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Shape optimization of a thick-walled power boiler component

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Abstract. This paper presents a methodology and successful application of structural optimization of a T-pipe under transient thermal and mechanical loads. In order to find the optimal shape of a thick-walled power boiler component, a parametric FE model and the evolutionary algorithm (EA) are applied. The power boiler start-up and shutdown curves are based on the TRD 301 guidelines. Maximum total stresses are assumed as optimization constraints. The obtained geometry is by about 18.6% lighter than the original one due to thinning of the walls. Maximum tensile and compressive stresses in the modified geometry are smaller than in the original one during the whole cycle. Additionally, lower total stress values are recorded during heating and cooling processes. Therefore, these transient processes can be accelerated and the shutdown and start-up losses can be reduced.

1 Introduction

High thermal stresses are created in power boiler components during their operation. The cyclic nature of these stresses causes low-cyclic fatigue and may lead to fracture. Manufacturers of such devices advise keeping the heating and cooling rates within prescribed limits. Nevertheless, extended start-up and shutdown times cause considerable losses in the electricity generation process. Rapid transient operations of power boilers, on the other hand, involve high temperature differences and thermal stresses in their components [1]. Methods of finding optimum operating parameters, which can ensure safe heating and cooling processes, can be found in [2, 3, 4, 5]. Reducing the thickness of the boiler component walls while maintaining the stress level during the cycle within established limits can bring material savings. Furthermore, it can improve the thermomechanical characteristics of the device. A component with thinner walls can be heated and cooled more quickly.

Most state-of-the-art methods now available for structural optimization of the power unit equipment refer to the steady state. Wen et al. [6] present an example of structural optimization of cellular cores of metallic sandwiches subjected to laminar forced convection at fixed pumping power. Xie et al. [7] employ a globally convergent method of moving asymptotes (GCMMA) in lightweight optimization of a corrugated sandwich panel using heat transfer and thermomechanical analyses.

This paper presents a method of lightweight optimization of the power boiler components subjected to transient thermomechanical loads. The power boiler start-up and shutdown curves are based on the TRD 301 code [8]. Maximum total stresses are assumed as optimization constraints.

2 Method formulation

The geometry of a power boiler component determines the level of stresses arising during its operation. A reduction in the component thickness may result in an increase in stresses caused by internal pressure but – on the other hand – it should reduce thermal stresses. This leads to the conclusion that the component wall thickness can be reduced without increasing total stresses. Using thinner walls in a component may produce material savings and make it possible to carry out transient processes at a faster rate.

In order to find the optimal shape of a thick-walled power boiler component, a parametric FE model and the evolutionary algorithm (EA) are applied (see Fig. 1). This method is based on mechanisms of biological evolution. The EA performs a multi-directional search of the solution space, protecting the population of potential solutions and the exchange of information between them. It is possible to adjust the EA operation depending on the characteristics of the optimization object. This can be done by changing the algorithm control parameters, such as the population size PJ, the parameter of probability of crossing operations p_c, the parameter of probability of mutation p_m, the number of generations N [9, 10].

The EA is designed to solve single and multicriteria optimization problems for nonlinear programming problems. The single-criterion optimization is formulated to find the vector of decision variables (e.g. dimensions of a thick-walled boiler component)

$$\mathbf{y} = \{ y_1 \quad y_2 \quad \dots \quad y_N \} \tag{1}$$

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Continuous Nonlinear Programming with the mapping from binary string to real number for the variable **y** is used. For all decision variables, the upper and lower constraints values are applied:

$$\underline{y_i} \le y_i \le \overline{y_i} \quad \text{for } i = 1,\dots,N \tag{2}$$

The optimal solution should satisfy the following inequality constraint

$$\max\left(\frac{\sigma_{3\min}(\mathbf{y})}{\breve{\sigma}_a}, \frac{\sigma_{1\max}(\mathbf{y})}{\hat{\sigma}_a}\right) \le 1 \tag{3}$$

where $\sigma_{3\min}(\mathbf{y})$ denotes the lowest third principal stress in the entire component during the heating process, $\sigma_{1\max}(\mathbf{y})$ is the highest first principal stress during the cooling process, $\breve{\sigma}_a$ and $\hat{\sigma}_a$ denote allowable stresses for the heating and cooling operations. Both values $\sigma_{3\min}(\mathbf{y})$ and $\breve{\sigma}_a$ are compressive stresses and are negative. Tensile stresses $\sigma_{1\max}(\mathbf{y})$ and $\hat{\sigma}_a$ are positive.

For the objective function, the structure volume $V(\mathbf{y})$ is chosen. The vector of decision variables **y** should be found to minimize the objective function $V(\mathbf{y})$.

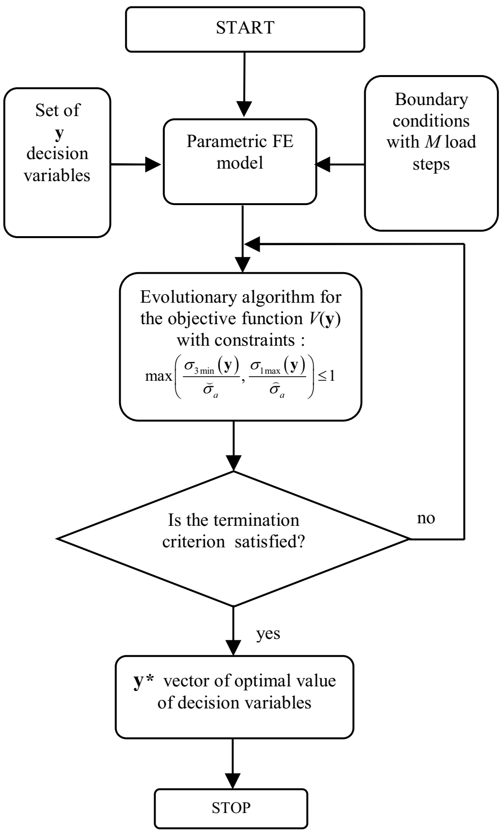

Fig. 1 Proposed methodology.

3 Numerical example

The proposed method will be used for a T-pipe shape optimization. The T-pipe is a part of a live and pre-heated steam pipeline in a 360 MW power unit. It was designed for pressure p_w=18 MPa and steam temperature T_w=540ºC. It weighs 1378 kg. The T-pipe geometry is shown in Fig. 2. The T-pipe is made of alloy steel 14 MoV63. Temperature-dependent thermal and mechanical properties are considered [11].

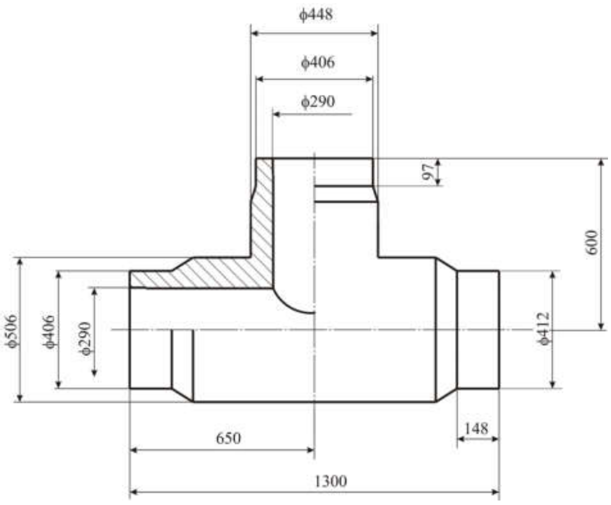

Fig. 2 T-pipe geometry.

TRD301 standards [8] can be used to propose the heating or cooling method for the selected T-pipe. They assume a quasi-steady state and one-dimensional temperature distribution through the wall of the structural element. The calculated allowable stresses for heating and cooling operations have values of $\breve{\sigma}_a = -214.4$ MPa , $\hat{\sigma}_a = 303$ MPa ,respectively. The following allowable rates of temperature changes are obtained:

$v_{T1(TRD)} = 2.119$ K / min - the beginning of heating,

$v_{T2(TRD)} = 2.993$ K / min - the end of heating,

$v_{T1(TRD)} = -2.119$ K / min - the beginning of cooling,

$v_{T2(TRD)} = -2.993$ K / min - the end of cooling.

If the pressure at the beginning of the heating process equals 0 MPa and the pressure at the end of heating equals p_w, the steam temperature can be calculated from the following relation:

$$\frac{dT_f}{dt} = v_{T1(TRD)} + \frac{v_{T2(TRD)} - v_{T1(TRD)}}{p_w} p_f(T_f) \tag{4}$$

where p_f is independent of temperature or should be calculated as saturation pressure at T_f., The steam temperature for the cooling process can be obtained in a similar way. The calculated steam temperature and pressure during heating and cooling operations are presented in Fig. 3.

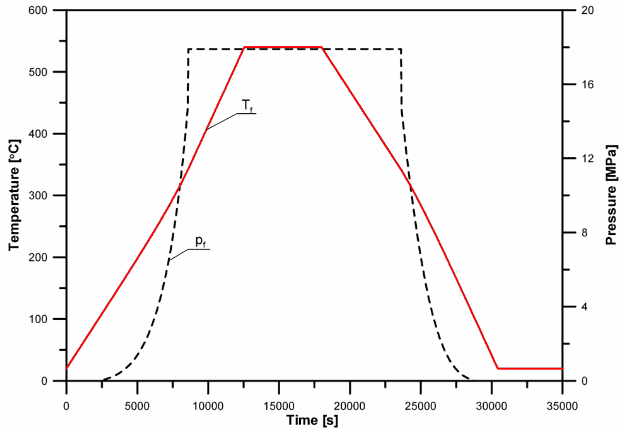

Fig. 3 Steam temperature and pressure based on TRD 301.

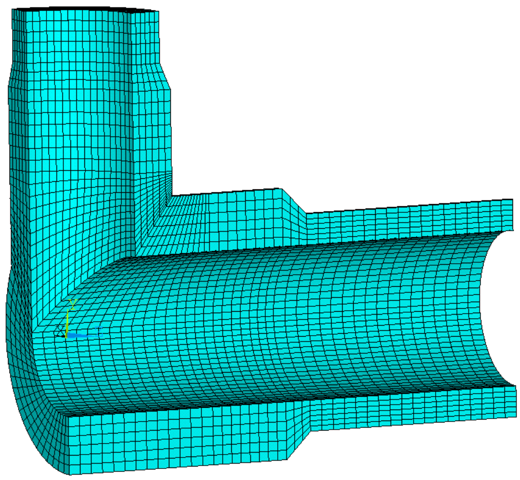

Fig. 4 T-pipe division into finite elements - original shape.

A quarter of the T-pipe, which is divided into 20-node finite brick elements according to symmetry conditions, is shown in Fig. 4. The temperature and stress distribution in this component is calculated by means of the finite element method using the ANSYS software [12]. The heat transfer coefficient on the inner surfaces of the T-pipe is assumed as 1000 W/m^{2}K [13]. The stress model is constrained to ensure symmetry conditions and it allows the possibility of unrestricted lengthening in the direction of the horizontal and vertical pipe. Additionally, the surfaces connecting the T-pipe to the pipeline must be plane.

The maximum value of equivalent stress (Huber-Mises-Henky) and the first principal stress caused by pressure and temperature occur after about 23100 s and equal $\sigma_{eq\,max}$=246.8 MPa and $\sigma_{1\max} = 229.2\ \mathrm{MPa}$, respectively. The maximum compressive stress can be estimated based on the minimum value of the third principal stress. The minimum value of the third principal stress occurs after about 10300 s and equals $\sigma_{3\min} = -139\ \mathrm{MPa}$. Throughout the entire cycle, the allowable stresses, as per the TRD 301 standard, $\breve{\sigma}_a = -214.4\ \mathrm{MPa}$, $\hat{\sigma}_a = 303\ \mathrm{MPa}$ are not exceeded. Principal stresses $\sigma_{1\max} = 229.2\ \mathrm{MPa}$ and $\sigma_{3\min} = -139\ \mathrm{MPa}$ will be assumed as the bounds of constraints during the T-pipe shape optimization.

The optimization problem is defined to minimize the volume (mass) of the structure assuming the following allowable stresses: $\breve{\sigma}_a = -139\ \mathrm{MPa}$, $\hat{\sigma}_a = 229.2\ \mathrm{MPa}$. The optimization process involves a search for the solution in a 6-dimensional design variable space (4 radii and 2 lengths of the T-pipe reinforcements – see Fig. 5).

The following range of variation is proposed for the decision variables: y_1 = {215 mm, 265 mm}, y_2 = {215 mm, 240 mm}, y_3= {450 mm, 550 mm}, y_4 = {450 mm, 550 mm}}, y_5= {190 mm, 215 mm}, y_6 = {190 mm, 215 mm}.

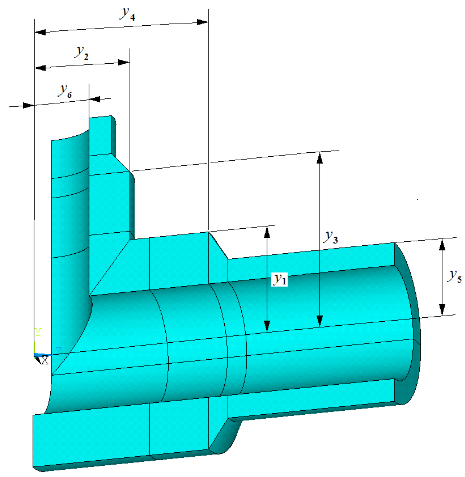

Fig. 5 6-dimensional design variable space of the EA shape optimization.

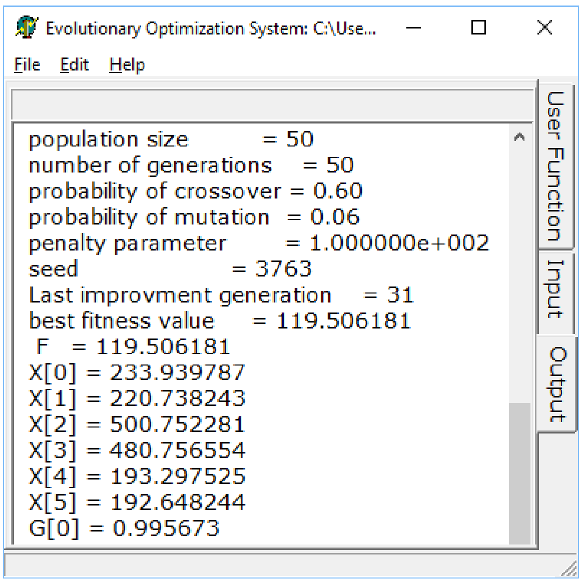

Fig. 6 Optimization result in the EOS.

The parametric FE model is subjected to the evolutionary algorithm (EA) using the Evolutionary

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Optimization System (EOS) software [9]. Moreover, the following steering parameters of the EOS are selected: population size PJ = 50, number of generations N = 50, probability of crossover p_c= 0.6, probability of mutation p_m= 0.06. The EOS software is connected to the ANSYS FE program to perform batch processing calculations of the objective function value and stress constraints. The best solution **y*** (y_1* = 233.94 mm, y_2* = 220.74 mm, y_3* = 500.75 mm, y_4* = 480.76 mm, y_5* = 193.3 mm, y_6* = 192.65 mm) is about 18.6% lighter than the original one. The calculated values are presented in Fig. 6, where F is the objective function $V(\mathbf{y})$ and X[0]...X[5] are all decision variables $y_1...y_6$. The variable G[0] is the constraint, which is defined by (3). The modified T-pipe can be seen in Fig. 7.

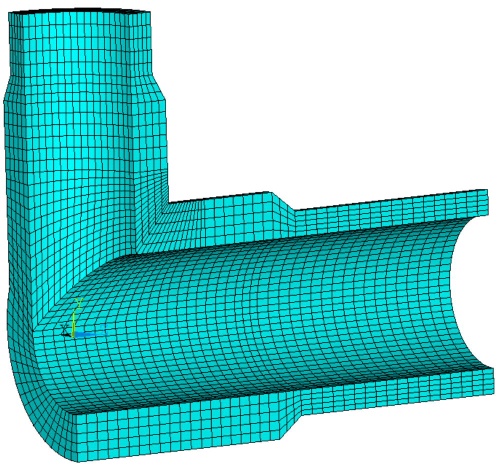

Fig. 7 Division of the modified T-pipe into finite elements - optimal shape.

Fig. 8 shows $\sigma_{1\max}(\mathbf{y})$ - maximum values of first principal stresses in the whole T-pipe over the load time history for the original and optimal shapes. Fig. 9 presents $\sigma_{3\min}(\mathbf{y})$ - minimum values of third principal stresses in the T-pipe for the two geometries.

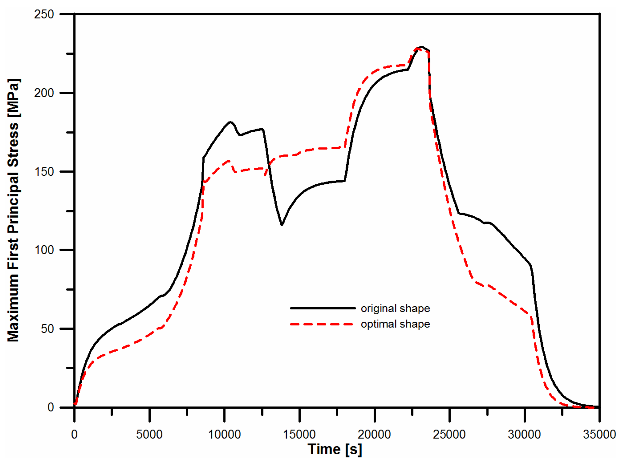

Fig. 8 Maximum first principal stress values in the entire structure depending on time.

The shape modification does not cause an increase in the maximum stress values in time and space. The highest recorded first principal stress for the optimal shape equals 228.2 MPa and is lower than the maximum value for the original shape, which equals 229.2 MPa. Additionally, the value of $S_{1\,max}$ for the modified geometry is lower than for the original one during almost the whole heating and cooling process (0 s – 12900 s and 22900 s – 35000 s). Maximum compressive stresses in the modified geometry are smaller than in the original T-pipe in the following time periods (0-12900 s and 23700-35000 s). Lower total stress values recorded during heating and cooling offer the possibility of speeding up these transient processes and reducing losses related to the boiler shutdown and start-up.

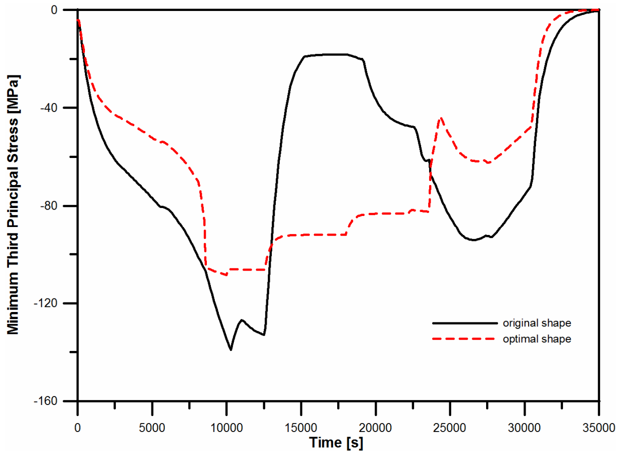

Fig. 9 Minimum third principal stress in the entire structure depending on time.

4 Conclusions

A methodology and successful application of structural optimization of a T-pipe under transient thermal and mechanical loads are presented. The obtained geometry is by about 18.6% lighter than the original one due to thinning of the walls. This result can be considered as significant, especially in view of the fact that this structure has been in use for many years and must have been optimized by designers before. Maximum tensile and compressive stresses in the modified geometry are smaller than in the original one during the whole cycle. Additionally, lower total stress values are recorded during heating and cooling processes. Therefore, these transient processes can be accelerated and the shutdown and start-up losses can be reduced.

The proposed methodology can be applied in any industry where heating and cooling processes take place.

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