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HEAT TRANSFER IN A WALL JET PROPAGATING IN A HIGHLY TURBULENT COCURRENT FLOW

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Heat transfer in a jet propagating in a cocurrent flow has been studied over wide ranges of the injection ratio ($m = U_s/U_0 < 1$ and $m > 1$) and flow turbulence ($Tu_0 = 0.2$–25%). It is shown experimentally that for $m < 1$, a 1% increase in turbulence leads to a 1% increase in heat transfer, and the wall adiabatic temperature and the relative heat-transfer function should be taken into account in heat-transfer calculations. For $m > 1$, the flow turbulence does not affect the heat transfer and the heat production can be calculated using the laws typical of jet flows.

Wall jets are frequently used in modern equipment and technologies. One important problem is to determine the heat transfer between the gas flow and the channel wall in order to determine the limiting temperatures and heat fluxes on the surface. The heat transfer in wall jets has been thoroughly studied for low-turbulence flow [1, 2] where the turbulence level does not exceed 4–5%. However, in real devices, the heat transfer often occurs at a high rate of velocity fluctuations. Thus, in the combustion chambers of gas-turbine and rocket engines, the turbulence can reach 30–40%.

The effect of slot turbulence on wall-jet propagation was studied in a number of papers [3–5]. It was found that the effect of turbulence extends mainly within the limits of the initial section. The heat transfer in the wall jet with elevated external turbulence ($\mathrm{Tu}_0 = 9.3\%$) was considered by Glazkov et al. [6]. A significant (up to 70%) effect of external turbulence on the initial part of the jet and a weak downstream effect were observed. The experiments of [6] were conducted under conditions where the main flow was the initial part of the jet. The laws of turbulence degeneration for jets, however, are significantly different from those for channels. The heat-transfer coefficient was determined for the injected-gas temperature equal to the main-flow temperature. The effect of flow turbulence on heat transfer in a cocurrent wall jet was numerically calculated by Spalding [7]. A steady two-dimensional problem with a modified model of the mixing length was solved using the method of Patankar–Spalding. For a maximum turbulence of 20%, the heat transfer increases by 70–100% ($U_s/U_0 = 0.5$ and $T_s/T_0 = 0.3$), which indicates a significant effect of flow turbulization on heat transfer. The maximum effect of the turbulence intensity on heat transfer for flows on a flat plate and in a pipe is known to be 20–50%.

The heat-transfer coefficient for gas injection into a boundary layer is determined from the difference between the wall temperature under heat-exchange conditions T_w and the adiabatic wall temperature for gas injection T_{wa}:

$$\alpha = q_w/(T_w - T_{wa}). \tag{1}$$

The integral energy relation for a boundary layer with injection is the same as for the case without injection if the energy loss thickness δ_h^{**} and the Stanton number $\mathrm{St} = \alpha/\rho_0 c_p U_0$ are determined with allowance for the adiabatic wall temperature. In this case, the experimental results are described by a power law of heat transfer [1]. Glazkov [6] obtained data on heat transfer for $T_s = T_0$, and Spalding [7] found the Stanton number from

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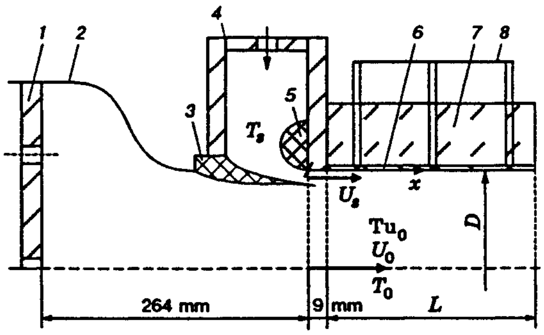

Fig. 1

the difference between the wall temperature T_w and the flow temperature T_0. Thus, there are no data on heat transfer that take into account the adiabatic wall temperature.

At the same time, it is important to know whether the power law of heat transfer remains conservative under conditions of elevated turbulence. Precisely these questions are considered in the present paper, which gives results of an experimental study of the influence of the initial turbulence of the main flow on the heat exchange between a cocurrent wall jet and a solid wall.

A sketch of the test section is shown in Fig. 1. The experiments were conducted in a cylindrical channel 6 (diameter $D = 80$ mm, length $L = 250$ mm, and wall thickness 2 mm). A wall jet was generated by supplying air from the injection chamber 4 through an annular tangent slot of height $s = 2$ mm. Separator 3 is made of caprolon, and shield 5 is made of textolite. The main-flow parameters are as follows: velocity $U_0 = 15$ m/sec, Reynolds number $\mathrm{Re}_0 = \rho_s U_0 D/\mu_0 = 8 \cdot 10^4$, temperature $T_0 \approx 300$ K, and turbulence level $\mathrm{Tu}_0 = 0.2$–20%. The wall-jet parameters are velocity $U_s = 3$–30 m/sec, Reynolds number $\mathrm{Re}_s = \rho_0 U_s s/\mu_s = 700$–6700, temperature $T_s \approx 363$ K, and turbulence level $\mathrm{Tu}_s = 5$–7%. The jet/flow velocity ratio is $m = U_s/U_0 = 0.2$–2. The turbulence generator (a perforated washer) 1 was mounted in the plenum chamber of the wind tunnel ahead of confusor 2 with a contraction ratio of 7.2, and the distance from the turbulizer to the test section was 264 mm. The turbulence level in the main flow was 7, 12, 15, and 20% for the number of orifices in the turbulizer 25, 13, 7, and 4, respectively (the orifice diameter was 14 mm). The turbulence was 0.2% if a fine grid was placed instead of the perforated washer. The velocity profile in the boundary layer of the main flow in the injection cross-section was approximated by a power law: the displacement thickness was $\delta^* = 0.37$ mm and $1/n = 1/7$ for $\mathrm{Tu}_0 = 0.2\%$, and $\delta^* = 2.6$ mm and $1/n = 1/14$ for $\mathrm{Tu}_0 = 20\%$. At the exit from the annular slot, the velocity profile was parabolic, and the boundary layer thickness was half the slot height.

In the experiments, the condition $q_w = \text{const}$ was satisfied, the heat flux was generated by electric heating of the channel wall, and the maximum heat-flux density was 5000 W/m^2. The test channel was made of stainless steel and coated with a layer of heat insulator 7. To reduce the streamwise heat overflow, grooves 1 mm deep were made in several cross sections along the perimeter over the cylinder length. The temperature was measured by Chromel–Copel thermocouples 8 made of a 0.2-mm diameter wire. The thermal e.m.f. was measured by an F30 voltammeter with an error of 0.1%. The linear dependence of the temperature on the thermal e.m.f. was approximated within 0.15°C using calibration data. The standard deviation of the temperature measurement within the range of 15–100°C was 3–0.3%. The power released by the electric heater was measured by a D57 wattmeter of accuracy class 0.1. The heat flux losses due to radiation and heat transfer through the insulator and the end surfaces of the channel were 6%. In determining St, a correction for the anisothermal character of the flow was introduced in accordance with [1]. The standard deviation of the measured heat-transfer coefficient was estimated as 5–7%, and that of the Stanton number was estimated as 6–8%.

The turbulence characteristics were determined by an automated complex based on a DISA 55M constant-temperature hot-wire anemometer [8]. The standard deviation of the turbulence level was estimated in [8] as 5–13% for the range Tu = 0.2–20%. The greatest contribution was made by the instability of the

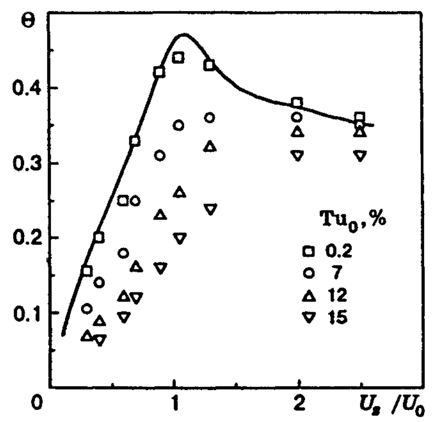

Fig. 2

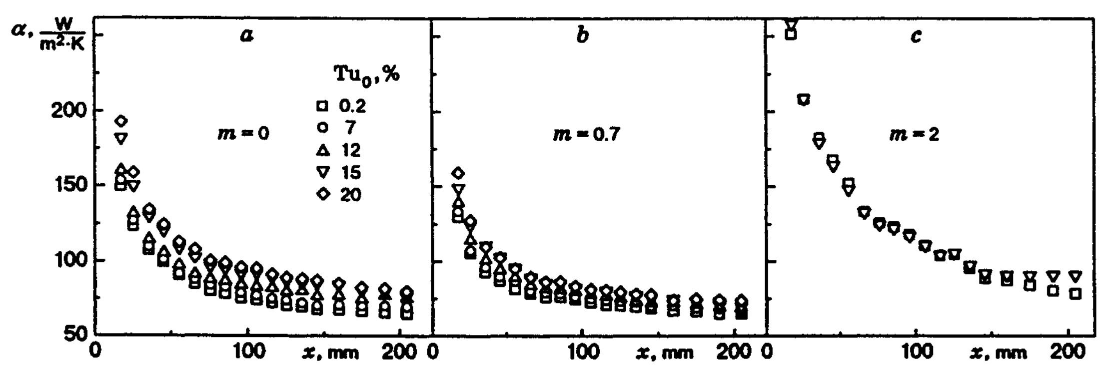

Fig. 3

calibration curve of the probe, the high turbulence level, and imperfection of the measurement equipment.

It was established [9, 10] that the turbulence intensity of external flow significantly affects the adiabatic surface temperature in the development of wall jets in channels. The quantitative effect of external turbulence on the mixing of the wall jet and the main flow is determined by the parameter m. Experimental data on screen efficiency are shown in Fig. 2. It is seen that in low-turbulent flow the efficiency $\Theta = (T_{wa} - T_0)/(T_s - T_0)$ increases monotonically as the secondary flow injection velocity increases to $m \approx 1$, for which the efficiency reaches a maximum value. As the parameter m increases further, the protective gas film efficiency decreases and asymptotically approaches the value for $m \approx 0.6$. The solid curve in Fig. 2 shows the calculated dependence obtained for low-turbulent flow [2] for $K_1 = (\Delta x/ms)\mathrm{Re}_s^{-0.25} = 14$, where $\Delta x = x - x_0$, x is the longitudinal coordinate, and x_0 is the length of the initial thermal section. It is seen that the experimental results for $\mathrm{Tu}_0 = 0.2\%$ are in good agreement with the calculated data of [2].

Elevated external turbulence significantly deteriorates the protective properties of the gas film over a wide range of m (compared to the data for $\mathrm{Tu}_0 = 0.2\%$). In addition, the behavior of the gas film in high-turbulent flow is qualitatively different: the gas film efficiency increases monotonically with increase in m over the entire measured range of the secondary flow injection velocity. Even for $m > 1$, in contrast to low-turbulent flow, the gas film efficiency continues to grow, approaching asymptotically the value of Θ for $\mathrm{Tu}_0 = 0.2\%$. Beginning from $m = 2$–2.5, the gas film efficiency for the turbulized external flow practically ceases to depend on the injected gas rate. Therefore, a further increase in the flow rate of the gas supplied for cooling does not lead to significant changes in protective properties of the gas film and, hence, these regimes are noneconomical in energy.

The effect of the flow turbulence level on the heat-transfer coefficient determined from formula (1) taking into account the wall adiabatic temperature is shown in Fig. 3 for three values of the injection ratio

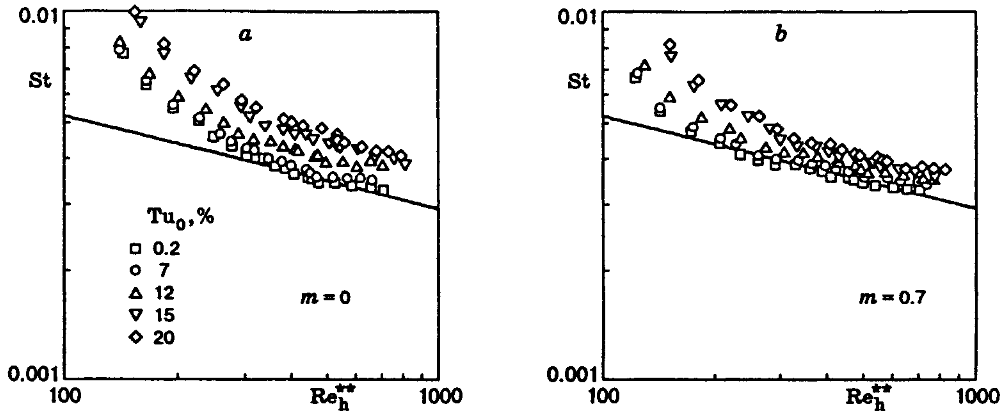

Fig. 4

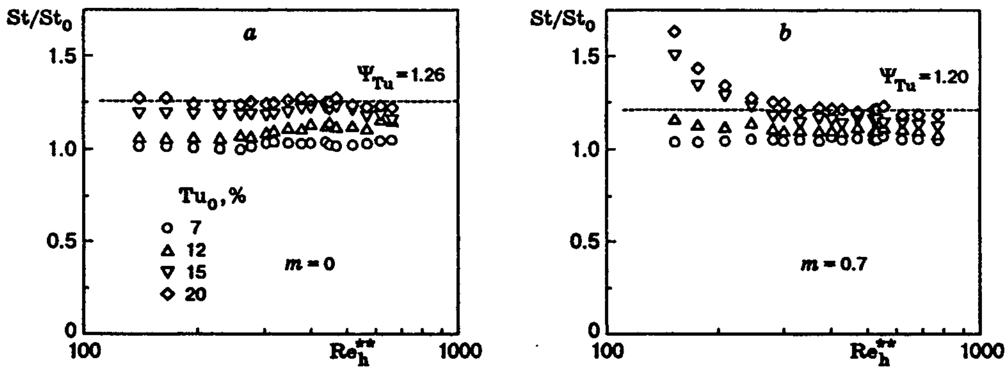

Fig. 5

(x is the streamwise coordinate reckoned from the injection cross-section). The experiments were performed in two stages. Initially, a wall jet of fixed m at temperature T_s was injected into a flow at temperature T_0, and the wall adiabatic temperature T_{wa} was measured. After that the cylinder was heated, and the wall temperature T_w was measured. The efficiency parameter Θ determined from T_{wa} agrees with experimental data obtained for a cylinder with heat-insulated walls [10] within the examined range of Tu_0. It follows from Fig. 3 that the flow turbulization increases the heat-transfer coefficient by 20–30% only for $m < 1$. If the jet velocity is higher than the main flow velocity ($m = 2$), the heat-transfer coefficient is almost independent of the flow turbulence intensity. This indicates that the turbulence induced by the wall jet acts as a buffer that prevents penetration of vortices generated by the main flow.

The effect of the main-flow turbulence on the heat-transfer law in the wall jet for $m < 1$ is shown in Fig. 4. In the experiments, the Stanton number was determined with allowance for formula (1), and the Reynolds number ($Re_h^{**} = \rho_0 U_0 \delta_h^{**}/\mu_0$) was found from the integral equation of energy conservation according to the formula $Re_h^{**} = \alpha x / c_p \mu_0$. The solid curve in Fig. 4 shows the power law of heat transfer for the wall turbulence [1]:

$$St = 0.0128 Re_h^{**\,-0.25} Pr^{-0.75}. \qquad (2)$$

For low-turbulent flow, the experiment is described by formula (2). Analysis of the experimental data for $m = 0$ and $Tu_0 = 0.2\%$ shows that the difference between the experimental and theoretical values of St is less than 5% ($Re_h^{**} > 250$). This value can serve to estimate the error of determining the Stanton number. The deviation of the experimental points from (2) for $Re_h^{**} < 250$ is due to flow prehistory. The change of the turbulence level from 0.2 to 20% increases the St by 24–28% for $m = 0$ and by 18–22% for $m = 0.7$.

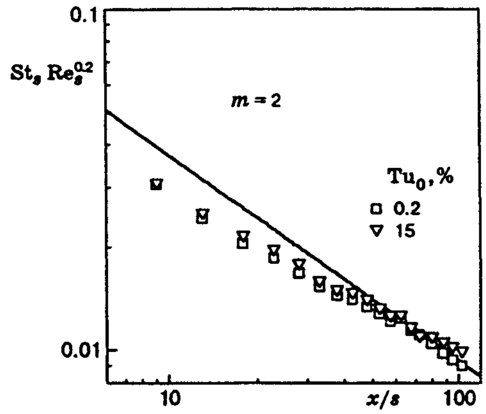

Fig. 6

A 18–28% the increase in the St for a 20% turbulence level is considerably higher than the measurement error of the Stanton number. The heat-transfer data near the screen ($m = 0.7$) are stratified with respect to the parameter Tu_0, although the heat-transfer law (2) takes into account the wall adiabatic temperature. Thus, although the St and Re_h^{**} in the heat-transfer law (2) are determined with allowance for T_{wa}, this is insufficient to generalize the experimental data with variation in the flow turbulence, and one should use the relative heat-transfer function which takes into account the flow turbulence.

The effect of external turbulence on the relative heat-transfer function $\Psi = (St/St_0)$ for Re_h^{**} = idem and two regimes is illustrated in Fig. 5. In the experiments, St_0 is the Stanton number for a 0.2% turbulence level. It is seen from the figure that a 20% increase in the turbulence intensity increases Ψ by 26% for $m = 0$ and by 20% for $m = 0.7$. In the region unaffected by initial conditions, the relative heat-transfer function does not depend on Re_h^{**}. Hence, in external turbulence, the relative heat-transfer law can be approximated by the function [1]

$$\Psi_{Tu} = (St/St_0)_{Re_h^{**}=\mathrm{idem}} = 1 + a Tu_0, \tag{3}$$

where $a = 0.013$ for $m = 0$ and $a = 0.01$ for $m < 1$.

It is known that for high injection ratios ($m > 1$), the jet behavior is the governing factor. The effect of elevated turbulence on heat transfer in this case is shown in Fig. 6, where $St_s = \alpha/\rho_s c_p U_s$ [α is determined from formula (1)]. The solid curve corresponds to the heat-transfer law for a wall jet [1]:

$$St_s = 0.12 Re_s^{-0.2}(x/s)^{-0.6}Pr^{-0.6}. \tag{4}$$

In this case, the turbulence intensity does not affect the heat-transfer law. The conservatism of the heat-transfer law (4) with respect to the external flow turbulence corresponds to the data on the efficiency for $m > 2$ (see Fig. 2). The free-stream turbulization for $m > 2$ affects only weakly both the cooling efficiency of the gas film and the heat-transfer law. In this case, the energy of the turbulent moles of the flow becomes significantly smaller than the kinetic energy of the wall jet. The near-wall boundary layer is fairly stable to free-stream disturbances, and the heat transfer is determined by the jet behavior.

The effect of cocurrent flow turbulence on the thermal mixing with the wall jet is ambiguous. On the one hand, turbulization significantly affects the mixing layer and can lead to more than a twofold change in the dimensionless temperature of the adiabatic wall. On the other hand, the near-wall heat transfer is more conservative, and its variation does not exceed 20–30%. The effect of the free-stream turbulence on heat transfer is significant for velocity ratios $m < 1$. In this case, the effect of turbulence on heat transfer should be taken into account through the wall adiabatic temperature in the heat-transfer law for near-wall processes (2) and the relative heat-transfer function in the form of (3). For $m > 1$, the heat transfer can be calculated by formula (4), which is typical of wall jets.

The effect of the main-flow turbulence and the injection ratio on heat transfer in the wall jet was studied in the present paper. As shown by Kutateladze and Leont'ev [1], the use of the power heat-transfer law (2) and the relative function Ψ for the near-wall turbulence allows one to take into account the joint action of several factors, for example, the streamwise pressure gradient, anisothermic character, and compressibiity of the flow. For injection ratio $m < 1$, for which near-wall turbulence laws prevail, the use of the relative function (3) helps to estimate the heat-transfer level for a wider class of turbulent flows (for example, for high-turbulent supersonic flow). It is known that the turbulence is characterized by several parameters, among which the turbulence intensity and the integral scale are the most important. From this viewpoint, the effect of the initial integral turbulence scale (it was 5–10 mm for the main flow) on heat transfer in the wall jet should be studied.

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REFERENCES

1. S. S. Kutateladze and A. I. Leont'ev, *Heat Transfer, Mass Transfer, and Friction in Turbulent Boundary Layer*, Hemisphere, New York (1990).
2. É. P. Volchkov, *Wall Gas Films* [in Russian], Nauka, Novosibirsk (1983).
3. S. C. Kacker and J. H. Whitelaw, "The effect of slot height and slot-turbulence intensity on the effectiveness of the uniform density, two-dimensional wall jet," *Trans. ASME, Ser. C, J. Heat Transfer*, **90**, No. 4, 469–484 (1968).
4. G. I. Sadovnikov, B. M. Smol'skii, and V. K. Shchitnikov, "Heat transfer on a flat plate in a wall jet with different initial turbulence levels," in: *The Study of Transport Phenomena in Complex Systems* [in Russian], Minsk (1974), pp. 81–91.
5. G. J. Sturgess, "Account of film turbulence for predicting film cooling effectiveness in gas turbine combustors," *Trans. ASME, Ser. A, J. Engin. Power*, **102**, No. 3, 524–534 (1980).
6. V. V. Glazkov, M. D. Guseva, B. A. Zhestkov, and V. P. Lukash, "Effect of the initial turbulence on the efficiency of cooling of permeable walls," *Inzh.-Fiz. Zh.*, **36**, No. 6, 965–971 (1979).
7. D. B. Spalding, "Boundary layer theory applied to film cooling processes," *Prog. Heat Mass Transfer*, **4**, 279–296 (1971).
8. V. V. Lemanov and S. Ya. Misyura, "Two-dimensional turbulent flow measurements by an automated hot-wire anemometer," *Izv. Sib. Otd. Akad. Nauk SSSR, Sib. Fiz.-Tekh. Zh.*, No. 3, 112–115 (1991).
9. V. P. Lebedev, V. V. Lemanov, S. Ya. Misyura, and V. I. Terekhov, "Efficiency of protective gas film with high injection ratios in a highly turbulent main flow," *Prikl. Mekh. Tekh. Fiz.*, **35**, No. 1, 48–52 (1994).
10. V. P. Lebedev, V. V. Lemanov, S. Ya. Misyura, and V. I. Terekhov, "Effect of flow turbulence on film cooling efficiency," *Int. J. Heat Mass Transfer*, **38**, No. 11, 2117–2125 (1995).