

## FLOW STRUCTURE AND HEAT TRANSFER BETWEEN A SWIRLED IMPINGING JET AND DIMPLE

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### ABSTRACT

Experimental results on investigation of turbulent flow structure and heat transfer at interaction of an air impact jet with a semi-spherical cavity are presented in this work. This research is continuation of investigations of turbulent jet interaction with complex surfaces and search for the method of heat transfer control. Experiments were carried out with fixed geometry of a semi-spherical cavity ( $D_c = 46$  mm), nozzle diameter ( $d_0 = 8.5$  mm) and swirl parameter ( $R = 0; 0.58; 1.0; 2.74$ ). The distance between the axisymmetric nozzle and obstacle was  $2 \div 10$  sizes over the nozzle diameter, and the Reynolds number varied within  $Re_0 = (1 \div 6) \cdot 10^4$ . It was found out that with an increase in swirling heat transfer intensity decreases because of fast mixing of the jet with ambient medium. In general, the pattern of swirl jet interaction with a concave surface is complex and multifactor.

### INTRODUCTION

Jets impinging onto heat-exchanging surfaces can be used in many technical apparatuses as very efficient heat-transfer intensifiers. Interest in this field was initiated many years ago, and since then ample data concerning this problem have been accumulated [1, 2]. Nonetheless, jets of interest still attract considerable attention, and extensive studies aimed at searching for new methods to control heat and mass transfer in such systems are presently under way. One strategy here implies modification of the surface onto which the jets impinge with obstacles. The authors of [3-5] examined heat transfer in narrow channels in which a system of jets impinging onto a surface modified with cavities was organized. It was shown that such systems exhibit good performance characteristics and can be used, in particular, for cooling turbine blades from inside. Yet, the physical reasons underlying the observed intensification of heat-transfer processes in the systems of interest have remained unveiled.

Cavities used as obstacles modifying the streamlined surfaces attract considerable interest as they proved capable of generating exotic self-oscillatory flow modes and intensifying heat transfer [6 - 9]. In the case of jets impinging onto heat-exchanging surfaces the interaction pattern of the jet flow with

the surface is much more complicated compared to a flat surface. For instance, vortex formation processes and local flow separations induced by obstacles prepared in the form of cavities may give rise to a complex, unstable flow involving large-scale low-frequency oscillations, Taylor-Görtler vortices, etc. Under such conditions, accurate prediction of heat-transfer processes becomes problematic, although this problem was addressed in several publications.

The aerodynamic structure of the flow and the heat transfer from hemispherical cavity versus the jet Reynolds number and as functions of the distance from the nozzle exit to the heat-exchanging surface were examined in [10]. The jet flow was not swirling, and the cavity-to-nozzle diameter ratio was  $D_c/d_n = 5.4$ . The average heat transfer from the cavity surface was found to be decreased twice compared to the case of a flat surface, all other conditions being identical. With decreasing the nozzle diameter, the heat transfer grows in value and, according to [11], at  $D_c/d_0 = 44$  its intensity may exceed the same quantity for flat surface. Moreover, in the region outside the cavity the thermal interaction between the lateral flow and the jet rapidly degenerates, making the heat transfer almost vanishing here [10]. In the cavity, pronounced vortex formations arise. These formations, resembling Taylor-Görtler vortices, give rise to coherent structures that can be easily registered by heat flow microsensors.

In [12], local heat transfer in hemispherical cavity was examined as a function of cavity-to-nozzle diameter ratio in the range  $D_c/d_0 = 11 \div 30$ . Here, unlike in [11], increase in the nozzle diameter was found to suppress heat transfer throughout the whole examined range of Reynolds numbers,  $Re_0 \sim 10^4$  to  $5 \cdot 10^4$ . Thus, the experimental data concerning the interaction of jet flows with curvilinear surfaces are still few in number and contradictory.

The problem becomes even more complicated if the jet flow impinging onto an obstacle has a rotational velocity component. So far, the heat transfer and the structure of flow between a swirling jet and a surface onto which this jet impinges remain poorly understood. Reported studies [13-17], also few in number, show the interaction of a swirling jet with

an obstacle to be a complex many-factor process. An increase in jet pre-rotation intensity enhances mixing processes in the system; as a result, the heat transfer of the surface interacting with the jet at large nozzle-to-surface separations ( $L/d_0 > 6$ ) turns out to be substantially reduced in comparison with non-swirling jets. At small nozzle-to-surface separations,  $L/d_0 < 6$ , the picture may be opposite, with additional maxima emerging in the radial distribution of heat-transfer intensity due to specific formation pattern of swirling jets with different pre-rotations, Reynolds numbers, and separations between the jet turning point and the surface onto which the jet impinges.

Expectedly, pre-rotation of the jet approaching a surface modified with a spherical cavity will complicate the problem. The fluid leaving the cavity will interact with the rotational jet flow impinging onto the obstacle, thus causing the formation of unsteady three-dimensional flows hard to predict numerically.

The purpose of the present study was to experimentally examine the time-average heat transfer from a surface with spherical cavity during impingement of a swirling jet onto this surface. Such flows are often encountered in low-temperature plasma generators and in power-plant cooling systems.

## NOMENCLATURE

$C_p$	[-]	Pressure coefficient
$d_0$	[m]	Nozzle diameter
$D_c$	[m]	Spherical cavity diameter
$L$	[m]	Distance between nozzle and bottom cavity
$K$	[W/mK]	Thermal conductivity
$Nu$	[-]	Nusselt number
$p_i$	[N/m <sup>2</sup> ]	Local pressure on the wall
$S$	[m]	Distance between nozzle and plate
$q_i$	[W/m <sup>2</sup> ]	Local heat-flux density
$R$	[-]	Jet swirl number
$Re$	[-]	Reynolds number
$Tu$	[%]	Turbulence intensity
$T_w$	[K]	Wall temperature
$T_0$	[K]	Jet temperature
$u$	[m/s]	Axial velocity
$r$	[m]	Radial direction
$y$	[m]	Vertical direction

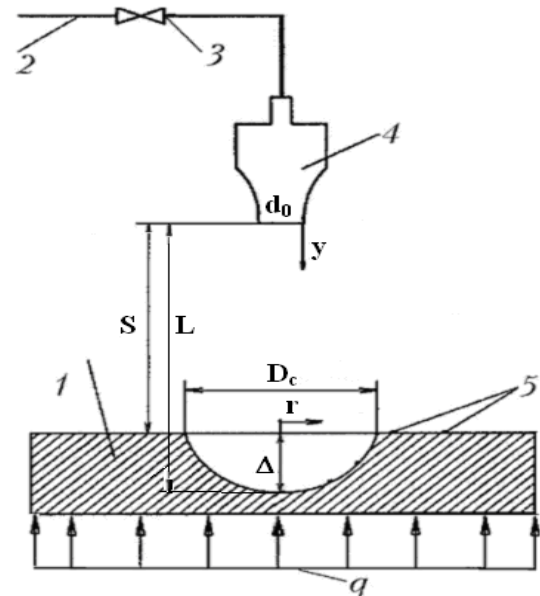
Special characters		
$\alpha$	[W/m <sup>2</sup> s]	Heat transfer coefficient

Subscripts		
$C$		Cavity
$i$		Local means
$0$		Jet parameter
$w$		Wall

## EXPERIMENTAL SET-UP AND MEASUREMENTS

The experiments were carried out on an experimental setup whose schematic diagram is shown in Figure 1. This setup was described in detail elsewhere [18]. A room-temperature axisymmetric swirling air jet emanated out of the nozzle whose diameter was  $d_0 = 8.5$  mm. The cavity diameter was fixed, equal to  $D_c = 46$  mm, and the cavity depth was either  $\Delta = 0, 12,$

or 23 mm. The gas-flow velocity  $U_0$  at the nozzle exit plane was varied in the range from 20 to 100 m/s, corresponding to Reynolds numbers  $Re_0 = U_0 d_0 / \nu = 10^4 \div 5 \cdot 10^4$ .



**Figure 1** Experimental setup. 1 – obstacle; 2 – air line; 3 – regulating valve; 4 – nozzle; 5 – heat-flux meters.

The distribution of flow velocity over the nozzle outlet was almost uniform, the turbulence number being  $Tu \cong 0.3\%$ . The jet approached the obstacle normally, and the nozzle-to-surface separation  $S$  was varied from 0 to  $10d_c$ .

The jet flow was made swirling by pre-rotating it with a flat swirler with variable slit inclination angle. The jet pre-rotation parameter, defined as the momentum ratio between the circumferential component of the jet flow and the jet flow along the jet axis, was  $R = 0, 0.58, 1, \text{ or } 2.74$ .

Measurements a velocity components (axial, tangential and radial) and turbulent pulsations were carried out with the help of 2 - components laser doppler anemometer LADO - 3 (Kutateladze Institute of Thermophysics production). A system of measurements was supply by 3-D automatic coordinate gage with minimal division value  $\sim 1$  mkm.

The test section, 50-mm thick and 190-mm in diameter, was prepared from copper. The desired temperature difference between the jet and the obstacle, about  $40^\circ$  in the majority of tests, was achieved with the help of an ohmic heater installed over the obstacle. The choice of copper, having a high value of thermal conductivity, as the obstacle material was motivated by the desire to organize the boundary condition  $T_w \cong \text{const}$  at the obstacle surface. The temperature of the streamlined surface and the flow temperature were measured by chromel - copel thermocouples.

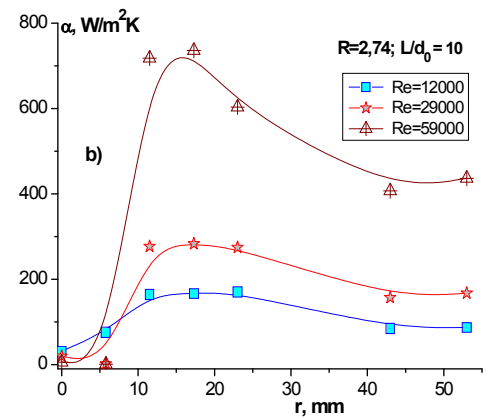
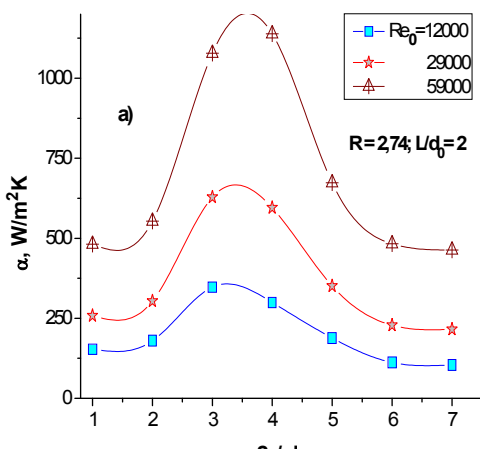
The heat fluxes were measured by gradient-type heat flow sensors glued onto the obstacle [19]. The characteristics of the sensors were as follows: in-plane dimensions -  $2.5 \times 2.5$  mm

(these dimensions were much smaller than the characteristic spatial scale of the flow under study), thickness - 0.2 mm, volt-watt sensitivity  $\approx 10$  mV/W, response time - 0.05 ms. In the course of measurements the arrays of instantaneous values of heat-flux densities were used to calculate the time-average heat fluxes, and also the heat-flow pulsations and spectra. The total number of realizations in one measurement was chosen to equal  $10^4$ . The time in which one measurement was taken was varied from 10 to 90 s; it was found experimentally that this time affected measured data insignificantly. A tailored multi-channel unit was used to amplify the signals from the thermocouples and heat-flow sensors; then, the signals were fed to a computer to be processed there. The coefficient of heat transfer was defined as  $\alpha_i = q_i / (T_w - T_0)$ , where  $q_i$  is the local heat-flux density measured by sensor, and  $T_w$  and  $T_0$  are respectively the temperature of the surface and the temperature of the air at the nozzle exit. An error analysis showed that the measurement inaccuracy for heat fluxes in our experiments was within  $0.3 \div 10\%$ . Simultaneously, the measurement inaccuracy for square-mean pulsations of heat fluxes was much greater, amounting to 30% for the obstacle with cavity. In processing the experimental data, experimentally determined losses of heat by radiation and free convection were taken into account.

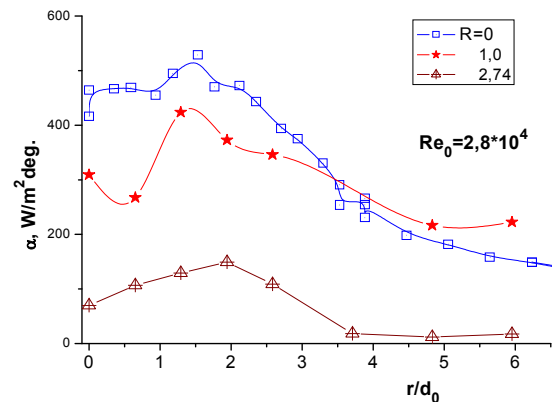
### INTERACTION OF SWIRLING JET WITH A FLAT PLATE. TESTING EXPERIMENTS

At the first stage of the study, we experimentally examined the heat transfer of flat obstacles interacting with eddying jets. A specific feature of such jets is their high ejective capacity due to which the jet divergence angle grew in value with increasing jet pre-rotation [14-16]. As a result, the heat-transfer maximum got displaced off the center (see Figure 2).

Depending on pre-rotation and nozzle-to-obstacle separation, this maximum could be either distinctly pronounced (Figure 2a,  $L/d_0 = 2$ ) or smooth (Figure 2b,  $L/d_0 = 10$ ). A similar tendency was also observed at other jet pre-rotations. Importantly, in the majority of our experiments the radial distributions of local heat transfer behaved similarly, the heat transfer increasing with increasing the Reynolds number (see Figure 2a).



**Figure 2** Distributions of heat-transfer coefficients over the flat surface interacting with a swirling jet.



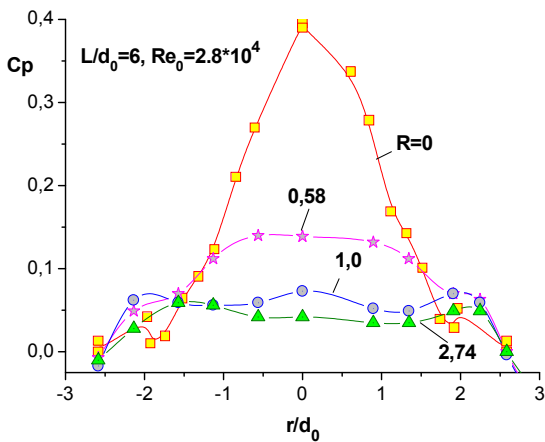
**Figure 3** Effect of jet pre-rotation on the heat transfer from flat surface,  $L/d_0 = 10$

Effect of a swirling angle of a impinging jet on a local heat transfer is shown on Figure 3. It is visible, that with increase swirling parameter of a jets intensity of heat transfer is considerably reduced. At large parameter ( $R = 2,74$ ) external region of flow heat transfer practically is absent, that speaks about intensive mixing impinging jets with an environment media.

### INTERACTION OF SWIRLING JET WITH A HEMI-SPHERICAL CAVITY. PRESSURE PATTERN

The value of the pressure coefficient was found as the pressure drop between the surface and the atmosphere normalized to the dynamic pressure of the jet flow at the nozzle exit plane,  $C_p = 2(p_r - p_a) / \rho_0 u_0^2$ . A typical distribution of pressure coefficient over the radius of spherical cavity is exemplified in Figure 4. Without pre-rotation ( $R=0$ ) the pressure profile exhibits a distinct axial maximum. At the periphery, an inflection point brought about by the separation of the flow in this region is observed. The mechanism underlying the

formation of flow in a cavity impinged by non-swirling jet was discussed in more detail in [10, 18, 19].



**Figure 4** Pressure coefficient in the cavity onto which a swirling jet impinges

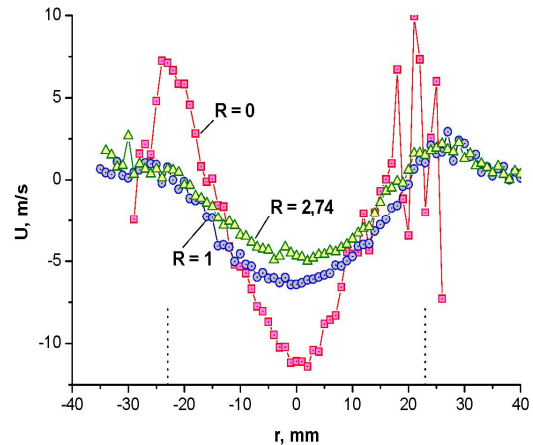
With increasing the jet pre-rotation parameter the pressure expansion of the jet flow and the reduction of the jet momentum in the axial region. For instance, at  $R=2.74$  the pressure coefficient is one order of magnitude smaller in comparison with non-swirling jet.

Another characteristic feature in the behavior of cavity pressure deserves mention. With swirling jets, the distributions of pressure in the peripheral region display no inflections. Moreover, a growth of pressure is observed in this zone due to the absence of flow separation and the action of centrifugal forces pressurizing the fluid here, as well as due to the influence of the surface curvature. Apparently, these factors will have influence on the formation of the temperature field in the system and on the heat- and mass-transfer processes proceeding there.

The important part of this work concerns experimental study of turbulent structure of the flow and turbulent fluctuations at swirl jet impinging on a surface with hemispherical cavity.

Systematic experimental data show extreme complexity of flow formation pattern in a swirl impinging jet interacting with a spherical cavity. Due to centrifugal forces the vortex jets expand faster, and the longitudinal component of velocity decreases with an increase in the swirl parameter. As an example, Figure 5 demonstrates distribution of longitudinal component of velocity in the impinging jet at variations of initial swirl angle in a plane located 5 mm away from an obstacle. With no swirl the jet penetrates into the cavity, turns around by 180 degrees and pools out along the lateral surface. This reverse flow mixes with the falling jet and practically does not interact with the flat surface surrounding the cavity. At that in peripheral part of the backflow jet we observed large-scale flow fluctuations which caused strong irregularities in velocity profiles.

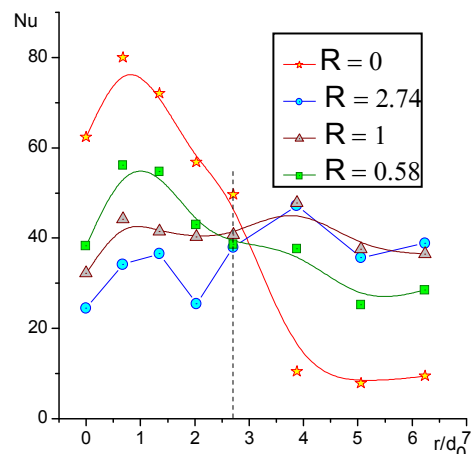
For swirling jets the backflows are expressed in a much less degree and the impinging jet area stretches far beyond the cavity, what leads to heat transfer enhancement in this area. The velocity value on a jet axis decreases by the factor of 2-3 with an increase in swirl intensity as compared to non-swirling jets. All mentioned peculiarities of hydrodynamics of swirling impinging jets testify previous data on heat transfer regularities.



**Figure 5** Distribution of longitudinal velocity at the cavity boundary at variations of swirl angle,  $Re = 12000$

### LOCAL AND AVERAGE HEAT TRANSFER IN A DIMPLE

The distribution of local heat-transfer coefficients in the radial direction is shown in Figure 6. The rate of heat transfer is maximal in the absence of jet pre-rotation. As the jet pre-

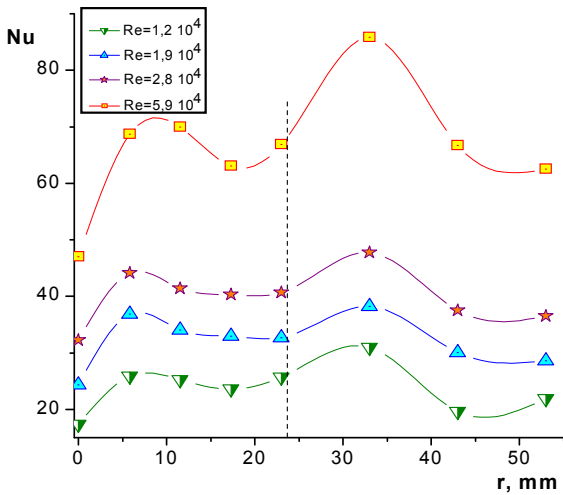


**Figure 6** Local heat transfer for an obstacle with spherical cavity impinged by a jet.  $Re_0 = 2.8 \cdot 10^4$ ,  $L/d_0 = 10$ . The dashed line shows the cavity boundary

rotation parameter  $R$  grows in value, the Nusselt number starts exhibiting a non-monotonic behavior both along the radius, with the formation of additional maxima, and versus  $R$ . In the vicinity of the cavity the heat-transfer intensity for swirling jets turns out to be more than twice decreased. On the contrary, in the cavity the heat transfer from the swirling jet turns out to be enhanced in comparison with the case of non-swirling jets, the radial distribution of the heat-transfer coefficient being more uniform here.

The latter points to a greater expansion angle of swirling jets, resulting in a growth of heat transfer in the region outside the cavity.

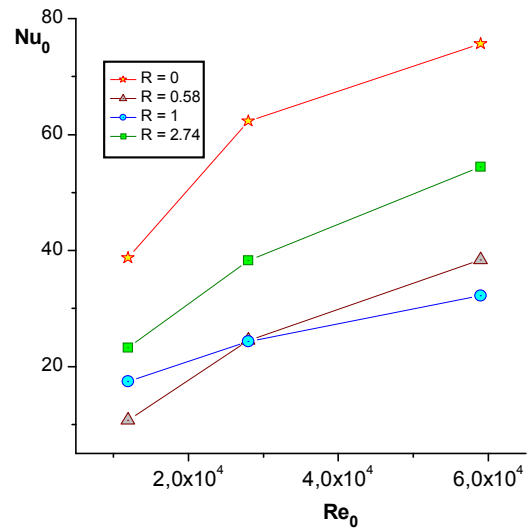
At fixed jet pre-rotation the heat-transfer data for different Reynolds numbers proved to be self-similar. This is evident from Figure 7. Only at large  $Re_d$  numbers the heat-transfer maxima are pronounced more distinctly, with their position over the radius being quite stable.



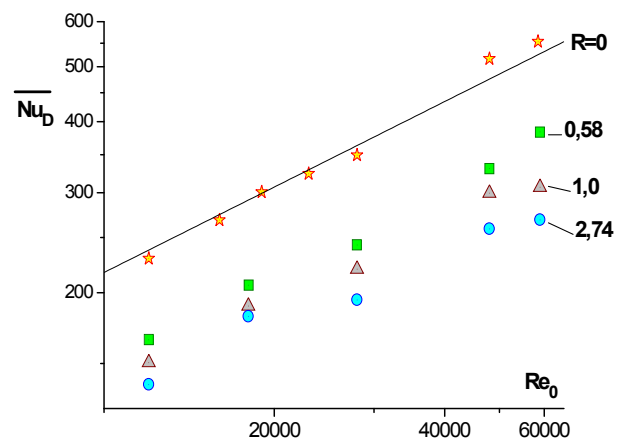
**Figure 7** Effect of Reynolds number on the heat transfer of swirling jet.  $R = 1, L/d_0 = 10$

An important characteristic of jets impinging onto surfaces is the heat transfer at the stagnation point. The experimental data  $Nu_0 = f(Re_0)$  for various jet pre-rotations are shown in Figure 8. Here, the most intense heat transfer is observed in the absence of pre-rotation ( $R = 0$ ), whereas the increase in the pre-rotation results in that the rate of heat transfer at the center of the cavity decreases. First of all, it is the formation of a complex aerodynamic structure during the interaction of jet with hemispherical cavity that underlies the complex behavior displayed by heat transfer.

From the viewpoint of applications, a most important point in such complex problems is the deduction of correlation dependences for the average heat transfer versus Reynolds number and other governing parameters. In view of axial symmetry, in the present study the average value of the heat-transfer coefficient over the entire surface of cavity was obtained by integrating the local distributions over the radius.



**Figure 8** Distribution of Nusselt number  $Nu_0$  at the stagnant point of the flow,  $L/d_0 = 10$



**Figure 9** The Nusselt number  $\overline{Nu}_D$  averaged over the cavity surface versus jet pre-rotation.  $L/d_0 = 6.0$

The average number  $\overline{Nu}_D$  versus  $Re_0$  at  $L/d_c = 6$  for various jet pre-rotations is shown in Figure 9. Here, the Nusselt number was calculated from the cavity diameter and from the mean value of the heat-transfer coefficient,  $\overline{Nu}_D = \alpha D_c / \lambda$ . With increasing the jet pre-rotation at all other conditions kept unchanged the heat-transfer intensity decreases in value. For a non-swirling jet the experimental data obtained at  $L/d_0 = 6$  can be fitted with the empirical relation

$$\overline{Nu}_D = 2.17 \cdot Re_0^{0.5}, \quad (1)$$

typical of laminar heat-transfer mode. As a matter of fact, the flow in the cavity is turbulent; yet, the specific character of the formed flow pattern, namely, the involvement of reverse flows, curved flow streamlines, large-scale structures, etc., has

resulted in the above relation for heat transfer. Apparently, relations of type (1) will undergo changes with distance to the heat-exchanging surface, and for this evolution to be clarified additional tests or numerical experiments are required. Although the rate of heat transfer in the cavity proper decreases in the case of swirling jets, the total heat transfer, including the area adjacent to the cavity may be increased. For this possibility to check, here again additional studies are required.

## CONCLUSION

An experimental study of heat transfer during impingement of swirling jet onto hemispherical cavity was performed. It is shown that, following jet pre-rotation, additional maxima appear in the radial distribution of heat transfer, brought about by specific features of mixing processes of eddying jets with ambient air.

The intensity of heat transfer in cavity for swirling jets is substantially, by a factor of 2-3, reduced in comparison with non-swirling jets. On the contrary, outside the cavity the heat transfer for swirling jets is more intense. During variation of jet pre-rotation parameter the local Nusselt numbers vary non-monotonically both at the stagnant point and at different points over the radius, pointing to a complex mechanism of hydrodynamic and thermal processes in swirling jets impinging onto spherical cavities. At fixed distance to the plate and at jet pre-rotation parameter kept unchanged the distributions of heat transfer are self-similar if considered versus the Reynolds number.

This work was supported by the Russian Foundation for Basic Research (Grant No. 10 - 08 - 00105).

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