Regular Article

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Thermal-strength analysis of a cross-flow heat exchanger and its design improvement

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Abstract: After a year of the heat exchanger operation in overload conditions, a number of cracks on the tube connections to the tubesheet have been observed. To explain the stress concentration and crack initiation, a finite element analysis is performed. A three-dimensional model is constructed and analyzed. To calculate more precisely the state of stress in the most loaded regions, a submodel is created. The maximum stress exceeds the allowable stress, and according to the standards, it can lead to ratcheting. To reduce stress concentration, all tubes should be shortened and corrugated tubes are installed in the high-temperature region from the side of the burner. A finite element model of the modified heat exchanger and a submodel are created. In the modified heat exchanger, ratcheting should not occur according to the applied standards. During the operation of the modified heat exchanger, there are no further problems with cracking.

Keywords: heat exchangers, thermal stress, thermal fatigue, numerical modeling

1 Introduction

Thermal-strength analysis plays an important role in the design of many machines and equipment. The thermal behaviors of ventilated brake discs using three different configurations were investigated at continuous brake conditions in terms of heat generation and thermal stresses with finite element analysis [1]. A high-temperature design and fatigue damage evaluation for a helical type sodium-to-air heat has been conducted in ref. [2]. The life prediction of

the brazed plate heat exchanger is presented in ref. [3]. FEM analysis can also show the performance [4,5] and cause of failure of the appliance [6]. The sources, consequences, and the probability of steam turbine components failure have been calculated in ref. [7]. Analysis of a shell-tube heat exchanger showed that the failure of the tube-to-tubesheet welded joint was induced by fatigue [8].

In this study, fracture failure of a cross-flow heat exchanger is investigated, when it has been operated in overload conditions. The submodeling technique [9] is adopted to simulate the stress distribution in the most loaded areas. The heat exchanger is designed to heat the air, which flows along the tubesheet and inside tubes, by gas burning. The burner is installed by a circular opening. The exhaust gas flows between successive baffles and exits through the round flange. A schematic of the analyzed heat exchanger is presented in Figure 1, and its main characteristics are shown in Table 1.

Materials used for the construction of this heat exchanger are indicated in Table 2. Thermal and mechanical properties of the used steels are presented in Figure 2. A description of physical phenomena, which led to the fracture formulations, will be used during the heat exchanger design improvement.

2 Visual inspection

The exchanger worked in the fifth climate zone. Less than 100 cycles started with an outside temperature lower than -20°C , whereas over 200 cycles started with an outside temperature between -10 and -20°C . After a year of the heat exchanger operation in overload conditions, a number of cracks on the tube connections to the tubesheet have been observed. The cracks are initiated at the root of the welds and propagated in the tube and weld tubesheet. Damaged areas have been repaired by tubes welding to the tubesheet. A fragment of damaged and then repaired heat exchanger is presented in Figure 3. Despite the repair, during a further operation, subsequent fractures occur.

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Figure 1: Schematic of a heat exchanger.

Table 1: Main characteristics of the heat exchanger

Tube diameters (mm)	DN 40, DN 50
Working medium	Natural gas/air
Ambient temperature (°C)	-20 ÷ 20
Design pressure (kPa)	3
Fuel	Natural gas
Burner type	Natural draft burner

Table 2: Materials used in the fabrication of the heat exchanger [1]

Tubesheet	1.4016 acc. to EN 10088-2 [10]
Flanges	1.4301 acc. to EN 10028-7 [11]
Baffles	1.4016 acc. to EN 10088-2 [10]
Thermal insulation	No

3 Failure analysis

Visual inspection showed that all the fractures of the heat exchanger occurred near the tube-to-tubesheet welded joints. To explain the stress concentration and crack initiation, a finite element analysis is performed. A three-dimensional model containing tubes, baffles, and tubesheet is constructed and analyzed. The created model is shown in Figure 4. It consists of 179,017 nodes and 173,371 shell elements. The model built in this way acts as a global model, which is used for a more precise analysis using the submodeling technique [9]. The Ansys Workbench software was used for the analyses [12].

The heat exchanger is located in the perpendicular channel, which allows to supply cold air and receive heated air. The air flows inside the tubes and partly between the tubesheet and the rectangular channel. The inlet and outlet air temperature equals -20 and 20°C, respectively.

Figure 2: Thermal and mechanical properties of used materials: (a) 1.4016 steel and (b) 1.4301 steel.

Figure 3: A fragment of damaged and then repaired heat exchanger.

The temperature distribution in the air flowing inside and outside of the heat exchanger is presented in Figure 5. The temperature distribution in exhaust gasses is shown in Figure 6. Due to the natural flow of the air and flue gasses, the value of the heat transfer coefficient is estimated as $17 \text{ W/m}^2 \text{ K}$ [13]. The calculated temperature distribution in the heat exchanger can be seen in Figure 7. A linear elastic strength analysis has been conducted using the isotropic Hooke's law. The calculated equivalent (Huber–von Mises–Hencky) stress distribution in the heat exchanger is presented in Figure 8.

Figure 4: Finite element model of the heat exchanger: (a) view from outside and (b) interior.

The highest stresses occur in the first row of tubes from the top. They are located in the upper parts of welds between the tube ends and the tubesheet. This is consistent with crack initiation places, which are presented in Figure 3. The working temperature of the heat exchanger in this area is about 600°C.

The stresses exceed the yield strength significantly, which equals 170 MPa at this temperature. According to the standards for the calculation of pressure equipment [14,15], stresses caused by thermal loads are included in the secondary group and they cannot exceed double yield strength (340 MPa). Exceeding this value may lead to ratcheting.

To calculate better the state of stress in the vicinity of the selected pipe a submodel is prepared [9]. The use of a submodel allows for more accurate analyses by refining the finite element mesh in the region of our interest, in terms of computation, with the optimal analysis time. The boundary of the local model (submodel) represents a cut through the global model. Displacements calculated on the cut boundary of the global model are specified as boundary conditions for the local model. It is shown in Figure 9. The submodel consists of 206,938 nodes and 75,587 solid elements (Table 3). It will be used for a detailed fatigue analysis.

Figure 5: Temperature distribution in the flowing air: (a) inside the tubes and (b) between the tubesheet and the rectangular channel.

Figure 6: Temperature distribution in the exhaust gasses.

The submodel reflects the global geometrical configuration of the structure but excludes the local structural discontinuities by weld toe. It includes the effects of large structural discontinuities such as branch connections, junctions, and thickness change. However, it excludes the notch effects of local structural discontinuities, which give rise to nonlinear stress distributions across the plate thickness. The submodel is used for a linear elastic strength analysis, and all the calculated stresses are elastic.

Figure 7: Calculated temperature distribution in the heat exchanger: (a) view from outside and (b) tubes.

Figure 8: Calculated equivalent stress distribution in MPa: (a) isometric view and (b) front view.

Figure 9: Submodel of the original heat exchanger.

Table 3: Number of nodes for a global model and submodel

Model type	Number of nodes
Global (shell elements)	179,017
Submodel (solid elements)	206,938

To estimate the highest stress in this area, the stress classification lines SCL1 and SCL2, which are presented in Figure 9, are defined. The membrane and bending stresses, normal to the supposed crack plane, are calculated for both classification lines.

The procedure of stress linearization through the plate thickness is applied. The membrane stress is equal to

$$\sigma_m^e = \frac{\sigma_{in}^e + \sigma_{out}^e}{2}, \quad (1)$$

where for the line SCL1 in = 1, out = 3 and for the line SCL2 in = 2, out = 4. The bending stress for both classification lines can be calculated from

$$\sigma_b^e = \frac{\sigma_{in}^e - \sigma_{out}^e}{2}. \quad (2)$$

All stresses in equations (1) and (2) are normal components of stress tensor to the supposed crack plane.

The structural hot spot bending and membrane stress in the analyzed region can be obtained from the following linear extrapolation [16,17].

$$\sigma^e = 1.67\sigma_{SCL1}^e - 0.67\sigma_{SCL2}^e \quad (3)$$

The bending and membrane stress ranges between time points $m t$ and $n t$, and the maximum, minimum, and mean stress can be calculated from the following equations:

$$\begin{aligned}
\Delta\sigma_m^e &= {}^m\sigma_m^e - {}^n\sigma_m^e \\
\Delta\sigma_b^e &= {}^m\sigma_b^e - {}^n\sigma_b^e \\
\sigma_{\max} &= \max [({}^m\sigma_m^e + {}^m\sigma_b^e), ({}^n\sigma_m^e + {}^n\sigma_b^e)] \\
\sigma_{\min} &= \min [({}^m\sigma_m^e + {}^m\sigma_b^e), ({}^n\sigma_m^e + {}^n\sigma_b^e)] \\
\sigma_{\text{mean}} &= \frac{\sigma_{\max} + \sigma_{\min}}{2}.
\end{aligned} \quad (4)$$

Elastically calculated structural stress and strain ranges have the following forms:

$$\begin{aligned}
\Delta\sigma^e &= \Delta\sigma_m^e + \Delta\sigma_b^e \\
\Delta\varepsilon^e &= \frac{\Delta\sigma^e}{E_a}
\end{aligned} \quad (5)$$

where E_a is the Young modulus for average temperature.

Elastically calculated equivalent (Huber–von Mises–Hencky) stress distribution in the submodel is presented in Figure 10. The stress and strain range is obtained from equations (1–5) and equals $\Delta\sigma^e = 546$ MPa and $\Delta\varepsilon^e = 0.00363$, respectively.

Assuming a local plastic deformation, the nonlinear structural stress and strain ranges are introduced.

They can be determined from Neuber's rule and the material hysteresis loop stress–strain curve model given by the following equations [16,17]:

$$\begin{aligned}
\Delta\sigma \cdot \Delta\varepsilon &= \Delta\sigma^e \cdot \Delta\varepsilon^e \\
\Delta\varepsilon &= \frac{\Delta\sigma}{E_a} + 2 \left(\frac{\Delta\sigma}{2K_{\text{CSS}}} \right)^{\frac{1}{n_{\text{CSS}}}}
\end{aligned} \quad (6)$$

where n_{CSS} and K_{CSS} are cyclic stress–strain curve data. These material data for 1.4016 and 1.4301 steel equal 0.09 and 450 MPa, respectively [14]. Nonlinear structural stress range $\Delta\sigma$ and strain range $\Delta\varepsilon$ can be calculated from a set of equation (6) in an iterative way. The following values are obtained: $\Delta\sigma = 465$ MPa, $\Delta\varepsilon = 0.00439$.

Figure 10: Calculated equivalent stress distribution in the submodel (MPa).

For low cycle fatigue, the calculated stress range is modified by following equation [14]:

$$\Delta\sigma = \left(\frac{E_a}{1 - \nu^2} \right) \cdot \Delta\varepsilon \quad (7)$$

and equals $\Delta\sigma = 726.7$ MPa. The equivalent structural stress range ΔS_{ess} is obtained based on the cycle type and the plate thickness. It equals $\Delta S_{\text{ess}} = 1026.6$ MPa.

The number of allowable design cycles N for the welded joint can be calculated from the following equation [14]:

$$N = \frac{f_1}{f_E} \left(\frac{f_{\text{MT}} \cdot C}{\Delta S_{\text{ess}}} \right)^{1/h} \quad (8)$$

where $f_1 = 1.076$ is the fatigue improvement factor, $f_E = 4$ is the environmental modification factor, $f_{\text{MT}} = 0.7638$ is the temperature adjustment factor, $C = 11577.9$ MPa, and $h = 0.3195$ are coefficients for the welded joint fatigue curves, which depend on a material type and a confidence factor. The calculated number of allowable design cycles equals $N = 227$. If the heat exchanger is heated and cooled only once a day, it can be damaged within 1 year of its operation.

4 Heat exchanger design improvement

To reduce stress concentration in tube-to-tubesheet welded joints, tube expansion compensation is needed. The use of commercially available expansion joints would significantly increase the cost of heat exchanger production. Corrugated steel tubes are applied. Manufacturer and service consequences when this design is put into practice

Figure 11: Proposed changes in the design of the heat exchanger.

Figure 12: Calculated equivalent stress distribution in MPa: (a) original heat exchanger and (b) modified heat exchanger.

are taken into account and solved. To successfully compensate for the heat exchanger expansion, all tubes should be shortened and corrugated tubes should be installed in the high-temperature region from the side of the burner. Details are shown in Figure 11.

Corrugated tubes are made of 1.4301 steel. A three-dimensional finite element model containing tubes, baffles, tubesheet, and corrugated tubes is constructed and analyzed. It consists of 1,042,914 nodes and 1,077,546 shell elements. The calculated temperature distribution in the modified heat exchanger is very similar to the temperature field in the original design. Next, a linear elastic strength analysis is conducted. The calculated equivalent stress (Huber–von Mises–Hencky) distribution in the modified heat exchanger is presented in Figure 12. The highest stresses are limited to the value of 333 MPa, which is lower than the allowable value of 340 MPa.

Figure 13: Submodel of the modified heat exchanger.

Table 4: Number of nodes for a global model and submodel with improvement

Model type	Number of nodes
Global (shell elements)	1,042,914
Submodel (solid elements)	555,033

To calculate more precisely, the state of stress in the vicinity of the selected pipe a submodel is created. It is shown in Figure 13. The submodel consists of 555,033 nodes and 246,380 solid elements (Table 4). Elastically calculated equivalent (Huber–von Mises–Hencky) stress distribution in the submodel is presented in Figure 14. Next, a detailed fatigue analysis is conducted by using equations (1–8). The stress and strain range is lower than in the original heat exchanger and equals $\Delta\sigma^e = 338.7$ MPa and $\Delta\epsilon^e = 0.0022463$, respectively.

The calculated value of allowable design cycles equals $N = 4,253$. After conducting the presented design modifications, the heat exchanger works without problems.

5 Conclusion

The visual inspection and the numerical modeling are presented in this article to explain the cause of fracture formulations in the cross-flow heat exchanger, which has been operated in overload conditions. With the help of finite element modeling, the stress concentration places, which are located in the upper parts of welds between the tube ends and the tubesheet, are identified. The maximum stress exceeds the allowable stress, and according

Figure 14: Calculated equivalent stress distribution in the modified heat exchanger submodel (MPa).

to the standards, it can lead to ratcheting. To calculate more precisely the state of stress in the most loaded regions, a submodel is created. It is used for a detailed fatigue analysis. The calculated value of allowable design cycles equals $N = 227$. If the heat exchanger is heated and cooled only once a day, it can be damaged within 1 year of its operation. High thermal stresses in the upper parts of tube-to-tubesheet welded joints are the reasons for fracture formulations. To reduce stress concentration between tube ends and the tubesheet, tube expansion compensation is necessary. To successfully compensate for the heat exchanger expansion, all tubes should be shortened and corrugated tubes should be installed in the high-temperature region from the side of the burner. A finite element model of the modified heat exchanger and a submodel are created. The presented results show that the maximum stresses in the exchanger with provided compensation do not exceed the allowable value. According to the standards for the calculation of pressure equipment in the modified heat exchanger, ratcheting should not occur. Additionally, the number of allowable design cycles is calculated, and it is 18 times larger than in the original heat exchanger. During the operation of the modified heat exchanger, there are no further problems with cracking.

Conflict of interest: Authors state no conflict of interest.

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