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Effects of flow turbulence on film cooling efficiency

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Abstract—Results of experimental investigations on film cooling effectiveness in high-turbulent flow are represented in this paper. The turbulence intensity changed within the range $Tu_0 = 0.2$–15% and the injection parameter was $0.1 < m < 2.5$. It has been established that at small injection parameters ($m < 1$) the high-turbulence intensity of flow results in a significant decrease (down to 400–500%) of the film cooling effectiveness compared to low-turbulence flow. At big injection parameters ($m > 1$) the turbulence effect weakens. The experimental data generalization and its comparison with the results of the calculation carried out by asymptotic theory are given.

INTRODUCTION

Working surfaces of many power plants experience the influence of high-temperature gas flows. Gas film cooling is widely used to create the film cooling of walls, with this a cooling gas is supplied along a streamlined surface. Wall cooling intensity is characterized by a gas film cooling effectiveness, which is determined by the distribution of the relatively non-dimensional temperature of an adiabatic wall from the place of a coolant supply downstream. The gas film cooling effectiveness for low-turbulence flows is well studied at the present time [1–7]. However, in practice when we deal with actual equipment we often observe high-turbulence flow. Thus, the turbulence intensity reaches 35% [8] in airflow wind tunnels of combustion chambers and in turbines of gas-turbine setups.

The flow turbulence significantly affects the heat and impulse transfer processes in near-wall flows. The data on the influence of high-intensity turbulence on cooling properties of gas film which are available in the literature are scanty and contradictory. In several works it is noted that the turbulence effect is strong. However, in others we run into the opinion that it is negligible. Thus, in ref. [9] the slot film cooling effectiveness in a high-turbulence flow decreases more than two-fold, and in refs. [10–11] it diminishes by less than 30%. In our early experiments [12], carried out with small injection parameters $m < 1$, we established the strong effect of external turbulence on distribution of the adiabatic surface non-dimensional temperature. The reasons for the discordance in experimental data may be explained by different conditions while performing the experiments. For example, in ref. [9] a wall made of a metal tube was used as a surface; the injection of coolant was carried out near a turbulizing grid where a uniform flow was less likely to be formed; and the turbulence intensity during experiments was determined by a calculation method. The contradictory results may also be conditioned by different initial turbulence intensities. In refs. [9–11] there are no complete data on mean and pulsation characteristics in the area ahead of a working channel and on its outlet, which also does not allow an accurate comparison of results obtained by different authors. It should be noted here that all the investigations concerning film cooling in high-turbulence flows have been carried out at small injection parameters ($m < 1$). At the same time it is known that gas dynamics of a wall co-current jet flow for $m < 1$ is greatly different from that for $m > 1$.

It is known [1] that a greater cooling effectiveness in low-turbulence flow is reached when the wall jet velocity is equal to the flow velocity ($m = 1$). In this case we observe a regime of minimum mixing between the flows. A gas flow rate decreases, and a velocity gradient on a jet boundary increases with the diminution of the injection parameter ($m < 1$). Consequently, a drop in the efficiency value is observed. With the increase in cooling gas supply ($m > 1$) the intensification of mixing processes between a wall jet and a free stream due to the velocity gradient growth starts playing a greater role, which also results in an efficiency drop.

It is evident that the mechanism of the high-turbulent pulsation effect on the process of flow mixing at big injection parameters may be significantly different from a flow at small injections. For $m < 1$ a velocity profile in a wall layer accepts a power law quite quickly. At $m > 1$ a wall jet develops in the vicinity of the wall, and a velocity profile has the maximum bend near the wall. Thus, at $m < 1$ there are two characteristic areas—a wall area with a power velocity profile and an external jet area. Undoubtedly,

NOMENCLATURE

- d diameter of opening in a turbulator
- D_1 diameter on a confusor inlet
- D_2 diameter on a confusor outlet
- E voltage of current
- k air flow compression factor in a confusor, D_1/D_2
- K generalizing parameter, $(Re_{\Delta x}/Re_S^{1.25})(\mu_0/\mu_s)^{1.25}$
- L integral turbulence scale
- M power index in equation (2)
- m parameter of gas injection, $\rho_s u_s/\rho_0 u_0$
- N power index in equation (1)
- n power index for velocity profile approximation
- $Re_{\Delta x}$ Reynold's number for free stream, $\rho_0 u_0 \Delta x/\mu_0$
- Re_s Reynold's number for a secondary air flow, $\rho_s u_s s/\mu_s$
- Re_T^{**} Reynold's number, built from a thickness of energy loss, $\rho u \delta_T^{**}/\mu$
- St_0 Stanton criterion in standard conditions, $\alpha_0/\rho_0 u_0 c_p$
- St Stanton criterion with a disturbing factor, $\alpha/\rho u c_p$
- s height of a slot
- T_w temperature of adiabatic wall
- T_0 temperature of free stream
- T_s temperature of a secondary air flow
- Tu_0 turbulence intensity in free stream, $\langle u'^2\rangle^{1/2}/u_0 \cdot 100\%$
- u_0 velocity of a gas free stream
- u_s velocity of a secondary gas flow
- u velocity of air
- x longitudinal coordinate measured from a slot cross-section
- x_0 initial section where $\Theta = 1$
- Δx longitudinal coordinate, x–x_0
- x' longitudinal coordinate with the beginning at the place of the turbulator location.

Greek symbols

- β coefficient of deformation temperature field in boundary layer, $(\delta_T^{**})_a/(\delta_T^{**})_{T=\text{const.}}$
- δ^* displacement thickness
- $(\delta_T^{**})_a$ thickness of energy loss on adiabatic surface
- $(\delta_T^{**})_{T=\text{const.}}$ thickness of energy loss, on a wall with heat transfer and constant temperature
- μ dynamic viscosity of air
- Θ effectiveness of gas film cooling, $(T_w - T_0)/(T_s - T_0)$
- ρ density of air
- Ψ relative function of heat transfer with Re_T^{**} = idem, $(St/St_0)_{Re_T^{**}}$.

Subscripts

- a adiabatic conditions
- e effective length
- s parameters on a slot cross-section of a secondary flow injection
- w parameters on the wall
- 0 free stream, standard conditions, initial area.

such a difference must affect the character of the external pulsation penetration to the wall and, correspondingly, the heat transfer processes.

The purpose of this work is to study the high-intensity turbulence effect on the gas film cooling effectiveness in a broad range of gas injection parameter change.

EXPERIMENTAL EQUIPMENT AND SETUPS

The experiments have been performed in an aerodynamic tube which is shown in Fig. 1. A turbulizing grid (4) was placed directly in front of the confusor (5) that smoothed the nonuniformities of the current. The factor of flow compression in the confusor was equal to $k = 7.2$. The working channel was made of textolite, its inside diameter was 80 mm, and its length was 320 mm. A thickness of cylindrical section walls was 20 mm. In order to arrange film cooling in the setup, a chamber of the injected gas supply was provided. On the working section inlet there is an annular slot with a height of $s = 2$ mm through which a gas wall jet is introduced. The slot-lip separating the flows is made of caprolone, the thickness of the separating edge is 0.2 mm. The working gas is air.

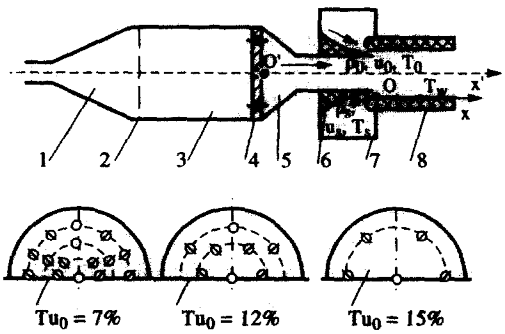

Fig. 1. Scheme of an experimental setup for investigation of cooling effectiveness in a high-turbulent flow: 1—inlet diffusor; 2—smoothing grid; 3—setting chamber; 4—turbulizing grids; 5—confusor; 6—separating slot-lip; 7—injection chamber; 8—experimental cylindrical section.

The experiments have been carried out within the following range of parameters: the velocity of free stream is $u_0 = 15$ m s^{-1}, the parameter of the secondary air flow injection $m = 0.2$–2.5, turbulence intensity of a running flow on the working section inlet varied within the range $Tu_0 = 0.2$–15%.

When determining film cooling effectiveness a gas, injected through the slot, was heated with the purpose of decreasing the errors in the experiment and arranging its performance in the most convenient way. A free stream was supplied at the environmental temperature. As numerous experiments, carried out by different authors, have shown, such an 'inverse' problem gives the same result for a nondimensional parameter of film cooling effectiveness as cold gas injection. In connection with the noted salient features, the temperatures of the free stream and injected gas were $T_0 = 292$ K, $T_s = 360$ K. The measurement of the internal flow temperature was performed by thermocouples placed on the axis of the setting chamber. The temperature of the secondary flow was found by thermocouples placed on the injection chamber outlet, and the temperature of the channel wall was measured by 19 chromel–copel thermocouples made of a wire with a diameter of 0.2 mm and placed flush with the surface.

Dynamic characteristics of the air flow were measured by a hot-wire anemometer of constant temperature DISA 55M with a standard bridge of 55M10 and a single-wire probe 55P11 with a wire diameter of 8 μm. The wire was placed perpendicular to the flow direction. To go from electric magnitudes to a velocity value we used the dependence [13]

$$E^2 = A + B \cdot u^N \tag{1}$$

where coefficients A and B were found by a least-squares fit from the experimental values of velocity u and voltage E. The performed analysis showed that an error in the value E^2 determined from the dependence (1) with power indices $N = 0.5$ (King's law) and $N = 0.4$–0.6 is the same. In this connection we used King's dependence for calibration.

Collection and processing of experimental data on measuring the temperature, mean velocity, turbulence intensity and other characteristics were carried out with the aid of the automated system [14].

RESULTS OF EXPERIMENTS AND DISCUSSION

First, the dynamic characteristics of the flow on the working section inlet were studied. With this purpose we measured the velocity field and the velocity pulsation field of the free stream with low-intensity turbulence in the slot cross-section. The measurements showed a good uniformity of these fields (the maximum nonuniformity did not exceed 2%). In this cross-section we measured velocity profiles in the boundary layer, which were adequately described by the power dependence with an index $1/n = 1/7$ characterizing the developed turbulent boundary layer. The thickness of the free stream boundary layer displacement in the slot cross-section constituted $\delta^* = 0.37$ mm.

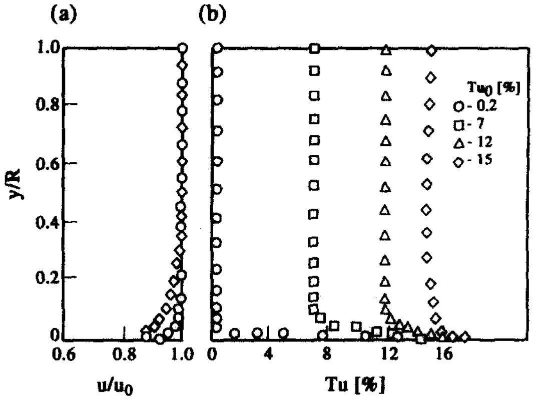

Fig. 2. Profiles of average velocity (a) and turbulence intensity (b) on the working channel inlet.

The high-intensity turbulence of the free stream was created by turbulence generators in the form of disks with openings. The scheme of turbulators is given in Fig. 1. The turbulators were manufactured according to the recommendations given in ref. [8]. The thickness of the disk is 4 mm, the diameter of the opening is $d = 10$ mm. Probably the two characteristic dimensions (the diameter of the opening and the distance between the openings) are decisive factors in the high-intensity turbulence generation. A different turbulence intensity was reached by changing the number of openings in the turbulator. There is no doubt that the confusor considerably deforms the largest vortices, affecting the turbulence scale change. The maximum turbulence intensity along a longitudinal component of velocity pulsations reached 15% on the working section inlet, in spite of the intensive suppression of pulsations in the confusor.

Thanks to the flow compression in the confusor, there are uniform profiles u_0 and Tu_0 (Fig. 2) in the slot cross-section. According to the results of the velocity pulsation longitudinal component measurements on the slot cross-section, a spectrum of turbulence power was built and the local maxima were absent. Based on the obtained power spectrum, the integral longitudinal turbulence scale L was determined. At $Tu_0 = 7$–15%, $u_0 = 15$ m s^{-1}, $L = 6$–10 mm, which agreed with the characteristic dimensions of the turbulators.

The character of the turbulence decay along the channel axis is given in Fig. 3. The initial part of the axis x' corresponds to the turbulator location (see Fig. 1). The value of $x'/s = 132$ corresponds to the slot cross-section coordinate in Fig. 3. A change of turbulence intensity on the axis along the channel wall in the flow with a different initial Tu_0 is shown at $u_0 = 15$ m s^{-1}: $Tu_0 = 15\%$ (seven openings in the turbulator); $Tu_0 = 12\%$ (15 openings); $Tu_0 = 7\%$ (25 openings). The light points describe a turbulence intensity change

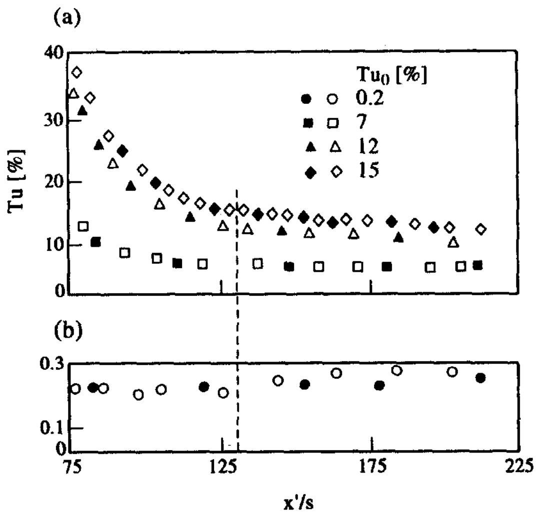

Fig. 3. Distribution of turbulence intensity with a high (a) and low (b) initial turbulence intensity along the channel axis. Light points—without jet injection, dark points—with a wall jet supply ($m = 2$).

without injection of a secondary flow, the dark points characterize a turbulence intensity change with injection at $m = 2$. It is seen that a considerable turbulence intensity decay takes place ahead of the slot cross-section. Further, Tu_0 changes little along the channel length. As is seen from the picture, a gas injection at $m = 3$ does not actually affect the turbulence decay on the channel axis. More detailed data on the behaviour of pulsation velocity components along a length of the confusor section and the channel is given in ref. [15].

The minimum turbulent intensity ($Tu_0 = 0.2\%$, Fig. 3, lower graph) was achieved by replacing the turbulator with a net. With this the turbulence intensity didn't actually change along the channel length.

Most of the published research on the turbulence decay has been performed with the utilization of turbulence generators in the form of grids. For such generators the turbulence pulsation attenuation within a short distance from the grid corresponds to the law [16]

$$1/Tu_0^2 = C \cdot (x'/d - x_e/d)^M \tag{2}$$

where C, M are constants, d is the size of a grid cell, x_e are coordinates of the effective measurement start. For the turbulators of the perforated plate type, which were used in the present work, a choice of dependence to describe the turbulence power decay is problematic, since turbulators of such type have a few characteristic dimensions, and each of them affects the generation and decay of Tu_0. However, as our investigations showed [12], the dependence (2) may be applied to the turbulators used in this work if the diameter of the turbulator's opening is taken to be of a distinctive scale d. The dependence coefficients (2) for turbulators with different numbers of openings were found from the experimental results by the least-squares fit. The coefficients are presented in the Table 1 (the experimental values were processed at $x'/s = 90$–250).

A change of the film cooling effectiveness parameter $\Theta = (T_W - T_0)/(T_S - T_0)$, depending on a non-dimensional longitudinal coordinate x/s, is shown in Fig. 4. The coordinate x was measured from the slot cross-section along the channel axis. The data are given for different initial turbulence intensities Tu_0 and for a different parameter of gas injection m. At $m = 0.6$ (Fig. 4a) the turbulence affects the effectiveness greatly; thus, with Tu_0 growing from 0.2 to 15% the value of effectiveness decreases by approximately three times. A greater effect is observed in the case when the velocity of the jet and co-current flow are close to each other ($m = 0.9$, in Fig. 4b). When the injection parameters are large (Fig. 4c), the turbulence effect on the protective properties of film cooling diminishes. Therefore, the large parameters of gas injection make the film cooling more stable to the actions of the external flow turbulent pulsation, and at $m = 2$ the stratification of experimental data for a different intensity of Tu_0 is insignificant.

The way the elevated turbulence affects the quality decrease in the heat protection of the surface along the channel length for different injection parameters is more clearly demonstrated in Fig. 5. Here, the values of gas film cooling effectiveness Θ with the initial high-intensity turbulence are related to the values of effectiveness Θ_0 with a low initial turbulence intensity ($Tu_0 = 0.2\%$). The comparison was made for the same parameters of gas injection which constituted $m = 0.6, 0.9, 2.0$ (Fig. 5a–c, respectively). This graph clearly demonstrates that high-turbulence cooling, for all considered m and x/s is less effective compared to low-turbulence cooling. A significant decrease in cooling effectiveness on the initial section of the channel $0 < x/s < 100$ should be noted. Further downstream, strongly mixed gas film cooling remains less effective, and the parameter of effectiveness changes weakly. With this, the small parameter of injection ($m = 0.6$) and a film cooling effectiveness decrease at $Tu_0 = 15\%$ amounted to 300%, at $m = 0.9$ the Θ decrease was the biggest, and constituted 400–500%. Such a strong effect of the external flow turbulence on the process of mixing wall jet and co-current flow under conditions of their velocity equality is probably explained by the lack of transverse velocity gradient on the boundary of two flows and by their mixing, basically, due to pulsations from outside.

In contrast to these results, the high-intensity turbulence has little effect on the parameter of cooling effectiveness when the parameters of injection are big ($m = 2$, Fig. 5c). The decrease of Θ constitutes only 10–20%. With this, the diminution of Θ along a length is uniform and close to the linear dependence.

Since a different effect of high-intensity turbulence on Θ was noted, depending on big and small parameters of injection, let us make a more detailed analysis of the Θ dependence on m. The dependence $\Theta = f(m)$ with a different initial turbulence intensity

Table 1

Number of openings in turbulator	$Tu_0(\%)$	C	x_e, m	M
25	7	1270	0.059	1.1
13	12	69	0.058	1.6
7	15	72.4	0.068	1.4

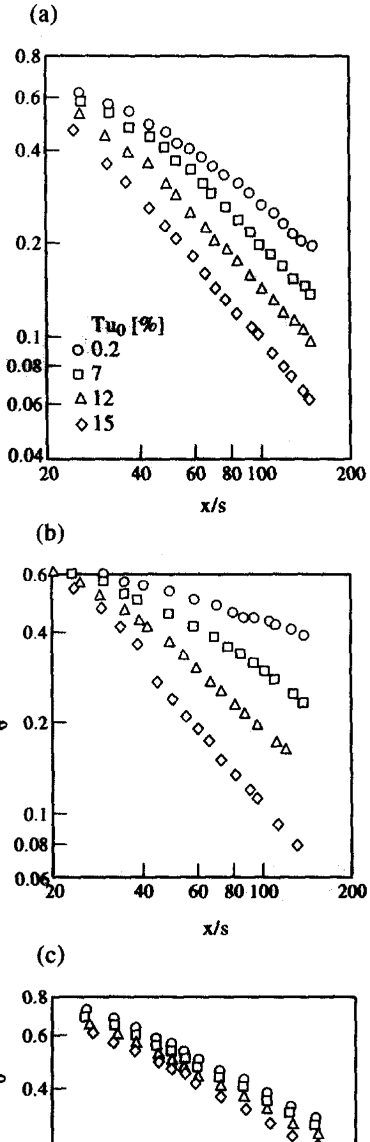

Fig. 4. Distribution of the effectiveness parameter along the channel length with different turbulence intensities and injection parameters: (a) $m = 0.6$; (b) $m = 0.9$; (c) $m = 2$.

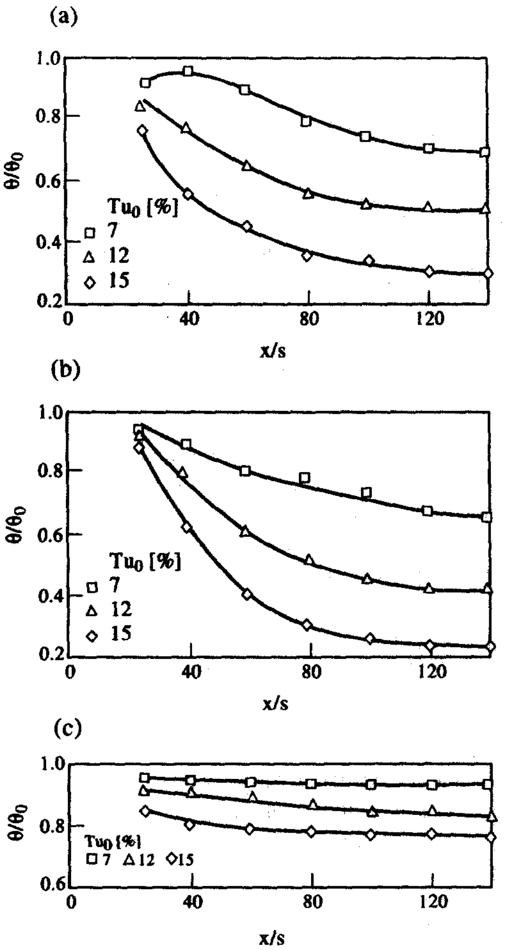

Fig. 5. Relative values of the wall jet cooling effectiveness with the initial turbulence change: (a) $m = 0.6$; (b) $m = 0.9$; (c) $m = 2$.

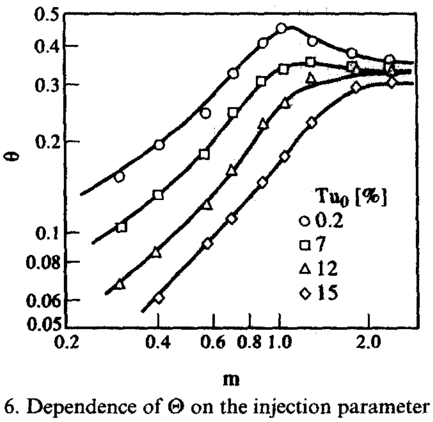

Fig. 6. Dependence of Θ on the injection parameter m.

is given in Fig. 6, in which the fixed parameter is $K_1 = 14$ ($K_1 = \Delta x/s \cdot Re_S^{1.25}$).

For the low Tu_0 the efficiency grows together with the increase in the parameter of a secondary flow

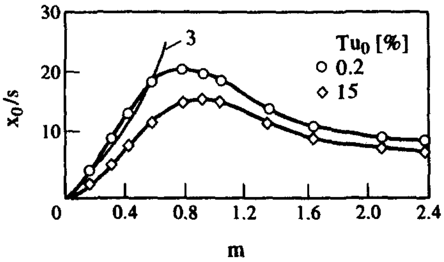

Fig. 7. Length of the initial section at different injection parameters.

injection to $m = 1$, where Θ has the maximum magnitude. With the further increase of m the effectiveness of film cooling diminishes, asymptotically approaching the value of effectiveness at $m = 0.6$. The elevated external turbulence significantly deteriorates the protective properties of a film cooling within a broad range (compared to the data for $Tu_0 = 0.2\%$). Besides, the behavior of a high-turbulent film cooling has a qualitative difference from the behaviour of a low-turbulent film cooling: Θ grows steadily with the increase in m throughout the whole measured range of the secondary air flow injection parameter. Even at $m > 1$, as opposed to the turbulent flow, the effectiveness of film cooling keeps on growing and approaches asymptotical the value of Θ at the low Tu_0. Beginning with $m = 2$–2.5, the effectiveness for the external turbulized flow actually stops depending on the velocity of injected gas. Thus, a further increase of the conveyed cooling gas flow rate will not result in a noticeable change of protective properties of the film cooling, and such regimes will not be economical of energy.

While generalizing the experimental data on the effectiveness of film cooling, we should usually know the length of the initial section x_0, i.e. the distance from the slot cross-section downstream where $\Theta = 1$. Let us analyse the change of x_0 under the investigated conditions of the flow. The initial heat section x_0 is determined from the graph $\Theta = f(x/s)$ by means of the coordinate $\Theta = 1$ intersection with the straight line going through the experimental points which well approximate themselves by a linear dependence in logarithmic coordinates [17]. The influence of injection on the length of the initial section is shown in Fig. 7 for low and high-intensity turbulence Tu_0. It is seen from the graph that, with turbulence degree growth, the length of the initial section actually diminishes throughout the measured interval m. The influence of the injection parameter on x_0 is not single valued. The length of the initial section has the maximum value at $m = 0.8$–1.0. With bigger and smaller values of m the magnitude of x_0 diminishes. With this, x_0 changes more profoundly if $m < 1.0$. In order to compare the data with the experimental information, the curve for $m < 1$ from ref. [18] is given

$$x_0/s = (0.112+0.036/m_1)^{-1}(m_1+1)/(m_1-1) \quad (3)$$

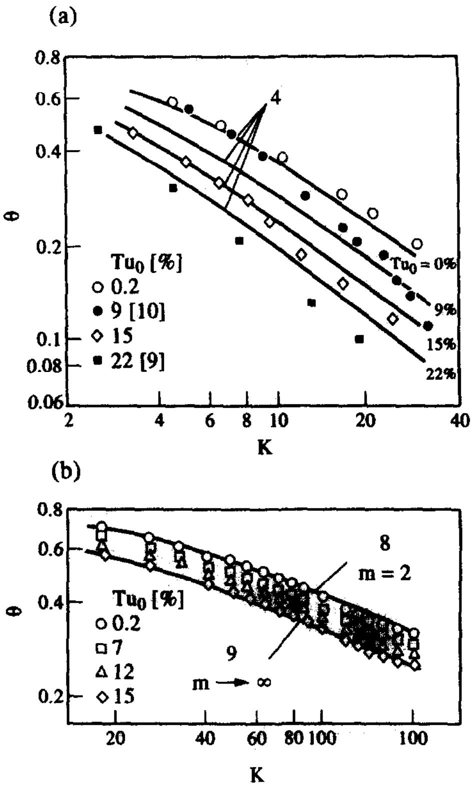

Fig. 8. Generalization of experimental data on effectiveness of high-turbulent wall cooling: (a) $m < 1$; (b) $m > 1$.

where $m_1 = u_s/u_0 \approx m$. It is obvious that the results of the calculation by this dependence describe adequately the experimental data in the field where $m < 0.6$ in the case of a low external turbulence.

GENERALIZATION OF EXPERIMENTAL RESULTS. COMPARISON WITH THE CALCULATED DATA

Since gas dynamics of co-current jet flow with $m > 1$ and $m < 1$ differs greatly, different dependences for data generalization on cooling effectiveness are usually proposed. In Fig. 8a the experimental data on the Tu_0 effect on the cooling effectiveness are presented in the form of generalized coordinates for $m < 1$. It can be seen that in the generalized coordinates the experimental points with the high-intensity turbulence lie lower than the points with a low-intensity turbulence.

In an effort to describe the results obtained during the experiments let us use the asymptotic theory of a boundary layer after Kutateladze–Leont'ev [2]. The expression for slot gas film cooling effectiveness on the adiabatic wall has the form

$$\Theta = [1 + A\cdot(Re_{\Delta x}/Re_s^{1.25})(\mu_0/\mu_s)^{1.25}]^{-0.8} \quad (4)$$

where $A = 0.016\beta^{1.25}\Psi$. The coefficient $\beta = (\delta_T^{*})_a/(\delta_T^{*})_{T=\mathrm{const.}}$ takes into account the temperature

Table 2

Tu_0 (%)	4	6	8	10	12	14	16	18	22
n	7.2	8.1	8.9	9.8	10.7	11.5	12.4	13.2	15.0
Ψ	1.03	1.05	1.07	1.09	1.10	1.12	1.14	1.15	1.19
β	9.2	10.1	10.9	11.8	12.7	13.5	14.4	15.2	17.0
A	0.26	0.30	0.34	0.38	0.42	0.46	0.51	0.55	0.66

field deformation during a boundary layer development on the adiabatic wall compared with the flow along the wall with heat transfer. At $x \to \infty$ a coefficient β tends to its maximum value [2]

$$\beta \to \beta_{\max} = \int_0^\delta \frac{\rho u}{\rho_0 u_0} \mathrm{d}y \bigg/ \int_0^\delta \frac{\rho u}{\rho_0 u_0}\left(1-\frac{T-T_w}{T_0-T_w}\right) \mathrm{d}y. \tag{5}$$

As is seen, with an incompressible quasi-isothermal flow β depends on velocity distribution in a boundary layer. Under the standard conditions of the flow (a gradientless incompressible isothermal flow with a low-intensity turbulence), a velocity profile has a power form with an index $1/n = 1/7$ and $\beta = n+2 = 9$.

A relative function of heat transfer $\Psi = (St/St_0)_{Re_T^{**}}$ takes into consideration the change of Stanton criterion under the influence of disturbing factors in the investigated conditions of a flow in comparison with standard factors. In the standard conditions of the flow, when we take into account that $\beta = 9$ and $\Psi = 1$ in expression (4), $A = 0.25$. The calculation from the equation (4) at $A = 0.25$ is shown in Fig. 8a. As we can see from the graph, when turbulent intensity is low, the experimental points are adequately generalized by this curve.

When a turbulence intensity is high, a deformation of a velocity profile in a boundary layer is observed—it becomes more rounded. The experimental data of various authors [10, 19–21], as well as the results of the present research one, may suggest the following interpolational expression for determining power index n in a velocity profile depending on the turbulence intensity

$$n = n_0 + bTu_0 \quad (n_0 = 5.5; b = 0.43). \tag{6}$$

As is known, external high-intensity turbulence causes the intensification of heat transfer processes in the boundary layer. In these conditions the relative function of heat transfer is different from one and may be derived from the dependence [21]

$$\Psi = \Psi_{Tu} = 1 + 0.0085 Tu_0. \tag{7}$$

Thus, in the case of high-intensity turbulence of a flow in equation (4), the coefficient β should be derived with due account of a power index independence in the velocity profile on the relationship (6), and the function Ψ should be calculated from the dependence (7). In Table 2 there are values of parameters n, β, Ψ and A, which were derived from the relationships (5)–(7) for different values of turbulence intensity Tu_0. As can be seen from the table, the coefficient β produces the greatest effect on Θ, and the influence of heat transfer function Ψ is negligible in the broad range of Tu_0.

The comparison of our experimental data with high-intensity turbulence ($Tu_0 = 15\%$) with the calculation by formulae (4)–(7) is given in Fig. 8a. It is obvious that the calculated dependence agrees adequately with the experiments.

The proposed method for consideration of the high-intensity turbulence influence on the effectiveness of film cooling was also evaluated by means of comparison with the experimental data of other authors. In Fig. 8a the experimental points from ref. [10] obtained at the turbulence intensity $Tu_0 = 9\%$ are compared with the proposed calculation. It can be seen that the calculation quite adequately describes the experimental data. Certain discordance of the curve with the points may be explained by experiment performance conditions. In this work the cooling was created with a long preincluded dynamic area which might result in a later penetration of pulsations to the wall cooling layer, as well as to its later wash-out. Besides, the gas injection is carried out at an angle to the surface which might also affect the mechanism of the non-dimensional wall temperature change along the channel length.

In ref. [9] such experiments were conducted with the highest turbulence intensity known in the literature, i.e. $Tu_0 = 22\%$. The experimental points from this work are also compared with the proposed method of calculation (Fig. 8a). It is seen that the calculated data quite adequately agree with the experimental results [9].

For injection parameters $m \gg 1$ the experimental data on the film cooling effectiveness, which depend on the integrating parameter K, are shown in Fig. 8b. The experimental data are given for different turbulence intensities $Tu_0 = 0.2$–15%. The initial turbulence increase results in the effectiveness decrease. However, such an influence is not very significant compared with the turbulence effect on gas film cooling at small injection parameters $m < 1$. This gives grounds to propose the dependences which are used for a low-turbulent cooling calculation to describe high-turbulence cooling. One of such dependences represents the curve for low-turbulent cooling [1] which is given in Fig. 8b:

$$\Theta = [(1+62.5/K)^{0.2}(1+62.5/(K|1-m|^{1.25}))^{-0.086}-1]^{0.8}/(1+0.016K)^{0.16}. \quad (8)$$

The curve was calculated at the injection parameter $m = 2$. From the picture it follows that for a low Tu_0 the experimental data adequately agree with the calculation. The experimental data for high-intensity turbulence $Tu_0 = 15\%$ lie a little lower than the curve throughout the entire measured range K. The other line in Fig. 8b represents the cooling effectiveness calculation at $m \to \infty$ [1]

$$\Theta = [\{1+(62.5/(\Delta x/(s \cdot Re_s^{0.25})+0.143))\}^{0.114}-1]^{0.8}. \quad (9)$$

As is seen, the calculated curve limits the experimental points from the bottom. Then the relationship (9) may be used to estimate the cooling effectiveness in a high-turbulence flow with big injection parameters in a first approximation with an error 10–20%. However, if we want a more correct description of the complicated process, we should consider in detail all salient features of the high-turbulence flow interaction with wall jet gas cooling.

CONCLUSIONS

The influence of the external flow high-intensity turbulence on the effectiveness of slot gas cooling has been experimentally investigated. It has been shown that the flow turbulization decreases cooling effectiveness. With this the value of the parameter Θ decreasing significantly depends on the relationship of the wall jet and co-current flow velocities. The external turbulence has a greater effect on cooling at small injection parameters ($m < 1$). The maximum decrease constituted 400–500% under experimental conditions ($Tu_0 < 15\%$). With greater injection intensities ($m > 1$) the influence of Tu_0 on cooling properties weakens. Thus, at $m = 2$ the diminution of Θ constituted only 10–20%. Such a weak effect is probably determined by the fact that in this case the energy of turbulent flow is a lot less than an average kinetic energy of the injected wall jet, so the wall boundary layer is quite stable to external disturbances.

The character of cooling effectiveness changes depending on the injection parameter. In the case of a high-turbulence flow it has some distinctions. In contrast with a low-turbulence cooling, when the effectiveness maximum is obtained if a jet velocity is equal to a flow velocity ($m = 1$), for a high-turbulence flow we observe a monotonous growth of Θ up to a certain asymptotic magnitude, which is reached with the injection parameter $m > 2$–2.5. These results are of actual importance, for they show that in the case of heat protection of surfaces in high-turbulent flows, a cooling quality may be increased even at injection parameters $m > 1$. Besides, the obtained result confirms that when the surfaces are cooled, the elevation of the parameter $m = 2$–2.5 is expedient, since the quality of cooling does not drop, and the coolant flow rate grows.

In order to describe the slot gas film cooling effectiveness with small injection parameters ($m < 1$), the calculation technique based on Kutateladze–Leont'ev theory has been proposed. With this, the high-intensity flow turbulence was considered through deformation of velocity profile and a relative function of heat transfer. The comparison showed an adequate accordance of the calculation with the experimental data of the present work and the experiments of other authors.

Taking into account the turbulence effect on Θ, the use of well known relationships for low-turbulence flows at $m \gg 1$ have been proposed for estimation of the high-turbulence effectiveness of cooling.

The turbulence intensity is not the only characteristic of a turbulent flow. At the present time there are no sufficient experimental data on the action of different characteristic turbulent scales on mixing processes. We need additional data on how the external turbulence interacts with the mixing layer turbulence and a wall turbulence. Perhaps such data will help to increase the calculation accuracy and to develop more efficient means of surface cooling.

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