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ABSTRACT

Experimental fatigue data for welded joints have been collected and subjected to re-analysis using the notch stress approach according to IIW recommendations. This leads to an overview regarding the reliability of the approach, based on a large number of results (767 specimens). Evidently, there are some limitations in the approach regarding mild notch joints, such as butt joints, which can be assessed non-conservatively. In order to alleviate this problem, an increased minimum notch factor of $K_w \geqslant 2.0$ is suggested instead of the current recommendation of $K_w \geqslant 1.6$. The data for most fillet-welded joints agree quite well with the FAT 225 curve; however a reduction to FAT 200 is suggested in order to achieve approximately the same safety as observed in the nominal stress approach.

1. Introduction

The notch stress approach for fatigue assessment of welded joints correlates the stress range in a fictitious rounding in the weld toe or root to the fatigue life using a single S–N curve. The notch stress is typically obtained using finite element models with the reference radius of 1 mm in order to avoid the stress singularities in sharp notches.

The approach has received much attention lately, due to the increasing available computational power. The approach is very flexible in the sense that all types of welded joints can be assessed using a single S–N curve. It does, however, require more modelling and analysis work than, e.g. the nominal or structural stress approaches.

Radaj et al. [1] presents a thorough review of the history of the approach. Fricke [2] gives practical guidelines for the notch modelling and stress analysis and Sonsino [3] proposes S–N curves to be used under different conditions. The approach is included in the IIW fatigue design recommendations by Hobbacher [4].

In this paper, we consider the notch stress approach according to the IIW [2]. The approach is based on the work by Radaj [5] and modified by Seeger and co-workers, see Olivier et al. [6,7]. The reference radius of $R1$ is determined as a mean value and the fatigue strength (FAT 225) is derived from experiments.

While many recent publications on fatigue experiments also discuss the results in terms of notch stresses [8–11], the amount of published experimental evidence of the reliability of the approach is very limited.

This investigation therefore presents a systematic re-analysis of fatigue data extracted from the literature and converted to the notch stress system. This provides an overview of the reliability of the approach and a basis for discussion of the observed limitations.

2. Extraction of fatigue data

Constant amplitude fatigue testing results have been extracted from the literature [12–35]. Table 1 presents an overview of the different test series, for which data have been extracted. All data are plotted in Fig. 1.

The data have been limited to small scale specimens from recent investigations. Only investigations with positive stress ratios are considered and only papers with thorough description of specimen geometry. The only difference in the data considered here is the specimen geometry, since this is the only parameter considered in the notch stress approach.

Only specimens failing from the weld toe is considered and only welded specimens of comparable quality in as-welded condition. The steel grades vary from S235 to S1100, specimen thickness varies from 5 to 25 mm and stress ratios vary from 0 to 0.5+. Run-outs are included as well, but the main focus of this study is on the finite life region.

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Table 1
Extracted experimental fatigue data series. K_t is determined according to the IIW recommendations for fatigue assessment using the notch stress approach [2].

	ID	Ref	S_y (MPa)	t (mm)	K_t	R	Loading	Process
T-joints	Haa1	[12]	420	20	2.80	0.1	Bending	SAW
	Bud1	[16]	550–690	16	2.64	0.0–0.5	Bending	SAW
	Sta1	[13]	420	20	2.91	0.1	Bending	SAW
	Gal1	[23]	700	5	1.99	0.1	Bending	?
	Gal2	[23]	355	6	2.03	0.1	Bending	?
	Ped1	[25]	700	6	2.03	0.1	Bending	MAG
	Tru1	[31]	420	20	2.73	0.1	Bending	?
Transversal attachments	Lif1	[17]	700–900–1100	8	2.35	0.2	Tension	MAG
	Fos1	[18]	355–460–690	12	2.49	0.1–0.5	Tension	MAG
	Fos2	[18]	355–460–690	12	2.57	0.1	Tension	MAG
	Fos3	[18]	690	25	2.69	0.1	Tension	MAG
	Son1	[26]	1100	8	2.20	0.0	Tension	MAG
	Man1	[15]	355–700	12.5	2.72	0.1	Tension	MMA
	Oht2	[30]	570	20	3.01	0.0–0.5	Tension	MMA
	Muc1	[32]	460	13	2.51	0.1	Tension	MAG
	Hut2	[34]	235–355	8	2.32	0.5	Tension	?
	Kud1	[35]	260	20	3.10	0.0	Tension	?
Butt joints	Lif2	[17]	700–900	8	1.60	0.2	Tension	MAG
	Wan2	[20]	235–390–700	8	1.60	0.0–0.1	Tension	?
	Hut3	[34]	235–355	8	1.60	0.5	Tension	?
	Dic1	[21]	318	24	2.05	0.0	Tension	MMA
	Oht1	[27]	284–579	20	1.95	0.0–0.5	Tension	SAW
	Nak1	[28]	431	20	1.95	0.0-Sy	Tension	MMA
Longitudinal attachments	Lif3	[17]	690–900–1100	8	3.42	0.2	Tension	MAG
	Lif4	[17]	690–900–1100	8	3.85	0.2	Tension	MAG
	Haa2	[19]	355–700	8	3.73	0.1	Tension	?
	Hut1	[22]	700	8	3.73	0.1–0.5	Tension	?
	Wan1	[20]	235–390–700	8	2.69	0.0–0.1	Tension	?
	Mad1	[24]	355	13	3.32	0.1	Tension	MMA
	Lop1	[14]	355–590	12	3.82	0.0	Tension	MAG
	Dim1	[33]	333	12.7	3.62	0.1	Tension	MAG
	Mor1	[29]	417	12	4.01	0.0	Tension	MAG

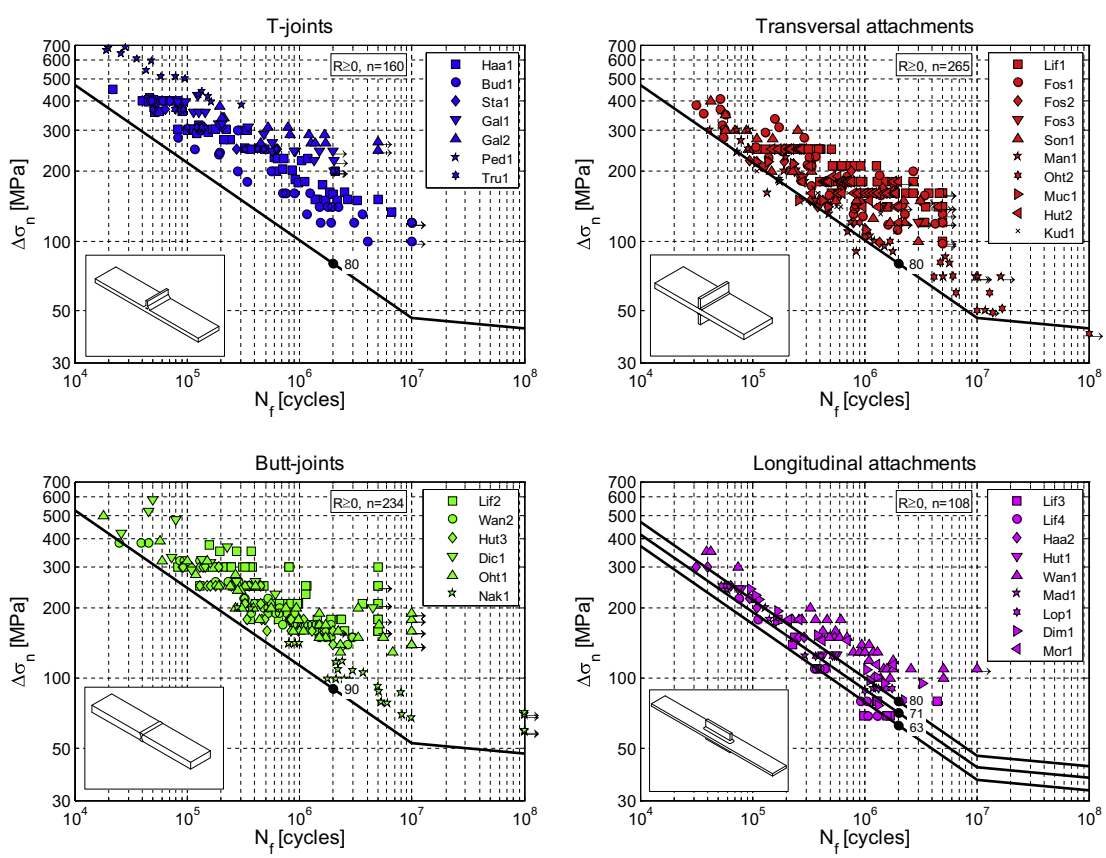

Fig. 1. Extracted fatigue data in the nominal stress system.

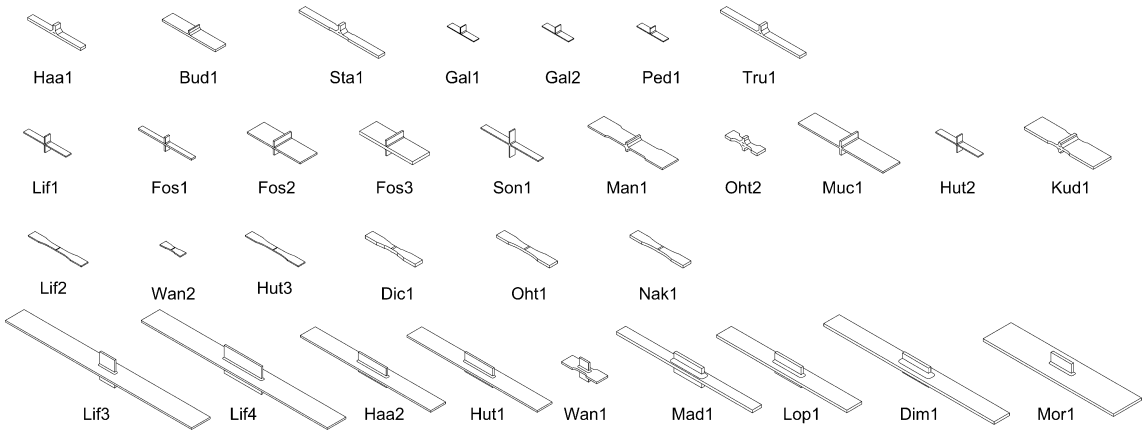

Fig. 2. Specimen geometry and associated ID.

In many cases, the fatigue data were listed in the references; otherwise software assisted extraction from the SN diagrams has been performed. The presentation of fatigue data in Fig. 1 is divided according to specimen type, in the nominal stress system. Only the four most popular specimen types are considered; *T*-joints, double sided transversal attachments (non-load carrying cruciform joints), butt joints and double sided longitudinal attachments.

The relatively large scatter in the results is explained by different thickness, weld quality, misalignment, stress ratio and so forth. The large scatter is considered positive in this investigation, since a more general overview can be achieved. Fig. 2 shows all specimens.

All data agree quite well with the FAT classes suggested by the IIW, and only very few data points fall below the design S–N curves. It is clear that the *T*-joints show much better results than suggested by the FAT 80 curve. This is expected though, since they are tested in bending, considering the positive effect of the steep stress gradient and little to none negative effect from misalignment.

3. Conversion to the notch stress system

Using the notch stress approach, the stress concentration factor (SCF) of an arbitrary welded joint can be determined using finite element analysis. Radaj et al. [1] explain how the geometric SCF K_t corresponds to the fatigue effective SCF K_f, due to the fictitious rounding of the notch.

$$K_f = K_t(r_{\text{ref}} = 1\ \text{mm})$$

The notch approach thereby uses an idealized geometry, which makes the above statement true. Conversion of the extracted fatigue data in the nominal stress system $\Delta\sigma_n$ to the notch stress system $\Delta\sigma_k$ for a given specimen can thus be accomplished as follows.

$$\Delta\sigma_k = K_f \cdot \Delta\sigma_n$$

The stress concentration factors of all specimens are therefore determined using complete FE models, according to IIW recommendations [2,4]. Here, the procedure is exemplified using the

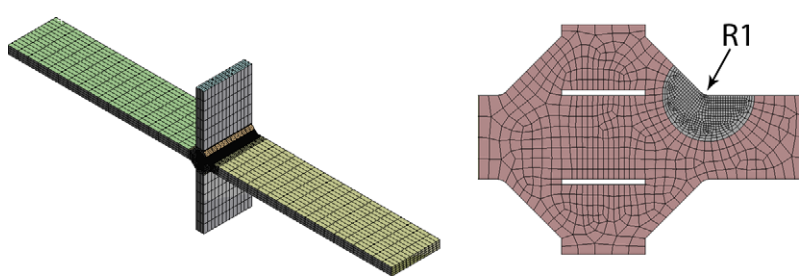

Fig. 3. Lif1-specimen: element size in the notch is 0.1 mm. Only the weld toe is considered here.

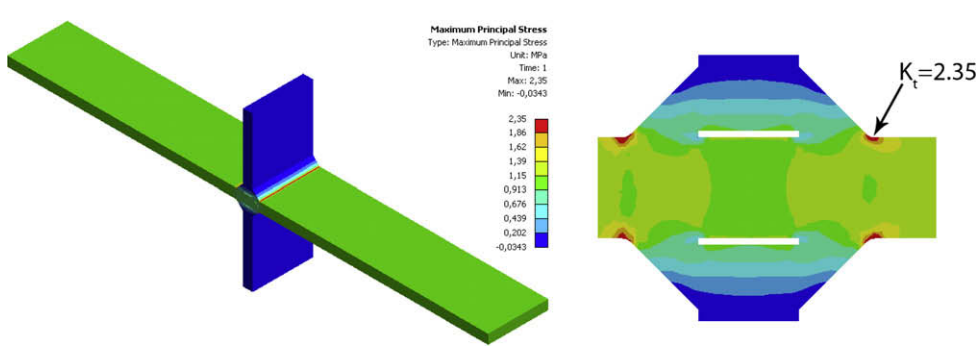

Fig. 4. The SCF of the specimen is determined by applying a nominal stress of 1 MPa to the FE model.

specimens from Lagerqvist et al. [17], (Lif1). The FE analysis was performed using mesh refinement in the area around the weld toe, as shown in Fig. 3.

A reference radius of 1 mm was used and flank angles of 45° for fillet welds and 30° for butt welds, as recommended in [4]. A tensile nominal stress of 1 MPa was applied to the specimen, such that the maximum principal stress observed in the notch corresponds to the SCF, see Fig. 4. The principal stress hypothesis is used for all notch stress analysis.

No misalignment is considered directly in the FE analysis for the determination of the SCFs. However, the SCF for the butt joints is multiplied by a stress magnification factor k_m, since these specimens are very prone to misalignment. Hobbacher [4] suggests $k_m = 1.10$ for butt joints made in flat position in shop. This value has been applied here.

4. Extracted fatigue data in the notch stress system

The converted fatigue data are plotted in Fig. 5. The immediate conclusion is that the results agree quite well with the FAT 225 curve for all fillet-welded joints, but not so well for the butt joints. As expected, the results for the *T*-joints are somewhat above the FAT 225 curve, which can be explained by them being tested in bending.

For the double sided transversal and longitudinal attachments, the results agree reasonably well with the FAT 225 curve, however, some data points fall below the curve. The reason for this is unclear, but the few specimens falling below the FAT 225 curve is assumed to suffer from some unfortunate conditions, e.g. misalignment or poor local weld toe profile. It is noted, that the FAT 225 curve is derived for "welds with relatively good quality toe profiles", according to Fricke [2], p. 13.

The results for many of the butt joints, on the other hand, lie significantly below the FAT 225 curve. This is expected to be due to the relatively mild notch present in butt joints with little overfill. The stress concentration factor determined for these joints were calculated to 1.6–2.0, whereas the stress concentration factor for the fillet-welded joints were in the range of 2.0–4.0. The problem is especially pronounced for thin butt joints, e.g. in 8 mm plate, which has a stress concentration factor of approximately 1.6.

It is well known that some low-SCF joints, such as butt joints, can be assessed in a non-conservative manner using the notch stress approach. Fricke [2] therefore suggests a correction for mild notches, i.e. assuming a notch factor $K_w = \sigma_k/\sigma_{hs}$ of at least 1.6, where σ_{hs} is the structural hot spot stress. However, in this investigation, the notch factor was above 1.6 for all butt joints, and this correction was thus not applied.

Fatigue assessment of butt joints by the notch stress approach is investigated further in the following section.

5. Notch stress assessment of butt joints

In order to obtain a more reliable basis for evaluating the notch stress approach for butt joints, additional test results from many investigations of butt joints were extracted from the S–N curve catalogue by Olivier and Ritter [36]. Only fully penetrated butt joints (I, V and X joints) and only toe-failures are considered. Extreme values are ignored, i.e. only series with a probability of occurrence in the interval 10–90% are considered here. Series outside of this interval are presumably affected by severe weld defects on the one side or a significant crack-initiation period on the other side. Run-outs are excluded as well.

In Fig. 6 the fatigue data for the butt joints are compared in the nominal (left) and notch stress system (right). The fatigue data is

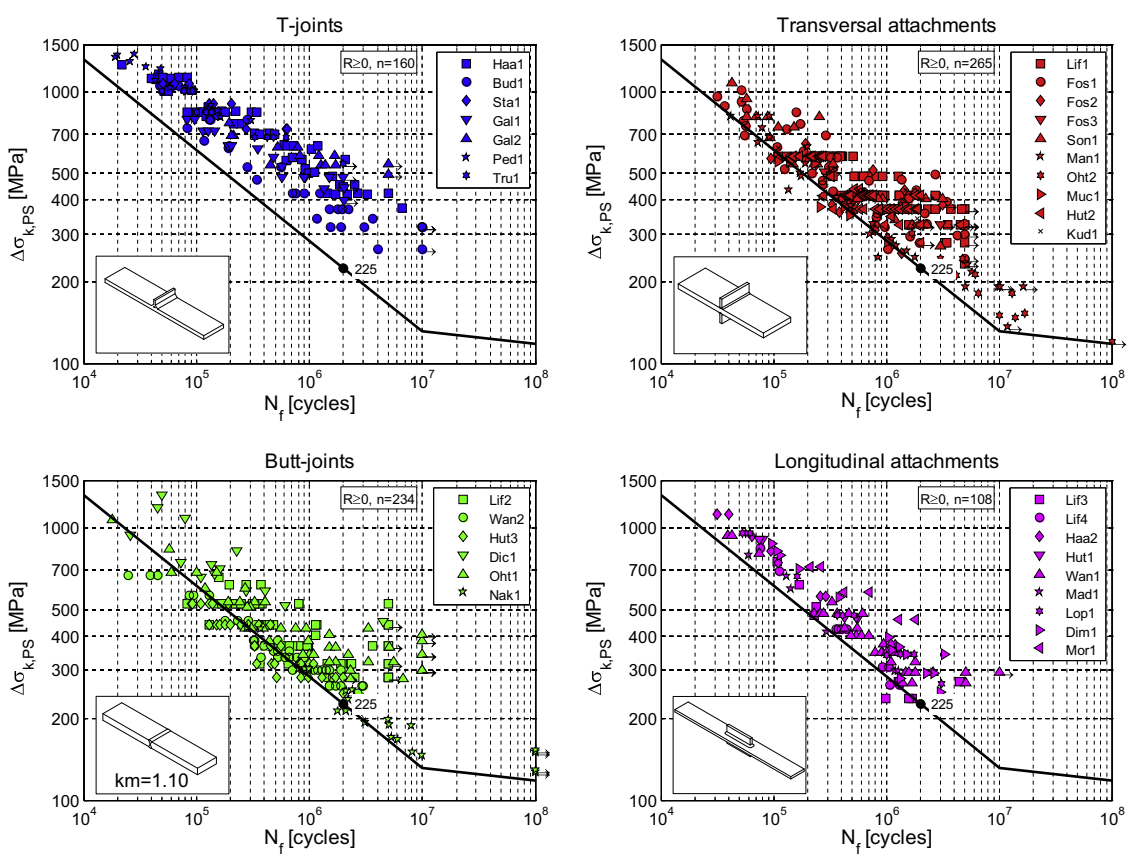

Fig. 5. Fatigue data converted to the notch stress system.

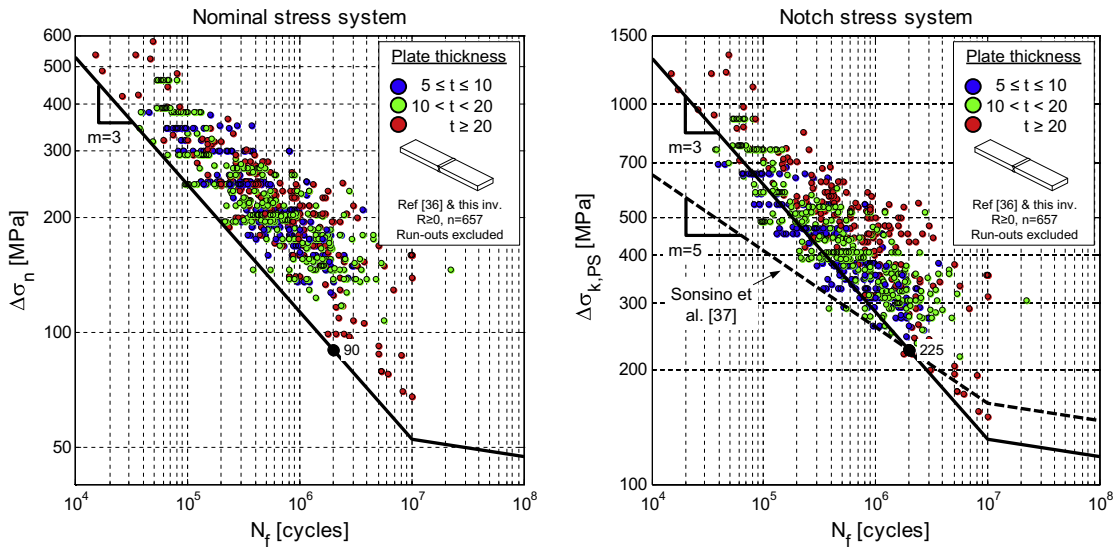

Fig. 6. In the nominal stress system (left) the fatigue strength seems independent of thickness. In the notch stress system (right), on the other hand, the low-SCF of thin specimens cause them to be assessed non-conservatively.

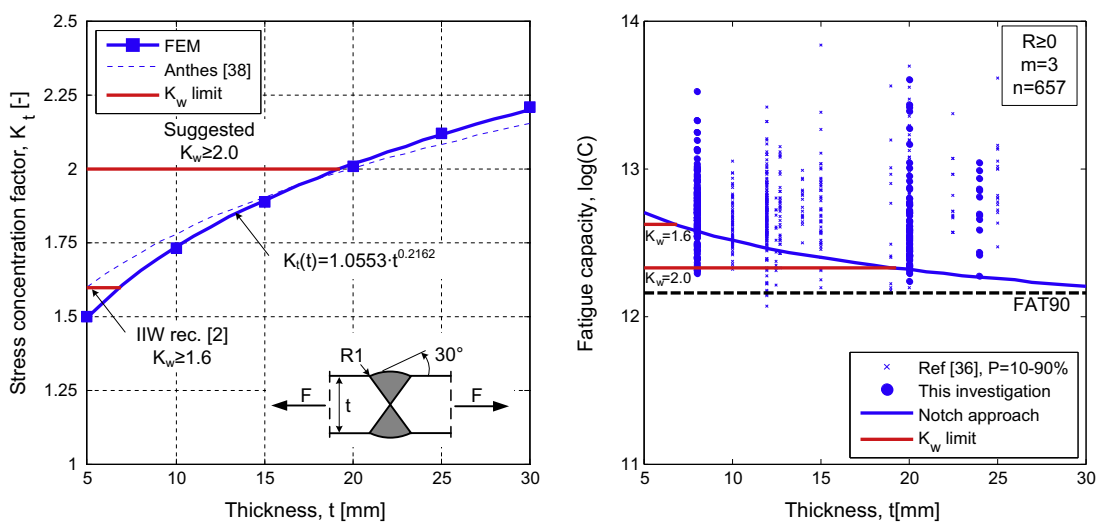

Fig. 7. The thickness has a large influence on the SCF of the idealized butt joint and thus the estimated fatigue strength. However, this estimation does not correspond well with experimental data.

converted using the formula for K_t in Fig. 7 (left) and $k_m = 1.10$ for misalignment. It is seen that the fatigue strength of all butt joints are approximately identical in the nominal stress system, regardless of the different specimen thickness. In the notch stress system, however, the thin joints are assessed non-conservatively because of the very low-SCF determined for these joints ($K_t < 2.0$).

Sonsino et al. [37] also reports problems with fatigue assessment using the notch stress approach considering thin/flexible welded joints, e.g. butt joints. They observed shallower slopes for these particular joints and therefore suggest the use of $m = 5$ while maintaining the FAT 225 value. As it is seen in Fig. 6 (right), this approach seems promising in the high cycle area, but too conservative in the medium-to-low cycle area.

In Fig. 7 (left), it is seen how the stress concentration factor decreases rapidly for thin butt joints. The stress concentration factor determined using FEM is compared to a formula by Anthes et al. [38], which shows a similar tendency. Accordingly, the notch stress approach will estimate very high fatigue strength for thin butt joints.

The fatigue capacity of all butt joint specimens from this investigation and [36] are plotted as a function of the thickness in Fig. 7 (right). The fatigue capacity is plotted instead of, e.g. the characteristic fatigue strength, in order to avoid uncertainties due to statistical treatment. The fatigue capacity is calculated assuming $m = 3.0$. For the notch stress assessment, misalignment is considered using $k_m = 1.10$.

It is seen, that the fatigue capacity estimated by the notch stress approach can become very non-conservative especially for thin butt joints. The experimental data in Fig. 7 (right) does not show a significantly higher fatigue capacity for thin joints. There do not seem to be any clear thickness dependency at all in this range. Or at least, there do not seem to be any support for the tendency suggested by the notch stress approach for thin butt joints.

Fig. 7 furthermore shows the effects of a minimum notch factor K_w limit of 1.6 and 2.0, respectively. It is seen, that the current recommendation of $K_w = 1.6$ only affects butt joints thinner than approximately 7 mm and has relatively little effect. If the minimum notch factor is increased to $K_w = 2.0$ on the other hand, a

much more conservative result is obtained for thin butt joints and other low-SCF joints.

In most standards and, e.g. the IIW recommendations [4], the thickness effect is only considered for welded joints in plate thicker than 25 mm. However, recent work, e.g. Gustafsson [39], reports higher fatigue strength in plates thinner than 25 mm. He reports a significant improvement in the fatigue strength for longitudinal attachments down to 3 mm.

Ohta et al. [40] have studied the thickness effect of butt joints. They report that the fatigue strength of butt joints in 9 and 40 mm plate was similar when testing by cycling down from the yield strength ($\sigma_{max} = \sigma_y$). On the other hand, if tested at $R = 0$, the 9 mm specimens showed significantly higher fatigue strength. They suggest that thinner butt joints have an apparent higher fatigue strength (when tested at low R-ratios), than thick butt joints. This is only indirectly due to the thickness, but is because butt joints in thicker plates better hold a high level of tensile residual stresses.

Conclusively, it seems that the fatigue strength for thin butt joints estimated by the notch stress approach is too optimistic and there is a need for guidance in order to alleviate this problem, e.g. by requiring an increased minimum notch factor of $K_w = 2.0$ or alternatively use a shallower slope of $m = 5$, as suggested by Sonsino et al. [37].

6. Further observations

In the following, a minimum notch factor of $K_w = 2.0$ is assumed for thin butt joints. By considering the fatigue data in the notch stress system, the difference in the geometry of the specimens can be disregarded to some extend. At least the effect of different stress concentration factors of the specimens can be disregarded; however, some specimens hold tensile residual stresses better than others and this effect cannot be disregarded. Still, observations can be made based on a larger amount of fatigue data than usual.

It is clear from Fig. 8, that the cutting off by the parent material curve FAT 160·K_w, $m = 5$ seems unnecessary. However, high strength steel must be applied for high stress ranges and the specimen edge roughness must be sufficiently fine. Sperle [41] showed that the fatigue strength even for thermally cut edges in high strength steel can be significantly higher, than the current IIW recommendation for parent material.

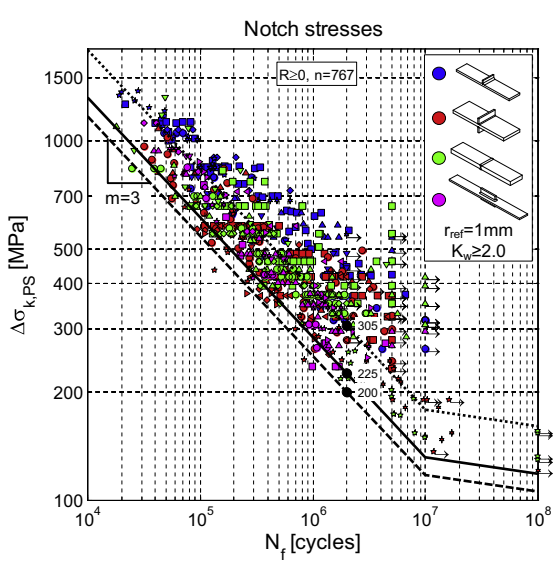

Fig. 8. Fatigue data in notch stresses for all series pooled together.

If excluding run-outs and using a slope of $m = 3.0$, the mean fatigue strength ($Ps = 50\%$) is FAT 305. Assuming a log-normal distribution, the standard deviation of log (C) is 0.28, which yields a design curve ($Ps = 97.7\%$) of FAT 199 (mean – 2 standard deviations).

Comparing the plots of Figs. 1 and 5, it seems that the nominal stress approach is more conservative than the notch stress approach, since less data points fall below the respective S–N curves. The reason for this is not clear, however, it seems that if the FAT 225 curve is reduced FAT 200, it gives approximately the same safety as observed in the nominal stress system, see Fig. 8.

7. Conclusions

The following conclusions are drawn based on re-analysis of a large amount of recent fatigue data in the notch stress system using the principal stress hypothesis.

1. For most fillet-welded joints, the experimental fatigue data agrees reasonably well with the current IIW guidance, i.e. using FAT 225 S–N curve, except for few data points.
2. FAT 200 is proposed for instead though, as it seems to give the same safety as observed in the nominal stress system.
3. The current IIW recommendations can cause non-conservative assessment for thin butt joints. Increasing the minimum notch factor from $K_w \geqslant 1.6$ to 2.0 is therefore proposed to alleviate this.
4. The parent material limit of FAT 160·K_w seems unnecessary if high strength steel is applied.

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