RESIDUAL STRESSES OF WELDED STRUCTURES AND THEIR EFFECT ON THE DYNAMIC BEHAVIOUR

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ABSTRACT

The vibration behaviour of local ship structures is highly influenced by residual stresses. In this paper, the experimental investigations necessary to validate an updating procedure of the dynamic FE-model of welded structures are presented. Extensive experimental investigations of several ship decks, built by different shipyards, were accomplished. For this purpose, welding stress measurements by strain gauges, residual stress measurements by hole-drilling and a modal analysis were accomplished. To account for the bandwidth of usual production techniques, different weld processes with varying energy input per unit length were used. The residual stresses due to welding were measured during the different production stages of the structures so that the origin of these stresses can be obtained. By using rosettes attached vice versa to the top-plate at different locations on the structures it was possible to determine the principal normal membrane stresses and the corresponding angles. These values correspond with the welding technique. These rosettes are also used to determine the overall residual stresses by means of the hole-drilling technique after full assembly of the structures. The results are at the same level as the measured stresses due to welding. The natural frequencies and mode shapes were gained by an experimental modal analysis. Additionally one structure was stress relief annealed and the dynamic behaviour was investigated before and after the treatment. The results differ to a great extent although the structures are geometrically very similar or even the same.

Introduction

The ship building industry is – among others – characterized by an increasing demand for comfort. This applies especially for cruise liners but also for cargo ships. Therefore, the vibration behaviour is of increasing interest. The vibration behaviour of local ship structures is highly influenced by residual stresses and predeformations. Measurements have shown that differences of up to 50% in natural frequencies are possible. Uncertainties of this magnitude involve a high risk regarding the resonance free operation of ships. This implies an economic threat in case of failure to fulfill contracts. Therefore an improvement of the prediction of local vibration behaviour is needed. Welding stresses have an essential influence on the dynamic behaviour of thin structures like ship decks. The additional nonlinear stiffness, induced by the welding stress, can change the modal parameters considerably. Despite this fact, welding stresses are mostly neglected during the design process of ships. The usually claimed frequency clearance of 20 % between natural frequencies and exciting frequencies cannot be assured neglecting the welding stresses. Although the computation of welding stresses is possible by a detailed thermal finite element model, this remains a time consuming and cost intensive task and is difficult to accomplish for most practical applications. Consequently, a simplified model to predict the influence of welding stresses to modal parameters is aimed. For this purpose, an inherent strain model is used. The parameters of this model are gained by a finite element model updating, using the natural frequencies and mode shapes of welded structures. To gain the necessary inherent strain parameters a modal analysis of structures similar to ship decks was worn out. The consistency of the inherent strain model - i. e. the capability of the model to reflect not only the dynamic behaviour but also the residual stress distribution - has to be validated. For this purpose residual stress measurements by the hole drilling method and by strain measurements during production have be worn out. Furthermore, an experimental modal analysis of the ship decks was accomplished to evaluate natural frequencies and mode shapes for updating purposes. A second experimental modal analysis was undertaken on one structure after stress relief annealing.
Experimental Results

In the following the results from three ship deck structures are presented. The geometry of the investigated panels was chosen according to typical superstructure decks. The dimensions of the three structures are similar (Figure 1). The structures consist of a 6.3 to 7mm thick top plate stiffened by bulge profiles. These parts were joined using continuous weld seams and chain intermittent weld seams by MAG welding. The remaining parts were joined by continuous weld seams by MAG welding.

![Figure 1: Drawing of the ship deck section](image)

Welding Stress Measurements

The residual stresses of a butt welded plate are caused by the shrinking of the welding seam. A characteristic residual stress distribution (longitudinal stresses) induced by welding, consists of high longitudinal tension stresses across the welding seam. Starting from the welding seam, the stresses decrease rapidly and switch over to compression stresses. The unsupported plate builds up a free transverse elongation, so that the transverse stresses are negligible.

![Figure 2: Used strain gauge](image)

Several strain gauges (see Figure 2) were applied double-sided to the top plate of the panels before assembly. As demonstrated in Figure 3 two different types of distribution were used.
During the welding of the stiffeners and the frame the strains were measured. Using these strains, the residual stresses caused by welding can be computed directly avoiding the impact of residual stresses not caused by welding. To determine the stresses, the strain gauges were attached at the same locations on opposite sides of the top plates, so that the resulting stresses can be divided into bending and membrane stresses. Because the bending component of the stresses has – apart of its influence on the predeformation – no influence on the dynamic behaviour, it is not reviewed in this paper.

The strains were evaluated after the welding of the profiles and after the welding of the frames. By this procedure the overall welding stresses can be divided into profile welding and frame welding induced stresses. The resulting minimal principal stresses can be found in Figure 4. As can be seen, there are compression stresses on every measuring point on the plate field of the structure.

The stiffeners of structure 1 were attached by a chain intermittent weld. A continuous weld seam was applied to structures 2 and 3. All welds were produced using metal active gas welding. The inserted weld energy (compare Table 1) was evaluated by measuring the weld voltage, current and velocity. Due to the different locations of the measuring points, the residual stresses cannot be compared directly. Nevertheless, the impact of the weld parameters on the residual stress can be seen very clearly. After welding the stiffeners, structure 2, yielding the highest energy input, shows the largest residual stresses.

The welding of the frame has a significant effect on all structures and leads to a further increase of the compression stresses.

| Table 1: Weld energies for the investigated structures due to the different welds |
|-----------------------------------------------|---------|---------|---------|
| Structure                                | 1       | 2       | 3       |
| Weld energy stiffeners          | 8,0 MJ (chain intermittent weld) | 27,1 MJ | 19,8 MJ |
| Weld energy frames                | 17,8 MJ | 21,6 MJ | 27,3 MJ |
| Weld energy overall              | 25,8 MJ | 48,7 MJ | 47,1 MJ |
Residual Stress Measurements

Nowadays, a well established experimental technique to evaluate the mean stresses in steel constructions is the hole-drilling method developed by Mathar in 1933. This technique uses a small, drilled hole at the surface of the specimen to allow a partial relaxation of the eigenstresses. While drilling the hole, the released strains are measured by an electric strain-gauge in three radial directions. By using the strain differences it is possible to calculate the eigenstresses before drilling.

Using a rectangular rosette with strain gauge angles of $0^\circ$, $45^\circ$ ($225^\circ$) and $90^\circ$ ($\varepsilon_a$, $\varepsilon_b$, $\varepsilon_c$) the angle of the principal stress are calculated by the equation:

$$\varphi = \frac{1}{2} \arctan \frac{\Delta \varepsilon_a + \Delta \varepsilon_c - 2 \Delta \varepsilon_b}{\Delta \varepsilon_c - \Delta \varepsilon_a}$$

(1)

where $\Delta \varepsilon_i = \text{measured change of the strain in the respective direction}$

The principal stress magnitudes are computed from equation (2) to (4).

$$\sigma_{1,2} = -\frac{E}{4A} (\Delta \varepsilon_a + \Delta \varepsilon_c) \pm \frac{E}{4B} \sqrt{\left(\Delta \varepsilon_c - \Delta \varepsilon_a\right)^2 + \left(\Delta \varepsilon_a + \Delta \varepsilon_c - 2 \Delta \varepsilon_b\right)^2}$$

(2)

using $E = \text{Young's modulus}$

$A, B = \text{constants, see below}$

$$A = \frac{a^2 (1 + \nu)}{2 r_i r_a}$$

(3)

and

$$B = \left(\frac{2 a^2}{r_a r_i} \right)^2 \left[1 - a^2 (1 + \nu) \left( \frac{r_i^2 + r_a^2 + r_i^2}{4 r_i^2 r_a^2} \right) \right]$$

(4)

where $a = \text{hole diameter}$

$r_a = \text{outer radius of the strain gauge}$

$r_i = \text{inner radius of the strain gauge}$

The residual stress measurements were accomplished after full assembly of the structures.

The strain gauges used for the hole-drilling method were the same as used for measuring the welding stresses. The type FRS-3 from TML® consists of three single gauges, length 3 mm and width 2.6 mm, in $0^\circ$, $45^\circ$ ($225^\circ$) and $90^\circ$ direction around a centrelime diameter of 10.26 mm. The nominal resistance of each gauge is 120 $\Omega$.

In [5] it is shown that the eccentricity of the drilled hole should be within a range of 0.02 mm to avoid any corrections during calculation of the principal normal stresses. The rosettes used in this case are comparable to the investigated 3/120RY21 rosettes from HBM® [5]. Estimating a hole eccentricity of max 0.02 mm of a hole 2.8 mm in diameter results there in a stress error of less than 2 % of the maximum principal stress. This result also applies for the used rosettes.

To avoid the effort for measuring the eccentricity and recalculating the results, it was decided to develop a device consisting of a measuring microscope and an XY-stage to match the necessary low eccentricity.
The strain values are recorded on both sides of the top plate, as shown in Figure 5. The residual stresses are calculated for each side considering the influence of the opposite side on the drilled rosette and are finally averaged to gain the membrane stresses.

The results are presented in Figure 6 and Figure 7.

As expected the direction of the maximum compression stresses are parallel to the stiffeners. The magnitude of the residual stresses is similar for the different structures. Nevertheless, the influence of the welding energy input on the residual stress distribution can be seen very clearly.
Modal Analysis

The dynamic behaviour of linear structures can be fully described by its modal parameters: natural frequencies, mode deflection shapes and modal damping factors. For Finite Element Updating purposes the modal parameters of the ship deck structures were determined experimentally using Output only Modal Analysis (OMA). This technique requires no information about the excitation, which must be only random in time and space. To retrieve the modal parameters from the collected data system identification techniques in time domain or in frequency domain can be used. Two such techniques, used in the frame of this study, are the Stochastic Subspace Identification (SSI, time domain) [8] and the Frequency Domain Decomposition (FDD, frequency domain) [9]. This new techniques are well suited for closely spaced modes and have shown very good agreement with other classical and more common input-output modal analysis techniques [10][11]. However, output only techniques are faster and easier to accomplish so that preference was given to these techniques.

When comparing modal parameters of different structures, the easiest and instinctive measure is the percentile difference of the natural frequencies. However to each natural frequency there exists one typical mode deflection shape. This means that it is not enough to measure the difference in frequency. The similarity in the mode deflection shapes has to be considered, too. Usually this is done by so called Modal Assurance Criteria (MAC). The MAC value measures the correlation of two mode deflection shapes.

The structures were mounted on springs at each corner. A computational modal analysis was performed to ensure that the elastic modes relevant for ship design (2nd mode and higher) of the structure's top plate are not affected by this mounting. The measuring grid consists of 169 points. 11 rows with 13 measuring points are located on the top plate and the remaining points are located at the bottom side of the frame. This fine grid was necessary to identify the complicated mode shapes properly. Piezoelectric accelerometers with a high sensitivity (10V/g) were used. Three transducers were used as references that allow the correlation of signals of different measurement datasets.

The structure was excited using airborne sound from a subwoofer, which was moved stochastically over the top plate during the measurement (see Figure 8).
Table 2 shows the natural frequencies of the analysed structures. Using LS 1 as a reference the mode shapes of the structures were analysed using the MAC-values. The correlated mode shapes were used for mode shape ordering of LS 2 and LS 3 in the table. The mode shape number indicates the order of appearance in frequency for each structure. The first mode shape is a result of the mounting and does not occur during ship operation.

Modes with a MAC-value less than 0.4 are not considered and are represented in Table 2 with an x. This low MAC-value was chosen, because the investigated structures are very sensitive to any changes in the fields between the stiffeners. But nevertheless these modes are visually very similar. It is possible to see, that some mode deflection shapes are flipped in order (e.g. mode 2 and 3 for LS1 and LS2). Also some mode shapes of LS 2 or LS 3 could not be assigned to any of LS 1. Even though the MAC-value is that low only 13 of 20, respectively 16 of 20 mode shapes could be correlated.

<table>
<thead>
<tr>
<th>Mode Shape</th>
<th>Frequency [Hz]</th>
<th>Mode Shape</th>
<th>Frequency [Hz]</th>
<th>MAC</th>
<th>∆f [%]</th>
<th>Mode Shape</th>
<th>Frequency [Hz]</th>
<th>MAC</th>
<th>∆f [%]</th>
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<tbody>
<tr>
<td>1</td>
<td>10.53</td>
<td>1</td>
<td>11.01</td>
<td>(0.65)*</td>
<td>-4.56</td>
<td>1</td>
<td>10.92</td>
<td>(0.72)*</td>
<td>-3.70</td>
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<tr>
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<td>30.26</td>
<td>0.94</td>
<td>0.59</td>
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<td>5</td>
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<tr>
<td>6</td>
<td>38.85</td>
<td>8</td>
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<td>1.13</td>
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<tr>
<td>7</td>
<td>43.36</td>
<td>x</td>
<td>x</td>
<td>x</td>
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<td>0.88</td>
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<tr>
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<td>14</td>
<td>52.84</td>
<td>0.7</td>
<td>-9.63</td>
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</table>

* Due to the used excitation via loudspeaker natural frequencies below 15 Hz are excited poorly. Therefore the analysis of mode shape 1 is insufficient and leads to faulty MAC-values.

While the geometry of the three structures is comparable, a significant difference in the top plate thickness exists. The top plate thickness of the structures 1 and 2 is 7 mm, the thickness of structure 3 is 6.3 mm. A FE-analysis showed that this difference is negligible for a residual stress free structure. A further FE-analysis was undertaken for the case that residual stresses are present. While most mode shapes and natural frequencies are marginally affected, the effect on local mode...
shapes is remarkable. The stiffness within the plate field is subject to a major decrease induced by residual stresses. Due to
the low resulting stiffness even slight changes in plate thickness have a strong effect on the vibration behaviour of local
modes. The effect on mode shape 2 was - compared to others - very distinct. This explains that structure 3 shows higher
frequency differences than structure 2 in comparison to structure 1 although structure 3 yields lower residual stresses.

In Figure 9 the mode deflection shape of the 3rd natural frequency is shown.

![Figure 9](image)

Figure 9: 3rd mode deflection shape of LS 1, LS 2 and LS 3

This mode shape is present in all structures and the line is highlighted in Table 2. An important fact is that this mode and its
local vibration of the top plate occur at a low frequency and not as expected at higher frequencies. The local modes depend
highly on residual stresses and therefore a change in the residual stresses results in a high frequency shift and in low MAC-
values.

Despite the similarity due to the structures’ geometry and the form of the mode shapes there are significant deviations in the
natural frequencies. (Note the deviation of the natural frequency between LS 1 and LS 3.)

Table 3 shows the dynamic behaviour of one structure before and after stress-relieve annealing. As an effect of the residual
stresses all natural frequencies beside the first one decrease. A visual examination of the mode shapes showed that the effect
of decrease in frequencies is higher for more local modes, while the effect is diminished for global modes including significant
vibration of the stiffeners. This is caused by the reduction of stiffness of the plate field, which has an essential influence on
local modes. The vibration behaviour of global modes is mainly characterized by the stiffness of the bulge profiles that is not
changed by residual stresses in a remarkable extent.

Table 3: First 10 mode shapes and natural frequencies of laboratory structure 2

<table>
<thead>
<tr>
<th>LS 2 stress-relieved</th>
<th>LS 2</th>
<th>LS 2 vs. LS 2 stress-relieved</th>
</tr>
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<tbody>
<tr>
<td>mode shape</td>
<td>fi [Hz]</td>
<td>mode shape</td>
</tr>
<tr>
<td>1</td>
<td>10.71</td>
<td>1</td>
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<tr>
<td>2</td>
<td>30.44</td>
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<td>3</td>
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<tr>
<td>10</td>
<td>55.86</td>
<td>4</td>
</tr>
</tbody>
</table>

* see footnote Table 2
Another important result is the correlation of mode shape 2 before annealing and mode shape 5 after annealing. This means the stress free calculated mode 5 will not be considered as a thread during ship design concerning a resonance free operation. But influenced by residual stresses the natural frequency shifts considerably and can now possibly be excited during ship operation.

Conclusions

In this paper, the influence of residual stresses on the local vibration behaviour of thin local ship decks is investigated. The investigations were worn out using 3 similar ship decks assembled at different ship yards and varying welding energy. Besides the welding energies, also residual welding stresses were measured during production. To evaluate the difference between welding stresses and overall residual stresses a residual stress measurement by the hole drilling method was accomplished. For the investigated structures, the welding stresses showed a distinct dominance. The energy input due to the welding process had a high influence on the build up of the residual stresses.

The vibration behaviour of the 3 structures was investigated using experimental modal analysis. Due to the varying residual stresses the structures showed up high deviations in natural frequencies. Furthermore, the effect on the mode shapes is remarkable leading to low MAC-values for the comparison of the structures.

As an attempt to exclude other effects than residual stresses on the vibration behaviour, one of the structures was stress relief annealed. The comparison of the dynamic behaviour before and after annealing shows similar changes in natural frequencies and mode shapes, so that the described effects on the vibration behaviour can be ascribed to the residual stresses.

Acknowledgments

This research was sponsored by the Bundesministerium für Bildung und Forschung, Deutschland (Federal Ministry of Education and Research, Germany).

References