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# DESIGN MODIFICATIONS OF PRESSURE BOILERS WITH FLAT ENDPLATES WITH STRESS RELIEF GROOVE

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***Abstract***

*Stress concentration is still present in the vicinities of structural notches in engineering structures. Zones with higher level of stress are particularly dangerous for structures subjected to fatigue or dynamic loads. The vessels with flat endplates with stress relief grooves are the common examples of such structures. Unfortunately, no clear evidence is given in existing codes how optimally to choose the circular groove radius and the optimal value of that radius depends on the dimensions of the boiler, operating conditions and the material used in manufacturing. Additionally, commonly used grooves with circular shapes are not optimal. Series of experimental tests, numerical or analytical studies presented in numerous articles confirmed these facts. In the presented article, the Authors proposed two-stage modification of the investigated vessel. The first step relies on change of the shape of the groove in the endplate, which provides certain reduction of stress concentration but still plastic deformations in the groove vicinity are not eliminated. The second step of the design modification is proposed for the boiler with the optimal elliptic groove configuration. In this step, some material is added around the top of the outer edge of the endplate. Two simple shapes are proposed for these parts - the short cylindrical ring or alternatively the circular ring welded on the top the endplate is used. For both concepts, the search of optimal dimension/parameters is performed. The numerical results of that study clearly show that the full elimination of plastic deformations near the groove is possible. The numerical analysis and optimizations were made with the well- established finite element software ANSYS, which is accepted by commonly used codes for designing of pressure vessels.*

***Keywords:*** *pressure boilers, flat ends, stress concentration, FEM analysis*

## 1. Introduction

Application of flat ends in pressure vessels is a certain alternative for well-established and commonly used dished ends. These flat ends can have various forms, suggested by respective codes [1, 2], and are shown in Fig. 1. In all recommended solutions the vessels, ends are joined with cylindrical or non-cylindrical tubes (carrying the load pressure) by means of welding. The rapid change of the shape near the head and tube connection results in relatively high stress peak, which is the subject of the extensive numerical and experimental studies throughout the years [4, 5, 7, 10-14]. In numerous numerical calculations, it was proven that the application of flat end with the stress relief groove provides the minimum stress concentration in the joint area. However, the level of stress still remains so high, that the reduction of the operating pressure is necessary.

The commonly used design codes [1, 2] suggest the application of the circular shape for the major part the stress relief groove, and this shape appears to be non-optimal. The calculation of the flat endplate dimension starts from establishing the endplate thickness, which is calculated as below [1]:

$$e_h = C_1 \cdot C_2 \cdot C_3 \cdot d_i \cdot \sqrt{\frac{p_c}{f}}\,. \tag{1}$$

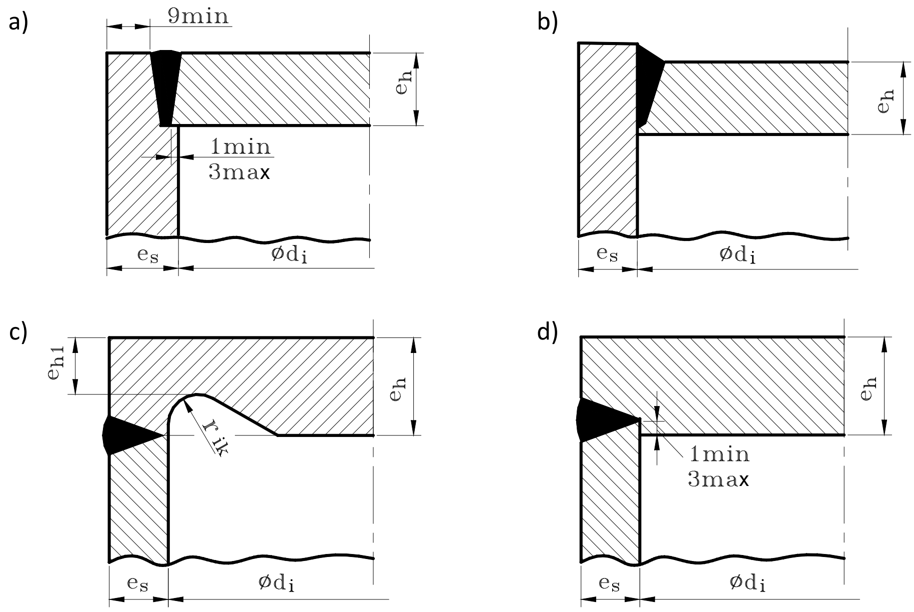


*Fig. 1. Four recommended by EN 12952:3 designs for flat ends of boilers*

Here, $d_i$ is the inner boiler diameter, $p_c$ is the maximum operating pressure calculated for cylindrical part of the boiler, $f$ stands for the admissible stress (this is valid for the same tube and endplate material). Constant $C_1$ depends on the ratio between applied pressure $p_c$ and the admissible stress $f$. It is taken form the plot included in the code. In fact, the value of $C_1$ varies from 0.41 to 0.82. The constants $C_2$ and $C_3$ are set to 1.0 in case of boilers with circular cross-section and endplate without cross-bore.

The parameters of the stress relief groove are set by a system of inequalities as follow:

$$\begin{cases} e_{h1} \geq e_s \\ e_{h1} + r_{ik} \leq e_h \\ r_{ik} \geq \max\{0.2 \cdot e_s,\ 5\ \text{mm}\} \\ e_{h1} \geq 1.3\left(\dfrac{d_i}{2} - r_{ik}\right) \cdot \dfrac{p_c}{f} \end{cases}. \tag{2}$$

The above set provides certain range for the groove radius $r_{ik}$ and the minimum endplate thickness $e_{h1}$ values. Here $e_s$ is the tube wall thickness and $d_i$, $p_c$, $f$ are defined in expression (1). The searched values usually cover certain polygonal area [11]. Such circumstance opens the problem of the optimal choice of the parameters defining the groove configuration. Different criteria and optimization techniques can be used in the considered case [8, 9]. Usually in structural problems, the objective function is proposed as the minimization of the maximum value of the stress concentration factor, which is expressed as follow:

$$F_\text{s} = \min\left\{\max \frac{\sigma_{eqv}}{R_e}\right\}, \tag{3}$$

here $R_e$ is the yield limit, while $\sigma_{eqv}$ defines the equivalent stress following the von Mises formulae. This approach is justified for the structures, in which only elastic response to the applied load exists. In case of structures, where elastic-plastic deformations appear and the hardening module is $E_t$ is rather low, the minimization of the maximum value of the equivalent plastic strain seems to be the most convenient:

$$F_\text{e} = \min\{\max \varepsilon_{pl\_eqv}\}. \tag{4}$$

Here, $\varepsilon_{pl\_eqv}$ is the equivalent plastic strain. This criterion can be alternatively expressed with the use of $\varepsilon_0$ – the maximum elastic strain defined as follow ($E$ – is the Young modulus): $\varepsilon_0 = R_e/E$.

Thus, the optimality criterion can take the form as follow:

$$F_e = \min\left\{\max \frac{\varepsilon_{pl\_eqv}}{\varepsilon_0}\right\}. \tag{5}$$

Due to the fact that only two designing parameters appear, the simple search method was used in order to find the optimal, if they exist, values for the radius of the groove and the minimum endplate thickness. In the considered problem no analytical solution can be given, so that the finite element model was used to get the approximate solution. In order to get the reliable results the ANSYS finite element code was used.

The performed numerical analysis of pressure boilers with endplates with circular stress relief groove resulted in several important observations and conclusions [12-14]. These are as follow:

- no influence of the inclination angle in the groove on the stress concentration is observed if its value is bigger than 60°,
- for each boiler the optimal pair of radius $r_{ik}$ and the minimum endplate thickness $e_{h1}$ values can be stated,
- the optimal combination of radius $r_{ik}$ and the minimum endplate thickness $e_{h1}$ values is such that: $e_{h1} + r_{ik} = e_h$, this means that the centre of circular groove lies on the bottom edge of the endplate,
- the optimal value of the radius $r_{ik}$ is usually close to the maximum, admissible value of the radius with respect to the remark given above.

## 2. Exemplary results of FEM calculations

The finite element model was prepared to perform the numerical analysis of the boiler. Due to the object symmetry (geometry and loads), only ¼ of the axisymmetric model was used including the half cross-sections of the considered endplate and the half-length of the cylindrical tube. This allowed reducing the full 3D numerical problem to the analysis of ¼ of axial cross-section of the boiler and application of axisymmetric finite element used in analysis of axisymmetric structures. The finite elements (PLANE183), which have quadratic displacement approximation, were used in analysis. Such elements are particularly efficient in modelling of structures with notches and irregular meshes. The mesh of finite element used in analysis was constructed in such a way that the solution error measured in stress energy was kept below 2.5%. As the calculations were made in a loop, this demanded the automatic modification of the mesh in the groove vicinity, each time adapted to the current values of the groove radius and the minimum endplate thickness. In the detailed FEM calculations, the material model (stress-strain curve) with linear elastic and linear hardening module was used. Such material model approximates fairly well the behaviour of the molybdenum constructional steel 16Mo3, which is widely used for the construction of pressure boilers and vessels. For calculation purposes it was assumed: the Young modulus: $E = 2.1\times10^5$ MPa, the hardening module: $E_t = 780$ MPa, the elastic limit $R_e = 270$ MPa, the tensile strength $R_m = 440$ MPa, and the maximum strain $\varepsilon_{\max} = 0.22$.

The presented below detailed results were obtained for cylindrical shell following the shape presented in Fig. 1c with the inclination angle 90°, the outside diameter Ø406.4 and the vessel wall thickness $e_s$ =20 mm. For these values the endplate thickness was set to: $e_h$ =63.22 mm while the maximum operating pressure was equal to: $p_{int} = 18.634$ MPa. The admissible, calculated with respect to the set (1), values covered the ranges for: $r_{ik} \in \langle 5.0, 43.22\rangle$ mm and $e_{h1} \in \langle 20.0, 58.22\rangle$ mm. The performed optimization procedure resulted in one minimum point over the whole admissible area for $r_{ik} = 31.34$ mm and $e_{h1} = 31.88$ mm, and the value of the objective function (in the sense of (5)): $F_e = 73.37\times10^{-5}$. The distribution of equivalent stress and equivalent plastic strain for optimal configuration is shown in Fig. 2. As it can be seen full elimination of plastic strains is not possible, which is unacceptable for the point of view of structure exploitation.

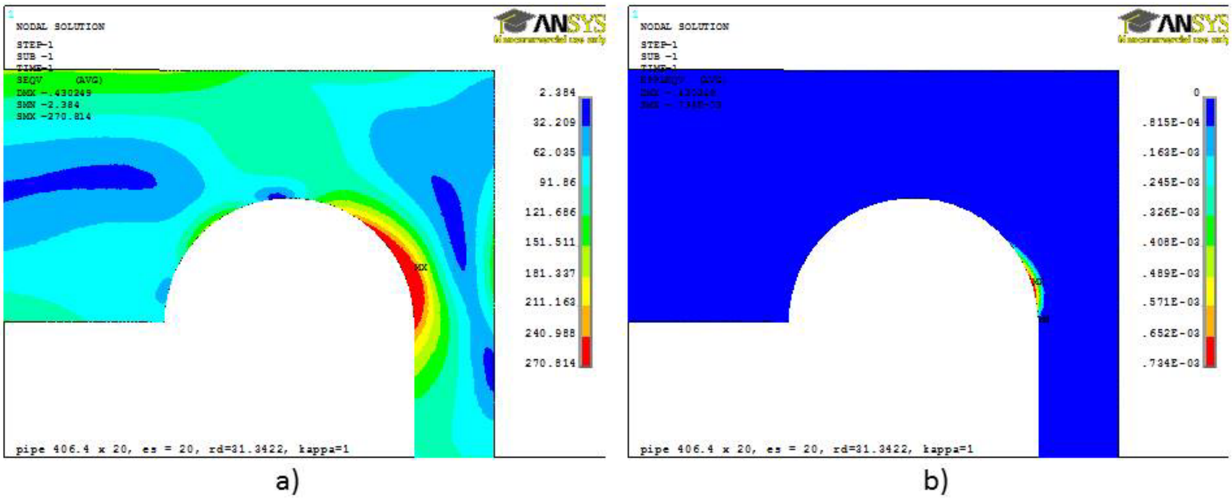


*Fig. 2. Equivalent stress (a) and equivalent plastic strains distribution (b) for optimal parameters of stress relief groove of circular shape*

It is well known [5, 8, 9] that the circular shapes for notches are not the optimal ones, so that certain improvement can be expected when the elliptic shape of the groove would be applied. The optimal choice of the groove, defined by the ellipsis semi-axes $a$, $b$, was searched in further numerical FEM calculations. The results of that study are presented in Fig. 3. A certain reduction of stress concentration in the notch area was observed. Namely, the maximum equivalent plastic strain was reduced to: $F_e = 26.63\times10^{-5}$. Such reduction was observed for the value of the vertical semi-axis of the ellipse: $b = 38.31$ mm and the horizontal semi-axis equal to $a = b\,/\,kappa =$ $= 20.27$ mm. However, the requirement of non-existence of plastic deformations in the analysed structure was still not fulfilled. The similar situation for different boiler diameters and wall thickness combinations is observed. So that, the modification of the shape of the groove from circular to elliptic does not fully eliminate the plastic deformations and only some additional steps must be undertaken to improve this situation or the reduction of the operating pressure will be necessary.

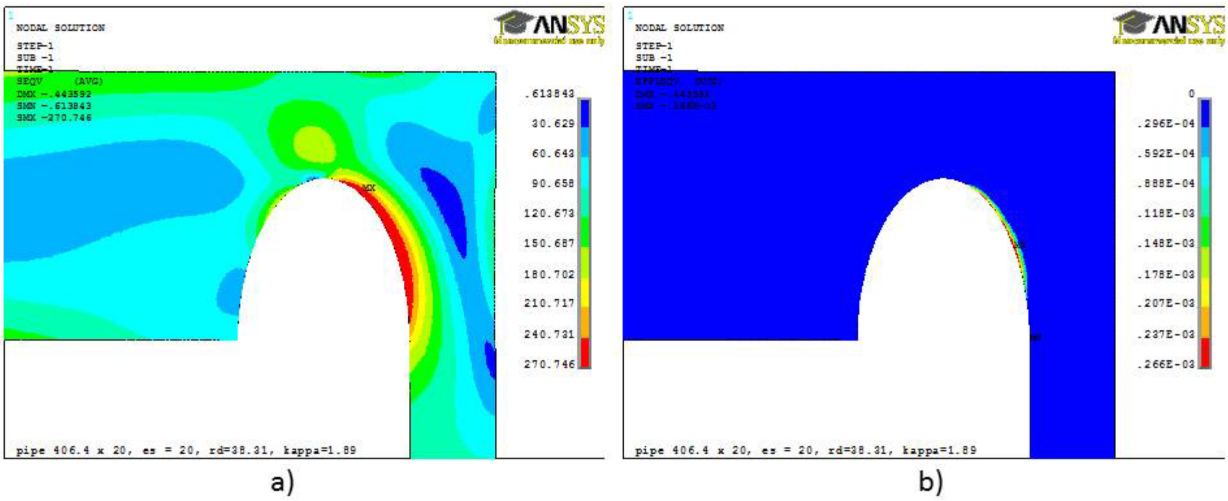


*Fig. 3. Equivalent stress (a) and equivalent plastic strains distribution (b) for optimal parameters of stress relief groove of elliptic shape*

## 4. Proposed modifications of vessel topology

Before the final, technologically justified proposal for the modified structure, the analysis of the structure topology and a standard trial of topological optimization were performed. For the purpose of such numerical finite element analysis, the final configuration of the vessels with optimal circular stress relief groove was used. The study performed in the ANSYS [3] code with the assumption that the only certain areas undergo the shape modification.

In Fig. 4, the details of that analysis are illustrated. Here, certain areas where no stress concentration is observed were excluded from the possible modifications. These areas are marked with violet colour in Fig. 4b. The results of that study are shown in Fig. 4c, where certain voids appeared. Such shape modification leads to certain improvement of the stress redistribution in the investigated area. However, the full elimination of the plastic deformations is not possible. What is more, the manufacturing of the void placed inside the endplate is not possible. Only the undercut from the outside of the boiler is easy to make. Also other shape modifications performed by the author [14], connected with the removal of the certain parts of the investigated boiler structure, did not lead to the substantial improvement of the stress redistribution and release of stress concentration in the investigated object. The conclusion of such observations was as follow: only addition of parts of the material in certain areas can improve the structure performance and lead to release the stress concentration. This concept was applied in the technologically justified form illustrated in Fig. 6. Here, two simple ideas were proposed. Both modifications relied on welding certain rings on the top of the endplate. In the first proposal, the short cylindrical ring with thickness equal to the wall thickness of the tube of vessel is welded around the boiler (Fig. 5 – Concept 1). Here the parameters to search are the height of the welded band and the radius of the weld (this approximates the fillet weld). In the second case (Fig. 5 – Concept 2), the ring made from rod with circular cross-section is welded with fillet welds around the boiler. In this approach, the radius of the rod and the radius of the weld are to establish.

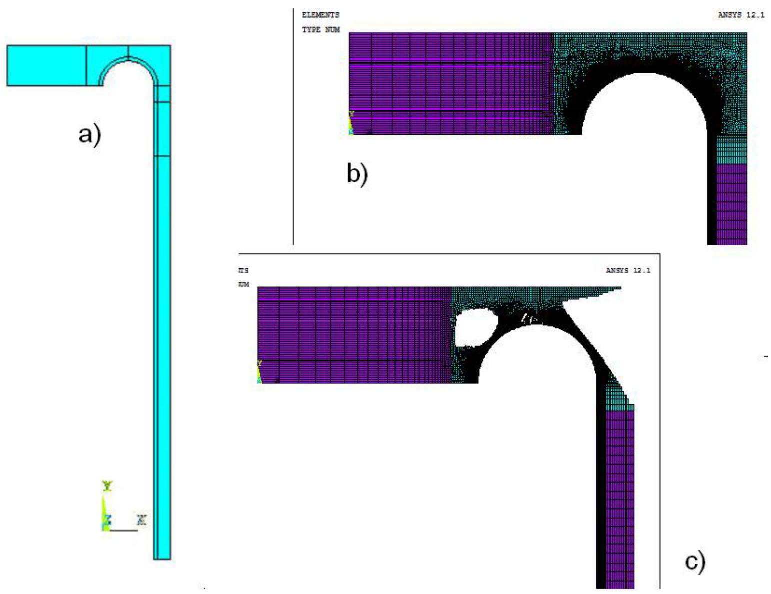


*Fig. 4. Topological optimization of the studied boiler; (a) the basic structure, (b) area (finite elements) to modify (remove) marked in blue, (c) final result of the topological modification*

## 5. Results of numerical optimization for modified structure

The Concept 1 and Concept 2 were in detail analysed in order to find such combinations of the defining parameters, which provide the full elimination of plastic deformations in the whole structure. The descriptions of geometry of additional parts are shown in Fig. 6. Before the final

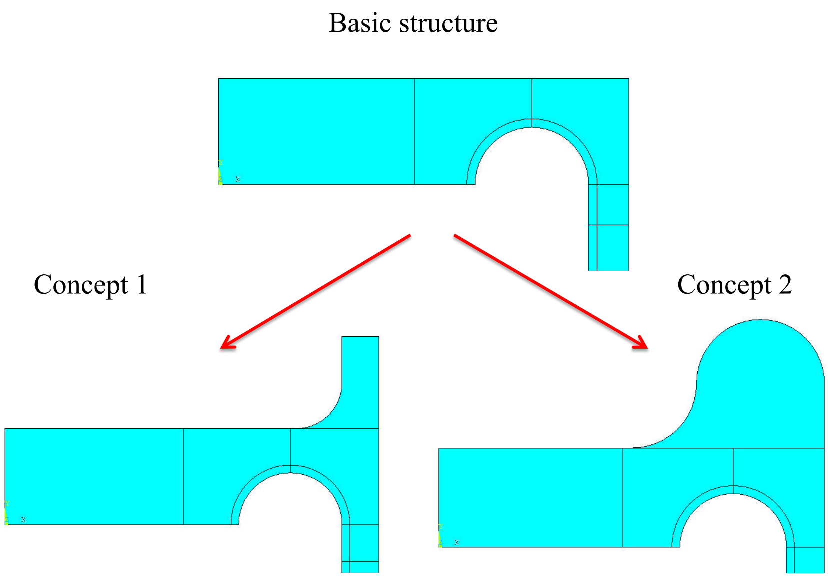


*Fig. 5. Proposed concepts of structure topology modification*

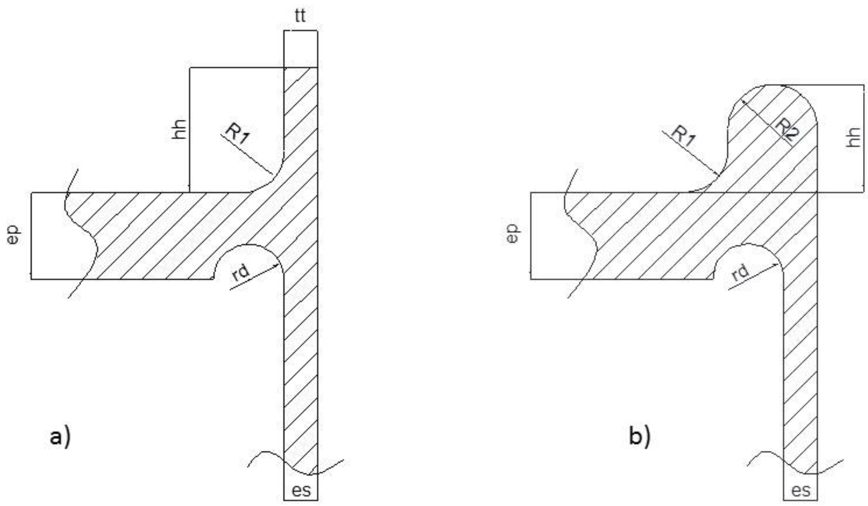


*Fig. 6. Definition of design parameters for proposed concepts of structure improvement*

analysis for the Concept 1 certain additional assumptions were introduced. These were as follow: in order to minimize the cost the added on the top of the endplate cylindrical ring should have the same thickness, namely *tt*, as the wall thickness of the tube, so that the radius *R1* and the height of the collar: $hh = k_1 \times es$ were the only unknowns. As a starting structure the vessel with optimal configuration for elliptic groove was assumed (vertical semi-axis: $b = 38.31$ mm, horizontal semi-axis: $a = b / kappa = 20.27$ mm). The result of the study is shown in Fig. 7.

Figure 7a shows the dependency between the height of the band $hh = k_1 \times es$ (expressed by $k_1$) and the value of the maximum equivalent plastic strain obtained in the finite element analysis of the modified boiler. As it can be seen for each of the analysed radius *R1* the value for the height of the ring, providing the elimination of the plastic strains was be found, but was different for different Radius *R1*. In Fig. 7b the distribution of the equivalent stress in the vicinity of the groove for modified vessel structure is presented. These plot was obtained for $k_1 = 2.05$ and $R1 = 0.7 \times es$ (the vessel tube original thickness, $es = 20$ mm). The similar study was performed for the Concept 2. In this approach, also the full elimination of the plastic deformation in analysed vessel is possible. However, the results are not so promising in the authors' opinion. In this case the elimination of plastic deformation was obtained for rods with diameters exceeding the *es*, namely for $R2 = 1.67 \times es$ when $R1 = 0.7 \times es$.

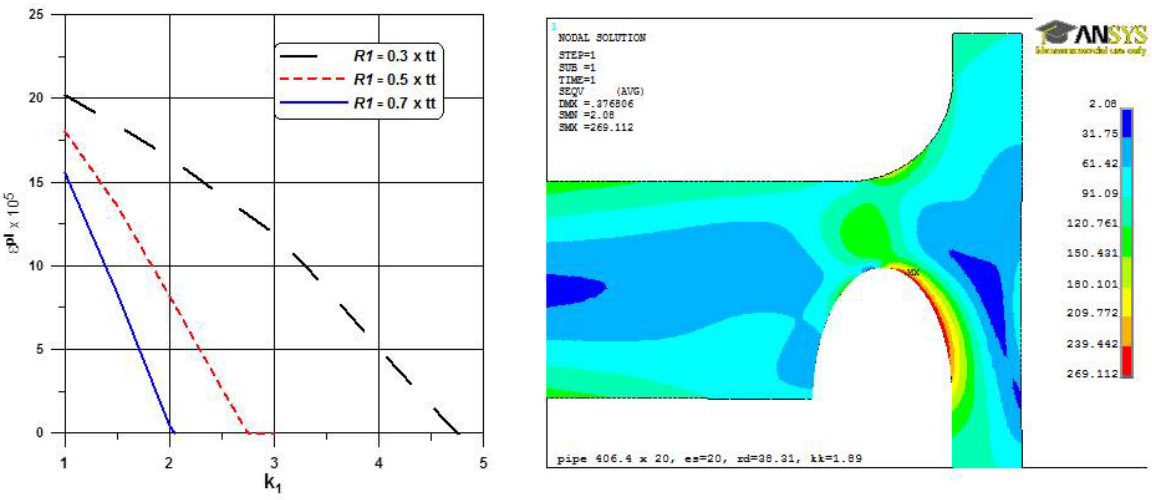


*Fig. 7. Results of Concept 1 study: (a) maximum plastic deformation in analysed structure for different values of radius R1 for concave face of weld and different height of band; (b) distribution of equivalent stress for R1 = 0.7 × tt and hh = 2.05 × es*

## 6. Conclusions

The performed study has shown that the proposed shape modification:

– change of the shape of the groove from circular to elliptic,
– welding of certain reinforcing rings placed on the top of the endplate and around the outer edge of the vessel,
– provides satisfactory results, namely full elimination of the plastic deformations in the groove area.

In the parametric optimization, process the height of that ring or the diameter of the circular ring welded around the outer part of the boiler can be respectively established. The numerical studies – performed with the assumption that no plastic deformations were present – for different combinations of the boiler, diameters and the tube wall thickness resulted in problem dependent optimal values:

– horizontal and vertical semi-axis of the groove,
– height of the applied additional band or diameter of the applied additional rod,
– radius of the concave face of the 1/2V butt weld applied to join the structure and additional part.

It means that the height of the band or the ring diameter must be established for each considered case individually. However, such numerical calculations do not demand large computational effort.

Further studies with other modifications of the flat endplate shape are still under consideration.

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