ZAMORNA ČVRSTOĆA SPOJA PRIKLJUČKA SPUSNE CIJEVI I PLAŠTA BUBNJA KOTLA

FATIGUE ANALYSIS OF DOWNCOMERS PIPE BRANCH AND DRUM SHELL JUNCTION AREA OF A BOILER DRUM

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Sažetak:

Proračun zamorne čvrstoće spoja priključka spusne cijevi i plašta bubnja, uslijed djelovanja cikličkog opterećenja koje će se pojaviti zbog gašenja i ponovnog pokretanja kotla (hladni start), tipičan je problem proračuna zamora konstrukcije. Na zamornu čvrstoću spoja priključka i plašta utječu prisutna mehanička opterećenja, radna temperatura, ali i utjecajni faktori zamora kao što su hrapavost površine, debljina stjenke, utjecaj srednjih naprezanja te utjecaj statistike i područja rasipanja S-N krivulje. Utjecaji promjene temperaturnog polja za vrijeme jednog ciklusa opterećenja i zaostalih naprezanja tijekom zavarivanja u ovome radu nisu uzeti u obzir. Geometrija bubnja diskretizirana je 3D konačnim elementima pomoću softvera *Abaqus 6.10*. Između dva hladna starta kotla mogu se izdvojiti dva ekstremna slučaja opterećenja spoja priključka i plašta. Za oba kvazistatička slučaja opterećenja izračunati su tenzori naprezanja na nivou čvora konačnog elementa. Dobiveni tenzori naprezanja analizirani su prema modelu troosnog cikličkog zamora. Zamorna čvrstoća analizirana je pomoću softvera *FEMFAT 4.8* za slučaj opterećenja hidrostatskim i radnim tlakom te ukupnom težinom koja opterećuje spoj priključka i plašta. Cilj istraživanja je utvrđivanje utjecaja mase na zamornu čvrstoću. Proračuni su izrađeni za minimalno potrebnu debljinu stjenke plašta bubnja i priključka (EN12952-3) te za još četiri veće debljine stjenke plašta, kako bi se utvrdio utjecaj debljine stjenke na zamornu čvrstoću.

Abstract:

Analysis of fatigue strength of downcomers pipe branch and drum shell junction area, loaded with cyclic loading which occurs as a result of shutting down and restart of a steam boiler (cold startup), is a typical fatigue strength analysis problem. Fatigue strength is influenced by applied mechanical loading, calculation temperature and fatigue influence factors, such as surface roughness, wall thickness, influence of mean stresses and influence of statistics and range of dispersion in S-N curve. Influence of temperature field change during one loading cycle and welding residual stresses are not taken into consideration in this paper. Drum geometry is discretized using 3D finite elements by software *Abaqus 6.10*. Between two cold startups of a boiler, two extreme loading cases of downcomers pipe branch and drum shell are extracted. For both quasistatic cases of loading, stress tensors are calculated for a single finite element node. Stress tensors are analyzed according to multiaxial cyclic fatigue model. Fatigue strength is analyzed using software *FEMFAT 4.8* for the case of loading with hydrostatic and calculation pressure, and overall weight which loads the junction area. The aim of the research is to determine the influence of the mass on fatigue strength. Calculations are made for minimal necessary drum shell thickness and branch thickness (EN 12952-3), and for four more junctions with increased thickness of drum shell, in order to determine the influence of wall thickness on fatigue strength.

1. INTRODUCTION

A water-tube steam boiler drum is a cylindrical pressure vessel loaded with the pressure of the working fluid, with its own mass, installed armature mass, mass of the working fluid and mass of the insulation. Primary function of the drum is to separate liquid phase from the steam, and continuus supply of the evaporator with technically prepared water. During the exploatation, the boiler is occaisonally stopped and restarted (cold startup). Thereby, the pressure in the drum is decreased to the atmospheric pressure, and if needed, the volume of working fluid in the drum is reduced. This kind of loading has a dynamic character. The part exposed to higher stress is the connection area of the drum shell and downcomers pipes (junction). The manufacturer usually guarantee 500 cold startups of a boiler without malfunctions and shutdowns. According to EN12952-3 [1], it is not neccesary to perform the fatigue calculation of the junction for less than 2000 cold startups. Fatigue calculation of the drum junction according to [1] is based on the high-cycle fatigue theory with the application of the stress concentration factor. Thus, this calculation is approximate and inapplicible for more detailed analysis. In this paper, on the example of a drum of the steam boiler Luzern [2], the influence of own mass, mass of installed armature, mass of the insulation and mass of the working fluid will be analyzed regarding fatigue strength of the branch and drum shell junction. According to static loading with working fluid pressure [1], the neccesary wall thickness for the branch and drum shell will be calculated, without additions for tolerance and corrosion. Thereby, the strength requirement will be satisfied [1]:

$$f_\mathrm{a} = p_\mathrm{c} \cdot \left(\frac{A_\mathrm{p}}{A_\mathrm{fs} + A_\mathrm{fb}} + \frac{1}{2}\right) \leq f_\mathrm{s}, \qquad (1)$$

where p_c is the calculation pressure, A_p is the pressurized area without allowances, A_{fs} is the effective cross-sectional area of body, A_{fb} is the effective cross-sectional area of the branch, f_s is the design stress for main body material (drum shell and branch material), and f_a is effective average stress. The drum shell and branch material is 15NiCuMoNb5-6-4. Yield stress wil be assumed for the calculation strength of the drum shell material and branch material at the calculation temperature of t_c = 287,2 °C, i.e. f_s = 365,82 MPa. Calculation pressure in the drum and the downcomers pipes is p_c = 71,5 MPa, the outer diameter of the drum shell is d_{os} = 1900 mm, and the outer diameter of the branch is d_{ob} = 406,4 mm. The minimal calculated wall thickness for the branch is e_{rb} = 42 mm, while for the drum shell it is e_{rs} = 43 mm. Fatigue calculation will be performed for four different boiler drums with enlarged wall thickness: $e_{rs}^{(1)}$ = 43 mm, $e_{rs}^{(2)}$ = 45 mm, $e_{rs}^{(3)}$ = 47 mm, $e_{rs}^{(4)}$ = 49 mm and $e_{rs}^{(5)}$ = 51 mm. The branch wall thickness will be equal for all cases, e_{rb} = 42 mm. The conventional static calculation [1] for dimensioning of the junction does not take into consideration the influence of the mass with installed armature and insulation. In this research, the influence of the drum shell, drum header, as well as the masses of the installed armature and insulation, will be taken into consideration. Mass of the installed armature is approximately 2000 kg, and it will be applied through the increase of the shell material density. Thus, the calculation weight will correspond to the weight of the drum shell, installed armature and the insulation. The mass of the working fluid (drum filled with water) will be applied as hydrostatic pressure. An attempt of proofing that the mass loading has no significant influence on fatigue strength of the drum will be made. It will be shown that the fatigue crack initiation in the connection area can occur at less than 2000 cold startups, if the static strength requirement [1] is applied for the dimensioning. Finite element analysis (FEA) was done on 3D model of a drum in Abaqus 6.10 [3]. Stress tensors were analyzed in FemFat 4.8 [4] with influence factors for transient multiaxial fatigue evaluation. The application of FEM enables a more detailed approach to the fatigue strength analysis of the branch and drum shell. Research of the pressure vessel branch fatigue strength has always been actual in the energetic field. Codes for the pressure vessels calculation offer approximate procedures for fatigue strength calculations [1,5,6,7,8] which can hardly satisfy the investors demands (material saving, influence of different loading, life cycle "extension", operating under altered working conditions, etc.). Scientific development of the fatigue strength calculation approach for the pressure vessel is based mostly on the application of FEM [9,10]. With temperature increase, the fatigue strength of material is changing. It is difficult to perform more detailed fatigue strength calculations, especially for the low-cycle fatigue (LCF), where the experimental research of the fatigue influence factors at high temperatures poorly follow the development of new materials. Fatigue cracks mostly occur at the surface of a material. Therefore, the surface condition is very significant for the fatigue behavior. In the crack initiation period, fatigue is a material surface phenomenon [11]. Due to surface roughness, small stress concentrations occur on the surface. Scatter of the fatigue strength is also significant for evaluation of the fatigue. Fatigue life of similar specimens or structures under the same fatigue load can be significantly different [11]. Crack initiation period is more sensitive to many influences in comparison with crack growth period. Scatter is low for fatigue crack growth because it is not dependent on the surface conditions [12]. Range of dispersion is the largest at high cycle fatigue (HCF) part of S – N curve, while for LCF, it is much lower, due to high stress amplitudes.

2. FINITE ELEMENT MODEL

Drum is discretized by 3D FE model. Due to symmetry conditions, only one quarter is modeled. Model consists of drum shell and downcomer pipe. Drum shell was meshed with solid hexahedral (C3D8) and quadratic tetrahedron (C3D10) elements. FE mesh was created very fine in the connection area (Fig. 1). As the most critical section, fillet was modeled with solid quadratic hexahedral elements (C3D20). For capturing realistic stress concentrations in fillet, element length along radius is 1 mm. First three element layers have thickness of 0,5 mm, and two additional elements have 1 mm thickness (Fig. 1). After inner radius on both sides, element lengths are 2, 3 and 3 mm, respectively. Fillet consists of 13200 elements and 64998 nodes. Complete models with drum shell, downcomer pipe and fillet in all five cases have similar number of elements and nodes, approximately 150000 elements and 300000 nodes.

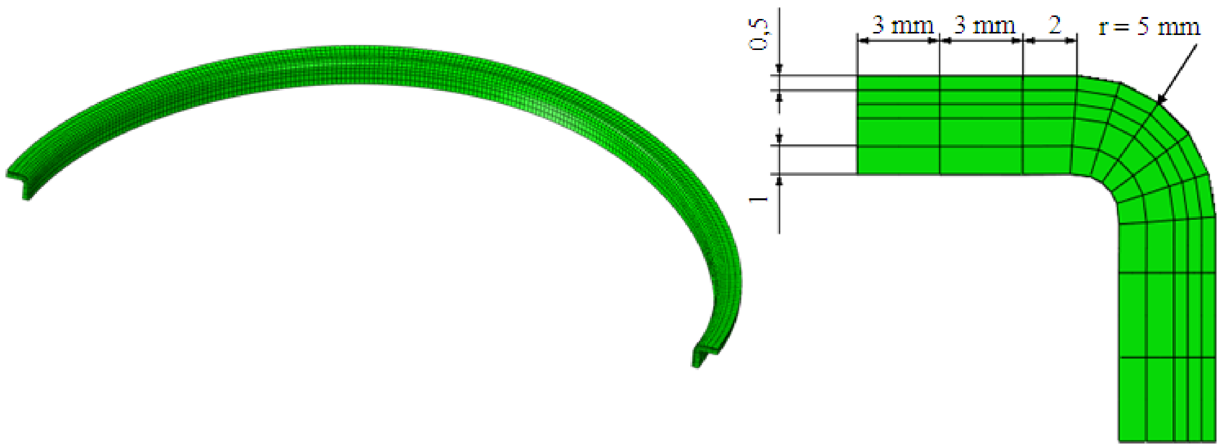

Fig. 1. *Drum fillet mesh with fillet mesh detail*

3. MATERIAL

For quasistatic steps, drum and downcomer pipe were modeled as homogeneous linear-elastic material. Modulus of elasticity at t_c = 287,2 °C is E = 186,22 GPa [8]. Yield strength for thickness between 40 and 60 mm, is $R_{P0,2}$ = 365,8 MPa (EN10028-2). Tensile strength at t = 20 °C is R_m = 610 MPa, while at t_c, R_{mt} = 415,38 MPa. It is calculated according to FKM [13] expression:

$$R_{\mathrm{mt}} = R_{\mathrm{m}} \cdot K_{\mathrm{Tm}}\,, \tag{2}$$

where K_{Tm} is calculated according to:

$$K_{\mathrm{Tm}} = 1 - 1{,}7 \cdot 10^{-3} \cdot (t\,[°\mathrm{C}] - 100). \tag{3}$$

A prerequsite for fatigue analysis in FemFat is the material data for unnotched specimen S – N curve for tensile/compressive loading with a stress ratio $R = -1$ and relative stress gradient $\chi = 0$. Material survival probability is defined as 50 %. This material is modified locally at each node of drum fillet FE model to obtain correct S – N curve at critical areas. Mean stress influence is taken into consideration using Haigh diagram, which is generated by polygonal lines. Fatigue strength is decreased for tensile mean stress and increased for compressive stress. Therefore, Haigh diagram is unsymmetric (Fig. 2). Alternating fatigue strength of steel for survival probability of 50 %, according to the investigation of Hück, Thrainer and Schütz [14] can be expressed as:

$$\sigma_{\mathrm{W}} = 0{,}385 \cdot R_{\mathrm{mt}} + 30. \tag{4}$$

After calculating R_{mt}, alternating fatigue strength σ_W = 190 MPa is obtained for infinite life at survival probability of 50 %. According to FemFat user manual [15] for higher strength weldable steel, cyclic exponent of hardening is $n' = 0{,}15$, and cyclic coefficient of hardening K' is defined by expression:

$$K' = 1{,}65 \cdot R_{\mathrm{m}}. \tag{5}$$

Mean and amplitude stress rearrangement is done according to Neuber-hyperbola with FemFat PLAST. Sigma – epsilon curve, Haigh diagram and S – N curve at survival probability of 50 % are shown in Fig. 2.

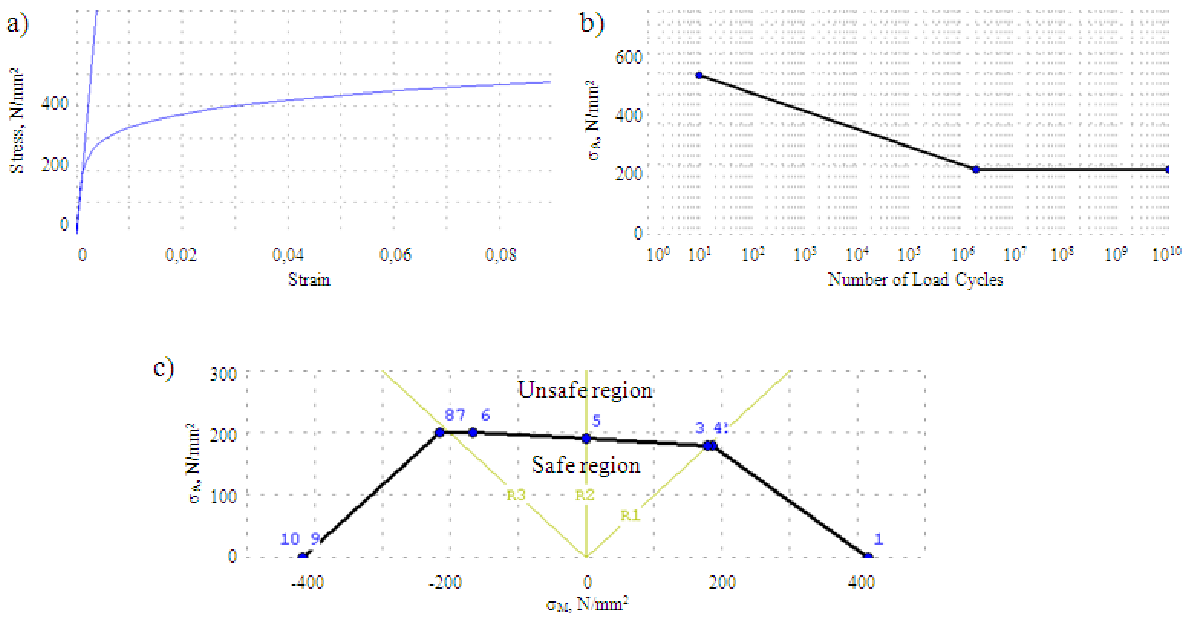

Fig. 2. *a) Sigma – epsilon curve for nonlinear material behavior, b) Haigh diagram, c) S – N curve*

Metal fatigue is a random process, and the consequent scatter of results complicates both the analysis of experimental data and their application to practical problem [16].

4. FATIGUE ANALYSIS

Loads which occur during lifetime of boiler drum are very complex with variable temperature and pressure. Critical plane approach was used for such drum loading. It is assumed that the plane is critical for fatigue failure at the critical node of the fillet, for which the accumulated damage exceeds critical limit. FemFat uses the Haigh diagram to define the most critical cutting plane angle and the damaging factor for each node. Cutting plane is defined at every 10° angle. Results of safety factors (*SF*) against endurance limit for interpreting in FemFat are for constant mean stress:

$$SF = \frac{\sigma_{\mathrm{Aall}}}{\sigma_{\mathrm{A}}}, \qquad (6)$$

where σ_{Aall} is the allowable stress amplitude and σ_{A} is the stress amplitude during fatigue load spectrum.

Fatigue influence factors used for drum fillet are:

stress gradients,
surface roughness, R_Z = 50 μm,
technological size influence, i.e. thickness of drum,
range of dispersion, T_S = 1,26 [15],
mean stress,
mean and amplitude stress rearrangement in case of material local plastification and statistical influence.

Surface roughness is very significant for fatigue, because the fatigue endurance stress limit decreases with rougher surfaces. Range of dispersion T_S, is defined as the ratio of the structure fatigue strength at 10 % survival probability to structure fatigue strength at 90 % survival probability. According to FemFat user manual, recommended and default value is 1,26 for steel, aluminium, grey and nodular cast iron, etc. Calculation of the fillet S – N curve at nodes was done with taking the defined survival probability of 99,99 %, as well as the range of dispersion. Gaussian normal distribution was assumed for calculation of the statistics influence variables. Effect of surrounding low loaded ares, appearing at the fillet with steep stress inclinations, was taken into consideration. The stress gradients at nodes were calculated from FE stress calculation. Stress gradient characterizes stress concentration at the fillet. From Hück, Thrainer and Schütz expression for alternating fatigue strength for infinite life at survival probability of 50 %, all influence factors are lowering the fatigue strength. Therefore, for the first drum model with minimal wall thickness of 43 mm, fatigue strength at the critical node is lowered to 119,2 MPa.

5. RESULTS

For fatigue evaluation, stresses due to gravity load were firstly calculated. After gravity load step, follows the step with overall loading, where hydrostatic and calculation pressure were applied on inner surface of drum and downcomer. Calculation and hydrostatic pressure were defined in way of pulsating loading. Equivalent von Mises stress distributions due to gravity load step and overall load step for the first model, i.e. with minimal wall thickness, are shown in Fig. 3. From stress distribution in the second step, maximal value was found on the drum fillet as 466 MPa.

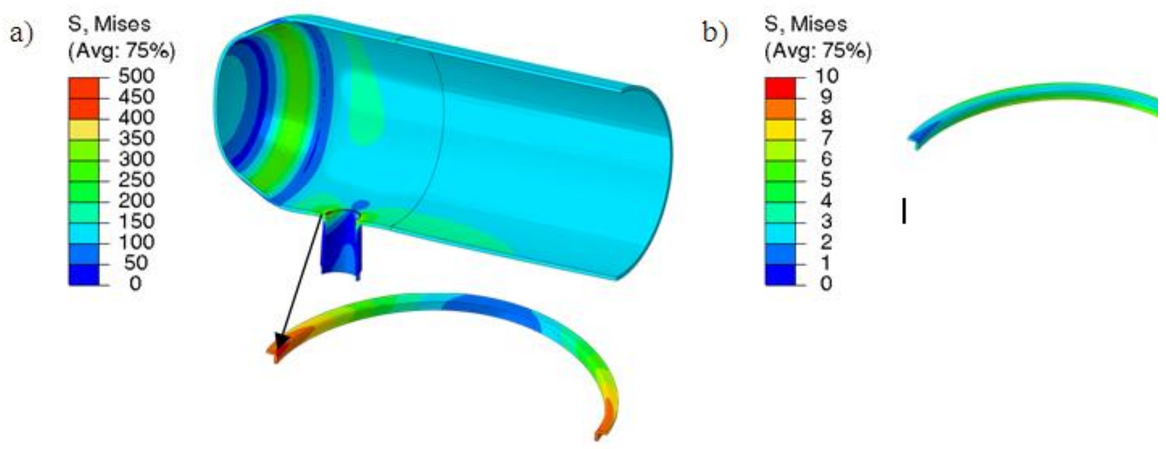

Fig. 3. *a) Equivalent von Mises stress for e_{rs} = 43 mm in the drum due to overall loading with fillet detail, b) von Mises stress detail due to gravity load for e_{rs} = 43 mm*

Since maximal stress is distributed on the drum fillets, only fillet nodes were analyzed for pulsating fatigue behavior. Therefore, stress tensors of the fillets from all 5 models were analyzed in FemFat for multiaxial transient fatigue with influence factors.

The results show that for the first model with minimal wall thickness of e_{rs} = 43 mm, minimal endurance safety factor is SF = 0,52 (Fig. 4) with σ_A = 228 MPa (Fig. 5) and σ_M = 76 MPa (Fig. 6). Number of cycles until crack initiation is 223. It can be concluded that with increase of the drum wall thickness, endurance safety factors are higher due to lower stress amplitudes. Model with the highest wall thickness of e_{rs} = 51 mm, result with minimal endurance safety factor of SF = 0,57, amplitude stress σ_A = 207 MPa, mean stress σ_M = 80 MPa and number of cycles until crack initiation of 785. Mean stresses are increased with the increase of drum wall thickness due to higher mass of the drum, and therefore, under gravity load the stresses have increased for a very small value. The results prove the fact that stress amplitudes are predominant for fatigue behavior, and that the mass of the drum has no significant influence on fatigue behavior.

Haigh diagrams for critical node at the fillet, for all five models, are shown in Fig. 7. The diagrams show that the increase of wall thickness lowers the stress amplitude, while mean stresses are higher, but not significantly. Every point in Haigh digram represents one cutting plane.

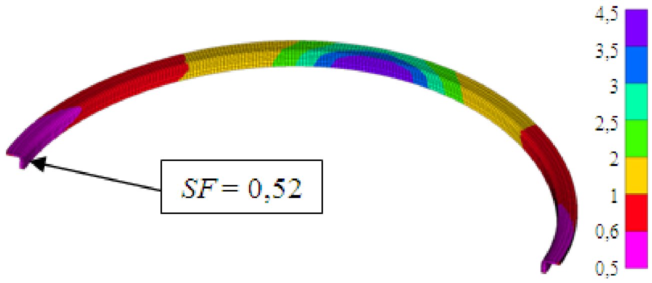

Fig. 4. *Endurance safety factors in drum fillets for e_{rs} = 43 mm*

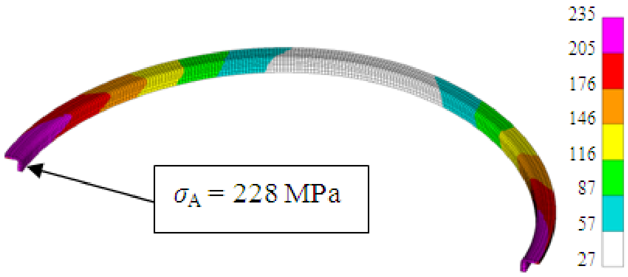

Fig. 5. *Stress amplitudes in drum fillets for e_{rs} = 43 mm*

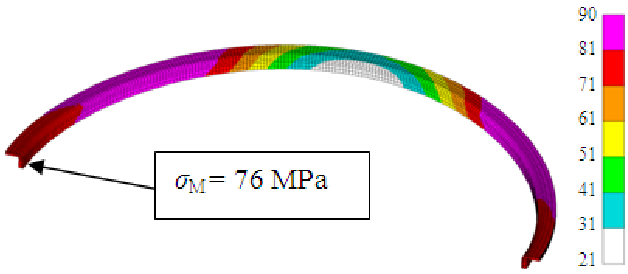

Fig. 6. *Mean stresses in drum fillets for e_{rs} = 43 mm*

The endurance safety factors, stress amplitudes and mean stresses are shown in Table 1.

Table 1 *Endurance safety factors, stress amplitudes and mean stresses for different e_{rs}*

e_{rs}= 45 mm	SF = 0,53	σ_A = 225 MPa	σ_M = 77 MPa
e_{rs}= 47 mm	SF = 0,55	σ_A = 218 MPa	σ_M = 78 MPa
e_{rs}= 49 mm	SF = 0,56	σ_A = 213 MPa	σ_M = 79 MPa
e_{rs}= 51 mm	SF = 0,57	σ_A = 207 MPa	σ_M = 80 MPa

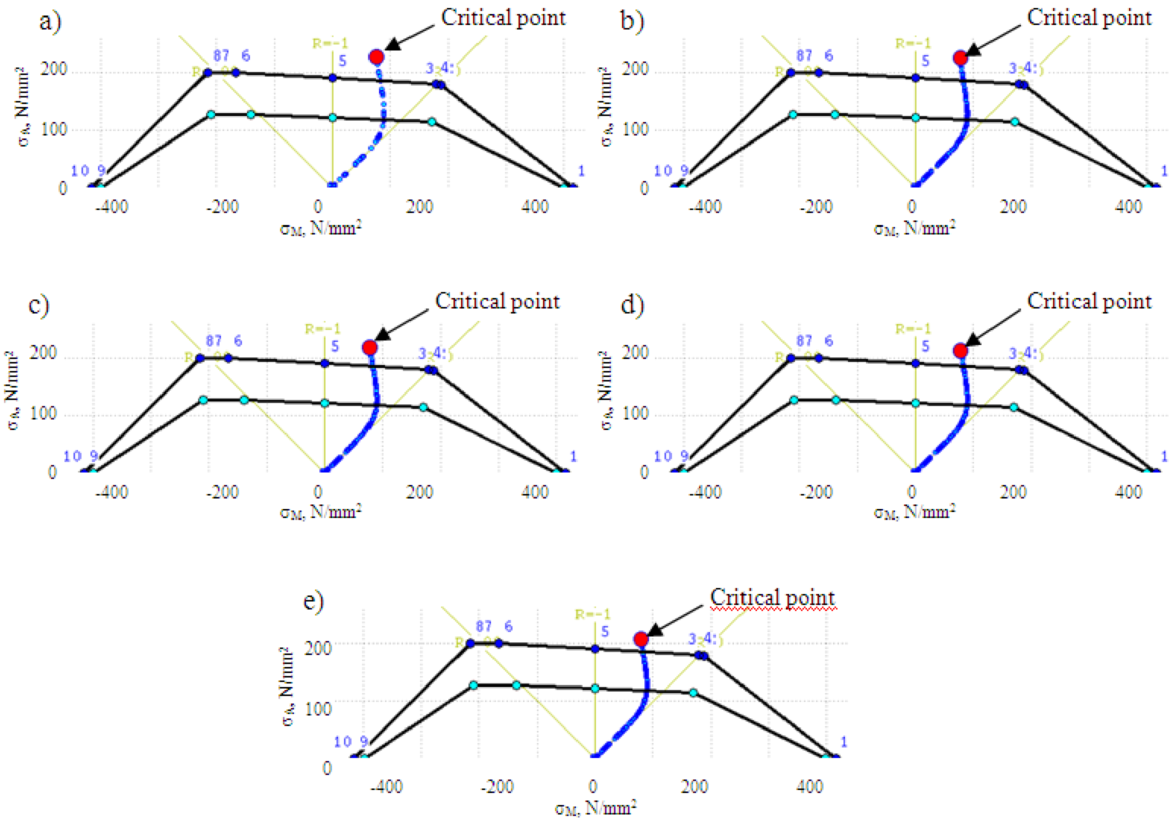

Fig. 7. *Haigh diagrams for drum fillet model for: a) $e_{rs} = 43$ mm, b) $e_{rs} = 45$ mm, c) $e_{rs} = 47$ mm, d) $e_{rs} = 49$ mm, e) $e_{rs} = 51$ mm,*

Endurance safety factors, cycles until crack initiation, stress amplitudes and mean stresses with respect to the thickness of the drum shell, are shown in Fig. 8.

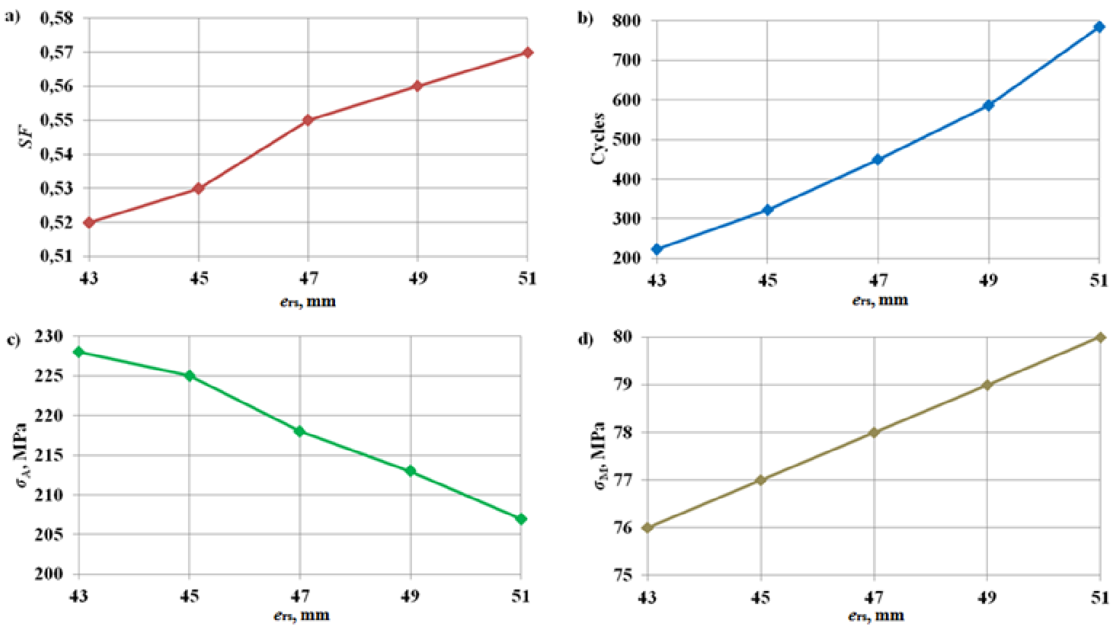

Fig. 8. *a) Endurance safety factors, b) cycles until crack initiation, c) stress amplitudes, d) mean stresses*

6. CONCLUSION

Results from the FEM and fatigue analysis of the drum with downcomer pipe show that drum thickness has very significant influence on fatigue behavior. According to EN12952-3 code, minimal drum thickness is 43 mm, which leads to only 223 cycles until fatigue crack initiation, with SF = 0,52 for infinite life. In practice manufacturers usually must guarantee 500 cycles, i.e. cold startups. Increased drum wall thickness result with lower stress amplitudes, which result in higher endurance safety factors and increased number of cycles until crack initiation. However, due to increased drum mass, mean stresses are sligthly higher. Maximal wall thickness of t = 51 mm result with number of cycles until crack initiation of 785 and SF = 0,57 for infinite life. To capture correct stress concentrations on the fillet, very fine solid quadratic hexahedral finite elements are recommended.

In this research, welding residual stresses, anisotropy of weld material and temperature field change through junction zone of downcomer branch and drum shell were not taken into account. Another assumption in this research was evaluation with stress based approach for lower number of cycles. Strain based approach should give more accurate results.

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