of 1600 ton-sec or a step force function of 8000 tons, both applied at the bow.

equal to the ordinary wave-induced stress plus one-half the variation in whipping stress.

Examination of the oscillograms discloses that the whipping was induced by a force whose time variation was complex but could be approximated by a step. The maximum response of the ship shown in Figure 8 could have been generated by a step of 8000 tons, which is equivalent to a pressure of approximately 8 psi acting over the forward 20 percent of the ship's length. The same peak response of the ship could have resulted from an impulse of 1600 ton-sec.

A calculation is being made of the hydrodynamic and buoyancy forces acting on the bow during the immersion and subsequent emergence of the bow flare. Preliminary results indicate a force of 8000 tons lasting about ¾ sec. A detailed description of these calculations will be published separately.

Analysis of the photograph of Figure 9 shows that, at the instant the photograph was taken, the pitch angle was practically zero and the water spray reached a height of at least
Figure 9 – USS ESSEX Entering the Roaring 40's

This photograph was taken from 07 level about 1400 GCT on 12 July 1957, shortly after crossing the fortieth parallel. (The "roaring 40's" are known throughout the maritime world as one of the roughest stretches of ocean in the world.)

The rough seas tore off the starboard bow catwalk and flattened the forward palisades shortly before this picture was taken. The height of the spray is at least 120 ft above still-water level. The palisades are about 7 ft high.

120 ft above the still-water level. An initial water velocity of 88 fps would be required to attain this height if the water was initially at the still-water level, although there is no way of knowing where and when the impact which generated the spray took place. In any case, the "slam" must have occurred 2 or more sec before the photograph was taken. It is believed that the ship was pitching upward at the time of the photograph.

Every oscillogram which was analyzed and which indicated whipping of the ship showed that, at the instant at which the peak value of the first cycle of whipping was observed amidships, the bow was nearly fully depressed and the wave hollow was amidships. Presumably, the bow had already entered the oncoming wave crest at this time. Our analytical study indicates that the maximum whipping stresses and motions were a consequence of the immersion of the bow flare. The maximum whipping response was observed for immersion in a wave 26 ft
high with apparent length of 1028 ft. The relative velocity of bow flare immersion was about 15 ft/sec. If the ship were to encounter higher waves of the same length, while making the same speed, one would expect an increase in whipping stresses and motions roughly in proportion to the wave height.

Empirical evidence suggests that the magnitudes of "whipping stresses" experienced in rough seas are larger for longer ships and for ships with bow flare. For geometrically similar ships operating in similar seas at the same speed-length ratio, Froude's scaling law suggests that the forces which cause the whipping motion increase as the length cubed, and the pressures as the length. One might therefore expect that very large ships, such as the super tankers, would be subject to large vibratory bending moments; roughly, the stress would be expected to increase as the ship's length.* (See Appendix.) A theoretical study of the effect of the ship's length and flexibility on the relative severity of whipping stresses appears worthwhile. Similarly, the effects of immersion of the bow flare on the "whipping" of ships should be investigated further. Preliminary studies indicate that the bow flare may, for certain ships, be the most important single source of hull-girder stresses.

The strain data obtained near the expansion joint (Gages 7, 9, 11) indicate that there is no appreciable localized variation in hull-girder stress.

**LOCAL BUCKLING STRENGTH OF STRENGTH DECK (HANGAR DECK)**

Detailed examination of the stresses in the longitudinal which is 3 ft 9 in. to port of centerline (Gages 3, 13, 15, 17, 19 in Figure 1) discloses that Gages 13 and 19 on the flange consistently show lower stress variations than Gage 3, which essentially responds only to the hull-girder stress.

The pattern of stress variation measured on the flange at the center of the longer spans (Gages 13 and 19) suggests that these spans have a downward eccentricity which results in adding a local bending stress to the uniform longitudinal hull-girder stress. This local bending stress is proportional and opposite in sign to the hull stress in the deck and to the total deflection at the center of the span. Thus the local bending stress will increase more rapidly than the hull stress, and a point will be reached where the local bending stress will exceed the hull stress. As a result there will be the somewhat unusual situation (at Gages 13 and 19) of increasing stress in the direction of tension when the main-deck stress is increasing compression. This analysis would explain the peculiar reversals of direction of stress observed on the records for Gages 13 and 19 at times of severe whipping. The analysis also explains the fact that the maximum vibratory stresses at these gage locations are observed several seconds later (at the time of maximum hogging bending moment) than those at gage locations (Gages 1, 3, 5, 7, 9, 11) where local stresses are negligible. It should be noted that the

---

*The assumption is that the ship structure is built to scale. In order to maintain the same stress for externally similar ships, the section modulus would have to vary as the fourth power of the length.
initial whipping motion appears to occur only when the wave hollow is near amidships (sagging) and the bow is almost fully depressed.

Examination of the strains recorded by Gage 15 at 0104 Z and 0114 Z on 11 July 1957* (Figures 6b and 5a) clearly indicates that plastic strains occurred incident to local yielding of the longitudinal at the web frame support. The zero of Gage 15 shifted an amount equivalent to a strain of 300 μin/in. The actual instant at which plastic flow occurred is shown at the beginning of the oscillogram in Figure 5a. At this instant the bow of the ship is fully depressed and the wave trough is amidships (wave crest to trough height, 26 ft). The direction of the local buckling stress (compression) is to be expected, in accordance with the general behavior of the structure described in the preceding paragraph.

It is significant to note that the locations of the largest deformations of the hangar deck in TICONDEROGA (Frames 100-104 and 107-111 centerline) are identical with those instrumented in ESSEX. Strain measurements on ESSEX indicate initial downward eccentricity and relative low buckling strength.** Deflection measurements made by ESSEX personnel indicate that the deck plating did have initial downward eccentricities, which probably occurred after the ship was built. It is suggested that tolerances be set for allowable eccentricities in stiffened panels and that the effect of these eccentricities on the buckling strength be considered in design.

The local buckling strength of the hangar-deck longitudinals in the areas between Frames 63 and 74 and Frames 96 and 115 should be increased by lowering the L/r ratio. Addition of a 12-in. by ¾-in. strip to the lower flange of the longitudinals will lower the L/r ratio to 68 (present ratio is 109) for those longitudinals with a span of four frames, and to 51 (present ratio is 82) for those with a span of three frames. This decrease in L/r ratio should increase the buckling strength of the deck sufficiently to prevent buckling failure for any foreseeable sea condition.

EXTREME VALUES

The extreme values of ship motion and wave-induced hull stress (excluding whipping stresses) to be expected for an ESSEX-Class carrier will be estimated statistically by the following formula derived in Reference 2 on the assumption of a Rayleigh distribution:

$$x_m^2 = E_m (y + \log_e N)$$

[1]

*The times given throughout this report refer to Greenwich Time.

**If the longitudinal, together with the proper width of deck plating, is considered as a column, then it is estimated that the spans between Frames 100-104 and Frames 107-111 can carry only about 60 percent of the axial load that the shorter spans between Frames 104-107 can withstand.
where

\[ x_{m_1} \] is the expected extreme value of any variable \( x \),

\[ E_m \] is the mean square value\(^*\) of \( x \) corresponding to the most severe operating conditions expected,

\[ N \] is the number of variations\(^**\) (peak-to-peak) of \( x \) expected during the life of the ship, corresponding to the most severe operating conditions, and

\[ y = \frac{x_{m_1}^2}{E_m} - \log_e N \] is the normalized variate of the extreme value distribution.

For a specified risk \( f \) of exceeding \( x_{m_1} \) one may look up the corresponding value of \( y \) in Table 1 of Reference 3 and then solve Equation [1] for the corresponding extreme value \( x_{m_1} \).

It will be assumed that the most severe operating conditions will occur 20 times during the ship’s life and that each time they will last for 4 hr. We will take \( f = 0.001 \); i.e., one chance in a thousand,\(^†\) that the estimated extreme value will be exceeded during the operating life of the ship. The corresponding value of \( y \), taken from Table 1 of Reference 3, is 7.0. Table 2 gives the corresponding estimated extreme values.

It should be noted that the bending moment due to whipping is superimposed on the still water, and the wave bending moments. The midship bending moment variations corresponding to the most severe stresses measured during the passage of a single wave, from Figure 5a, were 515,000 ft-tons (60 percent sag, 40 percent hog) for the ordinary wave stress and 1,230,000 ft-tons for the whipping stress. Assume that the bending moment variation is proportional to wave height, other conditions remaining constant; then an extreme whipping moment variation of 1,850,000 ft-tons may be expected, corresponding to a 39-ft high wave assumed for purposes of design. The design value for the ordinary wave bending moment variation is 1,600,000 ft-tons, from Table 2. The extreme midship bending moments that might be expected over the service life of these ships, incident to wave action would be of the order of

\[
\text{Sagging Moment} = 0.60 \times (1,600,000) + \frac{1}{2} (1,850,000) = 1,890,000 \text{ ft-tons}
\]

\[
\text{Hogging Moment} = 0.40 \times (1,600,000) + \frac{1}{2} (1,850,000) = 1,570,000 \text{ ft-tons}
\]

The maximum combined value of the moment, observed on 11 July (Figure 5a), was 920,000 ft-tons, sagging. This value approximates to the maximum designed sagging moment amidships.

\*\( E_m \) also is approximately equal to four times the area under the power spectrum of the variable \( x(t) \).

\**The statistical theory requires that these be \( N \) independent variations. The requirement of independence is not strictly satisfied here. This will result in a slight overestimate of the extreme value.

\†This is a nominal value of the risk taken. The Rayleigh distribution cannot be theoretically valid for the prediction of very extreme values. However, its use will probably yield fairly realistic estimates of the expected extreme values.
TABLE 2
Maximum Values of Ship Motions and Longitudinal Bending Moments for ESSEX-Class Carriers

All values given refer to peak-to-peak variation.

<table>
<thead>
<tr>
<th>Quantity</th>
<th>Ship</th>
<th>Location of Test Area</th>
<th>Conditions for Which Extreme Value Is Predicted (2)</th>
<th>Number of Variations per 4-hr Period</th>
<th>Number of Variations Expected During Operating Life of Ship Corresponding to $F_m$</th>
<th>Maximum Expected Value in One Storm (4-hr Operation)</th>
<th>Maximum Expected Value of Ship (C = 0.031)</th>
<th>Largest Measured Variation (3)</th>
<th>Maximum Variation for Design Purpose</th>
</tr>
</thead>
<tbody>
<tr>
<td>Roll Angle</td>
<td>ORISKANY</td>
<td>Cape Horn</td>
<td>&gt; 15 Quarter Head</td>
<td>61.0 deg$^2$</td>
<td>980</td>
<td>19,500</td>
<td>19.6 deg</td>
<td>32 deg</td>
<td>19 deg</td>
</tr>
<tr>
<td>Pitch Angle</td>
<td>ESSEX</td>
<td>Cape Horn</td>
<td>&gt; 24 Quarter Head</td>
<td>12.8 deg$^2$</td>
<td>1180</td>
<td>23,700</td>
<td>9.5 deg</td>
<td>14.9 deg</td>
<td>9.5 deg</td>
</tr>
<tr>
<td>Heave Acceleration</td>
<td>ORISKANY</td>
<td>Cape Horn</td>
<td>&gt; 20 Head</td>
<td>0.014 g$^2$</td>
<td>1350</td>
<td>27,000</td>
<td>0.32 gravity units</td>
<td>0.49 gravity units</td>
<td>0.3 g (USS VALLEY FORGE)</td>
</tr>
<tr>
<td>Longitudinal Bending Stress</td>
<td>VALLEY FORGE</td>
<td>North Atlantic</td>
<td>&gt; 18 Head</td>
<td>28.2 (kpsi)$^2$</td>
<td>1440</td>
<td>28,800</td>
<td>14.5 kpsi</td>
<td>22.3 kpsi</td>
<td>12.2 kpsi</td>
</tr>
<tr>
<td>Longitudinal Bending Moment Due to Whipping</td>
<td>VALLEY FORGE</td>
<td>North Atlantic</td>
<td>&gt; 18 Head</td>
<td>0.156 x 10$^{12}$ ton$^2$ft$^2$</td>
<td>1440</td>
<td>28,800</td>
<td>1,070,000 ft-tons</td>
<td>1,640,000 ft-tons</td>
<td>920,000 ft-tons</td>
</tr>
<tr>
<td>Bending Moment Due to Whipping</td>
<td>ESSEX</td>
<td>Cape Horn</td>
<td>&gt; 20 Quarter Head</td>
<td>-</td>
<td>-</td>
<td>52 cycles per minute when whipping occurs</td>
<td>-</td>
<td>-</td>
<td>1,230,000 ft-tons</td>
</tr>
</tbody>
</table>

(1) This is the average height of the larger, well-defined waves, as determined by visual observations.

(2) These are the conditions under which the largest values recorded at any time were obtained (peak-to-peak variation).

(3) These are the largest values recorded throughout seaworthiness tests on carriers, and cover about 2 years operation at sea.

(4) Stress, c.l. main deck, amidships. This is the ordinary wave-induced stress free of whipping stresses. The applicable section modulus = 167,000 ft-in$^2$.

(5) This bending moment is superimposed on the ordinary bending moment. The bending moments were computed from the stress by use of the design midship section moment of inertia, as calculated by Bureau of Ships.

* These values are estimated on the assumption that all variations are independent. This assumption is not strictly valid and results in a slight overestimate of the extreme value.
(BuShips). It should be noted that the bending moments are computed on the basis of the design midship area moment of inertia. It is believed that the design bending moment, for similar ships, should vary as the fourth power of the scale ratio if it is desired to maintain the same maximum sea-induced stresses.

![Graph showing hull stresses, wave heights, and ship speeds measured during passage around Cape Horn.]

Figure 10 – Hull Stresses, Wave Heights, and Ship Speeds Measured During Passage Around Cape Horn

Speed ordinate is obtained from Bell Log RPM data.
Figure 11 - Route of USS ESSEX for Voyage Around Cape Horn in July 1957

The ship's position is plotted for the period of rough weather. The wave height is listed for times (GCT) and dates shown on the chart.
SHIP OPERATION

Figure 10 is a plot of hull-girder stresses, wave heights, and ship speeds against time, corresponding to the most severe sea condition encountered on the route shown in Figure 11. It is evident that the commanding officer generally decreased the ship's speed as the wave heights increased, and vice versa, as was to be expected. It is likely that sustained operation of the ship in head seas at speeds somewhat greater than about 17 knots, during the period 1400-1800 on 12 July 1957, would have resulted in buckling of the hangar deck, similar to that on TICONDEROGA.

The ordinary wave-induced stress variations were not as sensitive to the sea and operating conditions as were the whipping stresses. It is noted that the maximum whipping stress was often much larger than the average of the largest stresses measured for the ten most severe occasions of whipping, the ratio of the maximum to the average being as large as 2.54. The ratio of the maximum ordinary wave-induced stress to the average of the ten highest wave stresses was only as high as 1.7, and the average value of this ratio was 1.23. The value 1.23 is also that which one would expect if the stress variations have a Rayleigh frequency distribution. The largest whipping stress occurred during operation at 16½ knots in heavy head seas. Although the ship speed was well within the range of Froude numbers for which bottom damage due to slamming is, according to Lehmann, likely to be encountered, the bottom of the bow did not emerge from the sea, as far as could be ascertained.

Examination of Figure 10 suggests restriction of the ship's speed to about 14 knots for operation in head or quarter head seas when the height of the average of the 10 percent highest waves* exceeds 20 ft. This restriction should be subject to change at the discretion of the commanding officer.

CONCLUSIONS AND RECOMMENDATIONS

1. The stresses incident to whipping of the ESSEX-Class hull in its fundamental mode are much larger than the ordinary wave-induced stresses. These vibratory stresses should be considered in hull design since they may be larger than any other single source of stress for long, limber ships.

2. There is empirical evidence to indicate that variations in bending stress incident to whipping become larger as ships become larger. For ships of the ESSEX class, allowance should be made in design for whipping bending moments of approximately \( \frac{1,250,000}{95} \) ft-tons. This value is about 95 percent larger than the maximum designed sagging moment. A theoretical study is needed to determine the effect of length and flexibility on the relative severity

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*This wave height is roughly comparable with the "average of the higher well-defined waves" as reported by the ship's aerological observers. Alternatively it would be a relatively simple problem to design and construct an indicating or recording meter which would give a reading of the "average of the 10 percent highest waves." Such a meter could then be installed.
of whipping stresses to be expected in heavy seas. Some effects of ship size on hull stresses can be studied by similitude relationships, as shown in the Appendix.

3. The bow forces necessary to cause the measured whipping stresses are unusually severe. A force of 8000 tons suddenly applied at the bow, and maintained for a duration longer than one-half second, would result in the measured whipping stresses. It can be shown that a mechanism generating these forces is associated with immersion of the bow flare.

4. Unfairness of the deck and supporting longitudinals may cause considerable additional local stresses in the longitudinals and contribute to premature buckling under whipping loads. It is recommended that limits of allowable eccentricity for stiffened plating be established and that these limits be included in the specifications for the design and construction of the hull structure. When tests or calculations are made to determine the buckling strength of stiffened panels, consideration should be given to the adverse influence of these allowable eccentricities.

5. The longitudinal buckling strength of the main deck near the centerline between Frames 100 and 111 and Frames 63 and 74 is inadequate. This is particularly true of the sections between Frames 63 and 67, 70 and 74, 100 and 104, and 107 and 111.

6. It is recommended that the longitudinals in the affected areas on ESSEX-Class ships be reinforced by adding additional material to the flanges, as suggested in this report.

7. The failure of the deck on TICONDEROGA is believed to have been due to insufficient buckling strength of the deck to cope with whipping stresses caused by heavy seas.

8. The expansion joint at Frame 62 does not introduce appreciable localized variation in hull-girder stress.

PERSONNEL

The reported measurements were made by J.T. Birmingham, assisted by J.L. Leahy, both of the TMB staff. The instruments were installed with the assistance of members of the Instrumentation Division and Trials Branch of the Taylor Model Basin. Planning and analysis were carried out under the direction of Dr. N.H. Jasper of the Model Basin. Captain E.R. Eastvold, Commanding Officer, Commander P.N. McDonald, Engineering Officer, and Commander J.G. Daniels, Executive Officer, all of ESSEX, gave their wholehearted cooperation to the program and thus made possible the successful results.
APPENDIX

SIMILITUDE CONSIDERATIONS FOR HYDRODYNAMICALLY EXCITED HULL STRESSES

It is of interest to determine the manner in which the maximum hydrodynamic forces and the resulting hull stresses vary with the length of the ship. To this end the method of similitude will be applied.

Assume that geometrically and structurally similar ships will operate in similar seas* at the same speed-length ratio as required by Froude's scaling law. Assume furthermore that viscous and compressibility effects may be neglected.

Let \( L \) be the length of the ship, \( F \) the hydrodynamic force acting on the ship, \( \tau \) a characteristic time interval which describes the time variation of the hydrodynamic force, \( p \) the hydrodynamic pressure, \( \sigma \) the hull bending stress, \( T \) the natural period of elastic vibration, and \( y \) the elastic deformation of the ship's neutral axis.

Then in accordance with Froude's law

\[
F = L^3; \quad p = L; \quad \tau = \sqrt{L}
\]

The response of the elastic ship to the hydrodynamic forces can be derived by consideration of the laws of mechanics and the laws of structural similitude.

The maximum elastic deformations of an elastic ship of mass \( M \) subject to an instantaneous impulse \( H \) and to a step force \( F \) are as follows:

For an impulse, \( y_H = \frac{HT}{M} \)

For a step force, \( y_S = \frac{FT^2}{M} \)

Structural similitude assures that \( T = L; \quad M = L^3 \).

The hull-bending stresses are \( \sigma = \frac{Ecd^2y}{L} = \frac{cd^2y}{dx^2} \), for constant \( E \).

*The probability of encountering "similar" severe wave configurations is, of course, much less for the larger ships.
The duration of the applied load is assumed to be small compared to the natural period of vibration. 

For the impulsive load

\[ F = \int Fd\tau \approx L^3 \sqrt{\frac{t}{T}} = L^{7/2} \]

\[ y_H = \frac{HT}{M} = L^{7/2} \frac{L}{L^3} \approx L^{3/2} \]

The elastic stress* is given by

\[ \sigma_e = \frac{cd^2y}{dx^2} \frac{L}{L^2} \approx L^{3/2} \]

For the step load

\[ F = L^3 \]

\[ y = \frac{FT^2}{M} = L^3 \frac{L^2}{L^3} \approx L^2 \]

The elastic stress* is given by

\[ \sigma_e = \frac{cd^2y}{dx^2} \frac{L}{L^2} \approx L \]

For static loads (ordinary wave-induced forces may fall into this category)

\[ F = L^3 \]

The stress associated with the rigid body motion is

\[ \sigma_r \approx L \]

Thus it has been shown that the wave-induced stresses in similar ships, operating at the same speed-length ratio in similar seas, vary as the length of the ship for a suddenly applied step load, and as the square root of the length for an instantaneously applied impulsive load. The ordinary, slowly varying, wave-induced stresses may be expected to vary as the length of the ship. It is to be noted that this analysis neglects the hydrodynamic forces associated with compressibility effects (shock waves in water). These may make an appreciable contribution. The stresses arising from the latter effects may be expected to vary as \( L^{3/2} \) for operation of similar ships at the same Froude number.

Let it be assumed that only those stresses are experienced which, according to the above analysis, vary as \( L \). Then the stress can be kept invariant with \( L \) by making the section modulus vary as \( L^4 \).

---

*There also will be a stress which is associated with the rigid body motion of the ship. This stress will vary as \( L \).
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3. Flight decks - Strains, ship motions, and wave heights were measured during a passage of USS Essex (CVA 9) around Cape Horn in very rough seas. The results of the measurements on Essex, together with those of the damage sustained by Ticonderoga (CVA 14) during a similar passage, indicate that the whipping motions are primarily caused by hydrodynamic forces associated with the impact of the bow flare. The report recommends strengthening of the main-deck longitudinal and consideration of whipping stresses in ship design.

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3. Ship hulls – Stresses – Trials
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The measurements show that hull stresses associated with intermittent whipping in the first flexural mode were much larger than hull-bending stresses induced at the frequency of wave encounter. The analysis indicates that the whipping motions are primarily caused by hydrodynamic forces associated with the immersion of the bow flare. The report recommends strengthening of the main-deck longitudinals and consideration of whipping stresses in ship design.

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