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ON THE OPTIMAL CHOICE OF STRESS RELIEF GROOVES IN FLAT WELDED ENDS OF PRESSURE VESSELS

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ABSTRACT

Flat ends with inside stress relief grooves are a certain alternative for dished ends commonly used in pressure vessels. The stress relief grooves proposed in different standards concentrate on grooves of circular shape, although other forms are also admitted. The code EN12952-3 introduces three parameters describing the groove configuration – the groove radius and the minimum endplate thickness under the relief groove and the chamfer angle. The respective formulas for the first two parameters are expressed in a form of inequalities, which means that certain range of their variation is admitted. The existing codes do not give the clear suggestion about the optimal choice of values of the groove parameters, providing minimization of the stress concentration in the boiler. This work tries to answer the question how the optimal choice of stress relief groove parameters should be performed.

Keywords: pressure vessels with flat end caps, stress relief grooves, finite element method, optimisation.

INTRODUCTION

Flat ends of pressure boilers have been applied in power industry and pipe work structures for many years and can have different shapes and forms. Several, allowable proposals are shown in Fig. 1, which follows the code EN 12952-3:2001.

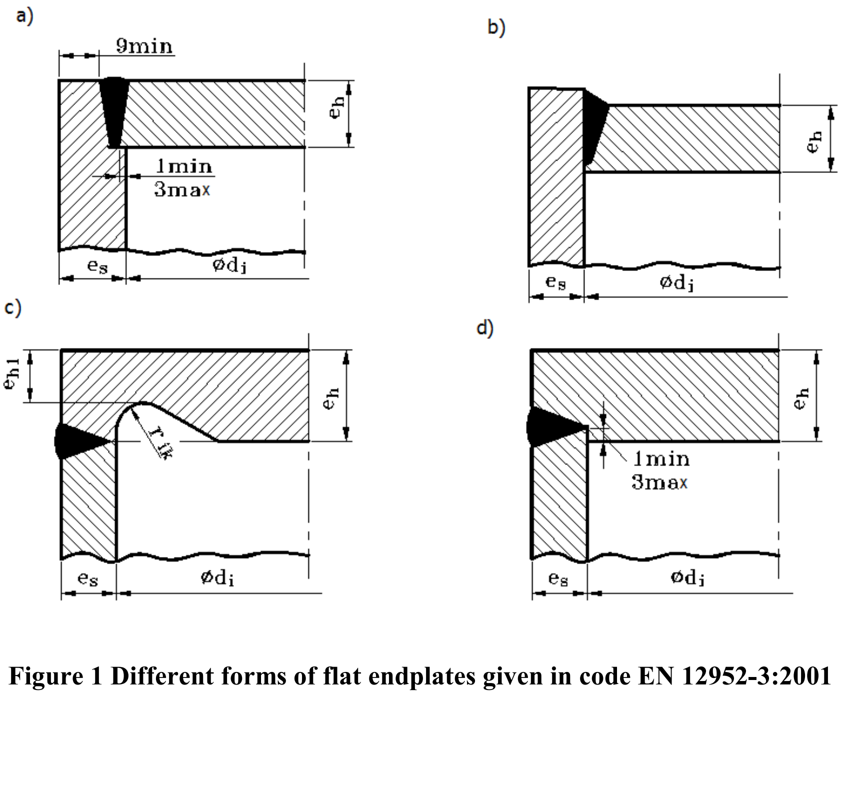

Figure 1 Different forms of flat endplates given in code EN 12952-3:2001

The common feature of the all presented above designs is the presence of the abrupt change of the shape in the vicinity of the cylindrical shell – flat endplate connection, which is the source of stress concentration. The consequence of the notch appearance may be the premature damage of the vessel or the presence of plastic deformations in the vicinity of the shell-endplate connection. This can be particularly dangerous in case of fatigue load existence. What is more, the observed failure incidents of boilers with flat endplates appeared as the breaks before leaks (Luedenbach 2004, Vilhelmsen 2000), which means that no visible evidences were spotted before the catastrophe. The usual practice, which protects against the boiler failures, is the reduction of the applied maximum, inner pressure when comparing it with that calculated for the cylindrical wall. On the other hand more attention could be paid to the proper choice of the groove parameters, which influences the reduction of the stress concentration.

The design shown in Fig. 1c is also recommended by the code EN 13445-3:2002. This code can be alternatively used in designing of pressure vessels with flat endplates (see Fig. 2). Both endplates shapes are not identical, and as it can be seen the proposal in EN 13445-3:2002 suggests the local increase of the shell wall thickness in the vicinity of the relief groove.

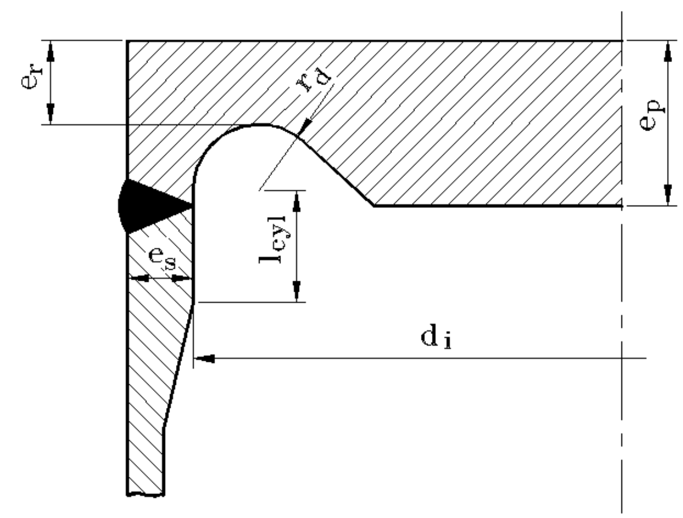

Figure 2 Flat endplate with stress relief groove in code EN 13445-3:2002

In both cases, two main parameters describing the groove configuration are the groove radius – r_d, and the minimum endplate thickness in the groove area e_r. The latter one can be alternatively replaced by the position of the groove centre – h (see Figure 3, EN 12952-3:2001). The chamfer angle α shown in Figure 3 can be regarded as the third additional parameter, but in further tests it will be proven its small influence on the numerical results. The parameters describing the admissible values of the groove configuration can vary within certain limits and the limiting conditions given in both cited codes slightly differ. The proposal given in EN 13445-3:2002 does not give the strict formulae for the assessment of the locally thickened shell wall e_s, which further influences the length l_{cyl}. These two values are suggested to set in the *design by analysis* procedure, which gives certain freedom for the designers and usually demands the application of finite element method in analysis. The more precise description for the groove parameters is given in code EN 12952-3:2001. Here the admissible values are set by means of system of inequalities, which take into account the shell wall thickness, the internal pressure and the endplate thickness. These formulas are as below:

$$\begin{gathered} e_r \geq e_s \\ e_r + r_d \leq e_p \\ r_d \geq \max\{0.2 \cdot e_s, \quad 5mm\} \\ e_r \geq 1.3\left(\frac{d_i}{2} - r_d\right) \cdot \frac{p_c}{f} \end{gathered} \tag{1}$$

where p_c the internal pressure in vessels is calculated using EN:12952-3, f stands for the admissible stress, and e_p is the endplate thickness calculated as follow:

$$e_p = C_1 \cdot C_2 \cdot C_3 \cdot d_i \cdot \sqrt{\frac{p_c}{f}} \tag{2}$$

Here, for the circular endplates constants C_2, C_3 are equal to 1.0, and the constant C_1 can be calculated with the use of the diagram given in the code. The minimum value for C_1 is 0.41, while its maximum values does not exceed 0.83. In the code there is also the analytical formula, which enables the calculation of C_1.

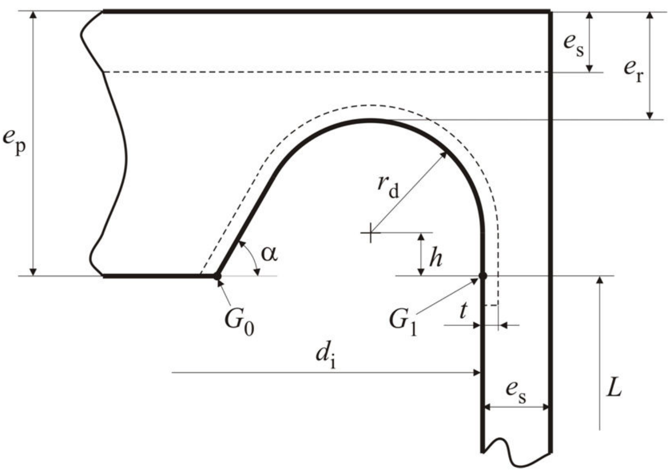

Figure 3 Configuration of circular stress relief groove with all parameters depicted (EN 12952-3:2001)

Figure 4 shows the admissible area for the groove parameters if the cylindrical tube ∅406.4×20 and the endplate with the groove are made from the 16Mo3 steel, with the yield limit: $R_e = 270\ MPa$. As it can be seen the admissible values for the groove radius and the minimum thickness of the endplate cover, in general, the quadrilateral area, which in certain cases (if the shell wall thickness increases) reduces to the triangular one. Several numerical calculations performed for different combinations of the groove parameters have clearly proved that the stress concentration strongly depends on the parameters values (Preiss, 1997, Szybiński, 2012). As a consequence the question how to choose the optimal values of the groove parameters should be raised. The problem of optimal choice of the stress relief groove parameters for arbitrary values of the cylindrical shell thickness has not been solved so far, however certain trials in this direction have been made. The simplified analysis of the flat endplate with stress relief groove and the cylindrical shell connection has been presented in papers of Kiesewetter (Kiesewetter 1989), Schwaigerer (Schwaigerer 1978) and Preiss (Preiss 1997). In the last mentioned paper it appeared that the most convenient location for the groove centre is the bottom edge of the endplate, but no indication how to choose the groove radius have been given. The assumption concerning the groove centre location on the bottom edge of the endplate was also used in next papers concerning the creep problems and the limit load assessment in pressure boilers with the flat endplates (Preiss and al. 1998, Vilhelmsen 2000). All these considerations concern the stress relief grooves with the circular shape in predominant part of the groove, and naturally the question about the application of other shapes can be stated. Several papers prove that the elliptical shapes applied for stress relief grooves are less severe notches than circular ones (Kristiansen and al. 1976, Pedersen and al. 2008, Pedersen 2008), so that the proposed area of investigation will concentrate not only on the optimal choice of the circular stress relief groove parameters but will also study the possible benefits coming from the application of elliptical notches. The above mentioned two shapes belong to the one family of curves, named super-ellipsis proposed by Pedersen:

$$\left(\frac{x}{a}\right)^n + \left(\frac{y}{b}\right)^n = 1 \tag{3}$$

here a, b, n are the design variables and if $n = 2$ and $a = b$ then we get the circular shape, if $a \neq b$ we get the elliptic shape, and if $n \to \infty$ than we get rectangular shape in the limit.

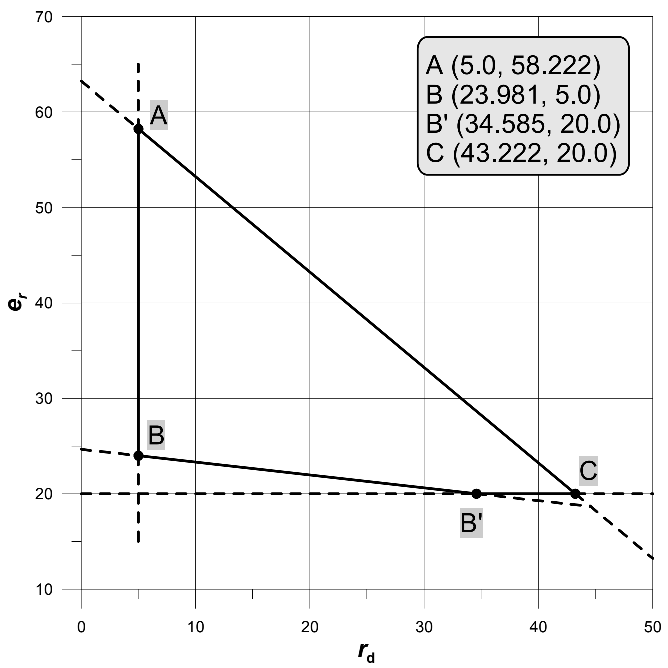

Figure 4 Admissible area for stress relief groove parameters for tube ∅406.4×20 made from steel with $R_e = 270\,\boldsymbol{MPa}$

The studied problem of the optimal choice of the stress relief groove providing the minimum stress concentration can be regarded as a parametric optimisation, in which the optimal vector of design variables is obtained on the base of analysis of the sequence of solutions obtained for different combinations of design variables. Various optimization criteria can be applied in search of the optimal groove parameters (Muc 2005, Muc and Muc-Wierzgoń 2012). In case of the elastic analysis the most common is the minimization of the maximum value of the stress concentration factor, which can be expressed as follow:

$$F_s = \min\left\{\max\frac{\sigma_{eqv}}{R_e}\right\} \tag{4}$$

here R_e is the yield limit, while σ_{eqv} defines the equivalent stress following the von Mises hypothesis. And the optimization should be performed over the whole admissible area. If the material exhibits the elastic – plastic deformations then besides the above criteria other forms of the objective functions can be introduced. One of them is the minimization of the maximum value of the equivalent plastic strain:

$$F_e = \min\{\max \varepsilon_{pl\_eqv}\} \tag{5}$$

Here, ε_{pl_eqv} is the equivalent plastic strain. This criterion can be also expressed with the use of ε_0 - the maximum elastic strain defined as:

$$\varepsilon_0 = \frac{R_e}{E} \tag{6}$$

Then it has the form:

$$F_e = \min\left\{\max\frac{\varepsilon_{pl\_eqv}}{\varepsilon_0}\right\} \tag{7}$$

In the presented study the numerical investigation were performed with respect to criteria defined in (5). For that purpose the linear elastic-plastic properties of the materials for the shell and the endplate are assumed (see Figure 5). The Young modulus is set to: $E = 2.1\times10^5$ *MPa*, while hardening module is equal to: $E_t = 780$ *MPa*. The numerical calculations were performed for the finite element model shown in Figure 6c (part of the structure). The choice of the model depends on the geometry of the structure, system of loadings and supports. In the analyzed problem the axial symmetry of the boiler geometry is used and the internal pressure is the only load applied. In such a case the axially symmetric model can be used with symmetric boundary conditions applied on all supported edges. This decreases the size of numerical task in a meaning way and helps to reduce the calculation time.

Certain additional comments concerning the proper choice of the optimization criteria are needed here. The Figure 7 shows the typical distributions of normal (σ_n), circumferential (σ_c), tangential to the boundary (σ_t) and equivalent stresses along the groove boundary obtained for certain combinations of the admitted groove parameters in elastic analysis. As it can be seen the circumferential and tangential stresses change its sign when moving along the groove boundary, it suggest that the difference between the absolute maximum and absolute minimum values for the respective stresses can also be chosen for the minimization procedure.

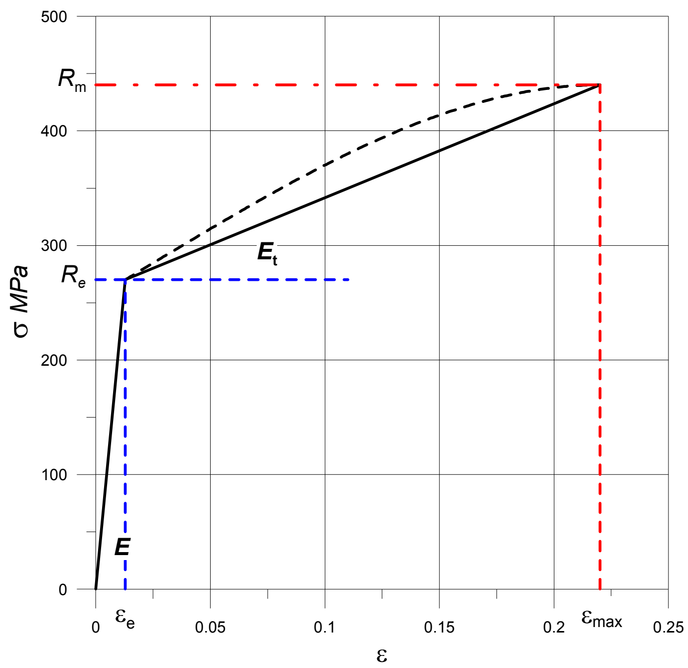

Figure 5 Stress – strain curve and approxiamtion with linear hardening

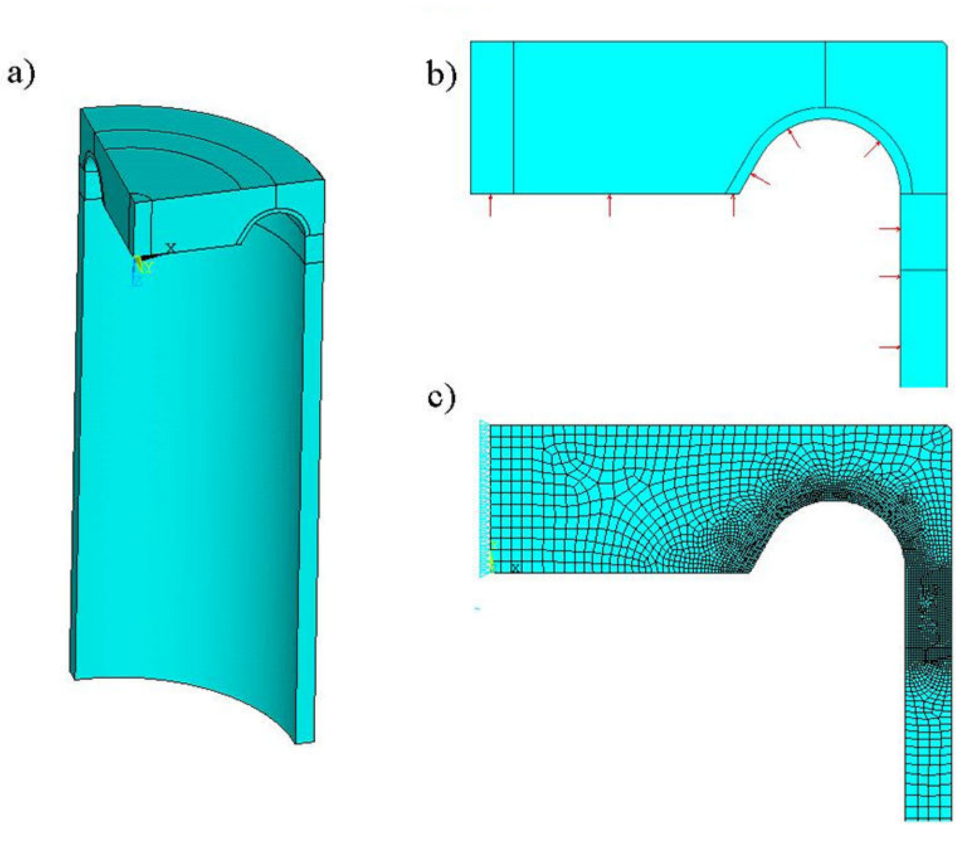

Figure 6 ¼ of half of cylinder with flat endplates with stress rlief circular groove (a), part of axisymmetric model subjected to internal pressure (b), exemplary finite element mesh (c).

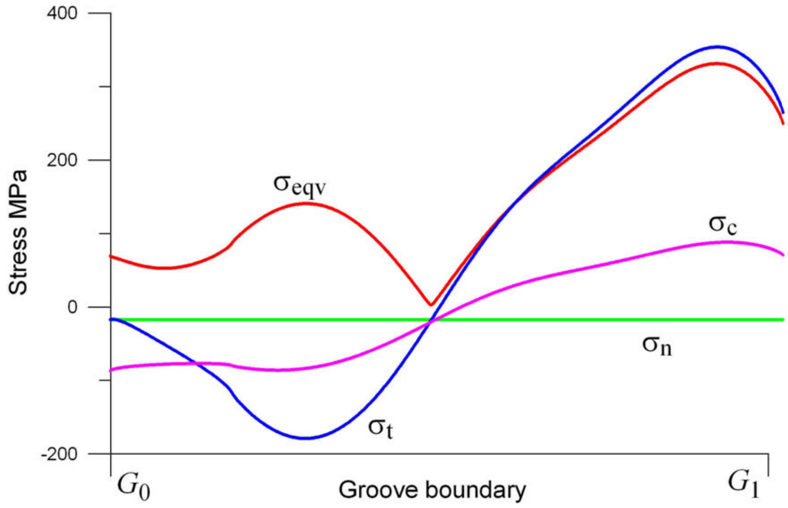

Figure 7 Distribution of stresses along the contour G_0G_1 of circular stress relief groove

NUMERICAL RESULTS FOR CIRCULAR STRESS RELIEF GROOVE

The first study, presented in the paper, concentrates on stress relief grooves of circular shape, and the exemplary numerical investigations were performed for the steel pipe ∅406.4×20 (material 16Mo3, the yield limit R_e = 270 *MPa*) and the material model shown in Figure 5. For the analyzed pipe the maximum calculated - following the code EN 12952-3:2001 - internal pressure was applied: $p_{int} = 18.633\ MPa$. On that base the minimum thickness of the endplate was calculated, and is equal to: $e_p = 63.22\ mm$. This value is valid only when the manufacturing tolerance and corrosion allowance are set to 0. The system of limiting conditions (1) for the groove parameters results in the following ranges for the groove radius:

$$\begin{aligned} r_d &= \langle 5.0 \div 43.22 \rangle mm \\ e_r &= \langle 20.0 \div 58.22 \rangle mm \end{aligned} \tag{8}$$

The detailed shape of the admissible area for both parameters is shown in Figure 4. The presence of the notch in the groove area results in appearance of certain plastic deformations in the zone of stress concentration. So that for the optimization the criteria (5) – minimization of equivalent plastic strains was chosen. In the performed analysis the chamfer angle α was set to 60 deg, as in traditional industrial applications. The influence of the chamfer angle α on elastic analysis results was studied in (Szybiński 2012) and it appeared that for values bigger or equal to the 60 deg only small reduction of stress concentration was observed for optimal configuration of design parameters. So that only two design variables - $(r_d, \quad e_r)$ - were the important optimization parameters. In such situation the simple search method was used to find the optimal configuration of the vector of the design variables, providing the minimum value of the maximum equivalent plastic strain in the whole analyzed part of the boiler. The results of that analysis are shown in contour plot shown in Figure 8. It presents the distribution of the maximum equivalent plastic strains over the whole admissible area. As it can be seen the proper choice of values for the design variables is the crucial one. Its influence on the resulting values of maximum equivalent plastic strains can be observed in

Figure 8, which vary between 60×10^{-5} and 400×10^{-5}. Additionally, one distinct minimum point can be observed. This minimum point appears along the edge AC, what means the centre of the circular lies at the bottom edge of the endplate ($h = 0.0$). This follows, to a certain extent, the results obtained by Preiss. The distribution of the maximum equivalent plastic strain along the edge AC is shown in Figure 9, where the minimum appears in point D. The optimal value for the optimization criteria is equal to: $F_e = 69.56$ and is reached for: ($r_d = 30.74mm,\quad e_r = 32.48mm$).

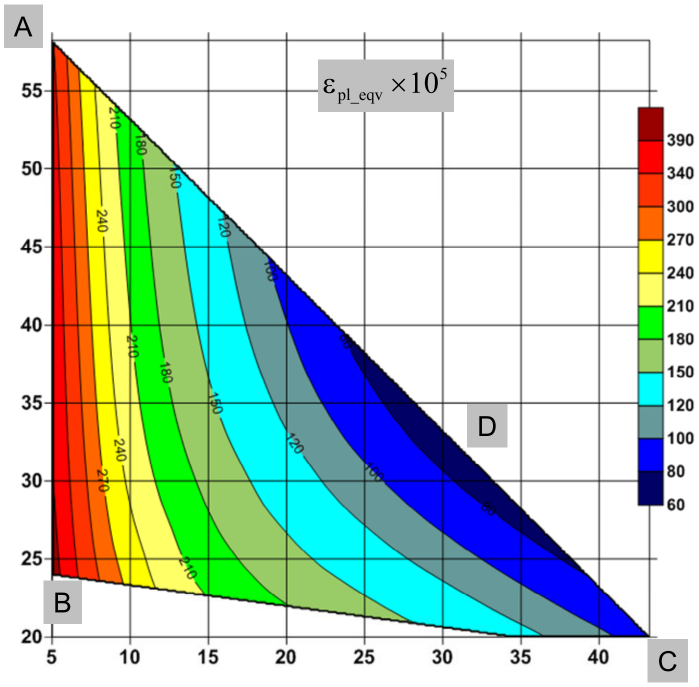

Figure 8 Contour plot of equivalent plastic strains over the parameters admissible area for circular stress relief groove

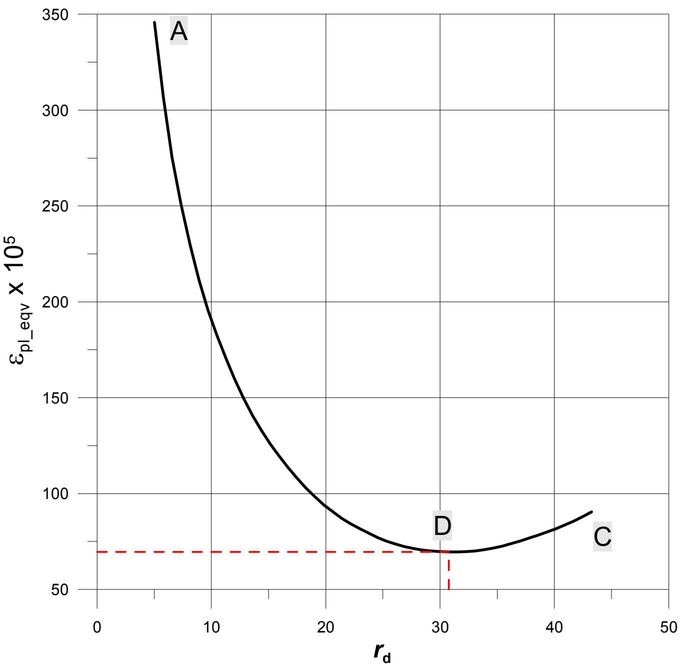

Figure 9 Distribution of equivalent plastic strains along the edge AC with optimal point D depicted

The similar analysis were performed for other typical tubes with outside diameters $\varnothing 406.4mm$. For the studied wall thicknesses, like: 22.2*mm*, 25.0*mm*, 28.0*mm*, 30.0*mm*, 32.0*mm*, 36.0*mm*, 40.0*mm* and 45.0*mm*, the optimal points, providing the minimum value for the maximum equivalent plastic strains, were located on the respectively modified edge AC with optimal point D approaching the corner C. The common conclusion for all these analyzed cases was the existence of one optimum point over the whole admissible domain for design parameters. This point was always located on the edge AC, rather in the close vicinity of the corner C. Unfortunately, no clear evidence was found how to choose the exact location of the optimal point D. This demands further thorough study.

NUMERICAL RESULTS FOR ELLIPTICAL STRESS RELIEF GROOVE

The motivation for this study were the results presented in numerous papers showing that notches with elliptical shapes can be less harmful for numerous structures than circular notches (Kristiansen, 1976, Pedersen, 2008, Pilkey 1997). Again as previously the same shell wall thickness and the same endplate thickness were assumed and the linear elastic – plastic material models were assumed for the shell and the endplate. In this approach at least three different optimisation parameters can be considered, namely a,b,h. The fourth parameter is the chamfer angle α, which is not illustrated in Figure 10 and is set to 90° but all these variables are not of the same importance. As it was proved in the elastic analysis, performed for the elliptic stress relief groove (Szybiński, 2012), the smallest stress concentrations were obtained for $h = 0.0$ and for chamfer angle $\alpha = 90^\circ$. This was profited in the current analysis and enabled to reduce the number of design variables to only two - a,b. In such a case again the simple search method over the whole admissible area was used. Here the limits for values a,b were set in a similar way as for the r_d in the previous analysis:

$$a,b = \langle 5 \div 43.22 \rangle mm \tag{8}$$

Here the limiting area covers a square. In Figure 11 the distribution of equivalent plastic strains over the whole area is shown. This surface plot was prepared on the base of grid point including the chosen values for a,b and the corresponding value of the maximum equivalent plastic strain obtained for the analyzed structure. Then the Kriging method was used to obtain the approximating surface (SURFER, 2009). As it was expected the most dangerous situation appeared for the smallest values of *b* and for the biggest values for *a*. The visible reduction of the maximum equivalent plastic strain value is observed for bigger values for *alb*. The next plot – Figure 12 – shows the distribution of the maximum equivalent plastic strain over the whole area in a form of the topographic map. One distinct minimum appears for the parameters as follow: $a = 20.06mm$, $b = 38.20\,mm$ then $\varepsilon_{pl\_eqv} \times 10^5 = 26.85$. This value is rather small and is more than two times smaller than in case of the optimal circular shape ($\varepsilon_{pl\_eqv} \times 10^5 = 69.56$). However, the full elimination of plastic deformations in the groove was not possible. Similar results for the elliptic groove were obtained with the first order optimization method existing in the ANSYS code, but in this case care must be taken when choosing the starting point for the optimization procedure and the tolerance of parameters when defining the design variables and objective functions. Here the best convergence to the optimal point was obtained when the starting point was set at the maximum value for *a* and the minimum value for *b*. The minimum value for the maximum equivalent plastic strain in analyzed structure was analogous to that one obtained with the simple search method. Additionally it is worth seeing that the solutions for maximum equivalent plastic strains presented in Figure 12 are not symmetric with respect to the diagonal joining points with minimum (5*mm*) and maximum (43.22*mm*) coordinate values for *a* and *b*. This observation conforms to earlier conclusion given in Pedersens papers and diagrams in Pilkey's monograph.

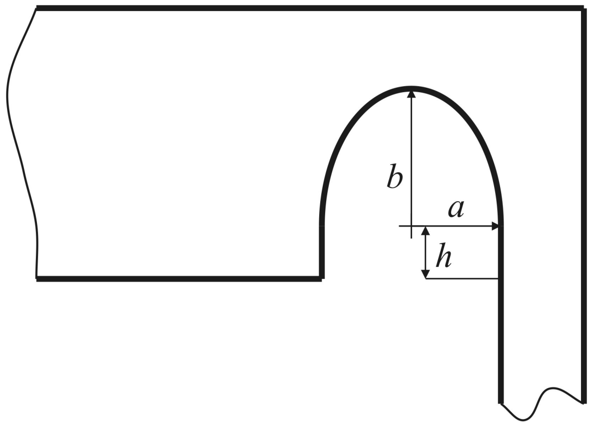

Figure 10 Elliptical stress relief groove with optimization parameters

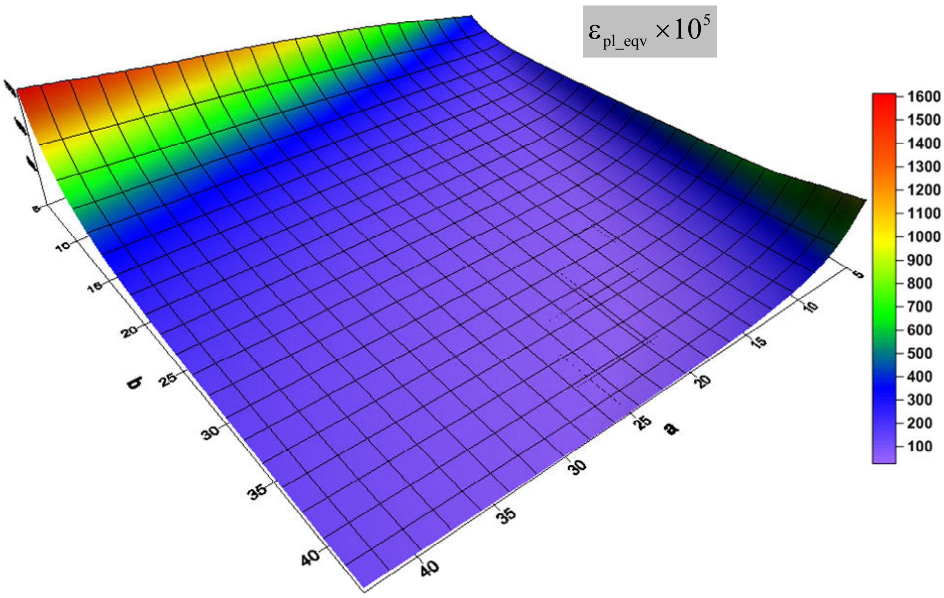

Figure 11 Distribution of equivalent plastic strain over the whole admissible area for *a* and *b* values

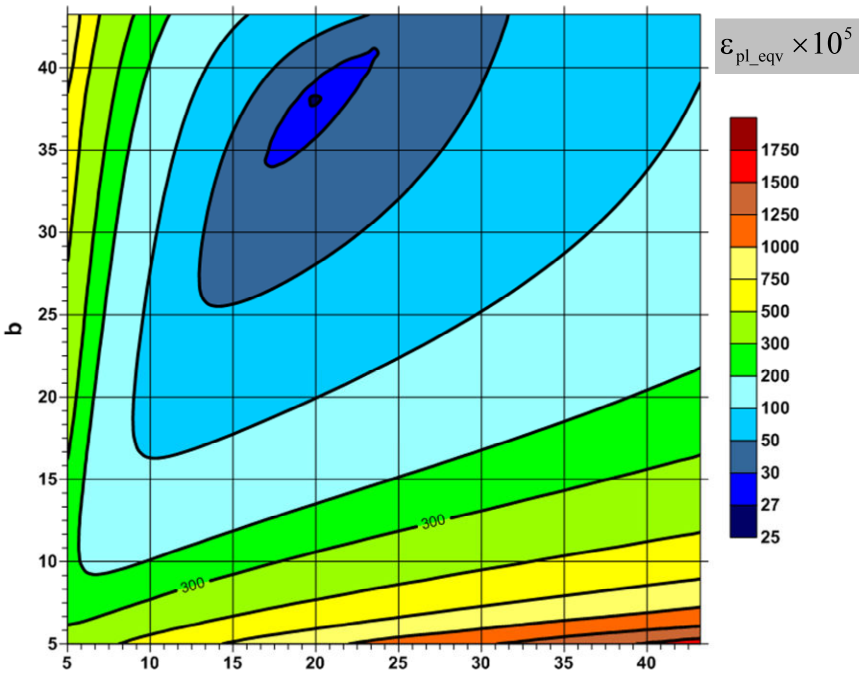

Figure 12 Contour plot of equivalent plastic strains over the parameters admissible area for elliptic stress relief groove

CONCLUDING REMARKS

The performed optimization procedure for two types of stress relief grooves for flat endplates in boilers has proved the existence of optimal values for the groove parameters, which provide the minimum stress concentration expressed by the maximum equivalent plastic strain. In case of only two optimization variables application of the simple search method seems justified and gives satisfactory results. This procedure fails when the number of design variables crosses two because the calculation time increases enormously, and is also difficult to illustrate the analysis results. As it was expected the elliptic shape of the groove has proven its superiority over the circular groove (see Figure 13), and should be recommended for use in industrial applications.

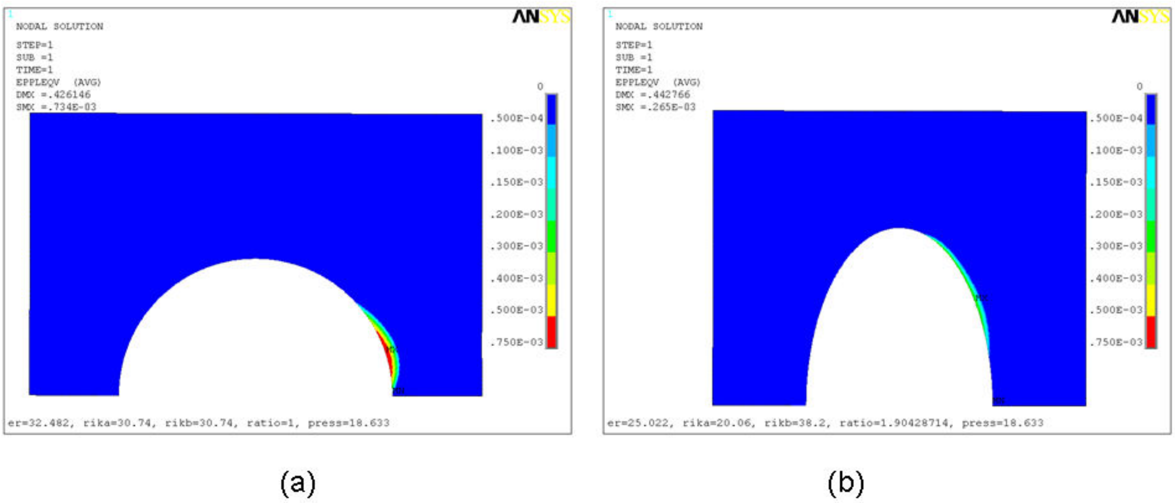

Figure 13 Contour plot of equivalent plastic strains in the vicinity of the stress relief grooves obtained for optimal configurations of circular groove (a) and elliptical groove (b)

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