JOINT SOLUTION OF THE GAS TURBINE ROTOR AND STATOR TEMPERATURE PROBLEMS

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The thermal interaction of a rotor and stator is investigated on the basis of tests conducted on an experimental gas turbine. It is shown that in calculating the thermal states it is necessary to obtain a joint solution for these units of the turbine: in steady-state regimes in order correctly to take into account the temperatures of the media bathing the components outside the blading; in transient regimes in order to take into account radiative heat transfer and natural convection between the rotor and the casing.

An analysis of the extensive literature (for example, [1–3]) on the determination of the temperature distributions in gas turbine components shows that, as a rule, the temperature states of the rotor and stator groups are treated separately, without allowance for their interaction. Since this procedure is in some cases dubious and unjustified, we have made a special investigation of the problem, the results of which are briefly outlined below.

Obviously, in a hot turbine the rotor and stator are always in a state of thermal interaction. The heat exchange between them is realized by convection and radiation, whose intensity, other things being equal, is the greater, the higher the level and the greater the difference of the temperatures of the corresponding rotor and stator surfaces. Moreover, the interaction is expressed by the effect of the rotor and stator on the temperature of the media that bathe their surfaces. In fact, whereas the change of gas temperature in the blading is almost independent of the stator and rotor temperatures, in the labyrinth seals and end cavities the temperature of the medium will vary as a result of heat exchange with both surfaces. Since the sections of the rotor and stator forming the air channel usually have different temperatures, they have different and sometimes opposite effects on the temperature of the medium. So far, in investigating the temperature state of the rotor or stator this factor has usually been disregarded, and the heating of the medium in the channel determined on the assumption that the temperatures of the channel walls are equal.

In order to determine the effect of the stator on the temperature state of the rotor of a gas turbine we conducted a series of experiments on the OGT-850 experimental turbine at the Central Boiler-Turbine Institute (for details see [2]), a longitudinal section of which is shown in Fig. 1a. In the built-up disk rotor of the turbine we installed about 100 thermocouples, the signals from which were transmitted through a brush-type slip ring unit to EPP-09 potentiometric recorders. Thermocouples were also attached to parts of the casing in order to measure the temperatures of the metal and the gas and air media bathing the rotor and stator of the turbine. Furthermore, we measured the temperature, pressure, and flow rates of the gas and air at the turbine inlet.

In analyzing the data obtained we performed the temperature field calculations on a USM-1 analog computer.

The thermal state of the components was investigated both in steady-state operation and in transient load-shedding and shutdown regimes.

Steady-state regime. On the basis of experimental data on the values of the gas and air flow rates, pressures, and temperatures (at the labyrinth chamber inlet) obtained in one of the turbine's steady-state operating regimes we determined the heat transfer coefficients for various sections of the rotor. For this purpose we used the criterial relations recommended by a number of authors [1,4].

We then made two calculations of the rotor temperature field, the results of which are plotted in the form of curves along the rotor surface in Fig. 1b. In one of the calculations the heating of the air in the leading and trailing seals and the end cavities was determined only from the heat exchange with the rotor (dashed curve). In the other calculation we also took into account the heat exchange with the stator components (solid curve). It is clear from the figure that the rotor temperatures are in good agreement within the blading and on the sections near the turbine bearings. In the region of the seals small discrepancies are observed in the neighborhood of the air supply chambers, where in both cases the air temperatures were naturally the same.

In the region of the leading seals neglecting the additional heating of the air by the stator components reduced the rotor temperature by 30–50° C. At the trailing seals, which in the OGT-850 turbine the air reaches after passing through the turbine rotor, it is cooled. In this case taking into account the heat exchange with the casing, which is at a lower temperature than the rotor, causes more intense cooling of the air and hence a decrease in rotor temperature.

It should be noted that these deviations in the temperature state of the rotor considerably exceed the guaranteed accuracy of the steady-state temperature field calculations, which is usually estimated at 2–3% of the maximum temperature.

Thus, the joint solution of the problem for the rotor and the stator probably does not give much improvement as regards the investigation of the tem-

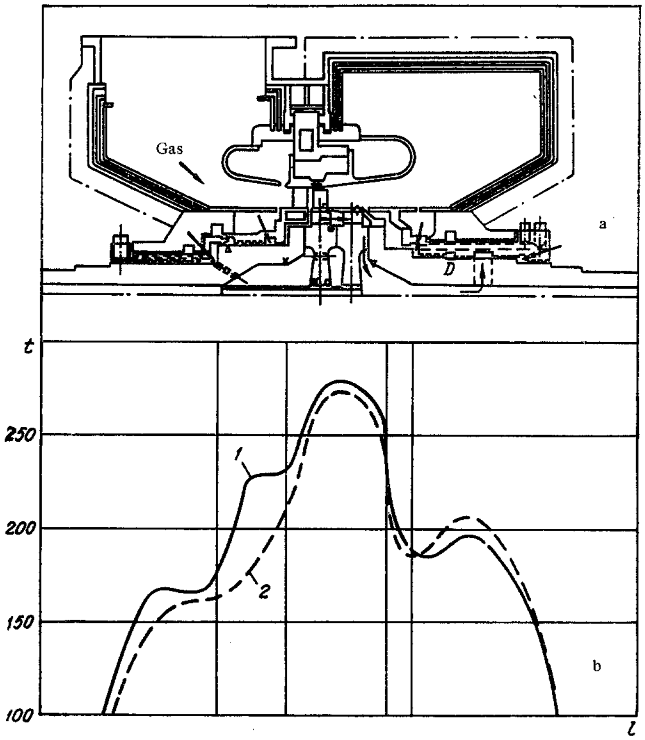

Fig. 1. a) The OGT-850 experimental turbine (the arrows indicate the points of admission of cooling air, the symbols points at which the metal temperature was measured) and b) the rotor surface temperatures in the steady-state regime (1 and 2—calculation with and without allowance for the casing).

peratures and temperature stresses in rotor disks. However, it is obvious that in determining the thermal displacements neglecting the interaction of the rotor and the stator through the air temperature may give considerable errors in the final results.

In investigating the steady temperature states we made two calculations, one of which took into account radiative heat exchange between the rotor and the casing while the other did not. The temperature fields obtained in both calculations were in good agreement with the experimental values of the temperatures and almost identical with each other. The greatest discrepancy was observed in the region of the blading and did not exceed 5° C. This indicates that in steady-state regimes with the gas turbine operating under load, heat exchange between the rotor and the casing does not have much effect on the temperature state of either the rotor or the casing.

It should be noted that the result obtained confirms the conclusion arrived at theoretically by other investigators [1,5].

Transient regimes. In the experimental investigation of nonstationary temperature regimes we concentrated our attention on the turbine cooling processes. We studied the temperature states of the rotor and stator in the presence of temperature drops at constant gas and air flow rates ($\alpha_c \simeq$ const), which for gas-turbine generators corresponds to the conditions that develop in the turbine when the load is reduced to the free-running regime. Moreover, we conducted experiments in which gas temperature drops were combined with reductions in the gas and air flow rates. These heat transfer conditions are characteristic of load-reduction regimes in propulsion gas turbines, and also of the shutdown of generator turbines after free running.

Figure 2 shows the variation of the principal parameters and metal temperatures in the rotor disk and casing in the region of the blading for two shutdowns

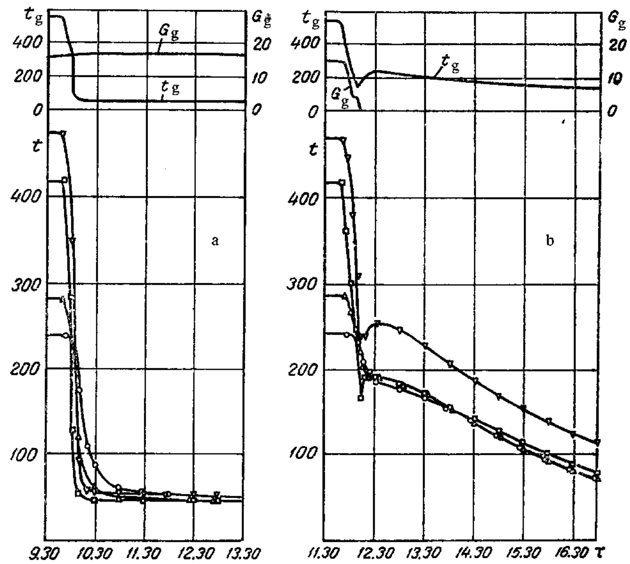

Fig. 2. Temperatures in the rotor and casing of the OGT-850 for shutdown at α_c = const (a) and α_c = var (b).

of the OGT-850 turbine. In these experiments the variation of the gas temperature ahead of the turbine $t_g = f(\tau)$ up to quenching of the flame in the combustion chamber was approximately the same, while the laws of variation of the flow rates (convective heat transfer coefficient) differed sharply. An analysis of the metal temperatures at various points on the rotor and stator showed that for each of these shutdowns the variation of the metal temperature $t = f(\tau)$ was approximately the same and chiefly determined by the way in which the convective heat transfer rate varied.

Thus, for example, in the case of shutdown with $\alpha_c \simeq$ const (Fig. 2a) the metal temperature curves fall smoothly from beginning to end, the rate of variation depending on the nature of the gas temperature drop and on the distance from the heat transfer surface (in this case the blading).

The curves showing the variation of the gas and metal temperatures for shutdown with α_c = var have sharp discontinuities and inflections (Fig. 2b). In this case the process as a whole can be divided into three stages. The first stage begins with the fall in gas temperature and continues until air ceases being supplied to the blading and cooling system, when forced-convection heat transfer remains at a high level and hence can exert an active influence on the temperature state of the turbine components by modifying the temperature of the medium (active shutdown). In the second stage following the end of intense convective heat transfer there is a redistribution of temperatures in the rotor (inactive shutdown). Heat is transmitted by conduction from the hotter to the cooler regions of the rotor, the tendency being to establish some average temperature level throughout the rotor. This is indicated by the gradual equalization of the thermocouple readings from different parts of the rotor. The existing heat loss from the rotor through the bearings and casing elements is then superimposed on the temperature redistribution due to heat conduction, as a result of which the thermocouple readings, continuing to approach each other over the entire region investigated, begin to fall smoothly. There then follows the third stage of shutdown, in which the turbine cools. The temperature difference between different points of the rotor is then so small that it does not exceed 20° C between the disk and the region of the labyrinth seals.

The behavior of the temperatures in the rotor and casing in the second and third stages of shutdown indicates the absence of intense convective heat transfer in the blading and cooling channels. Obviously, in addition to radiative heat transfer, free-convection heat transfer is established in the blading, in the labyrinth seals, and in the cooling channels after the supply of gas and air has ceased and the rotor has completely stopped. The relationship between the convective and radiant heat fluxes is different in the different stages of shutdown. As an example, Fig. 3 presents the results of a calculation of the specific heat flows into the blading in the region of the second stage for experimental shutdowns with α_c = const and

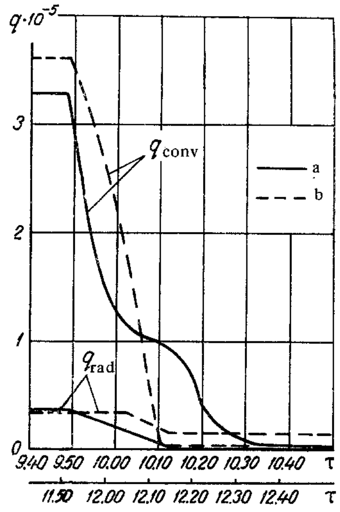

Fig. 3. Variation of specific heat flux in the blading of the OGT-850 turbine for shutdown with α_c = const (a) and α_c = var (b).

α_c = var. Strictly speaking, in Fig. 3 we have plotted the absolute values of the heat fluxes, since during shutdown temperature regimes may occur in which the heat flux changes direction. A graphical interpretation enables us to follow distinctly how during shutdown with α_c = const the convective heat transfer, starting from the steady state, always occupies a dominant position, although at the end of the cooling process both the convective and radiant heat fluxes probably tend to zero, and the rotor to a new stable steady state. The q_{rad} and q_{conv} curves in Fig. 3 to some extent explain the previously established and now confirmed weak effect of radiative heat transfer in the steady-state regime and show that it can probably also be neglected in the load-reduction regime or in the case of shutdown with $\alpha_c \simeq$ const. In the first stage of shutdown with α_c = var the picture is largely repeated, but following the end of the first stage, i.e., after transition from forced to natural convection, radiative heat transfer moves into the foreground. Even though its intensity is relatively low, radiative heat transfer between the rotor and the stator is obviously one of the principal factors determining the temperature state during cooling. The third stage of shutdown with α_c = var is nothing other than the regular thermal regime.

The table presents the results of calculations, based on the readings of various thermocouples, of the number m,

$$m = \frac{\ln \vartheta_1 - \ln \vartheta_2}{\tau_2 - \tau_1},$$

where $\vartheta = t - t_a$, and τ is time.

It is clear from the table that the number m is approximately constant in time and varies little at different points on the rotor from thermocouple to thermocouple in the case of shutdown with α_c = var. However, in the case of shutdown with $\alpha_c \simeq$ const, Table

The Number m for Different Shutdown Conditions

Measuring point	Shutdown with α = const			Shutdown with α = var		
	τ_1	τ_2	m	τ_1	τ_2	m
D	10 hr 12min	10 hr 15min	4.45	14 hr 20min	14 hr 30min	0.423
	10.22	10.29	5.52	14.40	14.50	0.543
	10.32	10.37	4.92	15.10	15.20	0.524
△	10.13	10.18	5.49	14.30	14.40	0.524
	10.18	10.23	10.07	14.50	15.00	0.422
	10.23	10.28	13.18	15.10	15.20	0.483
□	10.04	10.07	8.6	14.30	14.40	0.443
	10.07	10.12	5.88	15.00	15.10	0.422
	10.17	10.22	6.12	15.20	15.30	0.402
+	10.04	10.08	11.03	14.20	14.30	0.302
	10.08	10.13	11.55	14.40	14.50	0.363
	10.13	10.18	13.18	15.00	15.10	0.363
×	10.12	10.17	6.84	14.30	14.40	0.432
	10.17	10.24	6.0	14.50	15.00	0.463
	10.27	10.32	7.27	15.10	15.20	0.484
▽	10.12	10.17	3.84	14.40	14.50	0.362
	10.22	10.27	3.36	15.00	15.10	0.542
	10.32	10.37	3.92	15.20	15.30	0.482
∅	10.18	10.23	1.2	14.20	14.30	0.362
	10.28	10.33	4.8	14.40	14.50	0.362
	10.38	10.43	4.76	15.00	15.10	0.422

m is not constant and there is no basis for assuming the existence of a regular regime on the time interval investigated.

The results of the experiments described above show convincingly that the broken-line character of the variation of metal temperature during shutdown with α_c = var is a result of the sharp fall in the rate of convective heat transfer. Consequently, in constructing a calculation model in order to obtain reliable results it is necessary that the system of boundary conditions be capable of reflecting these sharp changes in the heat transfer coefficients during the process of active shutdown and realizing the transition to heat transfer by radiation and natural convection from the beginning of the inactive stage. A system of boundary conditions satisfying these requirements was incorporated in an analog calculation of the shutdown temperature states of the OGT-850 turbine rotor on an RC network.

The results of the calculation for several points on the rotor are represented by the dot-dash line in Fig. 4. Clearly, over the entire duration of shutdown with α_c = var the calculations are in perfectly satisfactory agreement with the experimental results (solid curves). The maximum deviation does exceed 10° C at a total temperature level of about 300° C. For comparison, the figure includes the results of a calculation of the same shutdown, but for boundary conditions in which the heat transfer coefficients remained constant at the level of the steady-state regime, whereas the temperatures of the media varied in accordance with the experimental data (dashed curves). The difference between the results of this calculation and the experimental values is much greater and clearly apparent during all stages of shutdown.

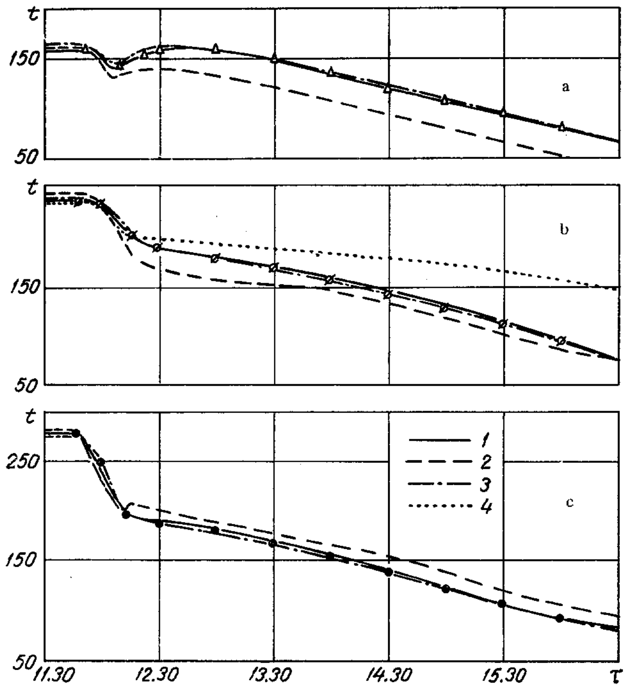

Fig. 4. Experimental and calculated temperatures in the rotor of the OGT-850 turbine during shutdown with subsequent cooling of the leading labyrinth seal (a), the center of the first stage disk (b), and the periphery of the first stage disk (c): 1) experiment with α_c = const; 2) calculation with α_c = const; 3) calculation with α_c = var and allowance for natural convection and radiative heat transfer during cooling; 4) calculation with α_c = var and without allowance for radiative heat transfer during cooling.

The effect of radiative heat exchange with the casing on the temperature state of the rotor during shutdown with α_c = var was checked in a calculation in which the system of boundary conditions was retained as far as convective heat transfer was concerned, but radiative heat transfer was not taken into account. It is clear from Fig. 4b that the results of the calculation are in good agreement with the conclusions arrived at above: radiative heat transfer has an important influence on the nonstationary temperature field of the rotor during shutdown with $\alpha_c = $ = var only after intense convective heat transfer in the blading and cooling channels has ceased.

These results permit the following conclusions:

1. In calculating the steady and transient temperature states of gas turbine rotors and stators, to assign the boundary conditions correctly it is necessary to make a joint calculation of the rotor-stator system.

2. In calculating the transient temperature states of the rotor and casing during shutdown with heat transfer coefficients that vary in time, the system of boundary conditions should take into account radiative heat transfer throughout the shutdown period and the variability of the heat transfer coefficient during the active stage of shutdown up to the end of rotor runout, and also take into account the transition from forced-convection heat transfer to natural-convection heat transfer at the beginning of the cooling process.

3. In view of the existing temperature levels and temperature differences characteristic of the rotor and stator, in investigating the steady-state temperature fields and also the transient temperature states of these turbine units associated with load reduction at constant heat transfer coefficients, radiative heat transfer can be neglected.

NOTATION

t is the metal temperature, °C; t_g is the gas temperature ahead of the turbine, °C; t_a is the temperature of the cooling air, °C; G_g is the flow rate of the gas through the turbine, t/hr; α_c is the forced-convection heat transfer coefficient, $W/m^2 \cdot deg$; q_{rad} is the specific radiant heat flux, W/m^2; q_{conv} is the specific natural-convectional heat flux, W/m^2; τ is the time, hr; l is the linear dimension, m.

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