LITERATURE CITED

1. N. A. Shakhova and G. A. Minaev, Inzh.-Fiz. Zh., 19, No. 5, 826; 19, No. 6, 1002 (1970).
2. N. A. Shakhova and G. N. Lastovtseva, Inzh.-Fiz. Zh., 25, No. 4, 581 (1973).
3. A. P. Lummi and A. P. Baskakov, Khim. Prom., No. 7, 522 (1967).
4. O. M. Todes et al., Khim. Prom., No. 6, 402 (1967).
5. G. S. Cherkez et al., in: High-Temperature Endothermic Processes in a Fluidized Bed [in Russian], Moscow (1968), p. 121.
6. A. D. Kalegin and E. V. Donat, Khim. Prom., No. 8, 619 (1970).
7. A. A. Sverdlov, Khim. Prom., No. 1, 63 (1974).
8. A. P. Baskakov, Heating and Cooling Metals in a Fluidized Bed [in Russian], Metallurgiya, Moscow (1974).

HEAT TRANSFER OF A CONICAL TUBE WITH INTERNAL COOLING BY A SWIRLING AIR JET

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The results of an experimental investigation of local heat transfer during flow of turbulent semibounded jets in a conical tube beyond an annular swirler installed at the inlet are presented.

In calculations of complex heat transfer in furnaces, in the nozzle apparatus of turbomachines, in ducts of variable cross section, etc., the problem arises of determining the temperature level of conical elements cooled on the inside by an annular swirling air jet. The few investigations of heat transfer and aerodynamics under conditions of a swirling motion of the coolant are devoted mainly to problems of the swirling flow in nozzles and diffusers [1-6].

To obtain the necessary mathematical dependences for the local heat transfer of a swirling jet stream in a conical duct, investigations were carried out on the experimental device shown in Fig. 1a. The main element of the test bench is a conical measuring section 1 with cylindrical entry and exit pipes 2 and 3. An interchangeable vaned swirler 4 with fairing 5 is installed in the entry pipe. The air flows through the swirler along the inside surface in the form of an annular swirling jet. The design of the test bench permitted installing cones with a different apex angle 2ψ (Table 1) and swirlers with a different angle of twist φ_0 and relative area of the hub β (Table 2). The swirlers were made in the form of annular cascades with straight vanes and constant angle of twist along the radius, as was used in the designs of the front devices of gas-turbine combustion chambers. The local heat-transfer coefficients were measured by means of built-in miniature electrical alpha-calorimeters 6 which were mounted on insulation boards 7 installed in longitudinal grooves (along four generatrices) of the cone flush with its inside surface. To prevent disturbances in the wall boundary layer owing to possible roughness at the site of installing the boards and calorimeters, their inside surface was machined jointly with boring of the conical part of the measuring section. The design of the calorimeter is shown in greater detail in Fig. 1b. The calorimeter consists of a copper housing 8 made in one piece with a bottom and a copper cover 9, which form a closed unit, inside which are located a thermocouple assembly 10 (copper—Copel) soldered to the bottom and an electrical heater 11. The thermocouple assembly and the heat losses of the calorimeter were calibrated preliminarily (Fig. 1c) and showed sufficient stability for the technology of manufacture and assembly of the calorimeters that was used.

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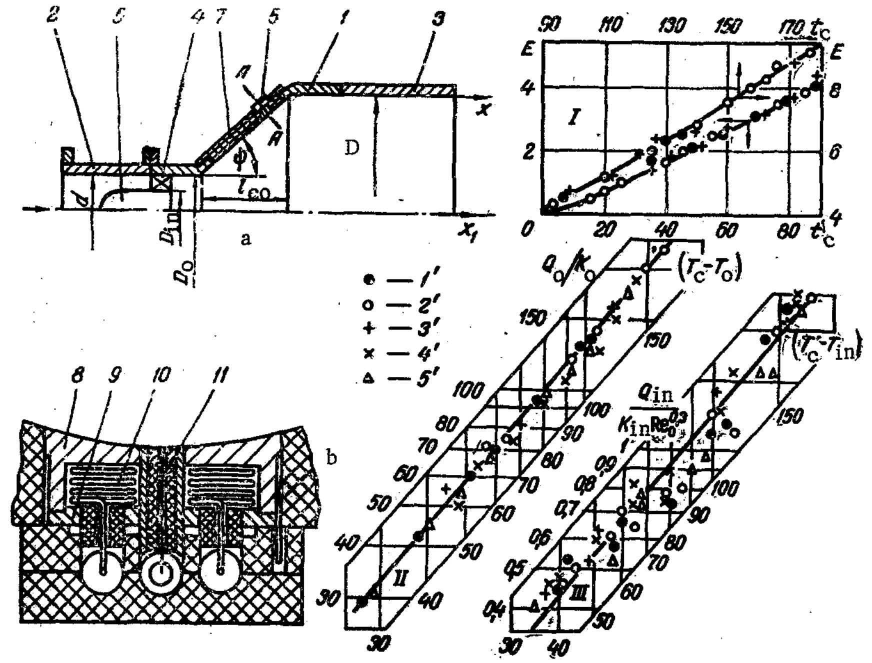

Fig. 1. Diagram of experimental device: a) measuring section; b) built-in alpha-calorimeter; c) calibration curves of calorimeter: I) dependence of emf of thermocouple assembly on temperature of measuring surface; II) heat losses from external surfaces of calorimeter into ambient environment; III) heat losses into the air flow in the tube (1'-5' are the numbers of the calorimeters). E in mV; Q_{in} and Q_o in W; t, $(t_c - t_o)$, and $(t_c - t_{in})$ in °C.

TABLE 1. Geometric Parameters of Cones

Cone	15	45	60
d, mm	56	56	56
D, mm	207,2	207,2	207,0
l_{co}, mm	287	76	43
$2\psi°$	30	90	120

TABLE 2. Geometric Characteristics of Swirlers

Swirler	24-25	31-25	31-47	31-72	44-25	44-47	44-72	66-25	66-47	66-72	74-25
D_o, mm	55,75	56,0	56,0	56,0	55,55	55,55	55,55	56,0	56,0	56,0	55,65
D_{in}, mm	28,2	28,4	38,21	47,31	28,16	38,21	47,31	28,3	38,21	47,31	28,32
n_v	43	18	18	18	14	14	14	8	8	8	4
α_{ef}°	66°15′	59°00′	59°00′	59°0′	45°40′	45°25′	45°20′	24°00′	24°00′	24°00′	15°40′
φ_0°	23°45′	31°00′	31°00′	31°00′	44°20′	44°35′	44°40′	66°00′	66°00′	66°00′	74°20′
β	0,25	0,25	0,47	0,72	0,25	0,47	0,72	0,25	0,47	0,72	0,25

Calibration was carried out in the same range of variation of the velocities and heat fluxes as the main experiments (Table 3). Two or three calorimeters were installed in each of the 13-20 measuring cross sections of the cone. Power supply to the calorimeters and all necessary measurements for determining the power being supplied to them and the temperature of the measuring surface were accomplished by a measuring unit that provided exact maintenance of the voltage at the electrical heaters of the calorimeters during the entire experiment and permitted recording the power of each calorimeter separately.

TABLE 3. Investigated Variants of Cones and Swirlers

Cone	15									
Swirler (φ_0—β)	24-25	31-25	31-47	31-72	44-25	44-47	44-72	66-25	66-47	66-72
G, kg/h	630—860	550—770	500—540	240—260	400—490	250—350	150—180	185—260	185—215	110
$q, \frac{\text{kcal}}{\text{m}^2 \cdot \text{h}} \cdot 10^{-4}$	2,1—4,0	2,3—4,4	2,1—4,5	2,3—4,0	2,2—4,3	2,0—4,4	2,1—4,2	2,4—4,6	2,2—4,4	1,8—4,1
T_c/T_{in} (maximum)	1,46	1,39	1,36	1,36	1,39	1,45	1,33	1,40	1,35	1,38
$c_c = \frac{Nu_x}{Re_x^{0,8} x^{n_c}}$	0,064	0,079	0,107	0,158	0,124	0,150	0,125	0,180	0,225	0,262
n_c	—0,61									
$c_m = \frac{Nu_x}{Re_{xm}^{0,8}}$	0,0260	0,0274	0,0299	0,0307	—	—	—	0,0298	—	—

Cone	45								60				
Swirler (φ_0—β)	31-25	44-25	44-47	44-72	66-25	66-47	66-72	74-25	44-25	66-25	66-47	66-72	74-25
G, kg/h	550—810	330—480	220—320	135—170	165—240	185—205	35—100	120	300—460	165—255	160	87	95—145
$q, \frac{\text{kcal}}{\text{m}^2 \cdot \text{h}} \cdot 10^{-4}$	1,3—4,9	1,9—7,2	1,5—6,3	1,5—5,2	1,5—6,4	1,4—7,0	1,2—6,2	1,9—5,8	1,1—6,4	1,2—7,6	1,2—4,9	1,0—3,6	1,4—7,9
T_c/T_{in} (maximum)	1,43	1,43	1,42	1,48	1,41	1,42	1,45	1,48	1,64	1,64	1,39	1,42	1,63
$c_c = \frac{Nu_x}{Re_x^{0,8} x^{n_c}}$	0,110	0,190	0,300	0,460	0,300	0,420	0,760	0,460	0,185	0,375	0,560	0,880	0,500
n_c	—1,0								—1,1				
$c_m = \frac{Nu_x}{Re_{xm}^{0,8}}$	—	0,0307	0,0332	—	0,0317	—	—	0,0350	—	0,0377	—	—	0,0390

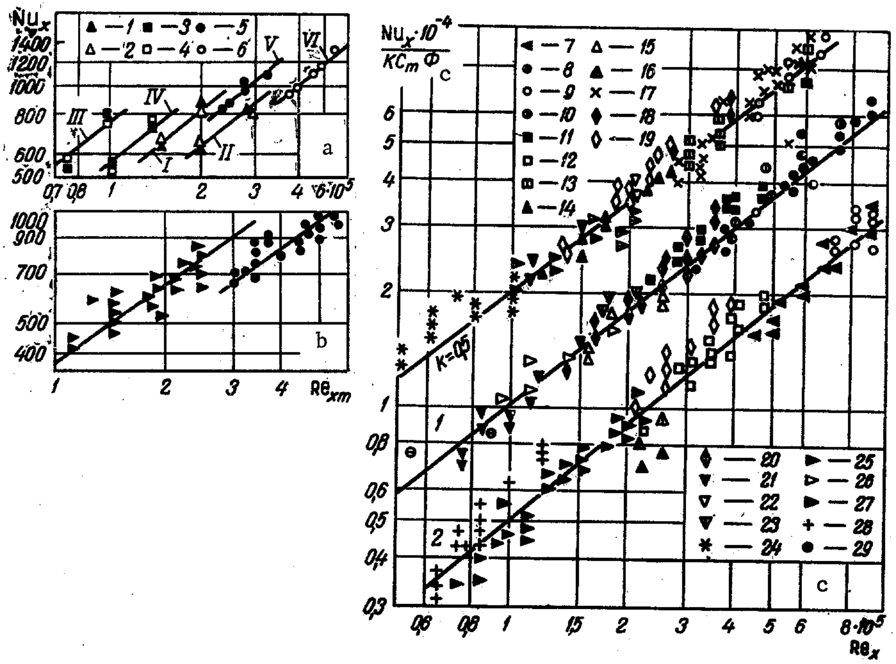

Fig. 2. Local heat transfer of a swirling turbulent annular air jet propagating along the inside surface of a cone: a) Effect of Gr for $\bar{x}$ = 3.82 (I) and 5.25 (II) when ψ = 45°, φ_0 = 44°, β = 0.25; T_c/T_{in} for $\bar{x}$ = 4.18 (III) and 5.25 (IV) when ψ = 45°, φ_0 = 66°, β = 0.25, and $\bar{x}$ for ψ = 15°, φ_0 = 44°, β = 0.25; b) Nu_x vs Re_{xm}; c) values of $Gr \cdot 10^{-9}$: 1) 1.1; 2) 4.6; values of T_c/T_{in}: 3) 1.17; 4) 1.37; values of $\bar{x}$: 5) 5.88; 6) 8.95; values of $\psi/\varphi_0 - 100\beta$: 7) 15/24-25; 8) 15/31-25; 9) 15/31-47; 10) 15/31-72; 11) 15/44-25; 12) 15/44-47; 13) 15/44-72; 14) 15/66-25; 15) 15/66-47; 16) 15/31-72; 17) 45/31-25; 18) 45/44-25; 19) 45/44-47; 20) 45/44-72; 21) 45/66-25; 22) 45/66-47; 23) 45/66-72; 24) 45/74-25; 25) 60/66-25; 26) 60/66-47; 27) 60/66-72; 28) 60/74-25; 29) 20/60-61 according to the data in [18]; —) according to Eq. (10).

The heat losses can be approximated by the functions

$$Q_0 = k_0 (T_c - T_0)^{1.1}\ \mathrm{W}\ ;\ k_0 = 0.0131 - 0.0161; \tag{1}$$

$$\left.\begin{array}{c} Q_{in} = k_{in} (T_c - T_0)^{n_1} \mathrm{Re}_0^{0.3}\ \mathrm{W}\ ;\ k_{in} = (0.144 - 1.075) \cdot 10^{-3} \\ n_1 = 1 + 0.0346\ \mathrm{tg}\,\psi \end{array}\right\}. \tag{2}$$

When processing the experimental data the value of the local coefficient was found from the expression

$$\alpha = \frac{Q_0 - Q_l}{f_c (T_c - T_{in})} k_x,\ Q_l = Q_0 + Q_{in}. \tag{3}$$

Expression (3) includes a correction factor $k_x = [1 - (x_0/x_1)]^{0.2}$ that takes into account in conformity with [7, 8] the effect on heat transfer of the relative thickness of the thermal boundary layer formed at the site of installing the calorimeter owing to the abrupt change of temperature of the surface exposed to the flow.

Before conducting the main series of experiments the method of measuring and processing of the experimental data was tested on such a classical case as heat transfer during turbulent flow of an unbounded stream past a plate. The data obtained in this test are generalized well by the known similarity equation [9]

$$\mathrm{Nu}_x = 0.0255\,\mathrm{Re}_x^{0.8}\ ;\ \mathrm{Nu}_x = \frac{\alpha x}{\lambda};\ \mathrm{Re}_x = \frac{W_{x0} x}{\nu}. \tag{4}$$

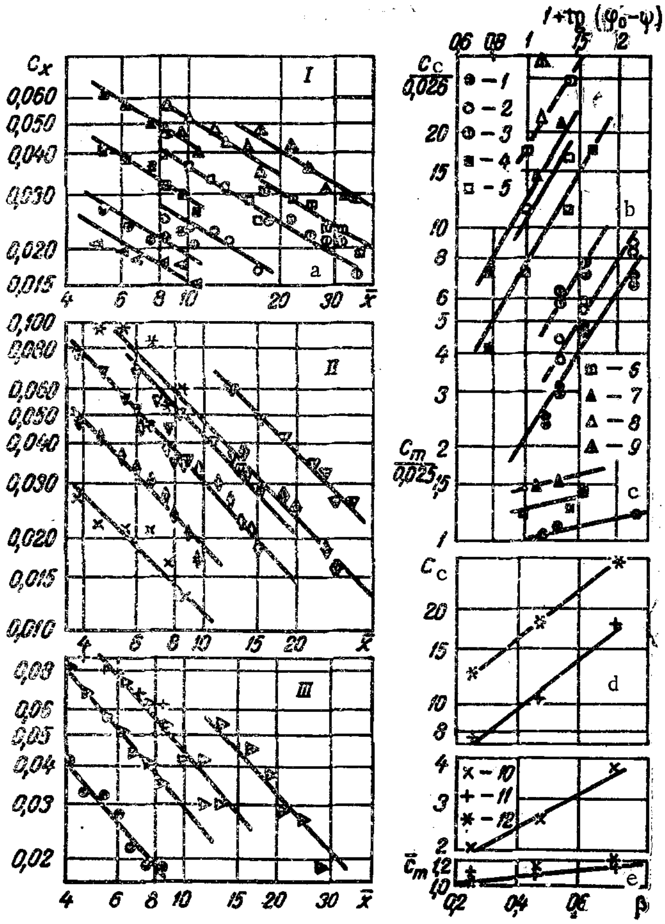

Fig. 3. Dependences: a) $c_x = f(\bar{x})$; b) $c_c = f(\varphi_0 - \psi)$; c) $c_m = f(\varphi_0 - \psi)$; d) $c_c = f(\beta)$; e) $\bar{c}_m = f(\beta)$. Symbols of curves for a) when $\psi = 15°$ (I), 45° (II), and 60° (III), see Fig. 2c. Values of ψ/β: 1) 15/0.25; 2) 15/0.47; 3) 15/0.72; 4) 45/0.25; 5) 45/0.47; 6) 45/0.72; 7) 60/0.25; 8) 60/0.47; 9) 60/0.72. Values of ψ: 10) 15°; 11) 45; 12) 60°.

In conformity with the theory of similarity and dimensionality [10] the Nusselt (Nu) number characterizing local heat transfer during flow of an annular jet along an inside cylindrical surface is a function

$$\mathrm{Nu}_x = f\left(\mathrm{Re}_x, \mathrm{Pr}, \mathrm{Gr}, \frac{T_c}{T_{in}}, \bar{x}, \beta\right). \tag{5}$$

If the jet moves over a helical trajectory in a conical tube, then we must add to the characteristic quantities the apex angle of the conical surface and a parameter characterizing the twist. As a rule, the Rossby number [11-14], Din criterion [15], or amount of twist [16] is taken for such a parameter. As shown in [6, 17], in the case of a constant angle of twist over the height of the swirler vane these criteria can be reduced to the ratio $(W_{\varphi 0}/W_{x0}) = \mathrm{tg}\varphi_0$. Then with consideration of constancy of the Prandtl (Pr) number for air Eq. (5) takes the form

$$\mathrm{Nu}_x = f\left(\mathrm{Re}_x, \mathrm{Gr}, \frac{T_c}{T_{in}}, \bar{x}, \beta, \psi, \frac{W_{\varphi_0}}{W_{x0}}\right). \tag{6}$$

As follows from specially set-up experiments (Fig. 2a), heat transfer in the case being considered, just as in the case of flow in a cylindrical duct [17], does not depend either on the temperature factor T_c/T_{in} (curves III and IV) or on the Grashof (Gr) number (curves I and II) calculated with respect to centrifugal acceleration. This reduces the number of arguments in (6), and as a result we have

$$\mathrm{Nu}_x = f(\mathrm{Re}_x, \bar{x}, \beta, \psi, \varphi_0). \quad (7)$$

On processing the experimental data in the form of the function $\mathrm{Nu}_x = f(\mathrm{Re}_x)$ for constant geometric parameters ψ, φ_0, and β the experimental points for all tested variants are stratified depending on the distance $\bar{x}$ (Fig. 2a, curves V and VI). The Nusselt (Nu_x) number is proportional to the Reynolds (Re_x) number to a power of 0.8, and the coefficient before Re_x varies with distance x according to the power law (Fig. 3a)

$$\mathrm{Nu}_x = c_x \mathrm{Re}_x^{0.8}, \; c_x = c_c \bar{x}^{n_c}. \quad (8)$$

The average flow velocity at the exit from the swirler is taken as the characteristic velocity in (8) and the distance x, which takes into account the increase of the boundary layer in the flow direction, is taken as the characteristic dimension; the physical constants were determined on the basis of the inside air temperature T_{in}. The values of c_c and n_c are given in Table 3. A physically substantiated analysis of the heat-transfer experiments requires knowledge of the kinematics of the flow — a representation of the total aerodynamic picture of the flow both in the entire volume of the investigated measuring section and in the wall regions most important from the viewpoint of heat transfer. Therefore, the majority of experiments to investigate heat transfer were accompanied by a detailed study of the aerodynamic structure of the flow. This made it possible to establish certain physical concepts of the phenomena occurring during jet cooling under conditions of the internal problem in deriving the experimental dependences for coefficient c_x.

The Nu_x number can be represented as a function of the Re_x number calculated on the basis of the local velocity determining heat transfer in the given cross section. As follows from Fig. 2b, the maximum axial velocity W_{xm} can be taken for the characteristic velocity. Then the experimental points are described by the relationship

$$\mathrm{Nu}_x = c_m \mathrm{Re}_{xm}^{0.8}, \quad (9)$$

wherein the coefficient c_m depends on the parameters φ_0, ψ, and β, and its experimental values for the tested variants are shown in Table 3.

Equating (8) and (9), we obtain

$$\mathrm{Nu}_x = c_m \Phi_c \mathrm{Re}_x^{0.8}, \; \Phi_c = \frac{c_c}{c_m} \bar{x}^{n_c}. \quad (10)$$

The structure of this equation allows a certain physical interpretation, since the function $\Phi_c \sim (W_{xm}/W_{x0})^{0.8}$ reflects the regularity of the change of the characteristic velocity along the measuring section and the coefficient c_m characterizes the effect of the apex angle of the cone, initial angle of twist (Fig. 3b,c), and relative thickness of the jet (Fig. 3d,e) on the intensity of heat transfer. Using the usual methods of deriving the empirical dependences and taking into account the qualitative structure of Eqs. (10), we obtain the following expressions:

$$\left.\begin{aligned} c_m &= 0.025\, a_m [1 + \mathrm{tg}(\varphi_0 - \psi)]^{0.22} \exp(0.208\,\beta), \\ a_m &= 0.92 + 0.32\,\mathrm{tg}\,\psi, \end{aligned}\right\} \quad (11)$$

$$\left.\begin{aligned} &\Phi_c = 3.24 a_c [1 + \mathrm{tg}(\varphi_0 - \psi)]^{1.38} \exp[1.07(0.778 + \sin\psi)\beta]\, a_\psi \bar{x}^{n_c}, \\ &a_c = 0.7\exp(2.66\sin\psi); \quad a_\psi = \frac{1}{2.87 + \mathrm{tg}\,\psi}; \; n_c = 1.17\sqrt{\sin\psi}. \end{aligned}\right\} \quad (12)$$

As follows from Fig. 2c, Eq. (10) together with (11) and (12) generalizes completely satisfactorily all experimental data obtained in the present study upon a change of the characteristic quantities within the following limits: $15 \le \psi \le 60$; $25 \le \varphi_0 \le 75$, $0.25 \le \beta \le 0.75$, $-15 \le \varphi_0 - \psi \le 65$, $3 \le \bar{x} \le 35$, $\mathrm{Re}_x = (0.8\text{-}10)\cdot 10^{-5}$. The experimental values of α and those determined by (10) coincide also for the few available data on local heat transfer obtained in an investigation of cooling of the transition cone of the front device of a triple-damper gas-turbine combustion chamber during firing tests [18]. As follows from (11) and (12), the intensity of heat transfer increases with an increase of the initial twist of the flow, apex angle of the cone, and hub ratio of the swirler (with a decrease of the relative height of the annular jet). Whereas the effect of swirling can be explained by an increase of the velocity gradient across the wall boundary layer [15, 19], the same such effect of the other two factors is due to a change of the aerodynamic structure of the flow during gradual transition from the flow at the initial section of the duct ($\beta = 0$) to a semibounded jet flow ($\beta = 1$) [20].

The range of applicability of the relationships obtained is limited to values of $Re_x = (0.8\text{-}10)\cdot 10^5$, though the experiments were carried out and the data processed in wider limits. This is due to the fact that when $Re_x < 0.8\cdot 10^5$ a departure from the regularities of turbulent heat transfer (Fig. 2c) is observed in certain series of experiments. We note that the lower limit of applicability of the formulas is less than the corresponding value of Re_x for a turbulent flow regime along a flat plate, which is explained, first, by the increase of the degree of turbulence during propagation of the swirling flow under conditons of the internal problem [21, 22] and, second, by the artificial decrease of the actual values of the Re_x number due to choosing only the axial component of velocity rather than the total velocity as the characteristic velocity.

It must be emphasized that an increase of φ_0 and β leads simultaneously to a decrease of the degree of attenuation of the characteristic velocity along the flow, whereas an increase of ψ causes the opposite effect. As a result a marked drop of the characteristic velocity leads to a decrease of the heat-transfer coefficient despite an increase of transverse transfer upon an increase of the apex angle of the cone.

It should be noted that an excess of the angle of twist of the jet over the apex angle of the cone figures as arguments in (11) and (12). This value is selected on the basis of analyzing the data of an aerodynamic investigation and permits generalizing the experimental data on local heat transfer both for the case $\varphi_0 > \psi$ and for the case of preseparation flow past a wall when $\varphi_0 < \psi$.

NOTATION

D, d, 2ψ, l_{co}, maximum and minimum diameters, apex angle, and length of cone; D_o, D_{in}, b_o, $\beta = (D_{in}/D_o)^2$, outside and inside diameters, vane height, and relative area of swirler hub; φ_0, initial angle of twist of jet; α_{ef}, effective exit angle at the average diameter; n_v, number of swirler vanes; x, $\bar{x} = x/b_o$, distance from swirler exit along surface exposed to the flow; d_c, f_c, $x_0 = x - d_c/2$, $x_1 = x + d_c/2$, diameter and area of measuring surface of calorimeter and coordinates characterizing its location on the surface exposed to the flow; $T_c = t_c + 273$, T_o, T_{in}, absolute temperatures of the calorimeter surface, outside air, and internal air flow; W_{x0}, W_{φ_0}, W_{xm}, axial and tangential velocities at the swirler exit, maximum axial velocity; Q_o, Q_o', Q_{in}, Q_l, heat released by the electric heater, heat losses due to conduction and radiation through the outside and inside surfaces, total heat losses; E, electromotive force; α, λ, and ν, coefficient of heat transfer, thermal conductivity, and kinematic viscosity; $Nu_x = \alpha x/\lambda$, $Re_x = W_{x0}x/\nu$, $Re_o = W_{x0}2b_o/\nu$, Pr, Gr, similarity numbers; n_c, c_c, c_m, a_m, a_c, $\bar{c}_m = c\ /0.025\alpha_m[1 + \tan(\varphi_0 - \psi)]^{0.218}$, $\bar{c}_c = c_c/0.026[1 + \tan(\varphi_0 - \psi)]^{1.6}$, experimental coefficients.

LITERATURE CITED

1. N. N. Kaura, Indian J. Technol., 8, No. 3, 107 (1971).
2. I. L. Batson and R. H. Gforzini, J. Spacecraft Rockets, 7, No. 2, 159 (1970).
3. H. F. Renaud, Compt. Rend. Acad. Sci., 270, No. 220 (1970).
4. A. P. Nazarchuk and N. S. Repalov, "Hydroaeromechanics and elasticity theory," in: Republic Interdepartmental Scientific-Technical Collection, No. 11 [in Russian] (1970).
5. A. T. McDonald, R. W. Fox, and R. V. Dowolstine, AIAA Paper, No. 84 (1971).
6. V. K. Shchukin, P. P. Polikarpov, V. A. Filin, and A. P. Yakshin, in: Transactions of the Kazan Aviation Institute, No. 154 [in Russian] (1973).
7. G. S. Ambrok, Zh. Tekh. Fiz., 27, No. 9 (1957).
8. V. M. Kapinos and N. N. Nikitenko, Teploénergetika, No. 8 (1963).
9. H. Schlichting, Boundary Layer Theory, McGraw-Hill, New York (1960).
10. L. N. Sedov, Similarity and Dimensionality Methods in Mechanics [in Russian], Gostekhizdat, Moscow (1957).
11. R. R. Long, Quart. J. Appl. Math., 9, Part IV, 385 (1957).
12. V. N. Kalashnikov, Yu. D. Raiskii, and L. E. Gupkel', Izv. Akad. Nauk SSSR, Mekh. Zhidk. Gaza, No. 1 (1970).

13. I. S. Allen, J. Fluid. Mech., 46, Part I (1971).
14. Johnston, Theoretical Principles of Engineering Calculations, No. 2 [Russian translation] (1973), p. 131.
15. V. K. Shchukin, Heat Transfer and Hydrodynamics of Internal Flows in Mass Force Fields [in Russian], Mashinostroenie, Moscow (1970).
16. Z. Lavan, H. Nielsen, and A. A. Fejer, Phys. Fluids, 12, No. 9, 1747 (1969).
17. A. V. Sudarev, in: Transactions of the Leningrad Ship-Building Institute, No. 14 [in Russian] (1968).
18. E. G. Narezhnyi, Author's Abstract of Candidate's Dissertation, Leningrad Ship-Building Institute, Leningrad (1958).
19. A. V. Sudarev, in: Transactions of the Leningrad Ship-Building Institute, No. 7 [in Russian] (1967).
20. A. V. Sudarev, Énergomashinostroenie, No. 9 (1967).
21. F. Kreith, Trans. ASME, 77, No. 8 (1958).
22. H. Geh, Paper ASME No. SA-22 (1957).

PROBLEMS OF HEAT AND MASS TRANSFER IN A VAPOR–GAS PHASE DURING EVAPORATION OF A FLUID

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UDC 536.423.1

An analysis is made of the experimental data of a number of investigators on evaporation of fluid from a surface and a comparison is made with theoretical solutions and data on the injection of gas through a permeable surface.

It has been shown [1] that in the relative representation of dimensionless coefficients of heat and mass transfer as functions of permeability factors, the experimental data on evaporation from the free surface of a fluid correlate well with the corresponding theoretical solutions, particularly those based on the asymptotic laws of Kutateladze and Leont'ev [2] for a turbulent boundary layer at a permeable surface. The experimental data of a number of investigators on evaporation and injection are shown in Fig. 1 along with some theoretical solutions. Figure 1 indicates that all the data are sufficiently alike for a quasiuniform boundary layer and Le = 1 (the latter limitation is only for heat transfer) where the permeability factor B < 0.1. The coefficients of heat and mass transfer decrease as the mass flow from the surface increases, which is in accord with presently accepted views [3]. It should be noted that for evaporation it is necessary to use a Nusselt (or Stanton) number based on diffusion mass flow [4]. For evaporation of a fluid in a vapor–gas medium, it is $Nu_D W_{2i}$ ($St_D W_{2i}$).

As is clear from Fig. 1, we have the relations $Nu_D W_{2i} = Nu_{D_0}$ and $Nu = Nu_0$ when $B < 0.1$, i.e., a similarity is observed between jointly occurring processes of heat and mass transfer at low intensity and heat transfer without mass transfer. Here Nu_{D_0} and Nu_0 are the diffusion and thermal Nusselt numbers defined by the usual similarity equations such as $Nu_{(D)0} = f(Re, Pr_{(D)}, Ar)$ for separately occurring processes of mass transfer at low intensity and heat transfer without mass transfer. Experimental data confirming this hypothesis are presented in [5, 6]. However, several authors have expressed the opinion that the analogy between heat and mass transfer can break down during evaporation even at low intensity. In this regard, references are made to the experimental studies of Nesterenko [7], who was one of the first to perform sufficiently accurate measurements of temperature and concentration within a boundary layer. However, if one considers the original data of Nesterenko, breakdown

*Deceased.

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