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AIR COOLING OF A TURBINE GUIDE VANE FOR A STATIONARY GAS-TURBINE PLANT (GTP) WITH A HIGH GAS PRESSURE

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A guide vane for a proposed stationary GTP is calculated with π_r = 80, 100, and 126 and T_g^* = 1673°K. The unsuitability of the convective blade cooling used up to now is noted.

One of the most promising directions in the development of gas turbines is an increase in the temperature and pressure of the gas ahead of the turbine. The tendency of an increase in the gas temperature in gas-turbine plants (GTP) has developed most clearly in recent years: Whereas temperatures of 800-850°K predominated in the 1950's, in the 1970's the temperature level for a base-load GTP has reached 1050-1150°K, while in peak-load GTP it has reached 1100-1250°K. The mastery of gas temperatures on the order of 1500-1600°K is anticipated in the near future. And the achievement of these temperatures will depend mainly on the cooling of the turbine and primarily of its blade apparatus.

The research of the Moscow Higher Technical School devoted to the creation of high-temperature GTP [1] has shown the following:

1) The efficiency of a plant and its specific power grow with an increase in the gas temperature T_g^* ahead of the turbine when there is a corresponding increase in the degree of pressure rise π_r. The efficiency can reach 47-49% in nonregenerative gas-turbine plants with multistage compression and expansion when T_g^* = 1500°K and π_r = 130°.

2) The intensity of heat exchange between the gas and elements of the flow section grows sharply with an increase in the absolute pressure of the gas ahead of the turbine when the hydraulic, thermodynamic, and geometrical parameters of the stage are unchanged.

3) With appropriate design of the cooling system the gas temperature ahead of the turbine can reach 2000°K.

As shown by calculations made earlier, experiments, and tests of turbine operation, the highest coefficient of heat transfer from the gas to the wall of a blade is observed at the leading (inlet) and trailing (outlet) edges, and it is far lower on the side (concave and convex) parts of its surface. The use of deflector vanes with transverse flow of the cooling air yields the best results in cooling the narrowest parts of the vane surface, mainly the leading edge, since in frontal onflow the coefficient of heat transfer from the wall to the cooling air increases by two to three times in comparison with its value for turbulent flow in a smooth channel.

In the present article we present the results of a calculation of a deflector guide vane with T_g^* = 1673°K and π_r = 80, 100, and 126. We analyzed a vane whose wall is made of a promising heat-resistant alloy having a melting temperature of 1473°K and λ = 45.4 W/m²·deg. As a result of gasdynamic and strength calculations we determined the following: l = 0.0662 m, D_{av} = 0.66 m, s = 0.096, $\bar{t}$ = 0.65; z = 33.

In the calculation it was assumed that the stagnation temperature of the gas at the vane, the conditions of external heat exchange, and the distribution of cooling air along the length

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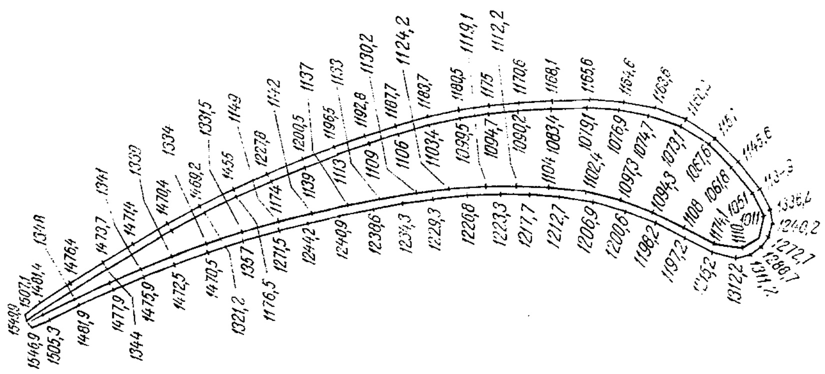


Fig. 1. Temperature distribution over vane surface; $T_g^* = 1673°K$, $\pi_r = 126$.

l are identical for all vane cross sections and that the meridional cross section and gas temperature are the same for all three of the modes indicated above.

The calculation was made in accordance with the method presented in [2]. The cross section at the mean diameter of the stage was taken as the calculating vane cross section.

The average value of the Nusselt number over the vane profile was determined from the conditions of onflow of the gas stream by the equation

$$Nu_g = A\,Re^{0.68}, \text{ where } A = f\,[(\alpha_0 + \alpha_1)/2]. \quad (1)$$

Actually, in the flow of a gas over a vane contour the local values of α_g differ considerably from the average values. The distribution of local values of α_g along the contour is subject to a calculation constructed on a solution of the integral equations for the boundary layer. The method developed at the Central Scientific-Research Institute for Boiler and Turbine Design and Construction (TsKTI) [3] for calculating the distribution of α_g over a contour is well known, for example.

From an analysis of the curve of local values of α_g obtained experimentally and by the TsKTI calculating method it follows that the entire vane contour can be divided into a series of characteristic sections, and in each section one can take α_g = const with sufficient accuracy for the calculation, namely:

at the inlet edge (section with an angle of 140°)

$$Nu_{g_{in.ed}} = 0.23\,Re_g^{0.625}; \quad (2)$$

at the outlet edge (section of length s/3)

$$Nu_{g_{out.ed}} = 0.0263\,Re_g^{0.8}; \quad (3)$$

on the remaining convex part (back)

$$\alpha_{g_{conv}} = 0.8\,\alpha_g; \quad (4)$$

on the remaining concave part (trough)

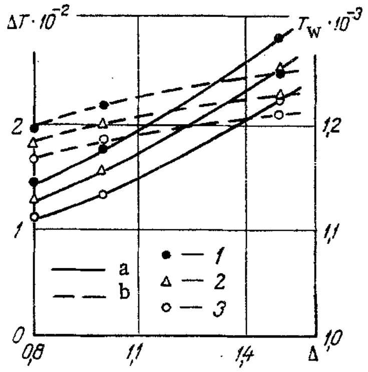
$$\alpha_{g_{conc}} \cong \alpha_g. \quad (5)$$

To determine the coefficient of heat transfer from the vane wall to the cooling air flowing onto the inlet edge we used the equation

$$Nu_a = 0.18\,Re_a^{0.8}. \quad (6)$$

With the above-indicated division of the contour into sections the calculation of the temperature field of the vane can be reduced to the calculation of the temperature distribution along the characteristic sections. In this calculation the temperature of the cooling air at the end of one section is taken as equal to the air temperature at the start of the next section. In this way the calculations over each of the sections are joined to each other.

The temperature field of the outer surface of the vane walls was calculated on an M-220 computer with allowance for the heat transfer in both the transverse and longitudinal (along the wall) directions. The temperature of the inner surface of the walls was determined from the law of heat transfer through a plane surface for the convex and concave parts of the



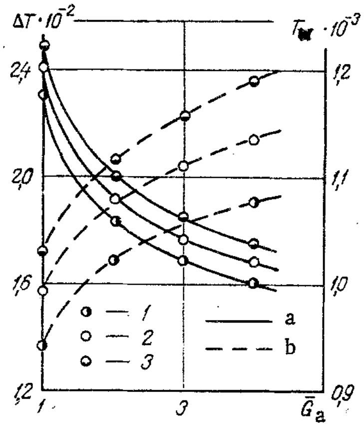


Fig. 2 Fig. 3

Fig. 2. Functions $\Delta T = f(\Delta)$ (a) and $T_w = f(\Delta)$ (b): 1) $\pi_r = 126$; 2) $\pi_r = 100$; 3) $\pi_r = 80$; Δ, m; ΔT, T_w, °K.

Fig. 3. Functions $T_w = f(\bar{G}_a)$ (a) and $\Delta T = f(\bar{G}_a)$ (b): 1) $\pi_r = 80$; 2) $\pi_r = 100$; 3) $\pi_r = 126$; $\bar{G}_a$, %; ΔT, T_w, °K.

contour and the outlet edge and from the law of heat transfer for a cylindrical surface at the inlet edge of the vane.

The calculated values of the temperature field at the surface of a deflector vane for the mode of $T_g^* = 1673$°K, $\pi_r = 126$, and $\bar{G}_a = 1\%$ are presented in Fig. 1. As is seen, under these conditions of turbine operation the wall temperature of the inlet and outlet edges of the vane is so high that none of the modern heat-resistant alloys is able to withstand it, and, in addition, there is a very pronounced temperature drop in the wall, which clearly affects the strength properties of the vane. The size of the temperature drop is in a direct dependence on the temperature and amount of the cooling air and on the wall thickness.

The increased heating of the inlet edge can be explained by its unfavorable shape. An increase in the radius of the inlet edge allows one to considerably decrease the temperature in this zone, and therefore in the present calculation the radius was increased to 0.005s, which allowed a decrease in the integral-mean temperature at the outer surface to 1280°K.

To clarify the influence of the thickness of the vane wall and the amount of cooling air on the temperature of the outer surface and the temperature drop in the wall we made calculations with $\Delta = 0.8$, 1.0, and 1.5 mm and $\bar{G}_a = 1$, 2, and 4% (Figs. 2 and 3). The size of the gap between the vane wall and the deflector was kept constant and equaled 0.5 mm. As the calculations show (Fig. 2), the wall thickness has little effect on the temperature variation of the outer surface (the calculations were made for the inlet edge), although the size of the temperature drop grows sharply.

In the GTP working mode of $T_g^* = 1673$°K and $\pi_r = 126$ an increase in the flow rate of cooling air to 4% (Fig. 3) decreases the temperature of the outer surface of the wall of the inlet edge to 1000°K, but the size of the temperature drop increases by more than 30%, creating additional thermal stresses in the vane wall. The turbine efficiency decreases at the same time.

Along with the analysis of the influence of the flow rate of cooling air on the temperature of the outer surface and the temperature drop in the wall of a vane, we also analyzed the influence of the initial air temperature on these parameters. The calculation was made by the equation

$$T_w = T_g^* - \Theta\left(T_g^* - T_a^*\right). \qquad (7)$$

In the calculation we took $\bar{G}_a = 1\%$. At such a flow rate of cooling air $\Theta = 0.4$ [3]. Such a high value of Θ at a relatively low flow rate of cooling air is explained by the fact that the vane apparatus of the turbine consists of vanes with a relatively large chord and at the usual values of $\bar{t} = 0.65$ their number is large ($z = 33$). It should be noted that while $\Theta = f(z; s)$, the relative depth of cooling depends on the flow rate of cooling air to a far greater degree, which follows from the equation

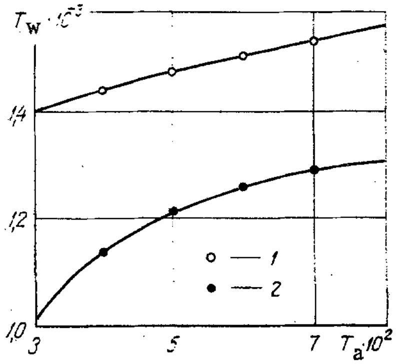


Fig. 4. Function $T_w = f(T_a^*)$: 1) at outlet edge; 2) at inlet edge.

$$\Theta = \left\{1 - \exp\left[-\frac{St_g K}{(1 - \alpha_g F_v/\alpha_a F_{cool})\bar{G}_a}\right]\right\} \frac{\bar{G}_a}{St_g K}, \tag{8}$$

where $St_g = \alpha_g F_g / G_g C_{pg}$ and $K = F_v/F_g$.

As shown in [4], the function $\Theta = f(\bar{G}_a)$ is, with the accuracy of the constant Reynolds number Re_g for the gas stream, a universal characteristic of a cooling system for turbine vanes.

As seen from Fig. 4, a sharp decrease in the initial temperature of the cooling air does not yield appreciable results in a decrease of the temperature of the outer surface of the vane wall, especially at one of the narrowest sections, the outlet edge. Thus, a decrease in the initial temperature of the cooling air does not yield the desired effect in convective vane cooling. An increase in the flow rate of the air with a simultaneous decrease in its temperature also is not a sufficiently effective measure, since the grain in Θ decreases with an increase in $\bar{G}_a$; i.e., the use of cooling systems with large reduced flow rates of air proves unjustified owing to the increase in the expenditures on cooling.

From the foregoing it follows that for high-temperature gas turbines with $\pi_r > 100$ the heat fluxes to the vane walls are so high that the convective method of cooling proves unacceptable, and one must employ the convective-blocking method.

NOTATION

C_{pg}, heat capacity of gas at constant pressure, J/kg•deg; D_{av}, average vane diameter, m; F_g, F_v, F_{cool}, areas of through cross section of vane array, of outer surface of a vane, and of cooling surface, m²; G_a, G_g, weight flow rates of cooling air and of gas, kg/sec; $\bar{G}_a = G_a/G_g$; l, vane length, m; S, chord length, m; $\bar{t} = t/s$, relative pitch of vanes; T_a^*, T_g^*, T_w, temperatures of cooling air, of gas stream, and of outer surface of vane, respectively, °K; ΔT, temperature difference between outer and inner surfaces of vane wall, °K; z, number of turbine vanes; $Nu_g = \alpha_g s/\lambda$, $Re_g = w_g s/\nu$, Nusselt and Reynolds numbers; St_g, Stanton number; w_g, velocity of gas stream, m/sec; α_0, α_1, angles of entry of gas stream into turbine and exit from it, deg; α, coefficient of heat transfer, W/m²•deg; α_a, α_g, coefficients of heat transfer to cooling air and from gas to wall, W/m²•deg; Δ, thickness of vane wall, m; $\Theta = (T_g^* - T_w)/(T_g^* - T_a^*)$; λ, coefficient of thermal conductivity of wall, W/m²•deg; ν, coefficient of kinematic viscosity of gas stream, cm²/sec; π_r, degree of rise of gas pressure. Indices: in.ed, out.ed, at inlet and outlet edges; conc, conv, at concave and convex surfaces.

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USE OF METHODS OF SOLVING INVERSE PROBLEMS OF HEAT CONDUCTION TO ESTABLISH THE COEFFICIENT OF HEAT TRANSFER IN JET COOLING

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The problem of determining the coefficient of heat transfer is analyzed as an inverse problem for the heat-conduction equation. The results of a calculation of the coefficient of heat transfer on the basis of experimental data on the jet cooling of metal plates are presented.

The jet cooling of metal (cooling of a continuous ingot, thermal treatment of manufactured articles and intermediate products, etc.) is widespread in metallurgical production. Data on the local (effective) coefficient of heat transfer α as a function of the surface temperature T^{sur} of an ingot [1] are required for the calculation and design of systems and the attainment of the optimum modes of cooling. The appropriate data in the literature are inadequate, however.

The experimental determination of the boundary temperatures or fluxes needed for the reconstruction of α and the temperature field in the entire volume of a body is strongly hindered and sometimes impossible owing to the high intensity of heat exchange and the sharp fluctuations in the heat load characteristic of jet cooling. Moreover, it is often impossible to reliably measure the temperature of one or several internal points of the body. In such cases the problem of reconstructing the unknown boundary conditions is formalized in the form of an inverse problem of heat conduction, which has raised great interest in recent years (for example, see [2, 3] and the literature cited there).

We considered three approaches to the solution of this inverse problem, corresponding to the reconstruction of one of three kinds of boundary conditions at the heat-exchange surface. The relative effectiveness of the proposed methods and of the regular-regime method [4], which is widespread in engineering practice, is studied.

Within the framework of the one-dimensional model the process of heat propagation in a solidified ingot (plate) being cooled can be represented as a boundary-value problem for the heat-conduction equation

$$c\rho \frac{\partial T}{\partial \tau} = D \frac{\partial^2 T}{\partial x^2}, \; 0 < x < L, \; 0 < \tau \leqslant \theta, \tag{1}$$

$$T|_{\tau=0} = T^0(x), \; 0 \leqslant x \leqslant L, \tag{2}$$

$$-D \frac{\partial T}{\partial x}\Big|_{x=L} = 0, \; 0 < \tau \leqslant \theta \tag{3}$$

with one of the following boundary conditions at x = 0:

$$T|_{x=0} = T^{sur}(\tau), \; 0 < \tau \leqslant \theta, \tag{4}$$

$$-D \frac{\partial T}{\partial x}\Big|_{x=0} = q(\tau), \; 0 < \tau \leqslant \theta, \tag{5}$$

$$-D \frac{\partial T}{\partial x}\Big|_{x=0} = -\alpha(\tau)\,(T|_{x=0} - T^{W}(\tau)), \; 0 < \tau \leqslant \theta. \tag{6}$$

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