OPTIMIZATION OF JET COOLING OF A ROTATING DISK

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The author presents empirical formulas for optimizing a jet system for cooling of a disk rotating in a housing, in terms of the heat transfer coefficient.

The method of cooling using air jets discharging from nozzles finds wide use in practice. In this case the maximum use of the cold potential of the cooler can be achieved by appropriate arrangement of the blowing jet. For example, the heat transfer for a disk rotating in a housing with jet blowing is a complex and strong function of the angle β at which the air nozzles are set [1].

It is evident that optimization by setting the air nozzles at the optimal angle β_{opt} giving the maximum heat transfer is one of the simplest ways to increase the efficiency of the jet system for cooling rotating disks. Therefore, the investigations of [1] were continued in the present work in order to obtain general correlations for calculating the value of β_{opt}.

The process of heat transfer to a disk rotating in a housing is accompanied by complex hydrodynamic phenomena in the end gap between the disk and the housing.

The action on the jet of the transverse induced flow due to rotation of the disk, the presence of axial and radial pressure gradients in the end gap, the interaction of the jet in the gap and on the disk surface, and also the extremely complex law for the formation of the boundary layer on the disk surface, in both the circumferential and radial directions in the flow of gradient character, are all factors which influence the heat transfer in the jet blowing zone.

It was shown from dimensional analysis in [1] that the heat transfer with jet blowing on a disk rotating in a housing is determined by six parameters: Pe, X, $\bar{d}$, $\bar{R}$, z, β. The laws for the combined influence of these parameters on the heat transfer can be found most reliably by experiment. The results of such an experiment, presented in the form of general empirical formulas for calculating the heat transfer from the disk in jet blowing, have been given in [1], which also suggested a technique for the correlation.

Below by a similar method we have obtained empirical formulas for calculating β_{opt}. For $Pe \leqslant 2.8 \cdot 10^4$:

$$\beta_{opt} = [2{,}23 - 0{,}12\,(1 - aX)\,(z-1)^b\,][50\,(\bar{d}_2 - \bar{d})\,(\varepsilon - 1) + 1], \tag{1}$$

where

$$a = 0{,}58\,[1 - 0{,}144\,(\mathrm{Pe} - \mathrm{Pe}_1)^{0{,}2}];$$

$$b = 0.55 - 0{,}0187\,(\mathrm{Pe} - \mathrm{Pe}_1)^{0{,}2};\ \ \varepsilon = B + (B_1 - B)\,\frac{z-1}{z_3 - 2};$$

$$B = 1.048 - 0.023\mathrm{Pe}\cdot 10^{-4} + \frac{0{,}74 - \mathrm{Pe}\cdot 10^{-4} + 0{,}278\mathrm{Pe}^2\cdot 10^{-8}}{\mathrm{Pe}\cdot 10^{-4}}X;$$

$$B_1 = 0{,}38 + 0.17\mathrm{Pe}\cdot 10^{-4} + (2.0 - 0{,}54\mathrm{Pe}\cdot 10^{-4})\,X - (1{,}08 - 0.32\mathrm{Pe}\cdot 10^{-4})\,X^2.$$

The constants are: $z_3 = 36$; $Pe_1 = 1.2 \cdot 10^4$. For $Pe \geqslant 2.8 \cdot 10^4$

$$\beta_{opt} = [1.047 + (0.0756 - 0{,}0271X)\,(z-1)^{0{,}528 - 0{,}0458X}]\,\tau, \tag{2}$$

where

$$\tau = 3.35\,(\bar{d}_2 - \bar{d})\,(1 - 5.6X + 3.1X^2) + 1.$$

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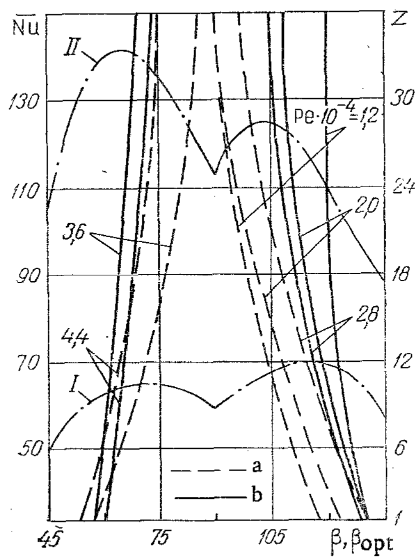

Fig. 1. The quantity β_{opt} (deg) as a function of the number of nozzles z for $\bar{d}$ = 0.04: a) X = 0.5; b) 1.6; I, II) $\overline{Nu}$ as a function of β for Pe = $2.8 \cdot 10^4$ and Pe = $5.5 \cdot 10^4$ (X = 0.5; d = 0.04, z = 9).

Equations (1), (2) are valid in the range of change of the parameters: Pe = $(1.2 - 5.5) \cdot 10^4$; X = 0.15 – 1.6; z = 1 – 36; $\bar{d}$ = 0.02 – 0.04. It can be seen that β_{opt} depends on all the quantities listed.

The results of calculating the quantities β_{opt} using Eqs. (1) and (2), shown in Fig. 1, show that β_{opt} is varied over a wide range as a function of the values of the parameters. For a more profound analysis of the heat transfer laws, Fig. 1 also gives the results of calculating $\overline{Nu}$ using the formulas of [1] and varying the values of β. The process of heat transfer to a disk rotating in a housing with jet blowing at an arbitrary angle, in contrast to an analogous case of blowing of a single jet over a stationary plate, investigated in [2, 3], is complicated by the influence of rotation and interaction of the jets. At a low disk rotation frequency the energy level of the induced flow is low, and its influence on the nature of the jet flow cannot appreciably change the laws for heat transfer in the jet blowing region.

For a high frequency of rotation the hydraulic processes in the end gap are characterized by strong interaction of the jets and the induced flow. In this case the heat transfer depends appreciably on the frequency of rotation of the disk.

For a qualitative analysis of the influence of the induced flow on the nature of the jet flow in the end gap, Fig. 2 shows a picture of the distribution of jets in the circumferential direction as a function of X and β.

The jet trajectories were calculated according to the recommendations of [4]. It was assumed that the circumferential velocity of the induced flow is constant over the width of the end gap, and is equal to 0.42U at radius R_0. It can be seen that at low frequency of rotation (X = 0.5) the action of the transverse induced flow is weak, and the absolute angle for flow of the jet over the disk surface differs very little from the angle at which the jets are set. In this case we have heat-transfer conditions determined by the jet blowing.

The latter point is also confirmed by test data. Analysis of Fig. 1 shows that for X = 0.5 and a small number of nozzles, when the interaction between the jets is weak, the quantity β_{opt} differs from β = 90°, and the dependence of $\overline{Nu}$ on β is qualitatively the same as was obtained in [2, 3] on a fixed plate.

In the work cited the maximum heat transfer was observed at angles β different from β = 90°. The reason for intensified heat transfer at a deviation of β from β = 90° in this case is an increase in the degree of turbulence in the flow because of breakdown of flow symmetry at the stagnation point [2]. The influence of the rotation in this regime appears mainly in

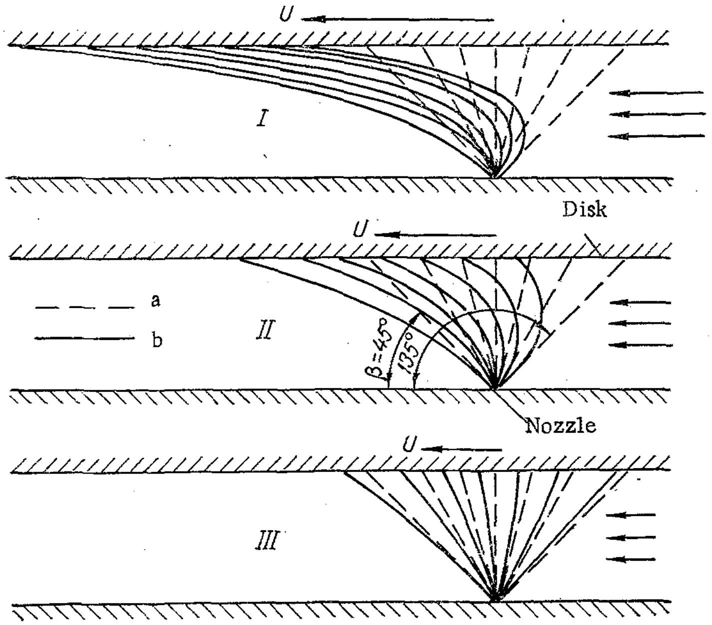

Fig. 2. Qualitative picture of the propagation of a jet in the end gap for different frequencies of disk rotation and nozzle angles β; a) X = 0; b) X > 0; I) X = 1.5; II) 1.0; III) 0.5.

a breakdown of the symmetry of the dependence of $\overline{Nu}$ on β (see Fig. 1) relative to β = 90°. With increase of z the heat transfer laws obtained in [2, 3] are not confirmed. In this case, the heat transfer begins to be influenced by interaction of the jet in the end gap and on the surface of the rotating disk.

It can be seen from Fig. 1 that the value of β_{opt} tends to β = 90° with increase of z.

An increase of the disk rotation frequency is characterized by a strong deviation of the jet in the direction of rotation (see Fig. 2), and in the limiting case with X = 1.5 the absolute angle of jet impingement on the disk surface is practically independent of the nozzle angle β, i.e., with increase of X we reach a heat-transfer regime governed by rotation. The heat transfer law in this regime is influenced by the complex hydraulic processes occurring in the end gap as a result of interaction of the jet and the induced flow.

The heat-transfer intensity on the disk surface in the jet blowing zone depends appreciably on the value of β. This is due to the strong influence of β on factors such as the kinetic energy of the jet at the disk surface, the relative velocity of the medium, and the radial pressure gradient in the end gap.

Analysis of the relations in Fig. 1 shows that with increase of the parameter X there is an increase in the deviation of the quantity β_{opt} from β = 90°. The intensification of heat transfer in the transition from β = 90° to β > 90° is determined mainly by deceleration of the circumferential component of the induced flow due to more intense exchange of kinetic energy between the jets and the induced flows. This leads to an increase of the heat transfer from the disk due to an increase of the relative velocity of the medium. However, the kinetic energy of the jet at the disk surface decreases, since the attenuation of kinetic energy in the jets blown into the induced flow increases with increase of β [4].

On the other hand, the transition from β = 90° to β < 90° promotes swirling of the induced flow and a decrease in the relative velocity of the medium. However, the kinetic energy of the jet at the disk surface increases, and this is the main cause of intensified heat transfer in this transition.

Thus, in the regime governed by rotation, the heat transfer is due mainly to two factors: the relative velocity of the medium and the kinetic energy of the jet at the disk surface. An increase of the relative influence of one of these factors in the dependence on the value of β leads to an increase of heat transfer in the transition from β = 90° to β ≠ 90°.

The limiting angles studied in the tests, $\beta = 45, 135°$, are characterized by a sharp fall in heat-transfer intensity. This is due to the fact that as a result of the strong deviation and attenuation of the jet by the induced flow, the jet blowing practically reduces to ventilation of the end gap by air jets.

With increase of Pe (of coolant flow rate) the maximum of heat transfer moves into the zone $\beta < 90°$ (see Fig. 1). This occurs because of an increase of the relative influence of the kinetic energy level of the air jets at the disk surface in the transition from $\beta > 90°$ to $\beta < 90°$ and the increase of the positive radial pressure gradient in the end gap. Tests have shown that with increase of Pe there is an increase of the radial pressure gradient for $\beta < 90°$. The opposite dependence is observed for $\beta > 90°$.

An increase of the positive radial pressure gradient in the flow going out to the disk perimeter promotes turbulence formation in the boundary layer and increased heat transfer.

NOTATION

d, nozzle diameter; R, R_0, ambient radius at the end surface of the disk, and jet inflow radius; z, number of nozzles; U, circumferential velocity at radius R_0; C, velocity of discharge of air from the nozzles; β, angle of the nozzles, formed by the velocity vectors C and U ($\beta = 0$ when the directions of the vectors coincide, and $\beta = 180°$ when the vectors are oppositely directed). Relative quantities are: $X = U/C$; $\bar{d} = d/R_0$; $\bar{R} = R/R_0$. Coefficients are: $\bar{\alpha}$, heat transfer averaged in the jet blowing zone; a, thermal diffusivity; λ, thermal conductivity. Similarity numbers: $Pe = Cd/a$; $\overline{Nu} = \bar{\alpha}d/\lambda$.

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TURBULENT TWO-PHASE JET AND ITS NUMERICAL INVESTIGATION

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Using a single-parameter model as a basis, investigators conducted a numerical study of a jet with heavy particles and compared the results with experimental data.

Turbulent jets with heavy particles, characterized by significant nonequilibrium of the flow, have recently become the subject of numerous theoretical [1-5] and experimental [6, 7] studies.

The initial system of equations describing such jet flow can be obtained within the framework of a model of interpenetrating and interacting continua [8] and should be closed with allowance for the effect of the particles on the turbulent structure of the jet. In [1, 4] this closure was effected within the framework of "mixing length" theory in accordance with the recommendations in [2]. Here, in [2, 4], an integral method was used to calculate the two-phase jet. In [1, 5] the finite difference method was used. It should be noted that the Prandtl formulas were replaced in [5] by the eddy viscosity transfer equation of a "pure" gas.

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