Experimental and theoretical research of a hot condition of the T–100–130 steam turbine

I V Kudinov, V K Tkachev, T B Tarabrina and R M Klebleev

Theoretical foundations of heat engineering and hydromechanics Department, Samara State Technical University, 443100 Molodogvardeyskaya street 244, Samara, Russian Federation

* uio1123@list.ru

Abstract. Based on thermal profiling of the cylinder of a high pressure cylinder (HPC) of Т–100–130 steam turbine detailed researched has been performed to study its temperature conditions during startup. Using experimental data about the temperature condition of an external surface of the cylinder, by way of solving the inverse problem of heat conductivity the average heat transfer coefficients have been determined during the period of startup on its internal surface (on the steam side). At the same time, approximated analytical solution of the heat conductivity problem has been applied for the cylinder’s two-layered insulation (thermal insulation – metal wall). Using the data of experimental and theoretical research thermal stresses have been identified in the cylinder wall, as well as the stresses due to the effect of steam pressure forces. It has been shown that in certain profiles of the turbine cylinder stresses are able to exceed the ultimate stress limit for that material. Using experimental data about the changes in certain parameters (temperature differential for the top- and bottom sections of the cylinder, differential in the shaft- and cylinder extension, vibration indicators, etc.) a theoretical method has been developed for forecasting their changes during a certain time interval as measured from the time of current measurement.

1. Introduction

When launching Т–100–130 steam turbine problems may arise relating with a significant pressure differential between the top and bottom sections of the cylinder, between its flanges and studs, as well as with the differential in extension of the turbine share and cylinder [1, 2, 3]. In cases where differences of the said parameters exceed the threshold limit values as defined by the startup regulations, the startup process must be stopped. The said problems are most typical for a high pressure cylinder, both operating at the highest temperatures as compared with other cylinders of the turbine. On the high pressure cylinder of Т–100–130 steam turbine six standard thermocouples have been designed; they are located in the fifth profile (figure 1) – one thermocouple on the top, one thermocouple at the bottom, in two flanges, and two studs. With the purpose of obtaining more detailed information about the temperature conditions and the temperature condition additional thermocouples have been installed on the external surface of the cylinder. Particularly, thermocouples have been installed in seven profiles (from the first S–1 to the seventh S–7) including a profile with standard thermocouples (sections between profiles are numbered with Roman numerals). The diagram for location of thermocouples is shown in figure 1. In profile 1 three thermocouples are installed: 1:1 – left flange, 1:2 – top, 1:3 – right flange. I profiles 2, 3, 4 one thermocouple is located in each (2:1, 3:1,
4:1) in the top section of the cylinder. In profile 5 (standard profile) 6 (six) thermocouples are located:
5:1 – left flange; 5:2 – top; 5:3 – right flange; 5:4 – left stud; 5:5 – right stud; 5:6 – bottom. In the sixth
profile – one thermocouple: 6:1 – top. In the seventh profile – three thermocouples: 7:1 – left flange;
7:2 – top; 7:3 – right flange.

![Diagram for locating thermocouples in HPC profiles.](image)

**Figure 1.** Diagram for locating thermocouples in HPC profiles.

The thermocouples installed are part of the information- and diagnostics system of the T–100–130
turbine generator installed at Samara CHP Plant, operating in real-time mode. The system is designed
for automated collection, processing, display, and archiving of information using personal computers.
Such information includes the turbine cylinder temperature; steam pressure and steam flow consumption;
deformation of the cylinder and shaft; number of revolutions of the turbine shaft; vibration indicators, etc.

The results of experimental research of studies of the temperature condition of HPC cylinder of
the turbine in the startup mode are shown in diagrams of figure 2.

![Distribution of temperature in the upper part of HPC during the startup (curves 1 – 8); 9 – stationary state.](image)

**Figure 2.** Distribution of temperature in the upper part of HPC during the startup (curves 1 – 8); 9 – stationary state.
Based on their analysis one can conclude that the sections II – V are heated up most intensely. The minimum temperature during the startup process is observed in the profiles S – 1 and S – 7. Using the experimental data of change in temperature over time, the method of forecasting of changes in any parameter has been developed. The method is as follows. First, they identify the regularity of change of that parameter over a certain time interval which precedes its current measurement. The value of the forecasted parameter (for instance, the difference of temperatures of the top and bottom section of the cylinder) is shown as

$$ \Delta T = A + Bt + Ct^2, $$

where \( t \) – time; \( A, B, C \) – unknown coefficients.

Let us assume that during the measurement performed at the moment of time \( t_3 \), the temperature differential was \( \Delta T_3 \), while at the moment of time \( t_1 \) and which preceded the time of measurement \( t_3 \), it was correspondingly equal to \( \Delta T_1 \) and \( \Delta T_2 \). Using the values of the parameters measured for the moment of time \( t_1, t_2, t_3 \), the ratio (1) can be recorded in the form of a system of three algebraic linear equations with three unknowns \( A, B, C \). After finding the unknowns the ratio (1) may be used in order to forecast changes in the value \( \Delta T \) over a certain time interval, counting from the time of its current measurement. It is, however, assumed that the law of changes in the forecasted parameter will remain the same as prior to the moment (time) of its measurement. Correspondingly, the accuracy of forecasting it increases as the time intervals before and after the current measurement decrease. Application of this methodology during the turbine startup shows that forecasting the change in temperature differential of the top and bottom sections of HPC cylinder over the time interval of 3 - 5 minutes with subsequent changes of that parameter is validated with a high probability, which allows for taking measures in order to avoid its getting out of the permitted values.

The experimental values of temperatures were used for approximated estimation of the heat transfer coefficients observed when transferring heat from the steam to the turbine cylinder, which were defined based on solution of the inverse problem of thermal conductivity. First, they found a solution of the direct problem for the two-layered wall (metal turbine cylinder – thermal insulation).

2. Statement of the problem

The mathematical statement of the problem in this case looks as follows (figure 3) [4, 5, 10].

$$ \frac{d\Theta_i(\xi, Fo)}{dFo} = \frac{a}{a_1} \frac{d^2\Theta_i(\xi, Fo)}{d\xi^2} \quad (2) $$

$$ (Fo > 0; \xi_{i-1} < \xi < \xi_{i}; i = 1, 2; \xi_0 = 0; \xi_2 = 1) $$

$$ \Theta_i(\xi, 0) = 1; \quad (i = 1, 2); \quad (3) $$

Figure 3. Diagram of the two-layered design.
\[
\frac{d\Theta_i(0,Fo)}{d\xi} + Bi_1\left[\Delta T - \Theta_i(0,Fo)\right] = 0
\]  
(5)

\[
\Theta_i(\xi_0,Fo) = \Theta_i(\xi_1,Fo)
\]  
(6)

\[
\lambda_i \frac{d\Theta_i(\xi_0,Fo)}{d\xi} = \lambda_2 \frac{d\Theta_2(\xi_1,Fo)}{d\xi}
\]  
(7)

where \( \Theta_i = \frac{(T_i - T_{f2})}{(T_0 - T_{f2})}; \) \( \xi = \frac{x}{\delta}; \) \( Fo = \frac{at}{\delta^2}; \) \( Bi_1 = \frac{a_1\delta}{\lambda_1}; \) \( Bi_2 = \frac{a_2\delta}{\lambda_2}; \) \( \Delta T = \frac{(T_{f1} - T_{f2})}{(T_0 - T_{f2})}; \)

\( \Theta_i \) – nondimensional temperature \((i = 1,2)\); \( Fo \) – Fourier number (nondimensional time); 
\( \xi \) – nondimensional coordinate; \( Bi_1, Bi_2 \) – Biot numbers; \( a \) – the smallest of the thermal conductivity coordinates \( a_i \), \((i = 1,2)\); \( T_i \) – temperature of the \( i \)-layer \((i = 1,2)\); \( x \) – coordinate; 
\( t \) – time; \( \delta = \delta_1 + \delta_2 \) – cumulative thickness of the two-layered wall; \( \lambda_i, a_i \) – heat conductivity coefficient and thermal diffusivity coefficient of the \( i \)-layer \((i = 1,2)\); 
\( \alpha_i \) – heat transfer coefficient on the steam side; \( \alpha_2 \) – heat transfer coefficient on the ambient side.

The approximated problem solving (2) – (7) is shown as \([4, 5]\)

\[
\phi_2(\xi) = \frac{(Bi_2 + 2)}{Bi_2}; \quad f(Fo) = C_{\phi_2} \left[ -6\left(\mu_4 + \mu_4 + \mu_4 + 2Bi_1a_2\lambda_1(\xi_1 - 1) - 2Bi_1a_2\lambda_1\right)_{Fo} \right]
\]  
(8)

\[
\phi_2(\xi) = \frac{(Bi_2 + 2)}{Bi_2}; \quad f(Fo) = C_{\phi_2} \exp \left[ -6\left(\mu_6 + Bi_2\xi_1^2\lambda_1(\xi_1 + \xi_1^3) - 4Bi_2\xi_1^2\lambda_2 + \mu_9 - \mu_9 \right)_{Fo} \right]
\]

where \( r_1 = Bi_1Bi_2^2T_0^2\xi_1(\lambda_1 - \lambda_2); \) \( r_2 = 2Bi_2Bi_2^2\xi_1(\lambda_1 - \lambda_2); \) \( r_3 = Bi_1Bi_2^2\xi_1(\lambda_1 - \lambda_2); \)

\[
\Theta(\xi, Fo) = 1.5(1 - \xi^2)\exp(-3Fo); \]  
(9)
3. Analysis of the results obtained

The analysis of solution (9) allows concluding that in the range of the number $0 < Fo < 0.1$ its difference from the precise solution [5] does not exceed 4%.

In figure 4 the time-temperature transformation curve is shown for the upper section of all the profiles as derived based on the data from figure 2, i.e. the values of the thermocouples 1:2, 2:1, 3:1, 4:1, 5:2, 6:1, 7:2 are shown, correspondingly, from the first profile until the seventh profile. By analyzing the curves one can notice that in the startup time frame of 13 – 30 hours to 20 – 30 hours they can be roughly approximated using a linear (curves 1, 4, 5) and quadratic (curves 2, 3, 6, 7) dependence on time

$$T(t) = v_1 + v_2 t; \quad T(t) = v_1 + v_2 t + v_3 t^2,$$

where $v_1, v_2, v_3$ – approximation coefficients.

The ratios (10) by way of solving the inverse problem of thermal conductivity were used to define the heat transfer coefficient ($\alpha_{1}$) on the internal surface of the turbine wall. As applied to the problem (2) – (7) the formulas (10) define the temperature in the point of contact of the layers $x = x_i$, i.e. $T(x, t) = T(x_i, t)$. By substituting the ratios (10) shown in nondimensional form into the left part of
the solution (8) relative to \( \Bi = \alpha \delta / \lambda \) for each profile we derive a transcendent equation, solving which we find \( \Bi \), and hence the heat transfer coefficient \( \alpha \). By using the solution (8) and the ratio (10) we found the average heat transfer coefficients over the time of startup from the steam aide in the point of the top section of HPC cylinder where the thermocouples had been installed. Their values from, correspondingly, the second profile to the seventh profile were the following ones: 12, 20, 36, 35, 11 \( W / (m^2 \cdot K) \).

The analysis of the results derived allowed concluding that the lowest value \( \alpha \) had been observed in the profiles where the diaphragm containers were located, in the space between which the velocity of steam movement was not significant (stagnant zones). The maximum \( \alpha \) were identified where there were high temperature values and high steam current velocities. The values found for heat transfer coefficients allowed to determine the temperature field across the wall thickness and thermal insulation. The results of calculations using the formula (8) as shown in dimensional form for the profiles 1, 2, 4, 5 are shown in the diagrams of figure 6. Their analysis allows concluding that the highest pressure differential across the wall thickness which was 8.2 °C, could be observed in the fifth profile, i.e. where the wall thickness was the greatest (\( \delta = 0.108 \) m).

The thickness of a metal wall of the turbine cylinder is different in various profiles. And, particularly, profile 1: \( \delta = 0.042 \) m; profile 2: \( \delta = 0.07 \) m; profile 4: \( \delta = 0.08 \) m; and profile 5: \( \delta = 0.108 \) m (figure 6).

![Fig. 6. Temperature change in terms of thickness of the two-layered (metal – thermal insulation) HPC wall of the turbine in various profiles along the length of the cylinder (See figure 1).](image)

The results of experimental and theoretical research of the temperature condition of the turbine cylinder were used to calculate the temperature stresses and displacements. Analytical solutions of
thermoelasticity problems [10, 11] cannot be applied for multilayer bodies of complex shape; therefore, the finite element method was used, which is one of the most efficient numerical approaches [4, 6, 7]. In case of its application it is normally possible to satisfy all the main conditions, under which they derive well-conditioned matrices of coefficients with unknown systems of algebraic linear equations. The most important of them are the positive definiteness (members located on the main diagonal are the maximum and positive ones), the banded form of a matrix (members located outside of the band equal to zero), a symmetric property of the matrix in relation to the main diagonal. They derive the band-type matrix by way of the corresponding numbering of node points. The notion of band width has a great significance, i.e. the lower it is the better the matrix is conditioned. Width of the band is fully defined by automatic domain partitioning into elements and numbering of the nodes.

4. Calculations of the temperature stresses

The mathematical setting of the thermoelasticity problem as applied to the solving it using the finite element method looks as follows:

\[
\begin{align*}
\frac{d\sigma_x}{dx} + \frac{d\tau_{xy}}{dy} &= 0; \\
\frac{d\tau_{xy}}{dy} + \frac{d\sigma_y}{dx} &= 0; \\
\frac{d\varepsilon_x}{dy^2} + \frac{d\varepsilon_y}{dx^2} &= \frac{d^2\gamma_{xy}}{dxdy}; \\
\varepsilon_x &= \frac{1}{E}\left(\sigma_x - \nu\sigma_y\right) + \alpha_x; \\
\varepsilon_y &= \frac{1}{E}\left(\sigma_y - \sigma_x\right) + \alpha_y; \\
\gamma_{xy} &= \frac{2(1 + \nu)}{E}\tau_{xy}; \\
\varepsilon_x &= \frac{dU}{dx}; \\
\varepsilon_y &= \frac{dV}{dy}; \\
\gamma_{xy} &= \frac{dV}{dx} + \frac{dU}{dy};
\end{align*}
\]

where \(\sigma_x, \sigma_y\) – normal stresses; \(\tau_{xy}\) – tangential stress; \(\varepsilon_x, \varepsilon_y, \gamma_{xy}\) – deformations; \(U, V\) – displacements; \(\alpha_x\) – linear extension coefficient; \(E\) – modulus of elasticity; \(\nu\) – Poisson’s ratio; \(x, y\) – coordinates.

The boundary conditions are defined as the absence of stresses on the contour in the direction which is perpendicular to the tangency in this point of the boundary (provided there are no outside forces). Besides, conditions must be preset for the absence of displacements of any point by both axes of coordinates (in order to exclude movement of the object as a whole), as well as the absence of displacement in some other point along one of the axes of coordinates (in order to exclude rotation of the object). The band width of the system of algebraic linear equations is defined using the formula \(B = (R+1)Q\), where \(R\) – difference between the highest and the lowest numbers of nodes for the element, in which this difference is the highest; \(Q\) – number of the degrees of freedom (for the two-dimensional problem \(Q = 2\)).

The temperature field in the fifth profile of the HPC cylinder within 15–30 hours of startup and the results of calculation of the displacement and shown in figure 7. Their analysis allows to conclude that due to the significant variance in temperatures between the top–bottom and the area of higher temperatures (on the flanges) the displacements are greater (by 0.85 mm), due to which there are blisters on the cylinder. Such asymmetry of the cylinder deformation can result in emergency situations (opening of the cylinder in the area of flanges, the blade tips scraping the cylinder, occurrence of cracks on the cylinder, etc.).
Figure 7. Deformation of cross profile of the cylinder during its heating. — Dimensions of the profile in cold condition; ---- deformation of the profile during heating. Numbers of the arrows – extension of the cylinder, mm. The encircled numbers are temperatures in °C.

The analysis of results of calculations of the temperature stresses allows to conclude that the stresses $\sigma_x$ on the flanges and $\sigma_y$ in the top- and bottom sections of the cylinder reach their maximum values ($\sigma_x = 6 \text{ kg/mm}^2$ and $\sigma_y = 8 \text{ kg/mm}^2$) immediately in the central part of the profile – on the external and internal surfaces they are equal to zero. The stresses $\sigma_x$ on the flanges and $\sigma_y$ on the top- and bottom sections of the cylinder reach significantly higher values (up to $14 \text{ kg/mm}^2$), which as applied to the cylindrical profile are, in essence, circumferential stresses (figure 8).

Figure 8 Distribution of stresses $\sigma_x$ and $\sigma_y$. ⊗ – extension; ⊘ – compression. Distribution of temperature is shown in figure 7.

In figure 9 the results of calculations of temperature stresses and stresses occurring due to steam pressure forces are shown as defined using the following formulas [8]
\[
\sigma'_\text{\(o\)} = \frac{P_i r_i^2 - P_o r_o^2}{r_o^2 - r_i^2} + \frac{(P_i - P_o) r_i^2 r_o^2}{(r_o^2 - r_i^2) \rho^2},
\]
\[
\sigma'_r = \frac{P_i r_i^2 - P_o r_o^2}{r_o^2 - r_i^2} + \frac{(P_i - P_o) r_i^2 r_o^2}{(r_o^2 - r_i^2) \rho^2},
\]
\[
\sigma'_z = \frac{(P_i r_i^2 - P_o r_o^2)}{(r_o^2 - r_i^2)},
\]

where \(\sigma'_r\) – normal (radial) stress; \(\sigma'_\text{\(o\)}\) – circumferential (tangential) stress; \(\sigma'_z\) – axial stress; \(P_i, P_o\) – internal and external stress; \(r_i, r_o\) – the inner and outer radii; \(\rho\) – current radius (the initial data were as follows: \(P_i = 130\) kg/cm\(^2\); \(P_o = 1\) kg/cm\(^2\); \(r_i = 633\) mm; \(r_o = 710\) mm; \(E = 18000\) kg/mm\(^2\).

The analysis of results of the calculations allows concluding that on the internal surface of the cylinder the circumferential stresses \(\sigma'_\text{\(o\)}\) due to temperature forces and steam pressure forces, being insignificantly different in absolute magnitude, have opposite signs. Thus, the internal surface is practically relieved from circumferential stresses. The signs of temperature stresses and those resulting from steam pressure forces on the external surface of the cylinder turn out to be the same and, hence, they are summarized.

Axial temperature stresses \(\sigma_z\) were defined using the formula (9)
\[
\sigma_z = \frac{\alpha E}{1 - \nu} \left( \frac{2}{r_o^2 - r_i^2} \int T \, dr - T \right),
\]

where \(T\) – temperature which by the wall thickness was perceived in the form of the following linear function \(T = A_1 + A_2 r\), where \(A_1, A_2\) – coefficients of approximation as defined on the basis of experimental and computational data using the temperature on the inside and outside surfaces of the wall.
Based on the practice of operating steam turbines it is known that in the area of top section of the sixth profile (figure 1) on the external surface defects in the form of cracks may occur (in circumferential direction). In order to identify possible reasons of their occurrence a research of temperature- and hot condition has been performed in the area adjacent to the crack. This area was the top right section of the HPC (between the fifth profile and the seventh profile). Based on the temperatures derived based on the experimental and theoretical research, and by using the finite element method, the $\sigma_x$ and $\sigma_y$ temperature stresses have been identified in the subspace of the cylinder’s vertical profile as located between the fifth profile and the seventh profile (see figure 10, 11). Their analysis allows to conclude that in the area of the 6th profile the maximum (up to $35 \text{ kg/mm}^2$) are the temperature stresses of compression ($\sigma_x$) which develop on the internal surface. On the external surface the temperature stresses $\sigma_y$ have an opposite sign with the maximum value of $32 \text{ kg/mm}^2$.

The reason of such distribution of stresses is that the bulge in the area of the 52nd node point (figure 10) on the inside surface, being in the area of the highest steam temperatures, is heated up fast, while the outside surface of the cylinder, due to its high thickness (in this section of the cylinder – up to 20 cm), remains at a significantly lower temperature. In the metal, which expands due to heating, on the internal part of the cylinder compression stress builds up, and on its outside surface – extensions occur. Under certain temperature differential tensile strength, being summed up with stresses due to steam pressure forces, may exceed the ultimate stress limit for this material ($\sigma_f = 36 \text{ kg/mm}^2$).
5. Conclusions

- Thermal profiling of the high pressure cylinder of Т–100–130 steam turbine has been performed, and detailed experimental research has been conducted in terms of its temperature condition during the startup process, based on which and using the analytical solution of a stationary problem of thermal conductivity for the two-layered wall (thermal insulation – metal wall), and the average heat transfer coefficients during the startup have been found on the wall’s internal surface.

- Using the results of experimental and theoretical research of the temperature condition the temperature stresses and deformations have been identified, which allows for concluding on the reasons of occurrence of defects on certain areas of the cylinder’s external surface.

- A method of forecasting the change of certain parameters of the steam turbine during its startup has been developed. This method is based on defining regularities in the change of this parameter over a certain time interval preceding its current measurement by assuming that the law of changing the forecasted parameter on the subsequent time interval will remain the same as the one prior to the moment of its measurement.

Acknowledgments

This work was performed with financial support of the RF Ministry of Education and Sciences as part of a basic component of the state order for the Federal State Budgetary Educational Institution of Higher Education Samara State Technical University (Project No. 1.5551.2017/8.9).

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