FATIGUE STRENGTH OF THIN-PLATED BLOCK JOINTS WITH TYPICAL SHIPBUILDING IMPERFECTIONS

L. Eggert, W. Fricke and H. Paetzold

ABSTRACT

During design of block joints in ship structures it is assumed that the permissible tolerances for axial and angular misalignment given in shipbuilding standards are not exceeded. This is particularly important for fatigue assessments based on the nominal stress approach which implicitly incorporates stress magnification due to imperfections. Preliminary investigations of thin-plated ship structures ($t = 4$ mm) showed that (a) it is very difficult to keep the thickness-dependent tolerances for thin plates, and (b) even if the tolerances are kept, the fatigue strength might not be sufficient as the stress magnification incorporated in the nominal stress approach is far exceeded. The present paper highlights the problem in the beginning. The main part describes fatigue tests of components performed in a joint industry project. Stiffened panels were fabricated by different shipyards under realistic shipbuilding constraints. The partly large imperfections (buckles, axial and angular misalignment) were extensively recorded using photogrammetric technique in order to obtain the characteristics of typical pre-deformations. The component tests establishing the link to the fatigue strength were carried out with a stress ratio $R = -1$ which is typical for naval vessels. Concurrent nonlinear finite element calculations containing the whole test set-up were carried out to obtain relations between pre-deformation and stress increase. After description of the test results, a procedure for the assessment of imperfections is discussed which are measured in the usual way in shipbuilding.

IIW-Thesaurus keywords: Structure; Joint; Fatigue strength; Thin plate.

1 Introduction

The proof of sufficient fatigue strength is apart from the buckling analysis the most important strength check for thin-plated ship structures which frequently results in several changes during ship structural design. The fatigue strength assessment is mainly performed with the nominal stress approach as this allows the designer to quickly and effectively evaluate the fatigue strength of structural details. Similar to other proofs, the occurring stress may here be compared with a permissible one taking into account necessary safety margins. In the case of fatigue, the stresses are represented by stress ranges leading to the following format [1, 2]:

$$\Delta\sigma_{max} \leq \Delta\sigma_p \qquad (1)$$

Here, the highest occurring stress range $\Delta\sigma_{max}$ is determined in the nominal stress approach for an ideal, undeformed structure, as represented in construction drawings or product models without any fabrication-related imperfections. The transformation of the structure into an appropriate calculation model is performed either on the basis of the beam theory or the finite element method. The stress ranges are computed assuming realistic loads, e. g. due to waves.

The permissible stress range $\Delta\sigma_p$ depends mainly of the structural detail considered, the shape of the load spectrum and the number of load cycles expected and is normally based on the Palmgren-Miner rule. At the same time the required safety is to be kept which is determined by the design S-N curve such that a probability of failure of 2.3% is not exceeded for the detail. The fabrication-related imperfections mentioned lead to a stress distribution which differs from that in a perfect component. This difference which usually results in increased stresses is implicitly considered in the right side of equation (1). This means that the permissible stress range is smaller than it would have been applied to perfect, i. e. undeformed structures.

In practice, the deviation from ideal to fabricated structure must not exceed tolerances which are prescribed by relevant construction standards in shipbuilding. For the block joint considered here, the stress increase at the weld toe is mainly caused by axial and angular misalignments, see Figure 1. The Production Standard for the German Shipbuilding Industry [3] provides tolerance limits for butt joints of highly stressed components of 10% for axial misalignment (related to plate thickness) and of a depth of 5 mm for lateral distortion, which is the normal distance of the fabricated joint to a reference plane. Concerning

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the latter, the production standard does not define how to measure the imperfection, i.e. the reference plane is not clear. Also, it is not clear whether buckles as harmonic part of imperfections are included or not. The tolerances for axial misalignment are derived from ISO 5817 [4], which introduces quality levels for imperfections in general. ISO 5817 does not give limits for lateral distortion, the older version of ISO 5817 [5], which was modified in 2006 with a technical corrigendum, adhered tolerances for related angular misalignment. The limits of [3] are assumed for the present investigation, as the upper flange of the hull of a naval vessel is considered as reference structure, being also characterized by very thin plating which results from the demand for light-weight structures. Here, plate thicknesses down to 4 mm are used.

In a preceding unpublished project, a typical longitudinally stiffened plate field with a butt joint in the upper deck of a naval vessel was modelled with finite elements considering the axial misalignment as well as angular and lateral distortion. It was shown that under membrane stresses the idealized imperfections result in much higher stresses in the weld than considered in the rules with respect to permissible stress ranges. This finding is in contradiction to past experience showing that damages have not been observed in thin-plated block joints of ships in service.

This contradiction initiated a joint industry project which is described in the following.

2 Objectives

In the past, out-of-plane distortions were regarded primarily with respect to their influence on buckling behaviour, which is also taken into account in the latest Eurocode for design of steel structures [6]. Fatigue cracks resulting from distortions were investigated primarily in connection with "web-breathing" phenomena in steel bridge design, which was examined in various publications, e.g. [7, 8]. Up to the project start, distributions of pre-deformations occurring in practice were almost unknown with regard to thin-plated ship structures with thickness $t = 4$ mm and transversal butt welds. Therefore, the increased stresses determined in the preceding project have been based on idealized distributions of deflections which consist of buckles corresponding to the eigenforms of the whole plate field, of an angular distortion in form of a locally pronounced angle and of axial misalignment. The idealized pre-deformations resulted in rather high stress increase which exceeds the implicitly considered value in the rules by far. Therefore, one objective of the joint industry project was to get insight into the amount and distribution of pre-deformations occurring at block joints in practice.

So far, the effect of such imperfections on the fatigue strength of welded joints was regulated by Recommendation 705 of the German Society of Welding Technique (DVS), which classified rule-based tolerances into according FAT-classes for fatigue strength assessment [9]. Concerning butt welded joints, this correspondence was derived from the results of fatigue tests with specimens having thickness $t > 10$ mm [10]. It is obvious, that the mechanical behaviour of a pre-deformed panel with $t = 4$ mm is rather different from the behaviour of a panel with $t > 10$ mm, especially concerning the effect of lateral distortion on second order based stress increase. Therefore, results of corresponding measurements will be utilized to consider the effects of such imperfections in structural components on the fatigue strength. The relationship between typical measurements of imperfections in shipbuilding and their effects on the fatigue strength considering also the number of load cycles, residual stresses and stress ratios should be investigated experimentally and theoretically.

3 Experiments

In the following, the experimental investigations are described, where stiffened panels were designed and fabricated, from which test models for fatigue tests were extracted. These were clamped into a special frame and subjected to cyclic loads. In all steps the pre-deformations were measured and controlled ensuring that the tests are performed with the imperfections found in the stiffened panels.

3.1 Panels and test models

The design of the panels corresponds to a section of a longitudinally stiffened deck of a naval vessel, see Figure 2. The dimensions of the single plate fields are 1200 × 400 mm, the longitudinals were bulb flats HP 60 × 4 and the deck transverses were relatively low with a web 150 × 4 and a flange 50 × 6. The butt joint is located 150 mm apart from one deck transverse. The panels were fabricated under shipyard conditions for block joints. As

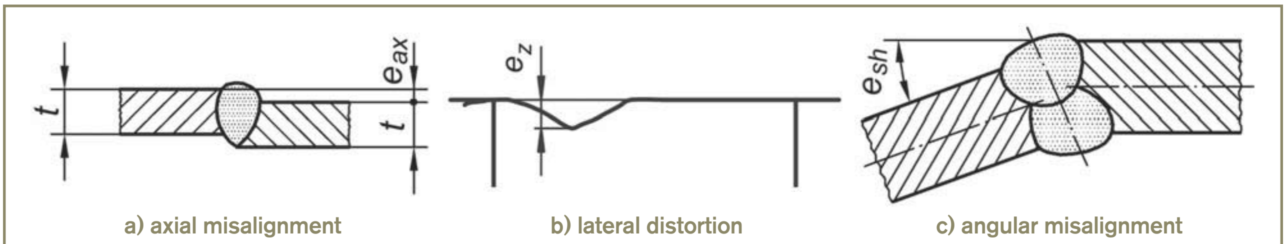

Figure 1 – Butt joint imperfections according to [3-5]

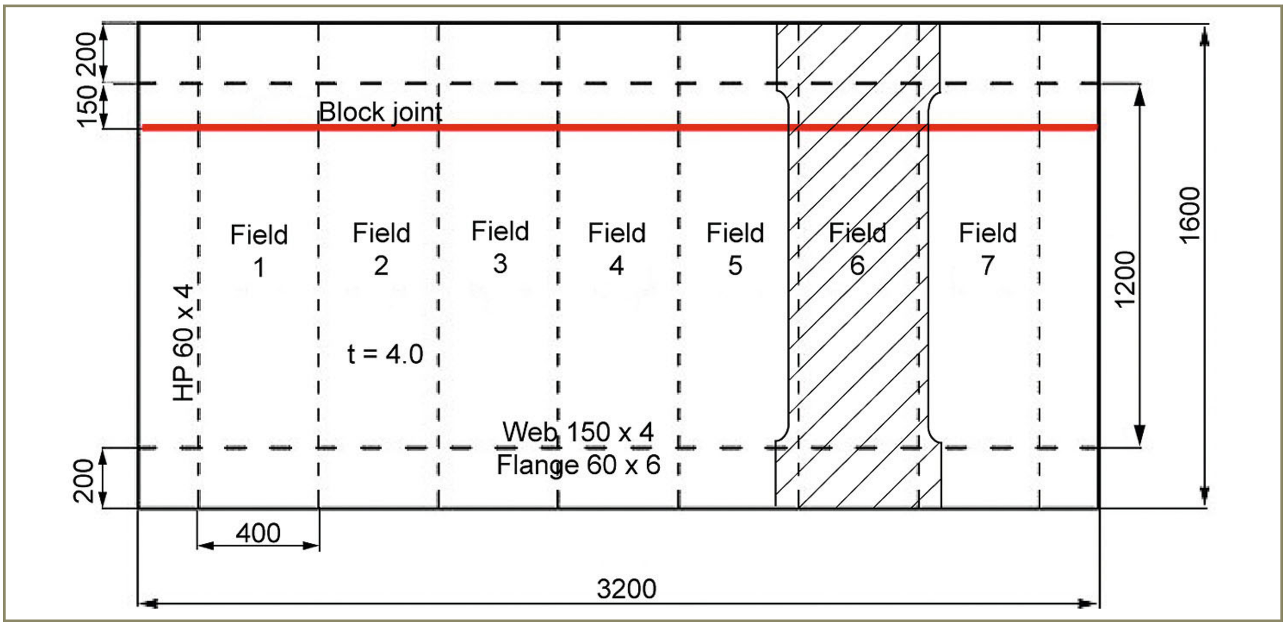

Figure 2 – Investigated panel with one test model indicated

a deck is considered, the bulb flats were welded in overhead position. In total four panels were available which were fabricated by three different shipyards.

From the panels having each seven single plate fields, 14 test models were extracted for the fatigue tests by flame cutting which are indicated in Figure 2 by hatching.

3.2 Measurement of imperfections

The distortions of the panels were measured by photogrammetry in order to record the often discussed characteristics of fabrication-related imperfections. Photogrammetric measurement yields 3D information of an object in space from 2D digital images. For this purpose markers were applied on the panel which are reflecting light when pictures are taken with a high-resolution camera. To transform the 2D into 3D information, at least three images are necessary from different viewpoints, however, about 20 pictures were taken from different angles, distances and particularly elevations. The accuracy (standard deviation) is about 50 μm for the object according to [11].

The grid for the measurement has been chosen with respect to a 2D representation of the imperfections and their evaluation. Important is which types of imperfection in which extent and shape have to be expected. The geometric imperfections consist of:

- lateral buckling distortion (distributed over the whole panel)
- axial misalignment (in way of the butt joint)
- angular distortion (in way of the butt joint).

Buckling distortions generally occur in stiffened panels and plate fields due to axial and angular shrinkage at the welds joining plate and stiffener or girder. They represent the basic shape of the expected imperfections. As characteristic buckling shape for the given geometry, three half waves in longitudinal and one half wave in transverse direction are expected. In order to record the geometry of the buckling shape with sufficient accuracy (regarding amplitude and distribution), the number of measurement points was chosen such that five points are arranged along each half wave.

This results in a basic grid with a = 100 mm. The distance was reduced to 50 mm in way of the weld in order to record the local axial misalignment and angular distortion. On each panel a total of about 800 markers were applied, Figure 3.

3.3 Test frame

A test frame was designed which had to fulfil the following requirements:

– Application of an alternating normal load with constant distribution via a bolted connection
– Choice of the load level such that fatigue cracks in the butt joint may be expected in the high-cycle fatigue domain

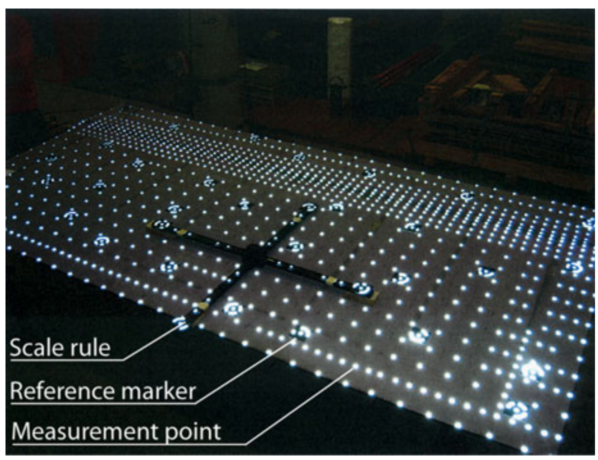

Figure 3 – Photogrammetric measurement

– Design of the welded details in the test frame with infinite endurance so that testing of several models does not cause fatigue damages in the frame
– Possibility to test the models at load ratio R = –1.

During development of the test frame, finite element calculations were carried out to ensure the durability of the frame itself and to find an arrangement of the specimen in the frame which leads to the desired load at the focussed butt joint.

Figure 4 shows the realized test frame with a clamped test model. The nominal load is introduced by a hydraulic actuator above the test model with a force of up to 250 kN. This was able to create a nominal load of about 400 kN in the test model considering the increased lever and the stiffness of the frame. The nominal stress is about 150 N/mm² in the cross section. For an estimated stress increase factor K_s = 2.0 due to misalignment, the maximum stress range is about 300 N/mm², which results in expected number of load cycles of 200,000 for a stress ratio R = –1 based on previous investigations. This lies in the high-cycle fatigue domain. The testing at R = –1 requires a pre-load during clamping of the test model by 130 kN (tension of the actuator) so that the model is subjected to alternating loads whereas the actuator is working in the tensile range (actuator force from +15 kN to + 245 kN). Alternating loads are thus avoided in the bearings.

4 Experimental Results

4.1 Fatigue tests

Figure 5 shows the crack locations for 13 of the totally 14 test models. One test was stopped after two million cycles without fracture which has to be regarded as “run-out”.

Crack initiation at the butt joint had occurred in half of the test models. Failure was defined as number of cycles

Figure 4 – Test model clamped in the test frame

N for a through-thickness crack, i. e. when the crack has reached the opposite plate surface. Other crack locations were sealing plates arranged at the stiffeners of some joints, the fillet weld at the deck transverse and the weld between plate and longitudinal.

Remarkable for the cracks at the block joint is that three of seven cracks initiated not in the plate field but close to the longitudinals. The reason is that the distortion results in a re-distribution of membrane stresses leading to increased stresses at the longitudinals. These are higher than the total stress within the plate field which consists of a membrane and bending portion.

In order to establish a relationship between the stress range and the number of load cycles at failure, finite element (FE) calculations of all test models have been performed. Here, a model of the test frame and pre-deformed test model has been created. A geometrically non-linear analysis was performed and controlled by comparison between measured and computed strains. The nonlinearity led to a straightening of the panel under tensile loads, while the deflection was enlarged under compressive loads. As the tests were carried out at R = –1, at a moderate nominal stress range, the nonlinear effects

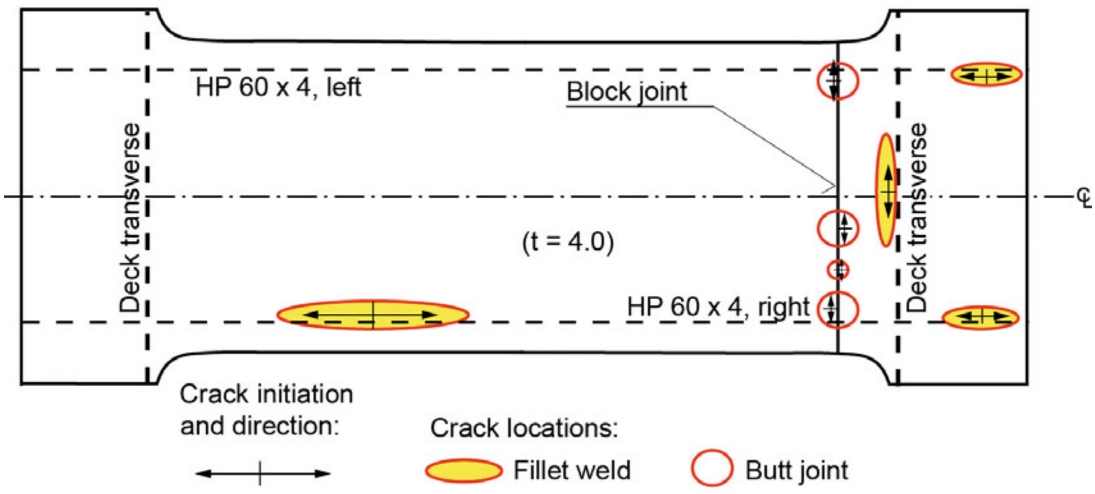

Figure 5 – Locations of crack initiation of 13 test models

on membrane stresses under tension and compression nearly compensate each other. The difference in membrane stress range for typical harmonic buckling shaped pre-deflections is less than 5%.

The mesh in way of the butt joint corresponds to the IIW recommendations for the structural hot-spot approach with extrapolation points at the distance from the weld toe of 0.5 resp. 1.5 times the thickness [10, 12]. This allowed the structural stress at the crack location to be evaluated so that the fatigue test results can be assessed with the nominal as well as structural hot-spot stress approach. The nominal stress simply corresponds to the force introduced into the model divided by the sectional area, whereas the structural hot-spot stress contains the effects of the pre-deformations.

Four out of seven fatigue test results based on nominal stress range are below the design S-N curve (FAT 80), see Figure 6. Remarkable is the result for Panel 2_6 which is far below the design S-N curve due to the fact that in this special case the angular distortion produced a "roof shape", which means, the joint was deformed in opposite direction of the stiffeners. All other test models were characterized by a downward distortion (as sketched in Figure 1 b), resulting only in a small deviation from the design S-N curve.

The nominal design S-N curve is based on a stress ratio R = 0.5 and contains a global allowance of 30% for stress increase due to angular and axial misalignment, see [10]. If the stress increase is directly included in the stress, the design S-N curve (or FAT class) is increased by 25%, in this case to FAT 100, see [10, 12]. This corresponds to the design S-N curve based on structural hot-spot stress. The test results containing the increased structural stress range at the crack location are compared to this design S-N curve in Figure 7. All results are now well above this line.

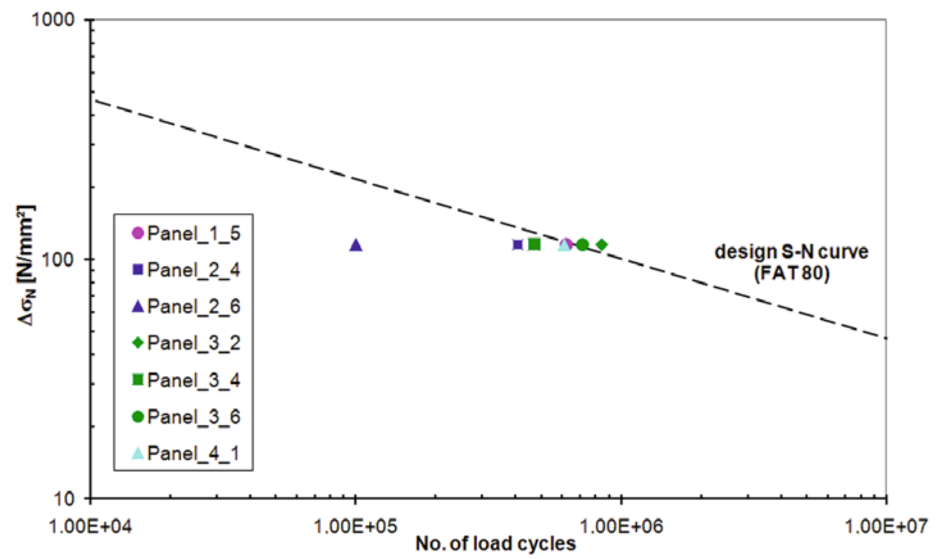

Figure 6 – Fatigue test results and design S-N curve based on nominal stress range

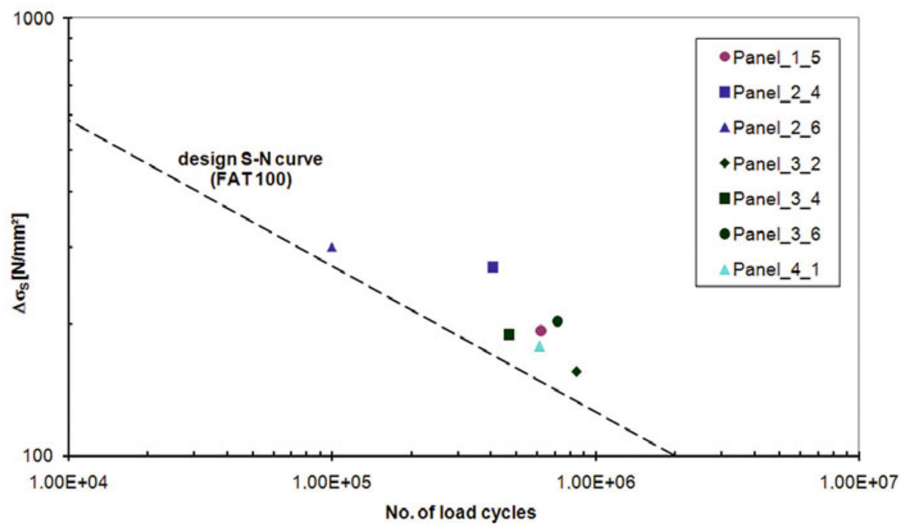

Figure 7 – Fatigue test results and design S-N curve based on structural hot-spot stress range

4.2 Distribution of imperfections

The results of the imperfection measurement are available as 3D ASCI-data in Cartesian coordinates. These allow a high flexibility as regards their evaluation.

The total distributions, which are dominated by harmonic buckling shapes throughout the whole panel and local misalignments at the butt joint, show different shapes in each panel. This is caused by shipyard-typical fabrication techniques as the four panels were fabricated by three different shipyards. The analysis of the shapes and their influence on the total mechanical behaviour of a panel under axial load will be described in future publications.

In shipyard's practice, however, measuring bridges are employed whereby the length of the applied bridge is not defined in the Production Standard mentioned. First investigations have shown that due to the buckling form the bridge length has a decisive effect on the gauge signal from which the lateral distortion is determined. In shipyard's practice, for a transverse butt joint, measuring bridges with a length according to the distance between the deck transverses (in this case 1200 mm) are used up to now, Figure 8. This measuring device of the height z_0 is arranged parallel to the deck longitudinals and depth measurements z_1 and z_2 are taken at both sides of the joint. The imperfections *axial misalignment* e_{ax} and *lateral distortion* e_z are calculated as the difference resp. the mean value of z_1 and z_2 while the latter has to be corrected by the bridge's height.

$$e_{ax} = |z_1 - z_2| \tag{2}$$

$$e_z = \frac{1}{2}(z_1 + z_2) - z_0 \tag{3}$$

The shrinkage angle e_{sh} was taken from a cubic spline function through the photogrammetric measurement points in front of and behind the weld, which allowed the angle to be determined by differentiation.

For each measurement point along the weld, lateral distortion, shrinkage angle and axial misalignment were analysed. This way it is possible to check if the usually assumed relationship between lateral distortion and shrinkage angle exists. This is illustrated in Figure 9 where the red line shows the tolerance limit according to [3].

A correlation between lateral distortion and shrinkage angle is observed only for panel 2. Even here, a significant scatter of the measured values can be observed. This has even resulted in a negative lateral distortion (≅ roof shape) in plate fields 1 and 6 (this is the test model with rather

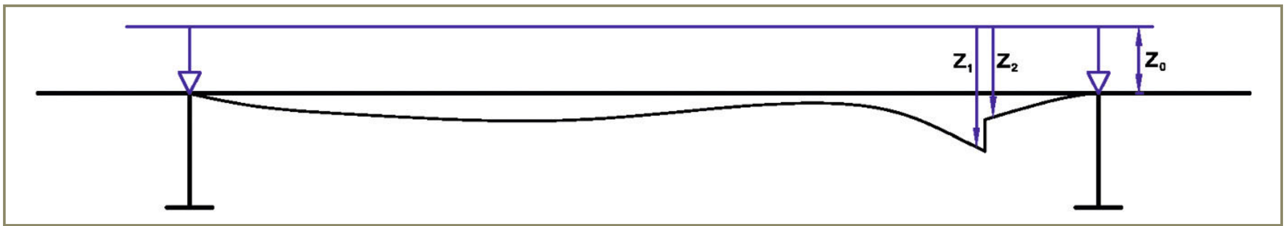

Figure 8 – Measurement of distortions in shipyard's practice

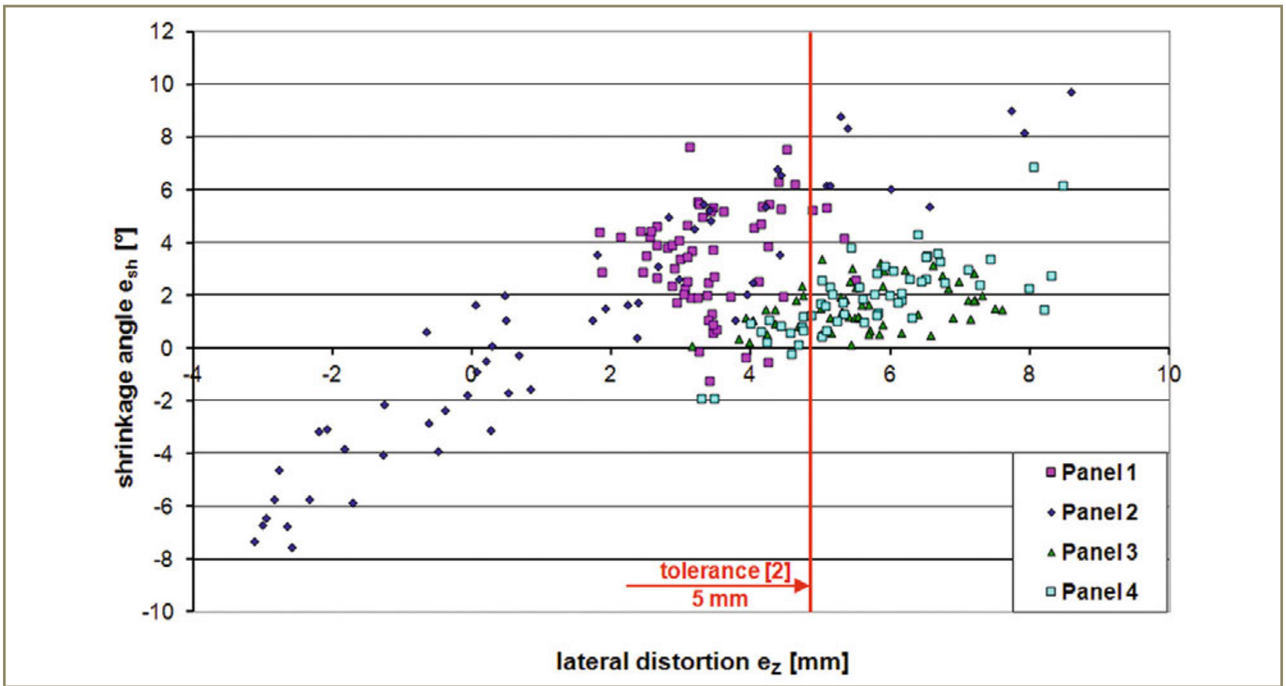

Figure 9 – measured lateral distortion e_z and shrinkage angle e_{sh}

early failure, see Figure 6). Figure 9 shows also that the tolerance limit for lateral distortion is exceeded by about 35% of all measurement points. A corresponding evaluation with respect to axial misalignment shows about 50% exceedence of the limit ($e_{ax}/t = 0.1$).

4.3 Relationship between measured imperfections and stress increase

FE models of the whole panels were created in order to assess the fatigue susceptibility of the block joints. The pre-deformations from the measurements have been incorporated and an axial unit stress $\sigma_N = 100$ N/mm² applied. The analysis results allow assessing how far a direct correlation between measured imperfections and the stress magnification factor according to [10] for 2D considerations at small-scale specimens is possible. This would mean that the measured lateral distortion and axial misalignment can directly yield the stress increase at the weld and herewith the fatigue behaviour. Figure 10 shows the relationship between stress magnification factor and lateral distortion.

A linear correlation between amount of lateral distortion and stress magnification can hardly be seen. Therefore, a discrete measurement of the lateral distortion at a butt weld is not sufficient for the assessment of the quality regarding fatigue. This is illustrated by the comparison of two distributions of plate field deflections in Figure 11.

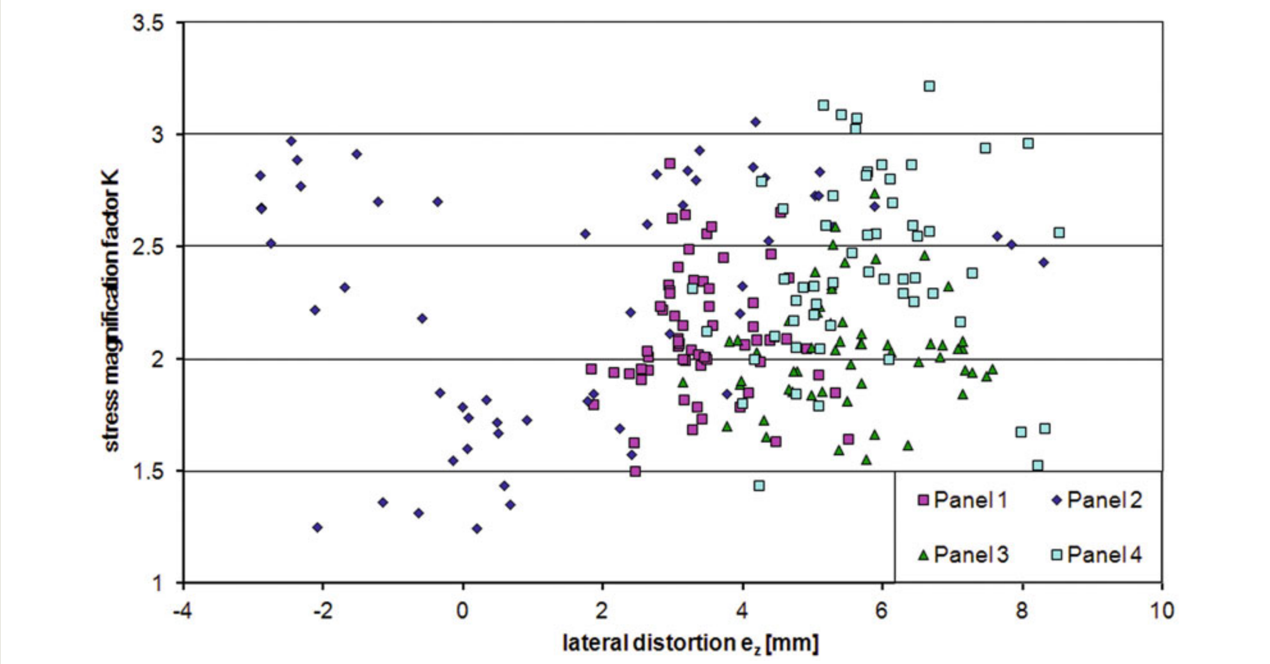

Figure 10 – Computed stress magnification factor vs. lateral distortion for the four panels

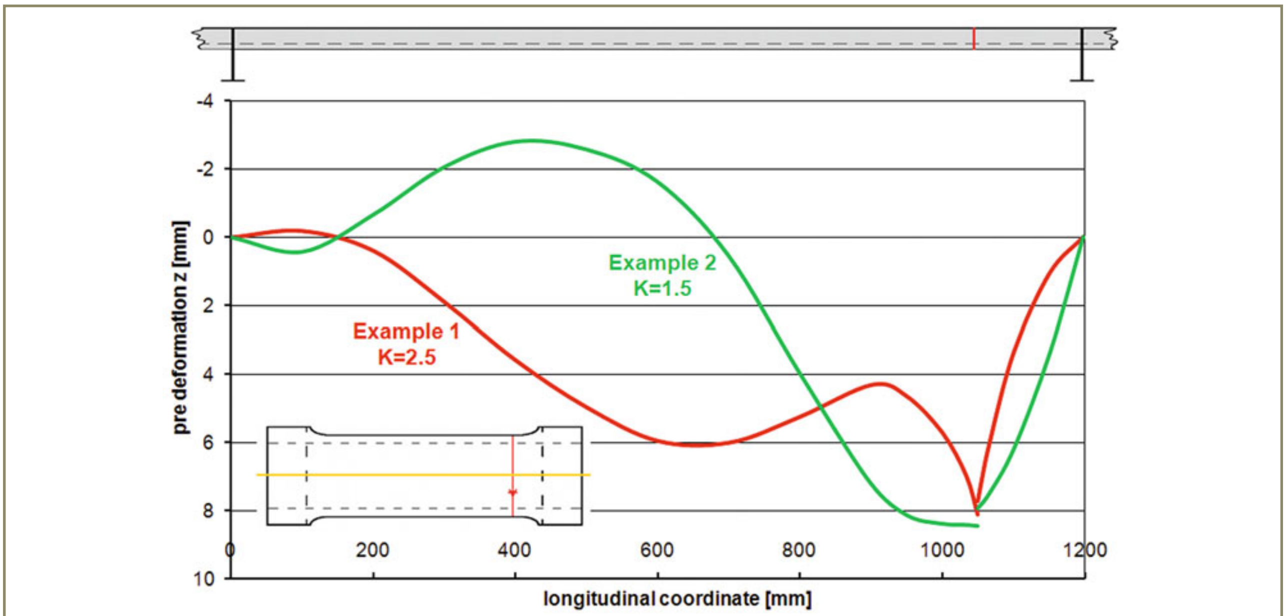

Figure 11 – Distribution of deflections of two single plate fields with same lateral distortion at the butt joint

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Both examples are characterized by the same lateral distortion of $e_z = 8$ mm, however, the shrinkage angles are quite different: 1.5° for example 2 and 9° for example 1. For such high lateral distortion it may be expected that the axial misalignment of $e_{ax}/t \approx 0.1$ plays a minor role. The FE calculations show that the stress magnification factor is $K_S = 1.5$ for example 2 and $K_S = 2.5$ for example 1.

Although these two panels are extreme examples, they represent two fundamentally different characteristics of the distributions of pre-deformations between panel 2 on the one hand and panels 1, 3 and 4 on the other. While in panel 2 a pronounced local angular shrinkage occurs, the distortion of the other three panels is mainly determined by a pronounced buckling shape with relatively small angles at the weld. Here, the maximum distortion is not observed directly at the weld, but 50 to 100 mm beneath the weld. The reason for the different distributions of pre-deformation can be found in the different fabrication methods of the shipyards. The obtained results indicate that a discrete measurement of lateral weld distortion and axial misalignment is not sufficient for the quality assessment with respect to fatigue behaviour.

5 Assessment of the experiments with the IIW fatigue recommendations

Imperfections such as angular distortion and axial misalignment can be considered e. g. according to the IIW Fatigue Recommendation [10]. Here, stress magnification factors are determined using measured axial misalignment and lateral distortion:

– for axial misalignment: $K_{axial} = 1 + 3e_{ax} / t$ (4)

– for lateral distortion: $K_{angular} = 1 + 3e_z / t$ (5)

These are combined to an effective stress magnification factor K_{eff}:

$$K_{eff} = 1 + (K_{axial} - 1) + (K_{angular} - 1) \quad (6)$$

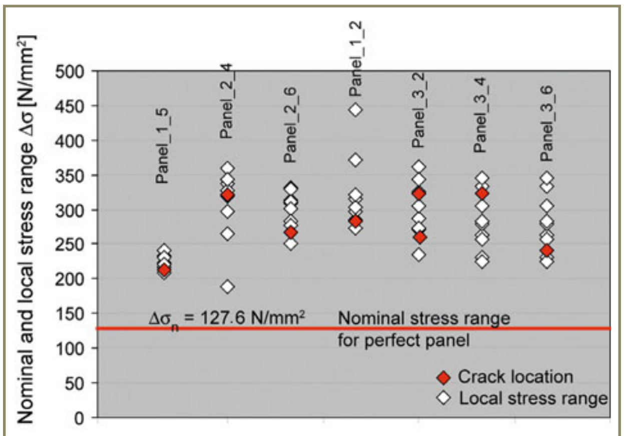

Figure 12 – Computed local stress range according to [10] along the butt joint and at the crack initiation point

However, no experience is available with the application of this recommendation to the structural components under consideration. The seven test results offer the applicability of the recommendation to be checked. Considered are the clamped models in the test frame. Available are imperfections measured along the butt joint at points 50 mm apart from each other between the two longitudinals HP 60 4. The reference length is again the distance of 1200 mm between the deck transverses.

The results of this evaluation are shown in Figure 12. Basis is the nominal stress range $\Delta\sigma_n$ in the panel, i. e. the total force referred to the sectional area. Above this, the local stresses computed with the magnification factor in equation (6) are plotted for all measurement points along the weld for the seven test models. The crack initiation points are indicated in red. It can be seen that the actual crack initiation points do not agree with the highest stress according to [10] which one might expect. The reason is the effect of a redistribution of membrane stresses to the longitudinal edges, when a pre-deformed plate is longitudinally loaded in plane. This 3D-effect, which comes notably into account for thin plate structures, is not included in the simplified equations (4) - (6), which are based on 2D-mechanics of a single plate strip.

A comparison with the local stresses obtained from the FE analyses had shown a much better agreement between the highest stress range and the location of crack initiation.

6 Estimation of the fatigue life under spectrum loading

The fatigue life of the test models was estimated also under spectrum loading applying the rules in [1, 2]. For each test model a component S-N curve is assumed having the fatigue life observed in the tests and a slope exponent m = 3. Using this S-N curve, the number of load cycles was determined for a straight-line spectrum with a maximum stress range $\Delta\sigma_{max} = 200$ N/mm² which is typical for naval vessels. The beneficial mean stress effect for alternating loads has already been considered already in the tests. The results are shown in Figure 13.

Except for the test model with roof-shaped distortion (panel 2_6), all test models would reach the number of required load cycles between $5 \cdot 10^7$ and $7.5 \cdot 10^7$ being typical for naval vessels as shown in Figure 13.

7 Conclusion

The fatigue test results for thin-plated block joints can hardly be associated with nominal S-N curves in the rules. The reason is the global consideration of 30% stress increase due to imperfections. These are exceeded in

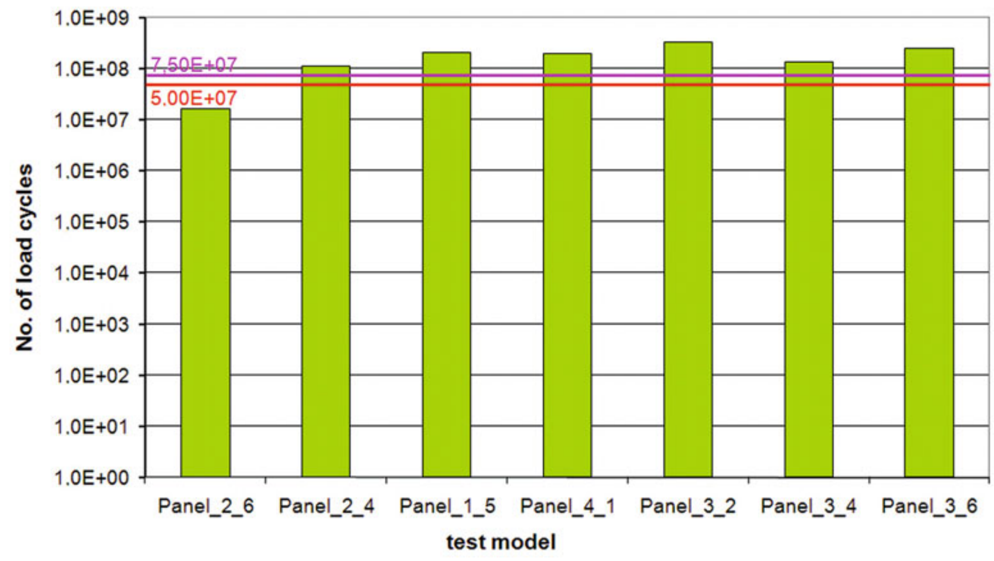

Figure 13 – Computed life for the test models subjected to a straight-line spectrum with $\Delta\sigma_{max}$ = 200 N/mm²

block joints of typical shipyard fabrication particularly due to large angular distortion of welds. It has to be noted that test models have been selected which show relatively high stress magnification and which were not rectified by fairing. Therefore, conclusions from the test models regarding global stress increases to be considered are in practice problematic.

A direct consideration of the stress increase obtained from FE calculations and comparison of the test results with the design S-N curve based on structural hot-spot approach shows good agreement. Therefore, the fatigue strength of the test models can be well assessed with the structural hot-spot stress approach. The estimation of the fatigue life of the test models for wave loads (straight-line spectrum) showed that this reaches the typical required fatigue life for naval vessels if the maximum stress range is restricted to ±100 N/mm². Higher stress ranges in combination with the typical stress ratio $R = -1$ are not possible for buckling reasons. Therefore, the buckling strength is a rather stringent design criterion in addition to the fatigue strength of the butt joint.

The application of the simple 2D formulae of the IIW Fatigue Recommendations [10] does unfortunately not yield the stress increase nor the crack initiation point correctly. Therefore, structural stress computations considering the directly measured imperfections are necessary. The aim is still a fatigue check based on the nominal stress approach. For this reason, further evaluation of the imperfection measurements and the derivation of a simple and practical relationship between the imperfections at the weld and the stress magnification are necessary. This would allow on the one hand direct assessments of shipyard-typical measurements and on the other the definition of tolerances in beforehand under consideration of the expected fabrication quality. This aim is followed in subsequent investigations so that fatigue checks will still be possible with the nominal stress approach.

As a conclusion of this project it is recommended to place much emphasis on the avoidance of local imperfections when welding block joints of 4 mm thickness and to keep the angular shrinkage as small as possible. A roof-shape distortion has to be avoided. It is highly recommended to extend the tolerance values in the production standards to angular distortion.

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About the authors

Dipl.-Ing. Lars Eggert (Lars.Eggert@luerssen.de) is with Fr. Lürssen Werft GmbH & Co. KG, Bremen and Prof. Dr.-Ing. Wolfgang Fricke (w.fricke@tu-harburg.de) and Dr.-Ing. Hans Paetzold (hans.paetzold@tu-harburg.de) are at Hamburg University of Technology (Germany).