European Conference on Computational Fluid Dynamics
ECCOMAS CFD 2006
P. Wesseling, E. Oñate, J. Périaux (Eds)

OPTIMIZATION OF HEATING AND COOLING OPERATIONS OF POWER BLOCK PRESSURE ELEMENTS

P. Duda*, A. Cebula* and R. Dwornicka‡

* Cracow University of Technology, Institute of Process and Power Engineering
Al. Jana Pawła II 37, 31-864 Kraków
e-mail: pduda@mech.pk.edu.pl, acebula@pk.edu.pl

† Cracow University of Technology, Institute of Computing Science
Al. Jana Pawła II 37, 31-864 Kraków
e-mail: dwornick@mech.pk.edu.pl

Key words: Diagnostic system, Stress, Control system, Heating optimization, Power plant, Fluid Dynamics

Abstract. *In this work, optimization will be done on the basis of a calculation of an initial medium temperature step and following that the maximum heating or cooling rate of temperature changes. The calculated results will have a very high practical significance because it will be possible to use them in power plants. Temperature step change can be easily implemented by suddenly opening hot water or steam supply into the interior of a pressure element, which has a lower initial temperature. Next, CFD calculations of the flowing medium in a power block element will be presented.*

1 INTRODUCTION

High thermal stresses are created during the operation of power block devices such as boiler drums, outlet headers, steam valves, turbines and heat exchangers. Due to a cyclic character of such stresses, a phenomenon of low-cyclic fatigue occurs, which may lead to the formation of fractures. Manufacturers of power block devices frequently advise the users to keep within the prescribed limits for maximum heating and cooling rates of elements [1]. Attempts are made to develop algorithms for operating power blocks [2, 3]. Thanks to these mathematical systems, one is able to extend the life of operating devices and shorten the duration of all transient operations.

2 THERMAL STRESSES AND STRESSES CAUSED BY FLUID PRESSURE

The T-pipe used in the fresh and pre-heated steam pipelines in a power unit of a 360 MW power plant is one of the most loaded elements in power block. The T-pipe is installed in a steam pipeline of BP1150 boiler with a steam capacity of 1150 t/h. It is exposed to high stresses caused by temperature and pressure given off by a flowing medium. The T-pipe was designed for the pressure, p_{in}=18 MPa, and steam temperature, T_w=540^{o}C. Its weight equals 1378 kg. The geometry of the T-pipe is presented in Fig.1.

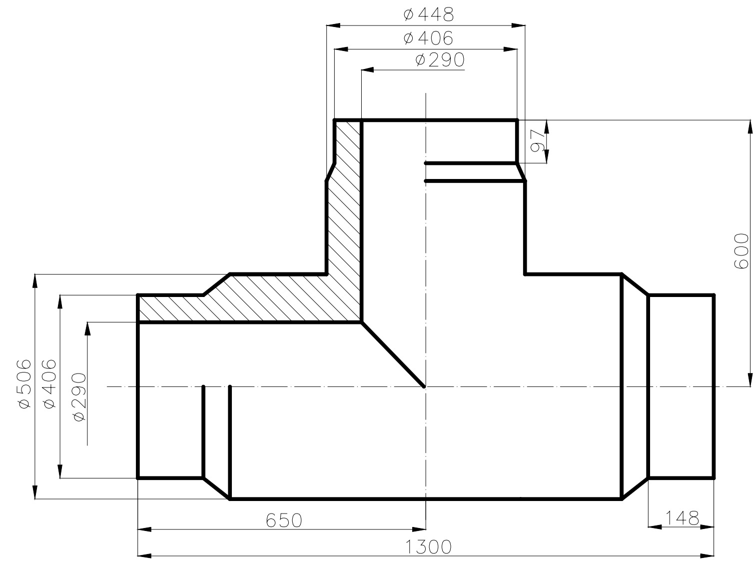

Figure 1: Geometry of the T-pipe.

It is necessary to construct pressure elements, which operate under high work parameters, out of materials that exhibit good mechanical properties. Materials should retain their properties within the wide temperature range in which they operate, especially at the yield strength R_e . The actual T-pipe was made out of an alloy steel with the purpose of working at elevated temperatures designated as 14 MoV63 (13 HMF). This steel has a ferritic structure and a high yield strength, R_e=206 MPa for the temperature t=500ºC. Thermal and mechanical properties are presented in Figs. 2-3.

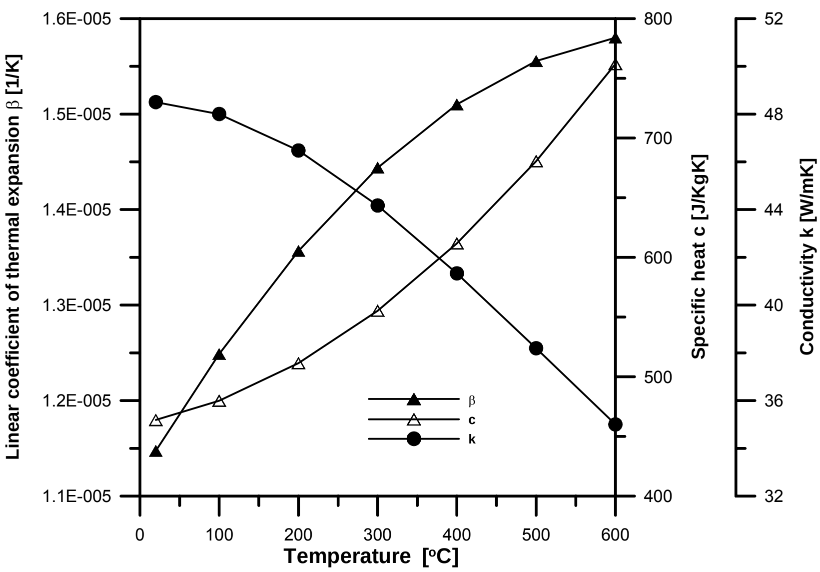

Figure 2: Thermal properties for the steel 14 MoV63 (13 HMF).

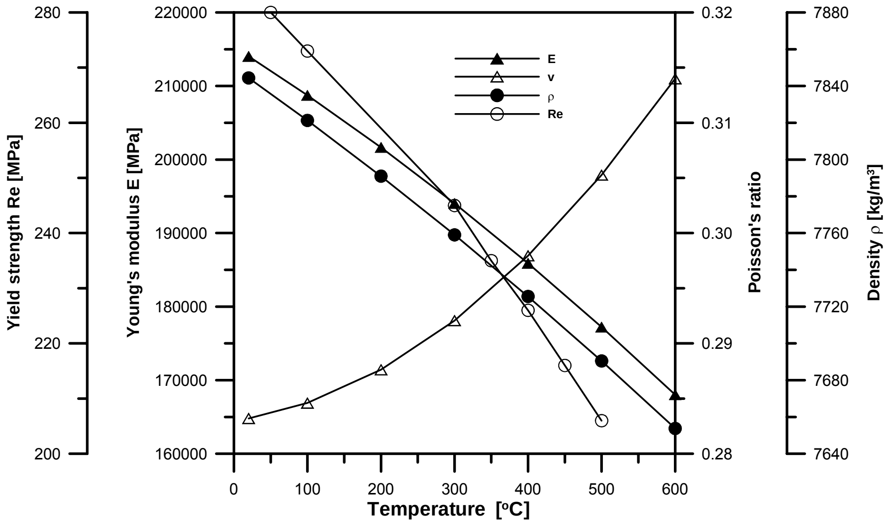

Figure 3: Mechanical properties for the steel 14 MoV63 (13 HMF).

Temperature and stress monitoring conducted for the T-pipe was presented in Fig. 1. Measured temperature and steam pressure histories are presented in Fig. 4.

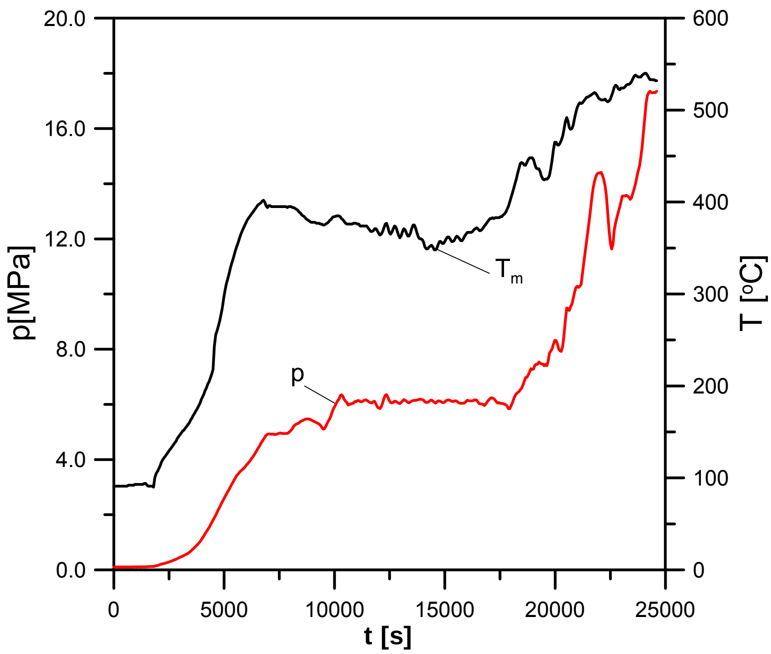

Figure 4: T-pipe –measured temperature transients and fresh steam pressure history

ANSYS software based on the finite element method was used for the calculation of time-space temperature and stress distribution. Assuming the symmetry, ¼ of the T-pipe area was divided into eight-node finite brick elements presented in Fig. 5. This is the most optimal mesh based on accuracy and solution time.

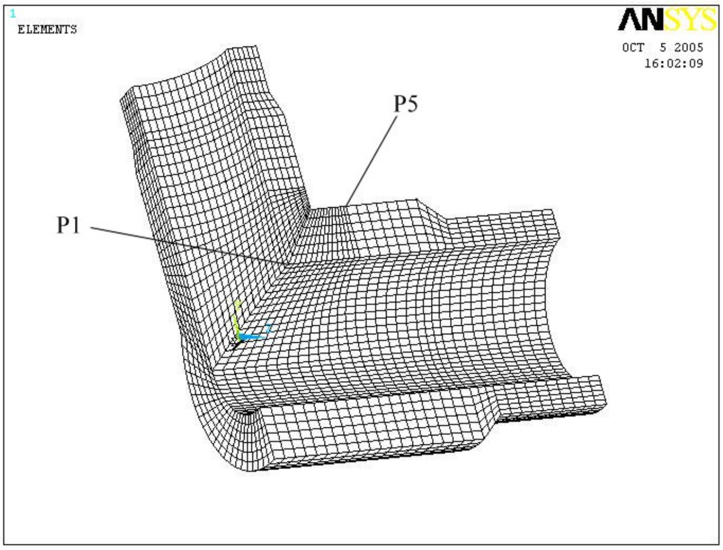

Figure 5: Division of T-pipe into finite elements

Temperature distribution in time and space are calculated on the basis of measured temperature and pressure histories while assuming that the heat transfer coefficient on the inner surfaces equals 2000 W/m^{2}K.

Next, stress analyses are carried out. In order to do so, the calculated temperature distribution in space and time is transferred to the stress model. Additionally, the stress model is constrained to ensure symmetry conditions and to allow for the possibility of free lengthening in the direction of the horizontal and vertical pipe. The basic assumption here is that the surfaces connecting the T-pipe with the pipeline must be planes. The most loaded region is located close to point P_1 presented in Fig.5. The calculated equivalent thermal and total stress histories are presented in Fig. 6.

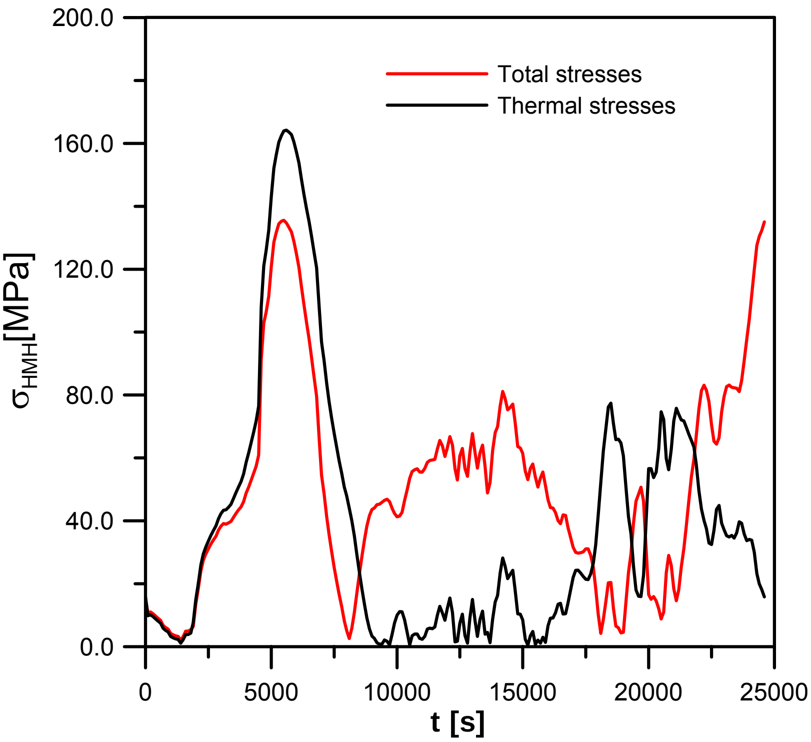

Figure 6: Equivalent thermal and total stress histories

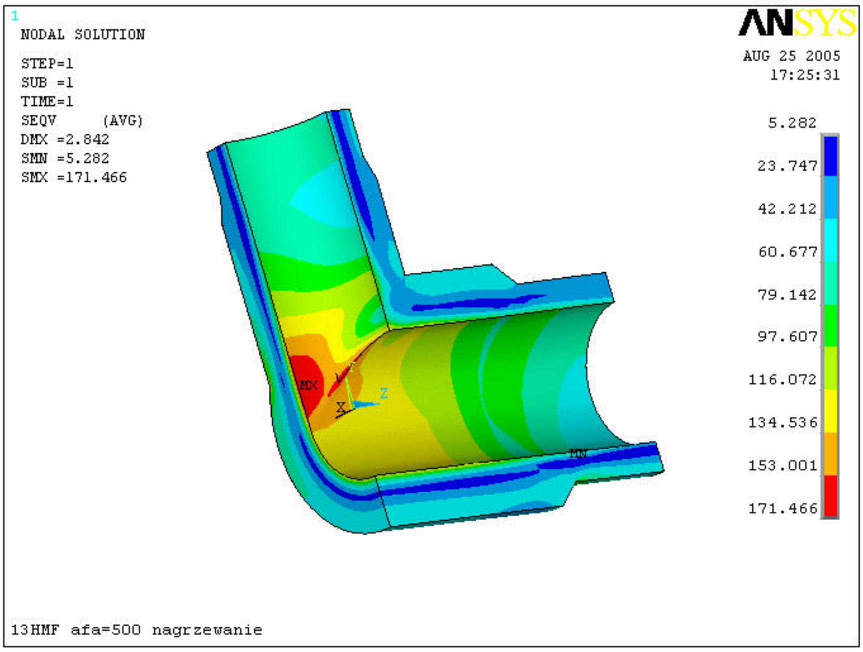

Figure 7: Equivalent stress distribution MPa at time t=5700 s

The highest stresses occur at 5700 s. They are caused by a sudden steam temperature increase. Steam pressure causes tension stresses with an opposite sign to thermal stresses. For that reason, total stresses are smaller than thermal stresses. Stress distribution in the whole T-pipe is shown in Fig. 7.

When assuming that the heat transfer coefficient on inner surfaces is h=2000 W/m^2K, the stresses do not exceed the yield point, which equals Re=245 MPa at a temperature of T=300^{o}C. When h is higher, the yield stresses on the inner surface are likely to occur. It is necessary to monitor this element in order to avoid yield stresses and the shortening of element life.

3 OPTIMIZATION OF THICK WALLED ELEMENT HEATING WITH RESPECT TO THERMAL STRESSES

As presented in the previous chapter, high thermal stresses occur in thick walled boiler components. If the rate of temperature changes is too high permanent deformations can take place in stress concentration locations, which leads to crack damages and the rapid decrease of component life. When the rate of temperature change is too low, the power boiler start-up and shut-down operations are extended in time. This increases start-up losses and extends the standstill period of the power unit. Therefore it is necessary to optimize the fluid temperature changes during power boiler start-up [4].

The optimal fluid temperature change, $T_f(t)$, will be calculated assuming that the maximum equivalent thermal stress should equal the maximum allowable stress σ_a.

$$max|\sigma_{HMH}| = \sigma_a \tag{1}$$

Initially, the T-pipe has a uniform temperature distribution, T_0=0^{o}C. Consider that fluid temperature changes from an initial temperature of $T_{f,0}$=20^{o}C at a constant rate. The heating process is transient and causes the rise of thermal stresses in the T-pipe. Fig. 8 presents three equivalent stress histories in MPa for the chosen rate of temperature change, v_T. Presented stresses are the maximum stresses in the whole T-pipe volume. It can be seen that for all three values of v_T, the thermal stresses approach three asymptotes when quasi-steady state is achieved.

The optimal rate of temperature change, v_T=4.65 K/min, was found for the asymptote σ_a=124MPa using the golden search method [5].

Next, consider that the fluid temperature changes suddenly at the beginning of the heating process by an initial fluid temperature step, T_s, and later rises with the optimal rate of temperature change, v_T=4.65 K/min. For large initial steps, high thermal stresses occur in the beginning of the heating process. Fig. 9 presents three maximum equivalent thermal stress histories for the chosen three initial steps. When the initial step equals 54.5^{o}C, the thermal stress moves quickly to the maximum allowable stress. The optimal value of the initial step was calculated by the golden search method.

The determined optimal fluid temperature history is presented in Fig. 10. Temperature and equivalent thermal stress distributions are presented in Figs. 11 and 12. It can be seen that the most loaded region is located close to the inner edge of T-pipe.

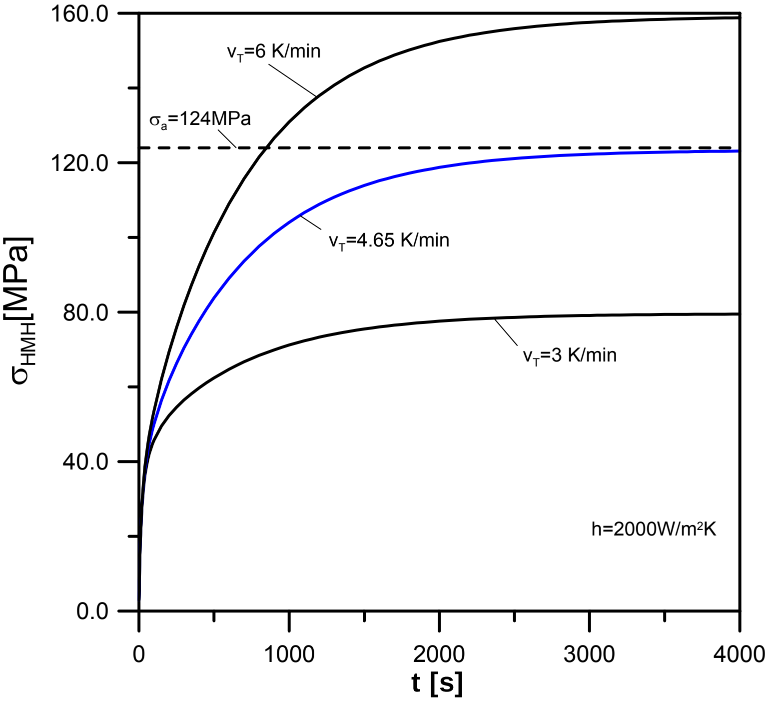

Figure 8: Comparison of equivalent stress histories MPa for three chosen rate of temperature changes v_T with the allowable stress σ_a

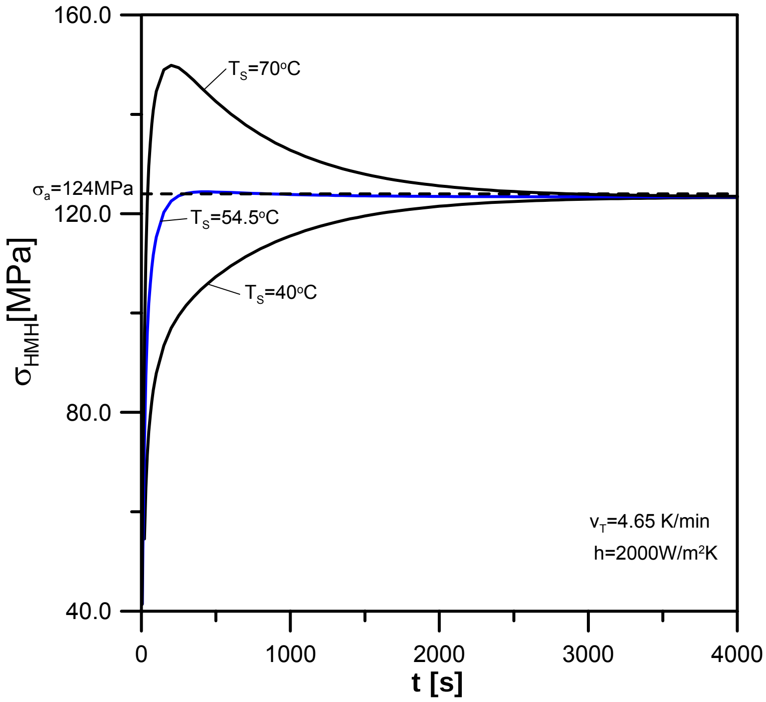

Figure 9: Comparison of equivalent stress histories MPa for three chosen initial medium temperature steps T_s with the allowable stress σ_a

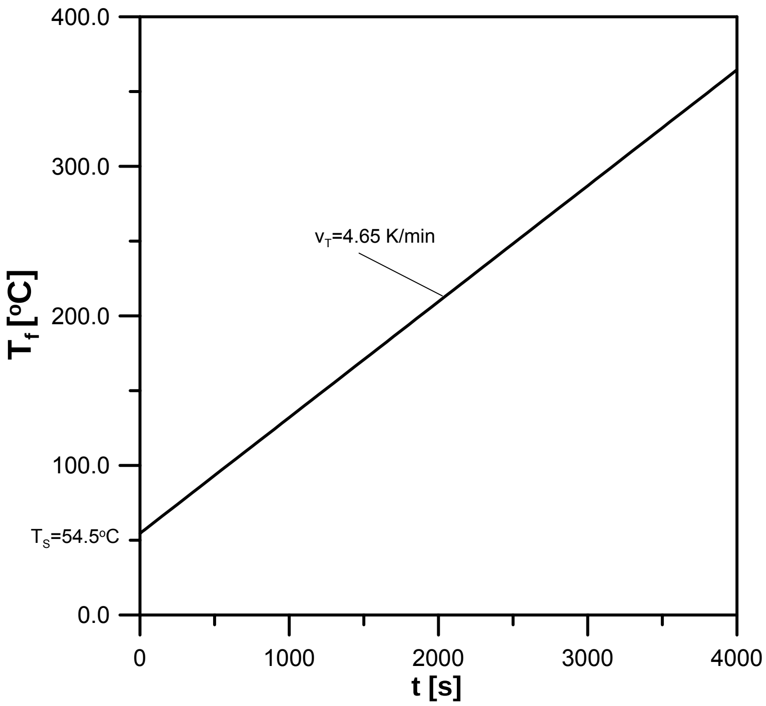

Figure 10: Determined fluid temperature history

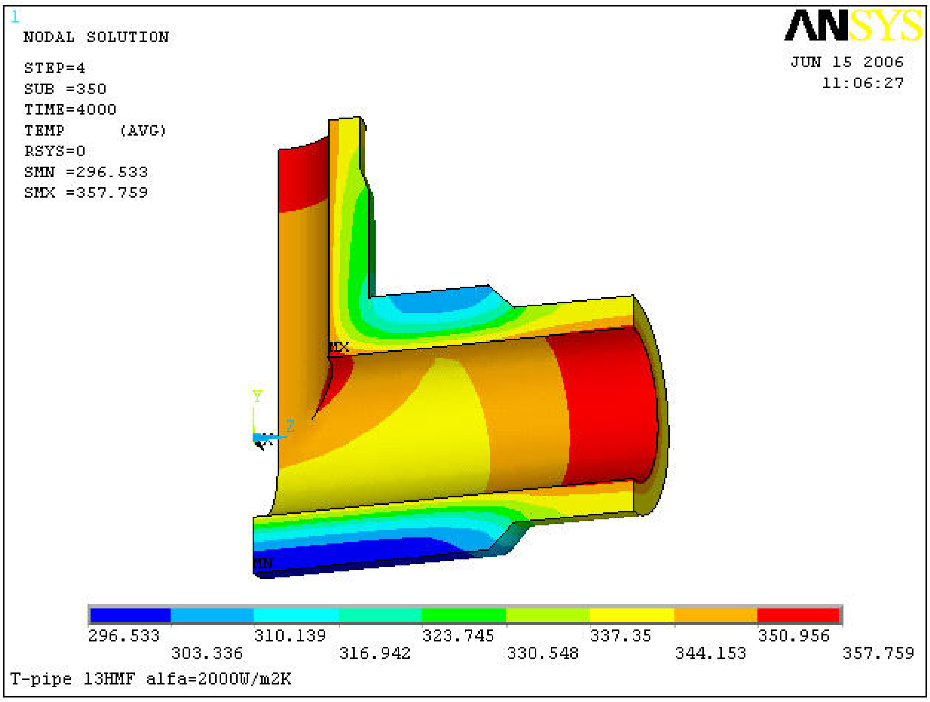

Figure 11: Temperature distribution at time 4000 s assuming the optimal fluid temperature

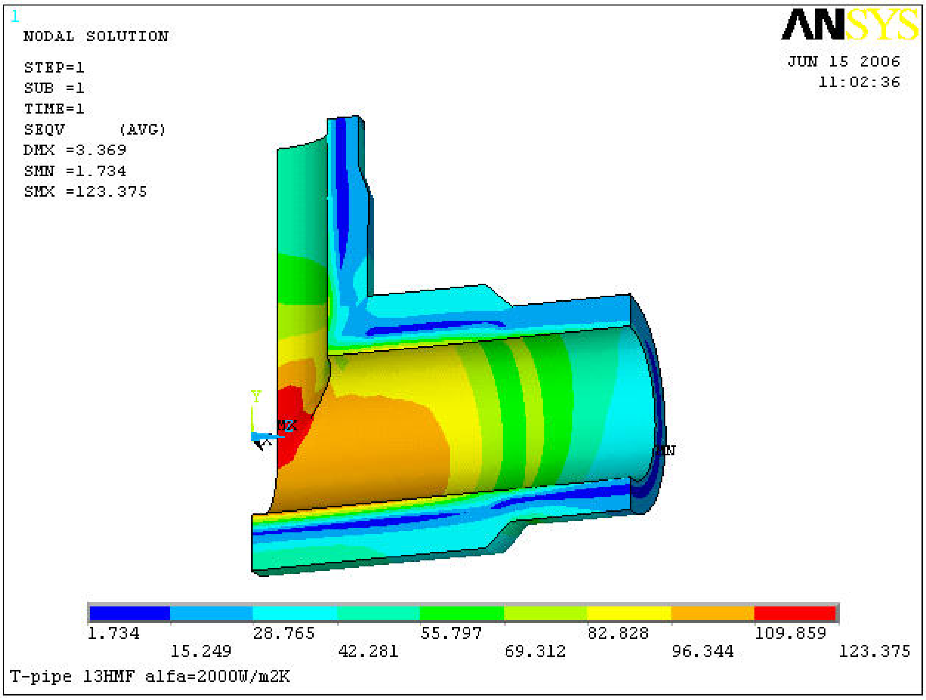

Figure 12: Thermal stress distribution at time 4000 s assuming the optimal fluid temperature

4 CFD ANALYSIS

The T-pipe 3D model was analyzed. Calculations were conducted with the use of Fluent software [6]. Only half the of the geometry was taken under consideration because of the symmetry of the T-pipe. A schematic of the problem and boundary conditions is shown in Fig. 13. T pipes consist of two pipes perpendicular to each other. The inner diameter for both pipes is 0.290 m. Other overall dimensions presents Fig. 1. Geometry was divided into tetrahedron volume elements, mesh consists of $85 \cdot 10^4$ cells.

Fluid is a steam, and the flow is classified as turbulent and transient. The mass flow rate at the inlet is known, and the model is used to predict the flow and temperature fields that result from convective heat transfer.

Mass flow rate at the inlet was set by unit step change of 31 kg/s. Turbulence intensity was set on the level Tu = 5%. Steam constants were obtained from steam data tables using a temperature of 261°C and a pressure of 2 MPa as the reference values.

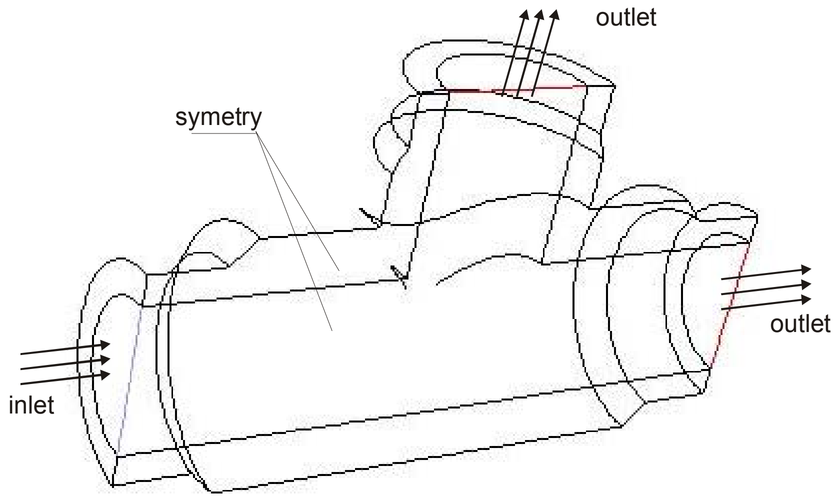

Figure 13: Schematic of the problem

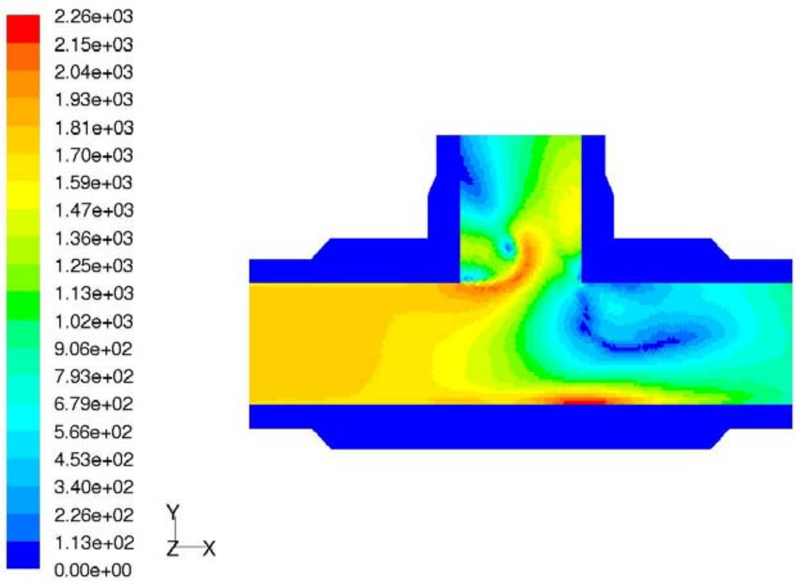

Figure 14: Velocity distribution m/s in cross-section of the T-pipe

Calculated velocity distribution is presented in Fig.14. The heat transfer coefficient obtained from CFD calculations is 2260 W/m^2K

This value is closed to assumed heat transfer coefficient in the thermal strength analyzes and in thermal stress optimalization algorithm. The similar value was obtained during the identification of the temperature field based on measured temperature histories at selected points in the T-pipe using the method for solving the inverse heat conduction problem [7].

5 CONCLUSIONS

- A simple method for determining the optimal fluid temperature history was presented.
- It limits the thermal stresses during power boiler start-up and shut-down operations.
- The calculated results have a very high practical significance because they can be used in power plants.
- The heat transfer coefficient obtained from CFD calculations is similar to the value which was obtained during the experimental identification of the temperature field.

REFERENCES

[1] TRD, Technische Regeln für Dampfkessel, *Carl Heymans Verlag, Köln, und Beuth-Verlag, Berlin*, (1986).

[2] J. Taler, B. Węglowski, S. Grądziel, P. Duda and W. Zima, "Monitoring of Thermal Stresses in Pressure Components of Large Steam Boilers", *VGB KraftwerksTechnik*, **1**, 73-78 (2002).

[3] P. Duda, J. Taler, E. Roos, "Inverse Method for Temperature and Stress Monitoring in Complex-Shape-Bodies", *Nuclear Engineering and Design*, **3960**, 1-17 (2003).

[4] J. Taler, P. Dzierwa, D. Taler, "Determining optimum temperature changes during heat and cooling of trick-walled components.", *Proceedings of the Sixth International Congress on Thermal Stresses*, *TS 2005*, 441-444 (2005).

[5] W. H. Press, S. A. Teukolsky, W. T. Vetterling, B. P. Flannery, *Numerical Recipes, The Art of Scientific Computing*, Cambridge University Press (1997).

[6] Fluent 6.0, Fluent Inc., Computational Fluid Dynamics Software, Centerra Resource Park, 10 Cavendish Court, Lebanon, NH 03766, USA

[7] P. Duda, P. Osocha, S. Grądziel, B. Węglowski, "Monitoring of thermal-strength state in thick-walled pressure elements", *Proceedings of the International Conference Heat Transfer and Renewable Source of Energy HTRSE-2004*, *8-11.09.2004, Międzyzdroje,* 485-492, (2004).