ce/papers Proceedings in civil engineering

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ORIGINAL ARTICLE

eurosteel AMSTERDAM 2023

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Different fatigue design stress methods for common constructional details in cylindrical shells under axial loading – Influence of the r/t-ratio and the shell thickness

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Abstract

Cylindrical shells are frequently used in industrial constructions, for example in pipes, towers and masts and are often subjected to fatigue loading. Two common welded constructional details were studied: i) detail 1, the wall thickness transition with a circumferential butt weld and ii) detail 2, the welded ring stiffener. Both details are loaded with axial stresses due to N and/or M. In the new Eurocode prEN 1993-1-9:2021 three different fatigue design stress methods are provided in general. For these two constructional details, beside the modified nominal stress method, the hot spot stresses and effective notch stresses are determined with Finite-Element-Analyses, with variation of geometric parameters, like shell thickness t, radius-over-thickness-ratio, t_2/t_1-ratio at the thickness transition and thickness t_s of the ring stiffener. Stress concentration factors for the different fatigue design stress methods are presented and the influence of the geometric parameters are investigated. Especially the difference between a plate with an infinite radius-over-thickness-ratio and the pipe with a finite radius-over-thickness-ratio is shown. Also, the influence of the shell thickness is investigated. For comparison of the three different methods the utilization factors of the hotspot stress method and the effective notch stress method are compared with those of the modified nominal stress method. A recommendation is given, which design stress method is appropriate for the two studied details, depending on the geometric parameters.

Keywords

cylindrical shells, ring stiffener, thickness transition, fatigue design stress methods, hot spot stress method, effective notch stress method

1 Introduction

Due to the significant developments of commercial software tools for Finite Element Analysis (FEA), the hot spot stress method and also the effective notch stress method get more attractive in engineering praxis for fatigue verifications. This was the main motivation for this research study to quantify the results of these both methods compared to the modified nominal stress method. Up to now, primarily constructional details with flat plates were studied in the literature. This investigation will widen the experience of these details also for the application in cylindrical shells. The term modified nominal stress is used in general in this paper, because for the studied detail 1 a stress concentration factor is used, to consider the eccentricity due to the different plate thicknesses (tapering to the outside). Two common welded details in cylindrical shells (with the inner radius r_i, see Fig. 1) were studied in detail. Cylindrical shells are frequently used in industrial constructions, for example in pipes, towers and masts and are often subjected to fatigue loading. Detail 1 is a circumferential, full penetration butt weld in a wall thickness transition (see Fig. 1a). This thickness transition is also known as tapered transition, studied for a tapering to the outside. The thicker shell has a bevel with an inclination of 1 : x (the value x is also varied). The inside of the pipe is assumed to be ground flush (usually used for penstocks and pressure conduits) and here also the outside is assumed to be ground flush. Detail 2 is also a constructional detail with a circumferential weld. This circumferential weld connects a ring stiffener to the pipe with a full penetration butt weld (see Fig. 2b). The height of the ring stiffener is named h_s and the width t_s. The cylindrical shells in detail 1 and 2 are only loaded with axial stresses $\Delta\sigma_x$ due to N and/or M. In [1] these two details are loaded with an internal pressure p_i (leads to circumferential stresses σ_φ),

https://doi.org/10.1002/cepa.2385

in addition to axial stresses $\Delta\sigma_x$. The focus in [1] is on the comparison of the accurate hot spot stresses, determined with the Finite Element Analysis (FEA) and with an analytical solution, for different load cases. Also, a study in [1] is presented, comparing the modified nominal and the hot spot stress method for different load cases. In this paper, a wider parameter range of the radius-over-thickness-ratio (r_i/t-ratio), of the t_2/t_1-ratio of the thickness transition for detail 1 and of the thickness and the area of the ring stiffener for detail 2 is investigated. The focus here is on the influence of these mentioned parameters on the stress concentration factors (SCFs) for the different fatigue design stress methods. Especially the wide range of the r_i/t-ratio, with the transition to a flat plate (very high r_i/t-ratio, $r_i/t = 500$), is investigated. The varied parameters for detail 1 and 2 are shown in Table 1. In this study, also thicker plates are investigated (t > 25 mm), to see whether the well-known size effect (reduced fatigue strength in thicker plates) is considered within each stress method.

Prepared hot spot stresses (SCFs) for practical application, determined with an analytical solution for different load cases for these two details are shown in [2].

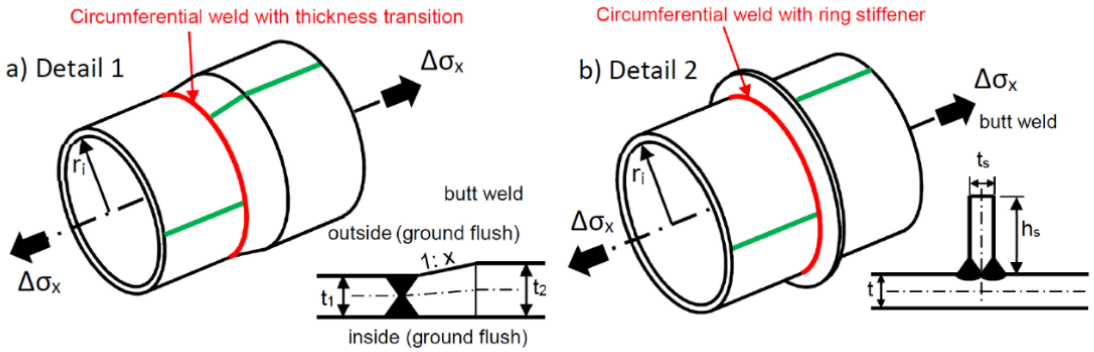


Figure 1 Investigated constructional details: a) detail 1 - Circumferential butt weld with a thickness transition (tapering to the outside), b) detail 2 – Circumferential full penetration butt weld to connect the ring stiffener with the pipe

Table 1 Varied parameters of detail 1 and 2 (abbreviations see Fig. 1)

Detail 1 – thickness transition			
r_i / t_1 [-]	t_2 / t1 [-]	1 : x [-]	t_1 [mm]
20 - 500	1.2 ; 1.5 ; 2.0	1:4 ; 1:5.5 ; 1:10	25 ; 50 ; 75
Detail 2 – ring stiffener			
r_i / t [-]	h_s / t_s [-]	t_s / t [-]	t [mm]
20 - 500	3.5 ; 7.0 ; 14.0	1.0 ; 2.0	25 ; 50 ; 75

In the traditional method of fatigue design, called nominal stress method in the literature, the difference in fatigue resistances of welded constructional details is considered by a catalogue of detail classes (detail categories). Each detail category is related to a different fatigue strength $\Delta\sigma_c$ (for $N = 2 \cdot 10^6$ stress cycles) and S-N curve respectively. Such a catalogue of fatigue resistance is e.g. contained in the present Eurocode 1993-1-9 [3] and in the new Eurocode prEN 1993-1-9:2021 [4]. In the nominal stress method, only the nominal stresses (in shell structures membrane stresses) are needed and no more detailed Finite Element Analysis (FEA) is necessary.

But, the stress calculations using Finite Element Analysis (FEA) are increasingly common, also in the fatigue design of welded constructional details. In these more advanced methods also local stresses are calculated in the area of the weld on the basis of a realistic Finite Element Model. With these realistic FE-models the local geometric stress increasing effects, which are the most relevant sources for different fatigue resistance of constructional details, can be considered and this largely eliminates the need for different detail categories. Here, the hot spot stress method and also the effective notch stress method are presented and the different stress concentration factors for each method are shown. These fatigue design stress methods are now provided in the new Eurocode prEN 1993-1-9:2021 [4]. The individual fatigue resistances for all three different stress methods in this new Eurocode [4] are the basis for the following investigations in fatigue design. Also, the new modifications for the consideration of the size effect (for t > 25 mm) are used. It should be noted, that the status for this Eurocode is a preliminary norm and is not officially released now.

2 Methodology – hot spot stress and effective notch stress concept

The hot spot stress method is documented in [4], Annex B, with different detail categories. The detail category in cases with stresses perpendicular to the weld axis, as for detail 1 and 2, ranges from 90 ($\Delta\sigma_c$ = 90 N/mm²) for a cruciform joint with load carrying partial penetration butt welds to 112 for a full penetration butt joint with both sides ground flush. Most constructional details have the detail category 100. It´s important to emphasize, that the hot spot stress method is only applicable to welded constructional details with a potential crack location at the weld toe and not for potential cracks starting from the weld root. Hot spot stresses are defined as theoretical stresses on the surface of a plate, which includes the membrane and bending stresses, but not the notch effect of the weld. Important hints for the Finite Element Analysis (FEA) can be found in the new Eurocode prEN 1993-1-14:2020 [5]. Usually a linear extrapolation to the hot spot is necessary to determine the structural stresses in the hot spot. In the prEN 1993-1-14:2020 [5] in chapter 8.2.3 the extrapolation rules for the hot spot stresses are documented (also a quadratic extrapolation is provided). They are identical to the rules of the IIW-recommendations [6]. In this paper, the linear extrapolation for a fine mesh is chosen with the extrapolation points at a distance of 0.4 · t and 1.0 · t from the hot spot. For these two investigated details the quadratic extrapolation leads to nearly the same hot spot stresses (deviation under 2 %). Misalignment (only relevant for detail 1, see Fig. 1) can be neglected, if the misalignment is smaller than 5 % of the shell thickness t [4]. In the FEA of detail 1 no misalignment was considered. This leads to stress concentration factors (SCFs), which only consider the detail geometry, but no additional geometric imperfections.

as for the effective notch stress method is used. The nominal axial stress $\Delta\sigma_{nom}$ for both details is 100 N/mm² to determine the SCFs at the shell with the thinner thickness t_1. In chapter 4, other nominal stresses $\Delta\sigma_{nom}$ are used for comparison of the different fatigue design stress methods. Fig. 2b shows detail 2 with the ring stiffener. The specified radius r = 1 mm for the effective notch stress method is used at the transition from the weld to the base material. With a path at the outer shell surface, shown in Fig. 2b, the notch stress is determined. For all FE-calculations (Software ABAQUS was used), axial symmetrical, quadratic elements with 8 nodes and reduced integration are used. As a result, it is a plane model, which also considers

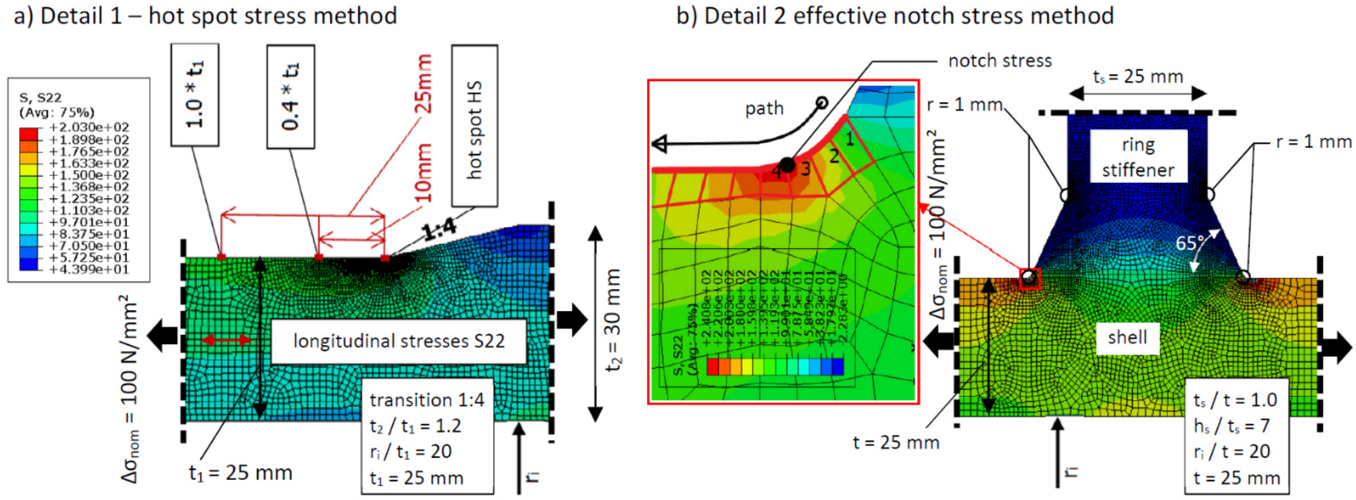


Figure 2 Axial symmetric FE-models of: a) detail 1 (hot spot stress method shown), b) detail 2 (effective notch stress method shown) with longitudinal stresses due to nominal stress $\Delta\sigma_{nom}$ = 100 N/mm²

The effective notch stress method is documented in [4], Annex C. With this method also welded constructional details with potential crack locations at the weld root can be analysed. In this method, not only the structural stresses (membrane and bending stresses) are considered, also the nonlinear stresses (notch stresses) at the weld toe or the weld root are taken into account. This is done by rounding the weld toe or root with a fictitious notch of radius r = 1 mm. A very fine mesh in the FEA is necessary to get accurate stress results near the rounding at the notch. In large FE-models sub modelling is required to fulfil the requirements for the FE-mesh. There are two different detail categories defined for the effective notch stress method, depending on the used stress components [4]. Detail category 200 is used for von Mises equivalent stresses, 225 is used for principal stresses. The effective notch stress method doesn´t cover the fatigue verification of welded details with mild notches (effective notch stress is less than 2 times the nominal stress), because unsafe results are expected. Mild notches can occur in detail 1, with a smooth transition and a small difference of the thicknesses t_1 and t_2 (see Fig. 1) and also in detail 2. These individual cases are marked in Fig. 3, 4, 5, 6 and 7, showing the calculated SCFs.

Fig. 2 shows some examples of the used Finite Element models to determine the hot spot stresses and the notch stresses, respectively. Fig. 2a shows detail 1 (thickness transition) with the relevant hot spot and the extrapolation points at a distance of 0.4 · t_1 and 1.0 · t_1 from the hot spot. For the hot spot stress method, the same FE-mesh

the circumferential stresses out of the plane. This hugely increases the computational efficiency. The material model of steel is ideal elastic with an elastic modulus E = 210 000 N/mm² and with Poisson's ratio ν = 0.3. No geometric imperfection (axial misalignment of the shell segments) was considered in this study.

3 Results from the FEA of detail 1 and 2

3.1 Stress concentration factors for detail 1 – thickness transition

In this chapter, the stress concentration factors (SCFs) for detail 1, loaded with axial stresses only, are presented. For detail 1, the decisive stresses are located at the outside of the shell (see Fig. 2). Fig. 3 shows the SCFs for the hot spot and for the effective notch stress method for different r_i/t-ratios with thickness t_1 = 25 mm. In the new Eurocode [4], the modified nominal stress method with appropriate SCFs has to be used for detail 1, to take the stress increasing effects of the thickness transition with plate eccentricity into account. These SCFs k_f of the modified nominal stress method (called k_f mod. nom. in Fig.3), calculated with Eq. (1) (equivalent to Eq. (D.6) in [4]), are shown in grey lines in Fig. 3. The inclination of the transition cannot be considered with Eq. (1), so the SCFs k_f of the modified nominal stress method are the same in Fig. 3a and Fig. 3b. Furthermore, Eq. (1) was originally developed for flat plates.

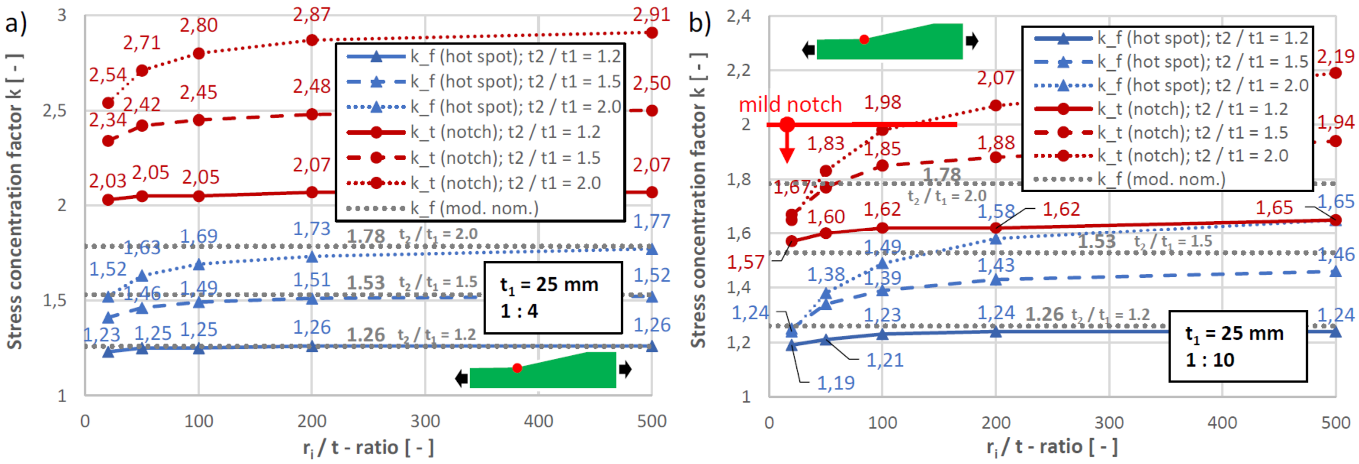


Figure 3 Detail 1 - Stress concentration factors (SCFs) for the hot spot (k_f hot spot), the effective notch (k_t notch) and the modified nominal stress method (k_f mod. nom.) for varying ratios r_i/t and t_2/t_1; thickness t_1= 25 mm; inclination of the transition: a) 1 : 4, b) 1 : 10

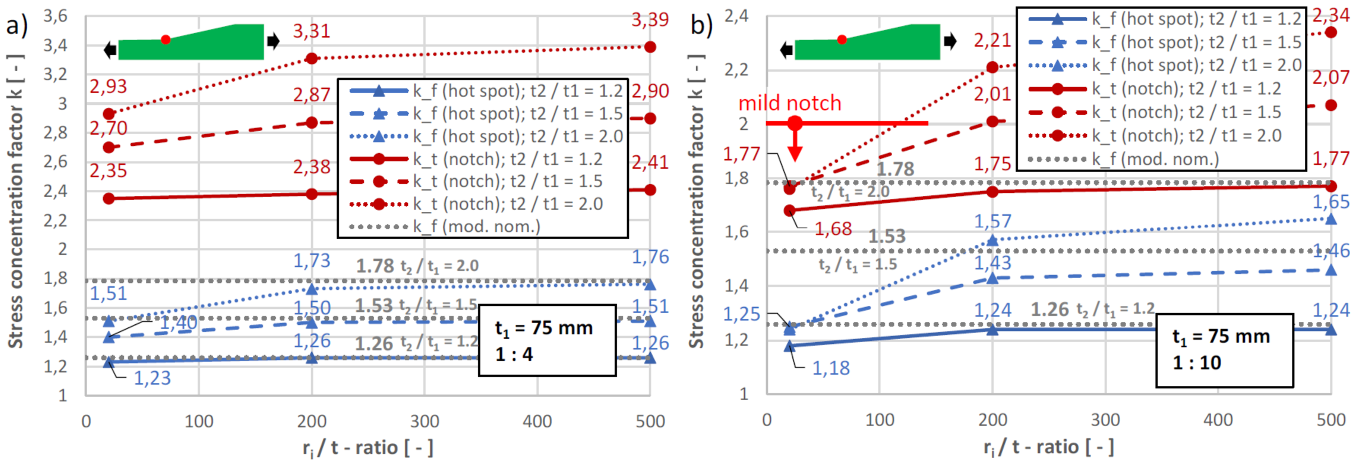


Figure 4 Detail 1 - Stress concentration factors (SCFs) for the hot spot (k_f hot spot), the effective notch (k_t notch) and the modified nominal stress method (k_f mod. nom.); thickness t_1 = 75 mm; inclination of the transition: a) 1 : 4, b) 1 : 10

Fig. 3a shows the SCFs for an inclination of the transition of 1 : 4. This Fig. 3a shows, that the SCFs k_f of the modified nominal stress method, calculated with Eq. (1), are always larger than the SCFs for the hot spot stress method. So, the SCFs k_f of the modified nominal stress method are very conservative for smaller r_i/t-ratios and larger t_2/t_1-ratios. The SCFs for the effective notch stress method have the same trend as the SCFs for the hot spot stress method, but they are significantly higher, as expected. Fig. 3b shows the SCFs for an inclination of the transition of 1 : 10. In Fig. 3b, the SCFs are smaller, compared with Fig. 3a, especially for smaller r_i/t-ratios and larger t_2/t_1-ratios. So, a smooth transition (smaller inclination) is very effective in fatigue design for small r_i/t-ratios and large t_2/t_1-ratios. In Fig. 3b it becomes also obvious, that the hot spot stress method leads to lower SCFs (also for r_i/t = 500 → behaviour almost like a flat plate), compared with the SCFs for the modified nominal stress method, because the inclination of the transition is not considered in Eq. (1). As marked in Fig. 3b, many SCFs for the effective notch stress method are smaller than 2.0 and are called mild notches [4]. Mild notches are not covered for fatigue verification with the effective notch stress method, because unsafe results are expected.

$$k_f = \left(1 + \frac{6}{t_1} \cdot \frac{t_1^{1.5}}{t_1^{1.5} + t_2^{1.5}}\right) \qquad (1)$$

Fig. 4 shows the SCFs for detail 1 with significant higher thickness t_1 = 75 mm, to see whether the size effect is considered in the SCFs or not. Fig. 4a shows the SCFs again for an inclination of the transition of 1 : 4, Fig. 4b for 1 : 10. The SCFs for the hot spot stress method and the modified nominal stress method (based on Eq. (1)) in Fig. 4 (t_1 = 75 mm) are almost identical to those in Fig. 3 (t_1 = 25 mm). Hence, the absolute thickness of the shell has no influence on the hot spot stresses. Comparing the SCFs for the notch stresses of Fig. 3 and Fig. 4, it becomes obvious, that those SCFs are influenced by the absolute thickness t_1 (k_t is higher in thicker shells). That is also a reason for the thickness factor k_s for thicker plates in [4], Annex B for the hot spot stress method, which reduces the fatigue resistance for thicker shells. This factor is shown in chapter 4 in Table 2. The SCFs for t_1 = 50 mm are not shown, because the SCFs k_f for the hot spot stress method are the same as for t_1 = 25 mm and t_1 = 75 mm. The SCFs for the effective notch stress method for t_1 = 50 mm are nearly identical to the mean values of the results for t_1 = 25 mm and t_1 = 75 mm. All SCFs, also for t_1 = 50 mm, are shown in [7].

3.2 Stress concentration factors for detail 2 – ring stiffener

Fig. 5 shows the SCFs at the weld toe of the ring stiffener (thickness t = 25 mm; h_s / t_s = 7). Fig. 5a shows the SCFs for a t_s/t-ratio of 1.0, Fig. 5b for a t_s/t-ratio of 2.0.

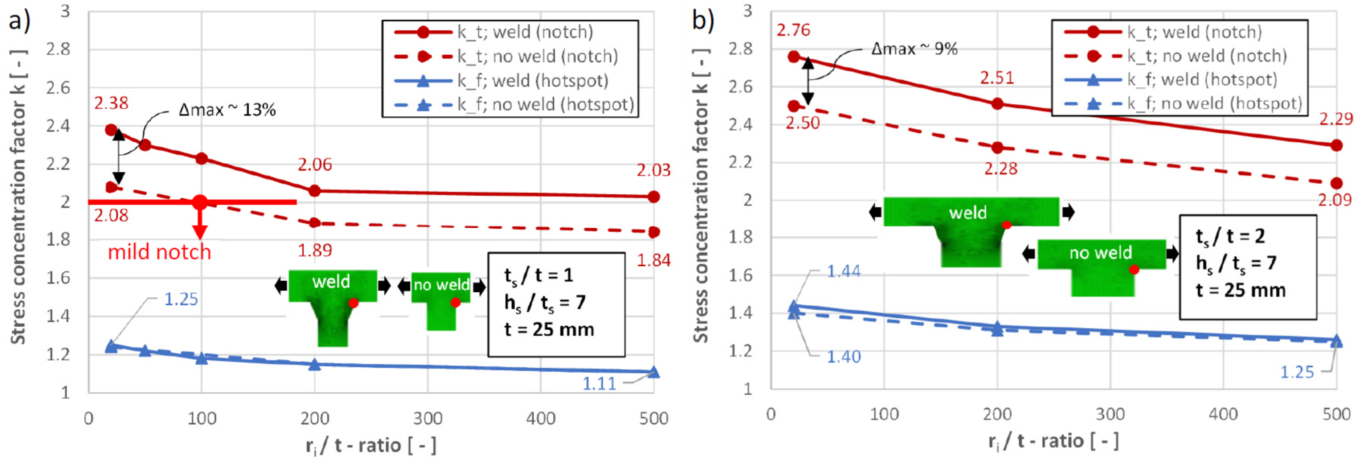


Figure 5 Detail 2 - Stress concentration factors (SCFs) for the hot spot (k_f hot spot) and the effective notch stress method (k_t notch); thickness t = 25 mm; h_s / t_s = 7; a) t_s / t = 1, b) t_s / t = 2

Two different types of weld geometry were investigated with the FEA. The first type (called "weld" in Fig. 5) considers the exact weld geometry (see Fig. 2) in the FEA. The second type (called "no weld" in Fig. 5) only considers the geometry of the ring stiffener and no widening of the weld. Fig. 5 shows, that the SCFs for the effective notch stress method k_t are more affected by the weld geometry. Higher SCFs k_t are obtained with the exact weld geometry ("weld", max. 13 % higher, see Fig. 5a). The SCFs for the hot spot stress method k_f are almost the same for both types of weld geometry. The tendency for the SCFs of detail 2 is different to those of detail 1. For detail 2 a different tendency is visible: The smaller the r_i/t-ratio the higher the SCFs.

If Fig. 5a and Fig. 5b, where cases with different t_s/t-ratios are compared with each other, it becomes obvious, that higher SCFs for the hot spot and the effective notch stress method are obtained with the thicker ring (t_s/t = 2), because the thicker ring stiffener causes a larger constraint for the axial membrane stresses and larger bending moments in the shell due to the larger constraint in radial direction of the cylindrical shell.

As for detail 1, the absolute shell thickness t of detail 2 has no influence on the SCFs for the hot spot stress method. This aspect is not visible in Fig. 5, because the SCFs for t = 50 mm and t = 75 mm are not documented here (see [7]). The SCFs for the effective notch stress method are again influenced by the absolute thickness t, as shown for detail 1. They are higher with larger shell thicknesses t (documented in [7]).

In general, the SCFs for a stockier ring with h_s / t_s = 3.5 are smaller (3 to 9 % smaller for the hot spot and effective notch stress method) than those for h_s / t_s = 7.0 and the SCFs for h_s / t_s = 14 are higher (2 to 8 % higher for the hot spot and effective notch stress method) than those for h_s / t_s = 7.0. These SCFs are not documented her, they can also be found in [7].

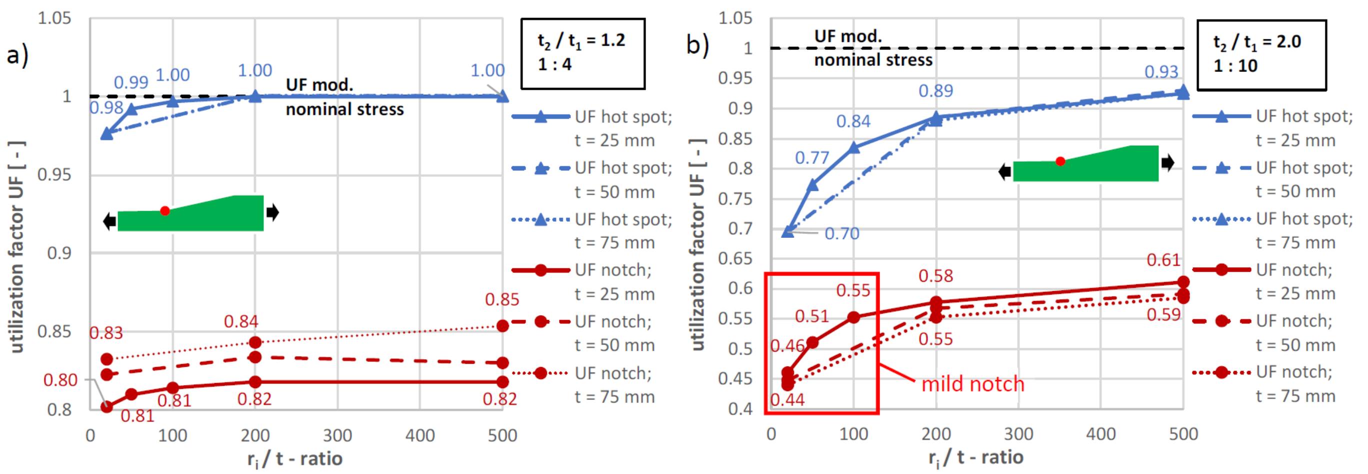


Figure 6 Detail 1 - thickness transition; Utilization factors (UFs) for detail 1 for hot spot- and effective notch stress method in comparison with the modified nominal stress method (UF = 1.0) for different plate thickness t = t_1; a) t_2 / t_1 = 1.2 and inclination 1 : 4; b) t_2 / t_1 = 2.0 and inclination 1 : 10

4 Discussion – Utilization factors for different fatigue design stress methods

4.1 Utilization factors for detail 1 - thickness transition

In this chapter, the utilization factors (UFs) for the three different fatigue design stress methods for detail 1 are shown. The nominal stress $\Delta\sigma_{nom}$ is chosen in such a way, that an UF of 1.0 results according to the modified nominal stress method (for $2 \cdot 10^6$ load cycles, see Eq. (2)). For example, the nominal stress $\Delta\sigma_{nom}$ for $t_2/t_1 = 1.2$ and $t_1 = 50$ mm is $\Delta\sigma_{nom} = 83.0$ N/mm². This results from the detail category $\Delta\sigma_{c,nom} = 112$ N/mm² for detail 1 (both sides ground flush, see Table 2) and the k_f-factor according to Eq. (1), leading to $k_f = 1.26$ ($t_2/t_1 = 1.2$) for the modified nominal stress method. Also, the wall thickness factor k_s in [4] (see Table 2), to consider the size effect, was applied ($k_s = (25/50)^{0.1} = 0.93$). This results in the aforementioned nominal stress of $\Delta\sigma_{nom} = 83.0$ N/mm² ($\Delta\sigma_{nom} = \Delta\sigma_{c,nom} \cdot k_s / k_f = 112 \cdot 0.93 / 1.26 = 83$ N/mm²). With Eq. (3), the UFs for the hotspot stress method can be determined. For this detail 1, the thickness factor k_s and the detail category for the hot spot stress method are the same as for the modified nominal stress concept (see Table 2). The stress concentration factor $k_{f\,(hot\,spot)}$ for the hot spot stress method is shown in Fig. 3 and 4. Eq. (4) shows the calculation of the UFs for the effective notch stress method with the SCFs $k_{t(notch)}$ (also shown in Fig. 3 and 4).

$$UF_{mod.\ nom.} = \frac{\Delta\sigma_{nom} \cdot k_f}{\Delta\sigma_{c,nom} \cdot k_s} = 1.0 \rightarrow \Delta\sigma_{nom} = \frac{\Delta\sigma_{c,nom} \cdot k_s}{k_f} \quad (2)$$

$$UF_{hot\ spot} = \frac{\Delta\sigma_{nom} \cdot k_{f(hot\ spot)}}{\Delta\sigma_{c(hot\ spot)} \cdot k_s} \quad (3)$$

$$UF_{notch} = \frac{\Delta\sigma_{nom} \cdot k_{t(notch)}}{\Delta\sigma_{c(notch)}} \quad (4)$$

Fig. 6 shows the UFs for detail 1 (thickness transition) for the different fatigue design stress methods, with variation of the thickness t_1 ($t_1 = t = 25; 50; 75$ mm). Fig. 6a shows the UFs for $t_2 / t_1 = 1.2$ and an inclination of 1 : 4, Fig. 6b for $t_2 / t_1 = 2.0$ and an inclination of 1 : 10. Fig. 6a indicates, that the modified nominal- and the hot spot stress method lead to nearly the same UFs. The UFs for the effective notch stress method are lower. They lie within 0.80 and 0.85. That means a service life based on the fatigue design up to 2 times longer ($\approx (1/0.80)^3$, based on a SN-curve with m = 3), compared to the other fatigue design stress methods.

In Fig. 6b with the smooth transition (1 : 10) the difference between the modified nominal and the effective notch stress method is even higher. Fatigue design with the effective notch stress method results in a service life up to 6 times longer ($\approx (1/0.55)^3$), compared with the modified nominal stress method. The SCFs for the effective notch stress method for r_i/t-ratio smaller than 100 are smaller than 2.0 (see also Fig. 3b and 4b; they are marked in Fig. 6b). So, they are mild notches and the effective notch stress method shouldn´t be used in these cases, according to [4], Annex C. In Fig. 6b, also, the UFs for the hotspot stress method are lower than those for the modified nominal stress method, because the SCFs for the latter one (k_f based on Eq. (1)) don´t take the positive effect of the smaller inclination 1:10 and the individual r_i/t-ratios into account.

The modified nominal stress method for high values of r_i/t ($r_i/t = 500$ → behaviour almost like a flat plate) should be seen as a reference for the hot spot and the effective notch stress method. Looking at the UFs in Fig. 6, the hot spot stress method should be the recommended method for fatigue design of this detail 1 in cylindrical shells. The effective notch stress method leads to very low utilization factors, compared to the modified nominal stress method and might be also on the unsafe side. Additional research activities seem necessary to clarify these beneficial results. Therefore, it is recommended to use the effective notch stress method for detail 1 only with appropriate misalignments. Misalignments is an often-discussed topic in the literature. Papers of Lotsberg [8] and Taras [9] show results for practical design.

Table 2 Detail category and thickness factor k_s for detail 1 for different fatigue design stress methods according to [4]

Detail 1 – thickness transition		
stress method	detail category	thickness factor k_s (k_s > 1.0)
modified nominal	$\Delta\sigma_{c,nom} = 112$	$k_s = \left(\frac{25}{t_1}\right)^{0.1}$ for $t_1 > 25\ mm$
	for both sides ground flush	for both sides ground flush
hot spot	$\Delta\sigma_{c,hot\ spot} = 112$	$k_s = \left(\frac{25}{t_1}\right)^{0.1}$ for $t_1 > 25\ mm$
	for both sides ground flush	for both sides ground flush
eff. notch stress	$\Delta\sigma_{c,notch} = 225$	-

4.2 Utilization factors for detail 2 - ring stiffener

In this chapter, the utilization factors (UFs) for the three different fatigue design stress methods for detail 2 are shown. The nominal stress $\Delta\sigma_{nom}$ is again chosen in such a way that an UF of 1.0 results according to the modified nominal stress method (see Eq. (2)). For this detail 2, the factors k_f and k_s in Eq. (2) for the modified nominal stress method are 1.0. So for detail 2 one gets $\Delta\sigma_{nom} = \Delta\sigma_{c,nom}$ and the results are equal to the nominal stress method. The detail category in this case is either 80 or 71 N/mm² for detail 2, depending on the length l (see Table 3). The length l is defined as the thickness of the stiffener t_s plus the widening of the weld. If l is smaller or equal 50 mm $\Delta\sigma_{c,nom} = 80$ N/mm², otherwise $\Delta\sigma_{c,nom} = 71$ N/mm². The accurate fatigue resistances $\Delta\sigma_{c,nom}$ are also marked in Fig. 7. With Eq. (3), the UFs for the hotspot stress method can be determined for detail 2, as for detail 1. Also, a wall thickness factor k_s has to be considered in the hot spot stress method, again based on the fatigue tests for flat plates (see Table 3, according to [4]). The SCFs for the hot spot stress method $k_{f\,(hot\,spot)}$ are shown in Figure 5. As for detail 1, Eq. (4) shows the calculation of the UFs for the effective notch stress method with the SCFs $k_{t\,(notch)}$ for detail 2 (also shown in Fig. 5).

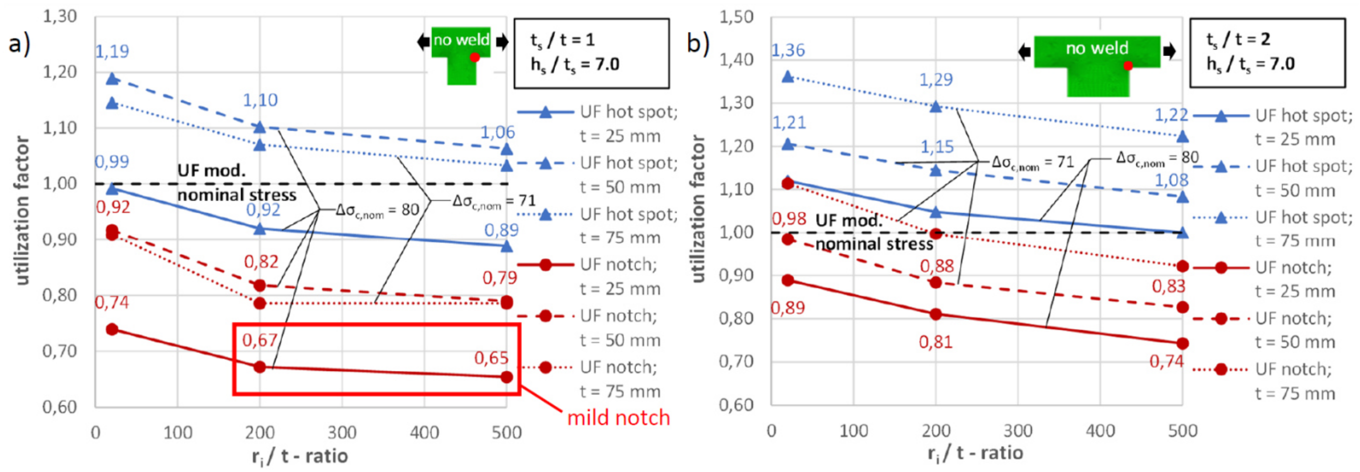
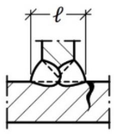


Figure 7 Detail 2 - ring stiffener; Utilization factors (UFs) for detail 2 with h_s / t_s = 7.0 for the hot spot- and effective notch stress method in comparison with the modified nominal stress method (UF = 1.0) for different plate thickness t; a) t_s / t = 1, b) t_s / t = 2

Fig. 7 shows the UFs for detail 2 (ring stiffener with h_s / t_s = 7.0) for the three different fatigue design stress methods. Fig. 7a shows the UFs for t_s / t = 1, Fig. 7b for t_s / t = 2.0. Please note, that the results for the modified nominal stress method are always UF_{nom} = 1.0. When Fig. 7a is compared with 7b, it becomes obvious, that the UFs for the hot spot stress method and the effective notch stress method are higher in Fig 7b with t_s / t = 2, because the thicker ring stiffener causes a larger constraint for the axial membrane stresses. This is in line with the reduced fatigue resistance of the nominal stress method for l > 50 mm. The increased thickness t_s is also the reason for higher stresses at the hot spot. The tendency of the UFs of detail 2 is different to those for detail 1. For detail 2 a different trendline holds: The smaller the r_i/t-ratio, the higher the UFs.

Table 3 Detail category and thickness factor k_s for detail 2 for different fatigue design stress methods according to [4]

Detail 2 – ring stiffener		
stress method	detail category	thickness factor k_s
modified nominal 	$\Delta\sigma_{c,nom}$ = 80 for $l \leq$ 50 mm $\Delta\sigma_{c,nom}$ = 71 for l > 50 mm	1.0
hot spot	$\Delta\sigma_{c,hot\,spot}$ = 100	$k_s = \left(\frac{25}{t_{eff}}\right)^{0.3}$ for $t_{eff} > 25\,mm$ $t_{eff} = min(14 + 0.66 \cdot l; t)$
eff. notch stress	$\Delta\sigma_{c,notch}$ = 225	-

In general, as for detail 1, detail 2 gives smaller UFs for the effective notch stress method, compared to the hot spot stress method. In this detail, misalignment has negligible influence on the UFs.

The SCFs for the effective notch stress method k_t for the marked cases in Fig. 7a are smaller than 2.0 (see also Fig. 5a).

So, they are mild notches and the effective notch stress method shouldn´t be used in these cases according to [4].

Fig. 7 also shows that different fatigue design stress methods lead to different UFs for detail 2. The thickness factor $k_s = (25/t_{eff})^{0.3}$ for the hot spot stress method [4] (see Table 3), which gives higher UFs, seems a little bit too conservative, compared with the modified nominal stress method.

5 Conclusion

In this paper, a study was presented in which two common welded constructional details in cylindrical shells (detail 1 – thickness transition and detail 2 – ring stiffeners) were analysed, with respect to the fatigue design of the three different stress methods. The modified nominal-, the hot spot- and the effective notch stress method were compared with each other according to [4].

Stress concentration factors (SCFs) for the hot spot and the effective notch stress method were presented over a wide range of geometric parameters. Especially, the influence of the r_i/t-ratio and the shell thickness on the fatigue design were investigated, also for higher thicknesses. It was shown, that the absolute shell thickness t has no influence on the hot spot stresses (for the same r_i/t ratios) for both investigated details, in contrast to the SCFs of the effective notch stress method. The SCFs of the effective notch stress method are influenced by the absolute shell thickness for the same r_i/t ratios and increase with increasing plate thickness t.

The results of the modified nominal stress method for fatigue design were compared with the hot spot- and the effective notch stress method in form of a comparison of the corresponding utilization factors (UFs), considering the different geometric parameters and also the size effect for thicker plates. Both details were loaded with axial stresses only. The modified nominal stress method for high values of r_i/t (r_i/t = 500 → behaviour almost like a flat plate) should be seen as a reference for the hot spot and the effective notch stress method. But, with the modified nominal stress method, the influence of the r_i/t-ratio of the cylindrical shells can't be considered. As presented, the r_i/t-ratio has different influences on the fatigue design of these two constructional details. For detail 1, a smaller r_i/t-ratio has a positive influence on the fatigue design, so smaller UFs for the hot spot and the effective notch stress method are obtained, compared with the modified nominal stress method. For detail 2, the influence of the r_i/t-ratio is different. If the r_i/t-ratio is small, higher UFs of the hot spot and the effective notch stress method are received.

The effective notch stress method leads to very small utilization factors, compared with the modified nominal stress method, especially for detail 1 with a smooth transition (inclination 1 : 10), also for higher r_i/t-ratios. Therefore, it is recommended to use the effective notch stress method for this detail 1 only with consideration of appropriate misalignments within the FEA. It seems, that the hot spot stress method is the best method for fatigue design for these two constructional details. In the hot spot stress method, the inclination of the transition for detail 1, the geometry of the ring stiffener h_s/t_s and the r_i/t-ratio for both details are directly considered in the FEA and the computational efficiency is much better as in the effective notch stress method.

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