HEAT EXCHANGE AND HYDRAULIC RESISTANCE IN RADIAL ROTATING CHANNELS

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We present the results of an experimental investigation into the effect exerted by rotation on the exchange of heat and on hydraulic resistance in the case of centrifugal and centripetal flows of a cooling medium in channels of circular cross section.

As is well known, in radial rotating channels both the Coriolis force and centripetal acceleration can affect heat exchange [1-3]. Under conditions of nonisothermal flow the centripetal acceleration may result in the appearance of thermal conduction. As the medium flows in the direction of the centrifugal force, a convection current is established in another direction, and under certain conditions this convection current is capable of intensifying the exchange of heat. Simultaneously with the increase in heat-exchange intensity, we find an increase in hydraulic resistance. Therefore, it is the goal of the present study to undertake an experimental investigation into the influence exerted by the fields of mass forces on heat exchange and resistance in the cooling channels of rotor blades insofar as this pertains to the values of decisive criteria that are close to the values for realistic operational conditions of gas turbines.

The experiments were carried out on an installation whose basic diagram is shown in Fig. 1. The fundamental operating element is a Duraloy disk 1 through which 48 radial channels 2 have been drilled, these channels 6 mm in diameter and 125 mm in length. The disk is attached to the shaft by means of a bracket and set into rotation by means of a pressure booster from a direct-current motor. The frequency of disk rotation is varied in the range from 2000-5500 rpm. Air is used as the working fluid. To produce the nonisothermal conditions of flow six electrical heaters 3 have been mounted in the disk, these heaters each exhibiting a power of 500 W. Moreover, to reduce heat losses provision has been made for compensation heaters 4, one on each side surface. Electric power is supplied to the heaters by means of a current collector, and the required power is regulated by means of laboratory autotransformers.

To prevent leakage of the supplied air into the structure of the installation, provision has been made for a system of supercharged labyrinth seals 5 and 6. With this purpose in mind, special chambers 7 and 8 have been fashioned in the labyrinth seals, through which air was supplied under an excess pressure, with the pressure drop across the outermost shields being measured simultaneously. A number of parameters were measured during the course of these experiments, said parameters characterizing the exchange of heat and the hydraulic resistance through the rotating channels. Among these we include the pressure and temperature of the air in the cavities at the inlet to and the outlet from the rotating channels, as well as the rate of air flow through the installation, the frequency of rotor rotation, the electric heater power, and finally the temperature of the wall and the temperature of the air at five cross sections along the length of the channel.

This experimental study included an examination of the effect of rotation in the case of both centrifugal and centripetal flows. In the case of centrifugal flow the air is supplied to cavity 9 and it is then directed to the rotating channels which are absolutely straight. In the case of centripetal flow the air is supplied to cavity 10 and from there, through inclined channels 11 which impart velocity to the air in the direction of rotation, the air is directed to cavity 12.

The experimental method involved the carrying out of tests, initially under static conditions with and without heating, and then identical tests under conditions of rotation. In view of the fact that the purpose of these experiments is to study the effect of mass

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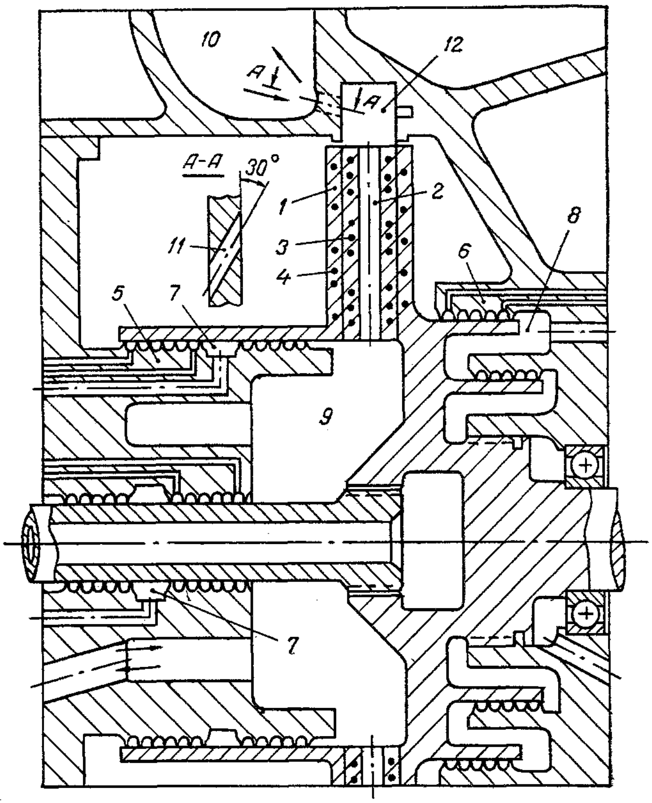

Fig. 1. Operational portion of the experimental installation.

forces on heat exchange and hydraulic resistance in rotor cooling systems with low pressure drops, the region of Reynolds number Re regimes was limited by values of $6 \cdot 10^3 \le Re \le 18 \cdot 10^3$, while in terms of the Grashof numbers Gr, by $10^6 \le Gr \le 3 \cdot 10^6$, respectively. These intervals of Re and Gr numbers were maintained with a change in the air flow rate G_a through rotating channels in limits of $30 \le G_a \le 80$ g/sec, with rotor rotation frequencies of $3500 \le n \le 5000$ rpm, and channel wall temperatures no higher than 90°C. This temperature level is achieved with a power on the order of 350 W from each of the heaters.

The method of processing the experimental data involved the following: using the measured flow rate of the air, the temperature at the inlet to and the outlet from the channels, and the rotational frequency, we determined the amount of heat supplied:

$$Q = G_a C_p (t_1 - t_0 \mp \Delta t_{rot}), \tag{1}$$

where

$$\Delta t_{rot} = (C_{u1} u_1 - C_{u0} u_0)/C_p.$$

We then calculated the average value of the heat-transfer coefficient:

$$\alpha = \frac{Q}{Z F_{ch} \Delta t_m}. \tag{2}$$

Subsequent to this we calculated the Nusselt, Reynolds, Grashof, Prandtl, and Rossby numbers:

$$\mathrm{Nu} = \frac{\alpha d_{ch}}{\lambda}, \quad \mathrm{Re} = \frac{4 G_a}{Z \pi d_{ch} \mu}, \quad \mathrm{Gr} = \frac{\omega^2 r_{av} d_{ch}^3 \rho_0 \left(1 - \frac{T_a}{T_w}\right)}{\mu^2}, \tag{3}$$
$$\mathrm{Pr} = \frac{\mu C_p}{\lambda}, \quad \mathrm{Ro} = \frac{w d_{ch}}{u_m}.$$

From the results of the measured pressure in front of the inlet and at the outlet from the channels, as well as from the measured air flow rate and the rotational velocity, we

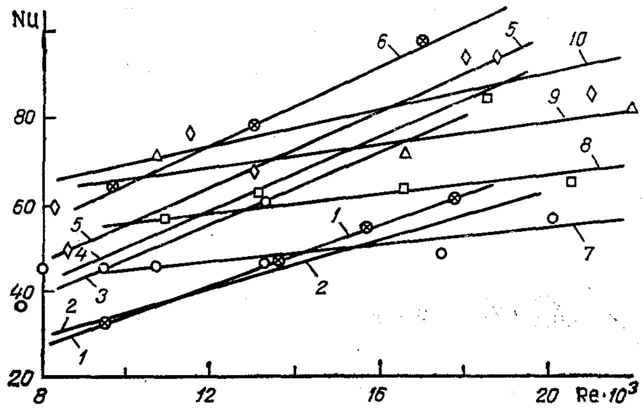

Fig. 2. Release of heat in centrifugal and centripetal directions of the flow: 1-5) centrifugal flow; 6-10) centripetal flow; 1, 6) n = 0 rpm; 2) on the basis of data from [5]; 3, 7) n = 3500 rpm, Ro = $(2.2\text{-}3.4)\cdot 10^{-3}$; 4, 8) 4500 rpm and $(1.6\text{-}2.8)\cdot 10^{-3}$; 9) 5000 and $(1.4\text{-}2.5)\cdot 10^{-3}$; 5, 10) 5500 and $(1.3\text{-}3.3)\cdot 10^{-3}$.

calculated the equivalent resistance factor ξ_r. For this purpose we used a numerical method to integrate the equations of motion in a channel of constant cross section, which was written in the form

$$d(p+\rho w^2) = -\xi_r \frac{\rho w^2}{2}\frac{dl}{d_{ch}} \pm \rho\omega^2 r\,dl. \tag{4}$$

On the basis of the given total momentum at the inlet section of the channel

$$(p+\rho w^2)_0 = \frac{G_a}{ZF} z(\lambda_0)\sqrt{2\,\frac{k+1}{k} RT_0^*} \tag{5}$$

we calculated the pressure at the outlet, which was compared to the value of p_1 given from the experiment. Since

$$T_0 = T^*\left(1 - \frac{k-1}{k+1}\lambda_0^2\right), \tag{6}$$

on the basis of the flow-rate equation we have

$$\lambda_0 = \sqrt{\left(\frac{k+1}{k-1}\,\frac{p_0 ZF 18{,}3}{2G_a R\sqrt{T_0^*}}\right)^2 \frac{k+1}{k-1} - \left(\frac{k+1}{k-1}\,\frac{p_0 ZF 18{,}3}{G_a R\sqrt{T_0^*}}\right)}. \tag{7}$$

The calculation algorithm was constructed so that on deviation of the calculated pressure value at the outlet from the channels in any given direction by a magnitude of greater accuracy than earlier specified we calculated a correction factor for the resistance coefficient ξ_r, and its initial value was thus calculated on the basis of the following expressions:

$$\xi_a = 0{,}3164\,\mathrm{Re}^{-0.25} \quad \text{when} \quad \mathrm{Re} \geqslant 2300 \tag{8}$$

and

$$\xi_r = \frac{64}{\mathrm{Re}} \quad \text{when} \quad \mathrm{Re} < 2300.$$

The described method of processing the experimental data enabled us to determine the true values of the coefficients of friction by carrying out two series of tests: with heating and without heating of the channels. Since the total losses are made up of the initial losses, the losses due to friction, and the outlet velocity, for kinematically similar regimes the increase in the effective resistance must be associated only with the elevated friction in the channels

$$\xi_{r.h} - \xi_{r.c} = \xi_h - \xi_0, \tag{9}$$

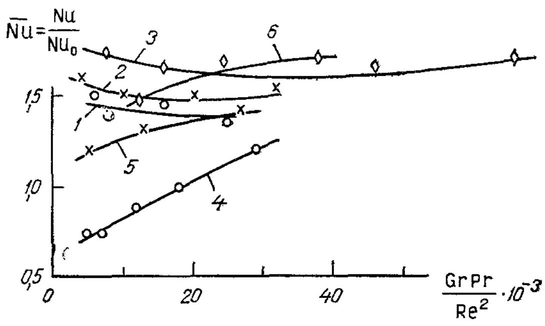

Fig. 3. Relative change in the intensity of heat release in radial rotating channels, as a function of the $GrPr/Re^2$ complex: 1-3) centrifugal flow; 4-6) centripetal flow; 1, 4) n = 3500 rpm, Ro = $(2.2\text{-}3.4)\cdot 10^{-3}$; 2, 5) 4000 and $(1.9\text{-}3.2)\cdot 10^{-3}$; 3, 6) 5500 and $(1.3\text{-}2.2)\cdot 10^{-3}$.

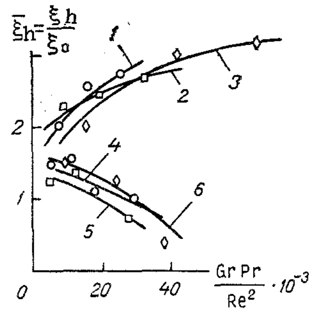

Fig. 4. Relative change in resistance in the channels, as a function of the $GrPr/Re^2$ complex: 1-3) centrifugal flow; 4-6) centripetal flow; 1, 4) n = 3500 rpm, Ro = $(2.2\text{-}3.4)\cdot 10^{-3}$; 2, 5) 4500 and $(1.6\text{-}2.8)\cdot 10^{-3}$; 3, 6) 5500 and $(1.3\text{-}2.2)\cdot 10^{-3}$.

where $\xi_0 = 0.023\sqrt{\omega d_{ch}/w} + 0.302\ Re^{-0.25}$ represents the resistance of the rotating isothermal channels [4]. Hence the relative increase in the coefficient of friction can be defined as

$$\frac{\xi_h}{\xi_0} = 1 + \frac{\xi_{r.h} - \xi_{r.c}}{\xi_0}. \tag{10}$$

To verify the validity of these measurements, we initially carried out test experiments on a fully stopped rotor, with and without heating. The data on heat exchange from these test experiments were compared with the results which are obtained on the basis of the following criterial relationship [5]:

$$Nu = 0{,}021\,Re^{0,8}Pr^{0,43}\varepsilon_l\left(\frac{Pr_f}{Pr_r}\right)^{0,25}. \tag{11}$$

Here ε_ℓ is a coefficient which takes into consideration the effect of the initial segment on the increase on the intensity of heat exchange (in our case for $\ell/d_{ch} = 21$, $\varepsilon_\ell = 1.13$). Comparison of the test-experiment results demonstrated that as the air flows in the direction from the axis of rotating to the periphery (centrifugal flow) the divergence from relationship (11) amounts to no more than +15 and −8% (Fig. 2). However, in the opposite (centripetal) direction of flow the intensity of the heat exchange is approximately greater by a factor of 1.9 than indicated by the data derived from relationship (11). Such a divergence, apparently, is associated with the various conditions prevailing at the inlet in radial channels. Thus, if in the first case the air enters the channels out of cavity 9, the latter being totally straight, then in the second case the air in cavity 12 exhibited a circular velocity component.

The strong influence exerted on the exchange of heat by the conditions prevailing at the inlet is confirmed by numerous studies. In particular, it has been demonstrated in [6] that if a 90° angle elbow is located at the inlet to the tube, the average value of the heat-transfer coefficient for the tube with ℓ/d_{ch} = 20 increases by a factor of 1.5-1.6. In the case under consideration, we observed a slightly greater increase in the intensity of heat exchange, which is fully explainable by the described condition at the inlet.

After the test experiments have been carried out, we carried out experiments dealing with centrifugal flow in rotating channels. The heat-exchange data expressed in Nu = f(Re) coordinates for various rotational velocities, including n = 0, are shown in Fig. 2. As we can see, the intensity of the heat exchange in the rotating channel in the case of centrifugal flow increases significantly. Thus, a change in the rotational frequency from n = 0 to n = 5500 rpm is accompanied by an increase in the heat-exchange intensity by a factor of 1.5-1.7, with the increase in the Re numbers in the case of an unchanged rotational frequency, leading to a reduction in the transfer of heat. A similar nature in the change of heat-exchange intensity has been observed in [7, 8].

One of the factors responsible for the intensification of the heat exchange in rotating channels may be associated with mass forces. With this purpose in mind, we processed the derived data in the form of $Nu = f(GrPr/Re^2)$ and these are shown in Fig. 3. During such processing we carefully monitor the influence of the $GrPr/Re^2$ complex on the relative change in the transfer of heat in these rotating channels.

The effect of the rotation on the change in heat-transfer intensity in the centrifugal direction of medium flow has repeatedly been observed and cited in the literature [4]. However, in certain papers reference is made to an observed increase [2, 8], and to a reduction in heat-exchange intensity [1, 9]. In this connection we should take note of the fact that both of these effects can occur and they can be explained by the unique features of current interaction, which leads to the appearance of variously directed convection currents.

An extremely accurate analog of the phenomena occurring within rotating channels is represented by the nonisothermal flow that occurs in vertical tubes. We know that with the appearance of variously directed convection currents in vertical tubes the intensity of the heat exchange may either increase or decrease [10]. The increase in heat-exchange intensity that is observed in experiments with rotating channels, however, was intensified by the fact that the inlet to the channel is fabricated so as to form a bend, and in this case the relative circumferential velocity of the air at the inlet increases with an increase in the rotational velocity.

In the experiments which we have carried out, in accordance with the increase in the intensity of the heat exchange, we also noted an increase in the channel resistance. The results of the processing of the experimental data are shown in Fig. 4 in the form of a relative change in the resistance, expressed in fractions of the resistance of rotating isothermal channels, as a function of the $GrPr/Re^2$ complex. As we can see, with an increase in this complex we have a significant increase in the resistance.

Analogous experimental studies were also undertaken with respect to centripetal flow. A unique feature of this case, as was noted earlier, is the tangential inflow of air, which affected the magnitude of the heat transfer for the case in which r = 0. With centifugal flow, in accordance with the change in the conditions at the inlet, as well as because of the specific influence of the mass forces, which led to the appearance of satellite convection currents, we observed a reduction in the intensity of heat transfer. The processing of the experimental data in the form of Nu = f(Re) is also shown in Fig. 2, from which it follows that in the case of centripetal flow the transfer of heat on the whole turns out to be substantially smaller than the amount of heat transferred in the case of a decelerated rotor.

The processing of experimental data on hydraulic resistance by a method analogous to that of centrifugal flow demonstrated that in the case of centripetal flow the range of variations in the relative resistance of rotating channels becomes significant even, and this is most important, as the increase in the $GrPr/Re^2$ complex leads to the observation of a reduction in the quantity $\bar{\xi}_h = \xi_h/\xi_c$ (Fig. 4). With values of $GrPr/Re^3 > 0.02$ the action of the mass forces causes the quantity $\bar{\xi}_h$ to become smaller than 1. This confirms once again that the fact observed in [11] of a reduction in hydraulic resistance in rotating radial channels in the case of centripetal flow was also governed by the influence of buoyancy forces.

NOTATION

Q, quantity of heat directed to the air, kJ/sec; G_a, mass flow rate of the air, kg/sec; C_p, heat capacity of the air at constant pressure, kJ/kg; t_0, t_1, temperature of the air at the inlet to and the outlet from the channels, °C; Δt_{rot}, change in the temperature of the air as a consequence of application or removal of energy in the rotating radial channels, °C; u_0, u_1, u_m, circumferential velocity at the inlet to, the outlet from, and in the midsections of the channels, m/sec; C_{u0}, C_{u1}, circumferential component of the air velocity at the inlet to and the outlet from the channels, m/sec; Z, number of radial channels in the rotor; F_{ch}, channel surface, m^2; d_{ch}, ℓ, diameter and length of the channel, respectively, m; Δt_m, mean temperature accumulation through the length of the channel, °C; α, coefficient of heat transfer to the air, $W/(m^2 \cdot K)$; λ, coefficient of thermal conductivity for the air, $W/(m \cdot K)$; μ, coefficient of dynamic viscosity for the air, $N \cdot sec/m^3$; ρ_0, ρ, average and instantaneous value of the air density, kg/m^3; ω, angular rotational frequency of the rotor, 1/sec; r_m, distance from the rotational axis to the midsection of the channel, m; r, r_0, instantaneous value of the radius and the radius of the inlet section of the channel, m; Pr, Prandtl number; Re, Reynolds numbr; Nu, Nusselt number; Gr, Grashof number; Ro, Rossby number; p, instantaneous value of pressure in the channel, Pa; w, w_0, instantaneous value of velocity and the value of the velocity at the inlet to the channels, m/sec; T_a, T_w, average temperature values for the air and the channel walls in length, K; T_0^*, total air temperature at the inlet, K; R, universal gas constant, $kJ/(kg \cdot K)$; k, adiabatic exponent; $z(\lambda)$, gasdynamic function; a_{cr}, critical speed of sound, m/sec; $\xi_{r.c}$, $\xi_{r.h}$, equivalent coefficients of resistance for cold and hot channels, respectively; ξ_0, ξ_h, coefficients of losses due to friction in cold and hot channels, respectively; ε_ℓ, coefficient taking into consideration the effect of the initial segments; $F = \Pi d_{ch}^2/4$.

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