Pipeline heating and cooling

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Abstract. The work aims to determine the undetermined temperature distribution of the medium and the pipeline wall using the finite difference method. Time courses of the temperature of the flowing medium and pipeline walls caused by a step change in temperature of the medium at the pipeline inlet, obtained by the numerical method, were compared with the courses calculated using strict analytical formulas. The numerical method of determining transient courses of the temperature of medium and pipeline wall can be used in the analysis of heating and cooling of heating or power pipelines with any changes in time of mass flow of the flowing medium or temperature of the medium at the inlet to the pipeline.

1 Introduction

During start-up and shutdown of power units, heating and cooling of pipelines, in particular, the pipeline connecting the boiler with the turbine, is an important issue. The temperature course of the medium is important not only because of the thermal stresses in the pipeline wall but also because of the stresses in the fittings installed on the pipeline [1-16]. In the case of steam or gases, the temperature changes along the pipeline length are higher due to the lower density and specific heat of the medium. Large wall thicknesses and considerable wall lengths also contribute to a significant reduction in the temperature of superheated steam in power pipelines.

2 Mathematical formulation of the problem

The heating of the pipeline by a flowing factor is described in the energy balance equation for the fluid [17]

$$\tau_{cz}\frac{\partial T_{cz}}{\partial t}+\frac{1}{N_{cz}}\frac{\partial T_{cz}}{\partial z^{+}}=T_{sc}\big|_{r=r_w}-T_{cz} \qquad (1)$$

and wall

$$\frac{1}{r}\frac{\partial}{\partial r}\left(r\frac{\partial T_{sc}}{\partial r}\right)+\frac{1}{L_r^2}\frac{\partial^2 T_{sc}}{\partial\left(z^{+}\right)^2}=0\,. \qquad (2)$$

Boundary and initial conditions are in the form

$$\begin{aligned}&-\lambda\frac{\partial T_{sc}}{\partial r}\bigg|_{r=r_w}=\alpha_{cz}\left(T_{cz}-T_{sc}\big|_{r=r_w}\right)\\&\frac{\partial T_{sc}}{\partial r}\bigg|_{r=r_z}=0\end{aligned} \qquad (3)$$

$$\begin{aligned}&T_{cz}\big|_{z=0}=T_0+f\left(t\right)\\&\frac{\partial T_{sc}}{\partial z^{+}}\bigg|_{z=0}=\frac{\partial T_{sc}}{\partial z^{+}}\bigg|_{z=L_r}=0\end{aligned} \qquad (4)$$

$$\begin{aligned}&T_{cz}\big|_{t=0}=T_0\\&,\\&T_{sc}\big|_{t=0}=T_0\end{aligned} \qquad (5)$$

where Eqs. (1-5) have the following designations: T_{cz} i T_{sc} – fluid temperature and wall temperature, respectively in °C, t – time in s, $z^{+} = z/L_r$ – dimensionless Cartesian coordinate, L_r – steam pipeline length in m, T_0 – constant initial fluid temperature in °C, $f(t)$ – fluid temperature changes in time at pipeline inlet in °C. The following formulae define the number of heat transfer units N_{cz}, and the fluid time constant τ_{cz}

$$\begin{aligned}&N_{cz}=\frac{h_{cz}A_{in}}{\dot{m}_{cz}c_{pcz}}=\frac{h_{cz}U_{in}L_r}{A_{cz}w_{cz}\rho_{cz}c_{pcz}}\\&\tau_{cz}=\frac{m_{cz}c_{pcz}}{h_{cz}A_{in}}=\frac{A_{cz}L_r\rho_{cz}c_{pcz}}{h_{cz}A_{in}}\end{aligned} \qquad (6)$$

where h_{cz} denotes heat transfer coefficient on the inner surface of the pipeline in W/(m²K), $A_{in} = U_w \cdot L_r$ – inner surface area of the pipeline in m^2, $\dot{m}_{cz} = A_{cz}w_{cz}\rho_{cz}$ – fluid mass flow rate in kg/s, ρ_{cz} – fluid density in kg/m^3, w_{cz} – velocity of the fluid flowing the pipeline in m/s, c_{pcz} – fluid specific heat in J/(kgK), $U_{in} = \pi\ d_{in}$ – inner circumference of the pipeline in m, $A_{cz} = \pi\ (d_{in})^2/4$ – cross-sectional area of the channel occupied by the fluid in m^2.

The initial boundary problem described by Eqs. (1-5) was solved with the control volume method.

The heating process is analysed on the example of a steam pipeline connecting a steam power boiler with a turbine. The pipeline wall is divided into eight layers. The control volumes in the area of heating medium - superheated steam comes into contact with the cells located on the internal radius of the pipe r_w. Due to axial symmetry, only half of the pipeline was analysed.

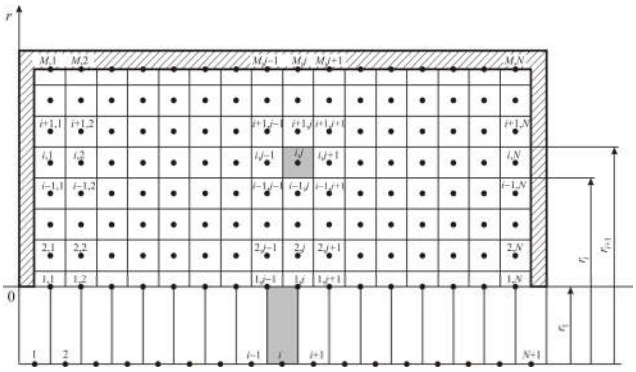

Fig. 1. Diagram of the pipeline division into control volumes

Temperature changes of the fluid and the pipeline wall describe the elementary balance equations recorded for the division shown in Figure 1. Below are presented exemplary heat balance equations for the cell (*i*,*j*) lying inside the wall and the cell (*i*) for the fluid:

$$\begin{aligned} & c_{sc}\rho_{sc}\pi\left(r_{i+1}^2-r_i^2\right)\Delta z\frac{T_{sc,(i,j)}^{n+1}-T_{sc,(i,j)}^{n}}{\Delta t}= \\ & =\pi\left(r_{i+1}^2-r_i^2\right)\frac{T_{sc,(i,j-1)}^{n}-T_{sc,(i,j)}^{n}}{\Delta z}+ \\ & +\pi\left(r_{i+1}^2-r_i^2\right)\frac{T_{sc,(i,j+1)}^{n}-T_{sc,(i,j)}^{n}}{\Delta z}+ \\ & +2\pi r_i\Delta z\frac{T_{sc,(i-1,j)}^{n}-T_{sc,(i,j)}^{n}}{\Delta r}+ \\ & +2\pi r_{i+1}\Delta z\frac{T_{sc,(i+1,j)}^{n}-T_{sc,(i,j)}^{n}}{\Delta r} \end{aligned} \tag{7}$$

$n = 0, 1, \ldots; i = 1, \ldots, M; j = 1, \ldots, N,$

$$\begin{aligned} & \tau_{cz}\frac{T_{cz,i}^{n+1}-T_{cz,i}^{n}}{\Delta t}+\frac{1}{N_{cz}}\frac{T_{cz,i}^{n}-T_{cz,i-1}^{n}}{\Delta z}= \\ & =\frac{T_{sc,(1,j-1)}^{n}+T_{sc,(1,j)}^{n}}{2}-T_{cz,i}^{n} \end{aligned} \tag{8}$$

$n = 0, 1, \ldots; i = 2, \ldots, N+1.$

In the Eqs. (7-8) the following notations were assumed: Δt - time step in s, Δz - spatial step in m, r - radial step in m, Δr - radial step in m, k_{sc} - thermal conductivity of the pipeline wall material in W/(mK), ρ_{sc} - density of the pipeline wall material in kg/m3, c_{sc} – specific heat of the pipeline material in J/(kgK), r_{in} - internal radius of the pipeline in m. The superscripts indicate the time step number. The assumption that the analyzed pipeline is ideally insulated on the external surface was made.

In order to ensure the stability of calculations of T_{cz} factor temperature and T_{sc} wall temperature using formulae (7-8), the Courant condition [17] must be met.

$$\frac{w_i\cdot\Delta t}{\Delta z}\le 1,\qquad i=1,\ldots,N\,. \tag{9}$$

and the condition of stability of the solution of the heat conduction equation using the explicit method [6].

In this case, the pipeline heating can be analyzed, in which the time changes in temperature of the flowing medium at the inlet to the channel is described by any function $f(t)$. In a particular case, when $f(t) = \Delta T_{cz}$. This problem has a strict analytical solution [18]:

$$\frac{T_{cz}-T_0}{\Delta T_{cz}}=e^{-(\xi+\eta)}U\left(\xi,\eta\right), \tag{10}$$

$$\frac{T_{sc}-T_0}{\Delta T_{cz}}=e^{-(\xi+\eta)}U\left(\xi,\eta\right)-e^{-(\xi+\eta)}I_0\left(2\sqrt{\xi\eta}\right), \tag{11}$$

where I_0 is the modified Bessel function of the zero order and the U is the function defined by the formula

$$U\left(\xi,\eta\right)=\sum_{n=0}^{\infty}\sum_{k=0}^{n}\frac{\eta^n\xi^k}{n!k!}\,. \tag{12}$$

In Eqs. (10-12) the following designations were assumed:

$$\begin{aligned} & \xi=\frac{z\cdot N_{cz}}{L_r}, \\ & \eta=\frac{t-z/w_{cz}}{\tau_M}=\frac{t-\left(z\ \tau_{cz}N_{cz}\right)/L_r}{\tau_M}\,. \end{aligned} \tag{13}$$

Eqs. (11-13) will be used to assess the accuracy of the finite difference method. In order to obtain a high accuracy of the analytical solution, the double series (12) was taken as $n > 400$ in the computer calculations, as the series is slowly converging.

3 Numerical simulation of pipeline heating

The steam pipeline to be analysed is made of 13HMF steel. Its dimensions are as follows: length $L_r = 48$ m, internal diameter $d_{in} = 0.217$ m, and wall thickness $g_{sc} = 0.028$ m. The physical properties of the pipe material are: density $\rho_m = 7650$ kg/m^3, specific heat $c_m = 519$ J/(kgK) and thermal conductivity $\lambda_m = 35$ W/(mK). It has been assumed that superheated steam with temperature $T_{cz} = 540$°C and pressure $p_{cz} = 11$ MPa flows suddenly into the pipeline. The steam thermal properties are – density $\rho_{cz} = 28{,}492$ kg/m^3, specific heat $c_{cz} = 2484$ J/(kgK) [19].

The initial temperature of the pipeline and fluid is T_0 = 300°C. The value of heat transfer coefficient on the inner surface of the pipeline, for the preset flow velocity of the fluid w_{cz}, was calculated from the Dittus-Boelter correlation formula. The distribution of fluid temperature T_{cz} and pipeline wall temperature T_{sc} as a function of time was determined from the solution of the a system of elementary balance equations and analytical solution described by Eqs. (10-11).

The analysis of Figs. 2 and 3 shows that the elementary balance method gives results with very good accuracy. It can be noticed that at higher steam speeds both the medium and the wall reach the steady state faster. The method of elementary balances can be applied for any changes in time of the medium temperature at the inlet and at temperature and pressure dependent thermal properties of the medium and wall material. In such cases, it is difficult or impossible to find an analytical solution.

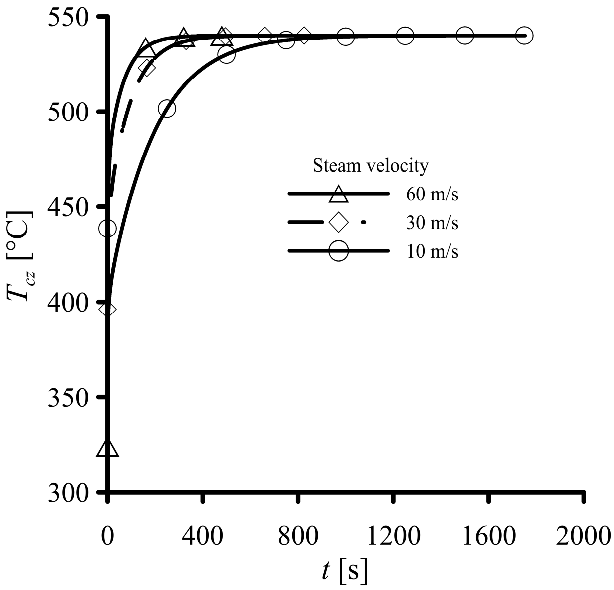

Fig. 2. Time changes of steam temperature for different steam flow velocities at the outlet from the pipeline; the points are the values obtained by a strict analytical solution and the solid line temperature runs determined by the finite difference method

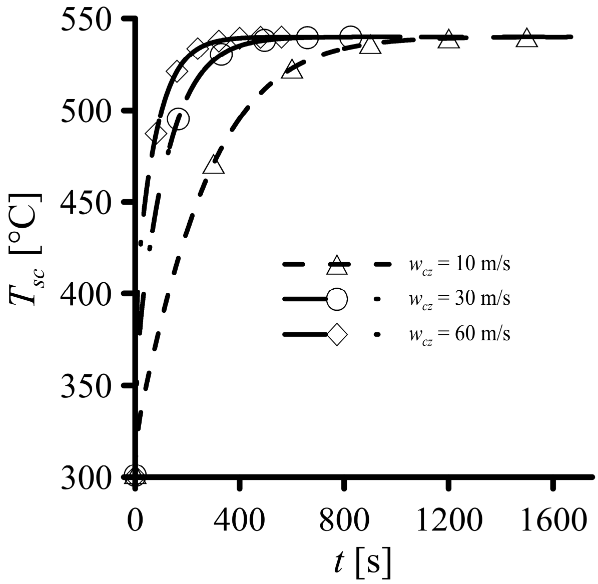

Fig. 3. Change in wall temperature over time at the outlet from the pipeline at different steam velocities; the points are the values obtained by a strict analytical solution, and the solid line temperature runs determined by the finite difference method

Figure 4 shows changes in temperature difference between the inner and outer surface of the pipeline at the end of the pipeline.

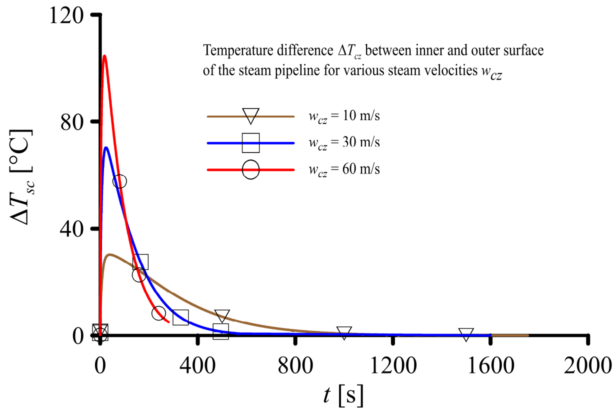

Fig. 4. The temperature difference between the inner and outer surface of the pipeline wall

The analytical solution, allowing to determine the change of heating medium temperature and pipeline wall temperature was derived assuming that the pipeline wall is a body with concentrated thermal capacity and without radial heat dissipation inside the wall. Therefore, in the next stage, the wall temperature as a function of time obtained from the tight solution (8) was compared with the average wall temperature of the pipeline (fig. 5) determined according to the following formula:

$$\overline{T}_{sc}\left(z\right)=\frac{2}{r_z^2-r_w^2}\int_{r_w}^{r_z}T\left(r,z\right)rdr\,. \tag{14}$$

where the temperature distribution $T(r, z)$ is obtained from a numerical solution.

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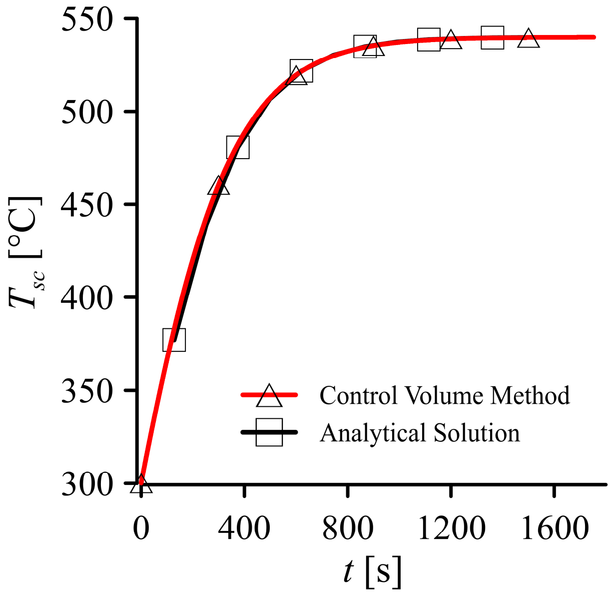

Fig. 5. Comparison of the wall temperature distribution at the end of the pipeline obtained from the analytical solution (8) with the value of the mean temperature calculated from Eq. (11) for the case when the steam flow velocity w_{cz} = 10 m/s**.**

4 Simulation of pipeline heating with FLUENT program

FLUENT [20] was used to determine the distribution of the temperature field in the cross-section of the steam pipeline under analysis. A two-dimensional model of the pipeline was built, which included axial symmetry.

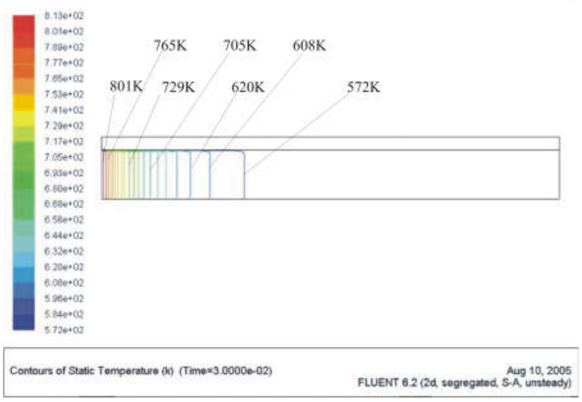

Fig. 6. The temperature distribution of steam obtained with the fluent program at time t = 0.03 s

Then as boundary conditions were set: steam velocity and temperature at the pipeline inlet, wall temperature, heat transfer coefficient on the internal surface of the pipe and ideal thermal insulation on the outer surface of the pipeline, i.e. dT/dr = 0. The distribution of the fluid temperature, at different times points during pipeline heating, is shown in Figures 6 and 7. The simulation was carried out for the following data: T_{cz} = 540°C, T_{sc} = 300°C, w_{cz} = 60 m/s, h_{cz} = 4269 W/(m^{2}K).

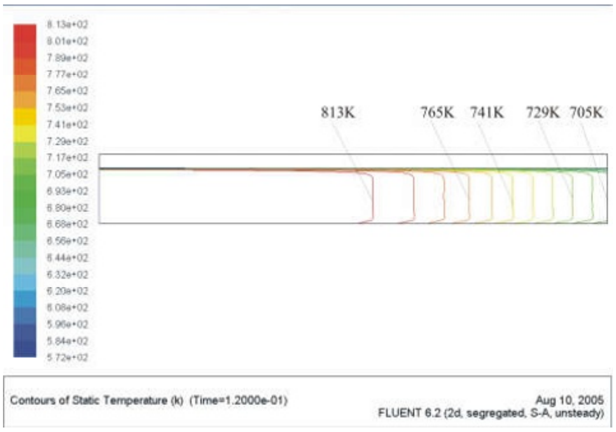

Fig. 7. Temperature distribution of steam obtained with the Fluent program at time t = 0.03 s

Conclusion

The explicit differential method presented in this paper can be used for the analysis of pipeline heating and cooling. A comparison of the calculated time histories of factor and pipeline wall temperature with a stepwise increase in factor temperature at the pipeline inlet with the histories obtained using strict analytical formulae shows that the accuracy of the differential method is very good. The finite difference method can be applied to any temporary changes in temperature of the medium at the inlet to the pipeline and depending on the temperature of thermo-physical properties of the medium and wall.

References

1. D.J. Littler (Ed.), *Modern Power Station Practice. Volume G – Station Operation and Maintenance. Third Edition* (Pergamon Press, Oxford, 1991)
2. L. Cwynar, *Rozruch kotłów parowych* (WNT, Warszawa 1981)
3. J. Taler, D. Taler, K. Kaczmarski, P. Dzierwa, M. Trojan, T. Sobota, Energy, **160**, 500-519 (2018)
4. T. Sobota. Heat Transfer Eng., **39**(13-14), 1260-1271 (2018)
5. T. Sobota, E3S Web of Conferences, **13**, (2017)
6. T. Sobota, E3S Web of Conferences, **14**, (2017)
7. J. Taler, B Węglowski, W. Zima, P. Duda, S. Grądziel, T. Sobota, A. Cebula, and D. Taler, Proc. IMechE **222** Part A: J. Power and Energy, 11-24 (2008)
8. J. Taler, S. Gądziel, T. Sobota, D. Taler, Archiwum Energetyki, **XXXVIII**(1), 97-122 (2008)
9. S. Lubecki, D. Taler, T. Sobota, Archiv. of Thermodynamics, **29** (4), 87-96 (2008)
10. J. Taler, Sz. Lubecki, T. Sobota, Energetyka, **2-3**, 680-681 (2011)
11. J. Taler, B. Węglowski, T. Sobota, D. Taler, M. Trojan, P. Dzierwa, M. Jaremkiewicz, M. Pilarczyk,

Proc. of the ASME 2016 Power Conference POWER2016, (2016)

12. T. Sobota, Testing station for computerized systems for continuous monitoring of power boilers' operation, in *Modern Power Systems and Units*, J. Taler (Ed.) (Wydawnictwo Politechniki Krakowskiej, Kraków, pp. 577-586 2007)
13. M. Jaremkiewicz, D. Taler, T. Sobota, Int. J. of Thermal Sci., **87**, 241-250 (2015)
14. M. Jaremkiewicz, D. Taler, T. Sobota, Appl. Thermal Eng., **29**, 3374–3379 (2009)
15. J. Taler, B. Węglowski, D. Taler, T. Sobota, P. Dzierwa, M. Trojan, P. Madejski, M. Pilarczyk, Energy, **92**(1), 153-159 (2015)
16. J. Taler, D. Taler, T. Sobota, P. Dzierwa, Archiv. of Thermodynamics, **32** (3), 103-116 (2011)
17. P.H. Oosthuizen, D. Naylor, *Introduction to Convective Heat Transfer Analysis* (McGraw-Hill, New York, 1999)
18. E.P Serov, B.P. Korolkov, *Dinamika parogienieratorov*. (Energoizdat, Moscow, 1981)
19. *ASME Steam Tables*. *Sixth Edition* (The ASME, New York, 1993)
20. *FLUENT 6*.0 (Fluent Inc., Lebanon USA, 2004)