Inverse Heat Transfer Problems in Structural Elements of Steam Turbines

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Abstract: The long and safe operation of high-power steam turbines depends on correct estimation of their main components thermal state and clearly identification of heat transfer conditions for the next thermal-stress analysis. The heat transfer conditions in structural elements of steam turbines are characterized by high complexity of processes. So the effective method of their identification is the numerical investigations by solving of inverse heat transfer problems. In work the structural elements of steam turbines (diaphragm and end sealings, exhaust hoods and inner space between cases of turbine K-325-23, 5) for which reasonable to make thermal researches were considered. The overview of solving methods for the inverse heat transfer problems in structural elements of steam turbines was carried out. On base of the conjugate heat transfer problems solving the methodology of boundary conditions identification was proposed. The mathematical model for considered problem was given. The flow structures which have influence on boundary conditions in each considered elements were detected. The results of identification of heat transfer coefficients on steam turbines’ rotors and cases surfaces in criterion equations were presented. There is possibility to use obtained results for repair of operating rotors and modernization of high power turbines’ constructions.

Keywords: Inverse heat transfer problem, exhaust hood, end sealings, heat transfer coefficient, steam turbine.

1. Introduction

During power plants operation the main equipment works with partial loading and it has big influence on thermal state of turbogenerators and on their efficiency and durability.

The thermal state of turbogenerators define by thermal boundary conditions on surfaces of their main elements [1]. The thermal boundary conditions in turbines’ flow-parts were studied enough [2]. But thermal processes in other structural parts of turbines (diaphragm and end sealings, exhaust hoods, inner space between cases etc.) were studied not enough.

There are two methods for thermal processes identification: The numerical calculation and experimental physical measurements. Each of them has own advantages and challenges. The difficulty of experimental measurements is caused by their high expenses, complex structure of turbines and their operating regimes. But these methods give more certain results unlike numerical methods which require verification before usage. The main advantage of numerical methods is possibility to consider any turbine elements at different operating regimes. But in any case for both methods is better to develop criterions equations which will allow defining the thermal boundary conditions during engineering calculations which will be more easy than carrying out investigations for each time when it necessary. This approach will allow decreasing time for numerical or experimental investigations preparation and postprocessing of experimental results.

Usually the expressions for calculating thermal
boundary conditions (heat transfer coefficients on surfaces) are formulated through operating regimes parameters (start-up graphics, time of takeoff switching on, steam flow rate through turbine flow-part and others). It’s caused by limited information which possible to receive during operation and some other factors like high pressure and temperatures, complex structure of turbine elements etc.

The complication of experimental investigations preparation cause limited volume of obtained data and limitation of measured temperature differences allow to define heat transfer coefficients at turbines start ups or at their work with partial loads. For obtained full information about thermal processes in different parts of steam turbines it is necessary to use numerical methods together with generalizations of physical experimental results.

More often the definition of heat transfer coefficients on inner surfaces of steam turbines which are in operating is carried out by solving of inverse heat transfer problems [3-7]. In this case the temperature of metal walls and temperature of main flow steam are using.

2. Problem Definition

Three elements of steam turbines will be considered in this work: End sealings, exhaust hood and inner space between cases of high pressure cylinder (Fig. 1).

The exhaust hoods of high-power steam turbines often have typical structure. The inner cavity of exhaust hood has close to toroidal form with additional elements which create flow twisting. The inlet surface of exhaust hood represented by ring surface. The outlet (one or several) is placed in bottom part of cylinder case. In this work the exhaust hood of turbine K-325-23, 5 [5] was considered (Fig. 2).

The space between cases is formed by inner case and outer case. The inner space between cases is joined to space between 9th and 10th stage by ring chink and joined to camera which formed by end sealings. In this work the inner space between cases of high pressure cylinder of turbine K-325-23, 5 [5] was considered (Fig. 3).

The heat exchange conditions in labyrinth sealings have high influence on thermal state of turbine’s stator and rotor. The complexity of flow in sealings doesn’t allow computing heat transfer coefficients by analytical methods. So in this case it is necessary to use experimental methods together with inverse heat transfer problems solving. In this work the two types of end sealings were considered (Fig. 4).

For identification of heat transfer coefficients on surfaces of turbines’ elements was investigated the rated duty and partial loading of turbine.
3. Methodology of Solving

The mathematical model of direct conjugate heat transfer problem consists of next equations:

1. For gas environment:
   - Continuity equation (mass sources and sinks are absent):
     \[ \frac{\partial \rho}{\partial \tau} + \text{div}(\rho \mathbf{v}) = 0 \]  (1)
     where \( \rho \) — density of gas; \( \tau \) — time; \( \mathbf{v} \) — velocity vector.
   - Equation of motion of viscous gas:
     \[ \rho \frac{\partial \mathbf{v}}{\partial \tau} = -\text{grad}(p + \frac{2}{3} \mu \text{div} \mathbf{v}) + 2\text{div}(\mu \mathbf{S}) \]  (2)
     where \( p \) — static pressure; \( \mu \) — effective coefficient of dynamic viscosity, \( \mathbf{S} \) — tensor of strain rates.
   - Energy equation:
     \[ \frac{\partial}{\partial \tau} (\rho E) + \text{div}(\rho E \mathbf{v}) = \text{div}(\lambda \text{grad} T) \]  (3)
     where \( E \) — total energy, carried to mass unit; \( i \) — enthalpy of gas; \( T \) — temperature; \( \lambda \) — heat conductivity coefficient (in consideration of turbulent constituent). In consequence of small absolute value of dissipative function there are not sources of heat here which were generated of mechanical work of gas flow.

2. For solid medium—the equation of heat conduction (the Fourier equation):
   \[ \text{div}(\lambda \text{grad} T) = c_p \rho \frac{\partial T}{\partial \tau} \]  (4)
   where \( \lambda \) — heat transfer coefficient; \( T \) — temperature; \( c_p \) — heat capacity of material.

For closure the differential equations system added of the equation of gas environment state is dependence the density of environment from temperature and pressure. For calculation of the turbulent components which were included to equations of moving and energy was used the Menter’s model of the shear stresses \( k-\omega \) [6]. The physical characteristics in equations were considered as functions of temperature and pressure in working interval of temperatures.

The inverse heat transfer problem is solved on base of the conjugate problems results [7].

Based on temperature field for the part of surface was calculated the heat flux \( q_i \) (W) through this part and its temperature \( T_{wi} \) (°C). In this case the heat flux calculates in the following way.

According to Fourier law the heat flux through element area \( dF \) in solid body is equal
\[ dQ = -\lambda \frac{\partial T}{\partial n} dF \]  (5)
where \( \lambda \) — heat conductivity of solid body; \( \frac{\partial T}{\partial n} \) — temperature gradient in direction which normal to area element \( dF \).

The equation of heat balance for volume element \((i,j)\) will have next form
\[ (dQ_x - dQ_x + dx) + (dQ_y - dQ_y + dy) = dQ_t \]  (6)
where \( dQ_x \) — the heat flux through left side of volume element; \( dQ_x + dx \) — the heat flux through right side of volume element; \( dQ_y \) — the heat flux through bottom side of volume element; \( dQ_y + dy \) — the heat flux through upper boundary side of volume element (through surface of body from element volume to operating environment); \( dQ_t \) — heat power, which accumulate by element volume during element time leg \( d\tau \).

According to finite-difference approximation this value can be calculated as
\[ dQ_{ik} = \rho c \frac{T^{(k)}_{ik} - T^{(k-1)}_{ik}}{d\tau^{(k)}} dx dy dz \]  (7)
where \( \rho \) — density of solid body; \( c \) — specific heat capacity, upper index \( k \) show that the value relate to \( k \)-th moment of time.

The substitution (6) to (7) and next finite-difference
Inverse Heat Transfer Problems in Structural Elements of Steam Turbines

approximation gives

\[
\begin{align*}
&\left(2\lambda \frac{T_{i+1j} - T_y}{dx_{i+1} + dx} + dy, dz - 2\lambda \frac{T_y - T_{i+1j}}{dx + dx_{i+1}} dy, dz\right) + \\
&+ \left(2\lambda \frac{T_{j+1i} - T_y}{dy_{j+1} + dy} dac_J, dz - dQ_{i+1j} dy\right) \\
&= \rho c \frac{T_{i+1j} - T_{j+1i}}{dz^2} dx, dy, dz
\end{align*}
\]

where \(T_y\) — temperature in element volume \((i, j)\); \(dx_i\) and \(dy_j\) — sizes of volume \((i, j)\); \(dz\) — permanent for all volumes the “fictive” size in direction \(z\).

After that HTC on each part of body surface calculated with used Newton-Rihman’s law as

\[
\alpha_i = \frac{q_i}{f_i(T_f - T_{wi})}
\]

where \(T_f\) — temperature of free flow near the surface; \(T_{wi}\) — temperature of \(i\)-th surface part; \(f_i\) — area of \(i\)-th surface part; \(q_i\) — heat flux through \(i\)-th surface part.

4. Calculation Data

Further the results of numerical investigations and the criterion equations for each turbine element are presented.

1. Exhaust hood

The steam velocity distribution (Fig. 5) in exhaust hood influence on the heat exchange conditions on its surfaces.

In toroidal camera the flow rate increases from section I-I to section IV-IV. In the bottom part of exhaust hood the steam moves from section VI-VI to section V-V.

In all points on inner surface of exhaust hood the Nusselt number \(Nui\) at increasing flow rate is changing monotonously. The local Nusselt numbers changing can be approximated by power law

\[
Nu_i = \frac{q_i}{f_i(T_f - T_{wi})} \frac{G_{12}^n}{G_{12}}
\]

where \(G_{12} = G_{12}/G_{12}; \ G_{12}\) — steam flow rate through 12-th stage at rated power of turbine.

The relative heat transfer coefficients \(\alpha_i\) circumferential variation on inner surface of HPC outer casing (Fig. 6).

In the most part of control points the relative HTC isn’t changed but in point near the exhausts the relative HTC has the big changes. These changes depend on structure of steam flow.

The changing of \(\alpha_i\) through generatrix of surface in meridional sections of exhaust hood camera (Fig. 5, for section III-III) can be approximated by linear dependence

\[
\alpha_i = \alpha_0 - K \cdot (1 - x)
\]

where \(K\) — coefficient which depends from turbine’s operating regime:

\[
\begin{align*}
\text{At } 0 < \overline{G} \leq 0.5, K &= 0.28 \cdot \overline{G} \\
\text{At } 0.5 < \overline{G} \leq 0.7, K &= 0.52 \cdot \overline{G} - 0.12 \\
\text{At } 0.7 < \overline{G} \leq 1.0, K &= 0.006 + 0.202
\end{align*}
\]

where \(\overline{G} = G_i/G_{max}; \ G_i\) — flow rate in the considering regime.

2. Space between cases

The structure of steam flow in space between cases is shown on Fig. 7. The flow is characterized by wavy
Inverse Heat Transfer Problems in Structural Elements of Steam Turbines

For stepped sealing the heat transfer conditions can be presented by dependence

$$\text{Nu} = 0.478 \cdot \text{Re}^{0.7} \cdot \text{Pr}^{0.43} \left( \frac{H}{\delta} \right)^{-0.56}$$  \hspace{1cm} (16)

In both dependences

$$\text{Nu} = \frac{\alpha_{av} \cdot 2\delta}{\lambda}, \quad \text{Re} = \frac{W \cdot 2\delta}{\nu}$$  \hspace{1cm} (17)

5. Conclusions

The uses of modern modeling methods of thermal and gas-dynamics processes which based on the solving of direct and inverse problems of heat transfer allows to identify of heat transfer conditions in highly engineered elements of turbine. These methods have high effectiveness at modeling processes in exhaust hoods, space between cases and end sealings of high-pressure and intermediate-pressure cylinders of steam turbines also.

The more results which received with analytical, numerical and experimental approaches are presented in [8].

The methodology of inverse problems usage together with numerical investigations can be used in other fields of turbines designing. In particular for calculations main parameters of small turbines which work with non-traditional operating mediums. This direction is very perspective for Ukrainian turbine industry.

References

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