vibration frequency; ζ_{0j}, ζ_j, displacements of the j-th surface from equilibrium position; ρ_i, β_i, dimensional and dimensionless densities of i-th fluid; δ, dissipation parameter; ξ_1, ξ_2, amplitude of interface vibrations; a, modulation amplitude; b, q, dimensionless modulation amplitudes; g, free-fall acceleration; x, y, z, Cartesian coordinates; h, H, dimensional and dimensionless thicknesses of liquid film; p_1, p_2, pressures; v_{0i}, $v = (u, v, w)$, fluid velocity vectors; t, time; ∇, gradient operator; i, l, n, indices; j, interface-numbering index; A, analog of Reynolds number; k, wave number, $k = (k_x, k_y)$, wave vector; b_*, critical modulation amplitude; k_*, critical wave number; U_0, U, potentials of repulsive and van der Waals compressive forces.

LITERATURE CITED

1. A. A. Baishulakov, M. N. Sokolov, and Yu. V. Malakhov, Ultrasonic Foam Suppression in Enrichment Plants [in Russian], Moscow (1967).
2. A. Sheludko, Colloidal Chemistry [in Russian], Moscow (1960).
3. L. D. Landau and E. M. Lifshitz, Fluid Mechanics, 2nd ed., Pergamon Press, New York (1987); Theory of Elasticity, Pergamon Press, New York–Oxford (1959).
4. V. G. Nevolin, Zh. Prikl. Mekh. Tekh. Fiz., No. 4, 53-55 (1981).
5. N. W. McLachlan, Theory and Application of Mathieu Functions, Clarendon, Oxford (1947).
6. V. V. Stepanov, Prikl. Mat. Mekh., 14, No. 3, 311-312 (1950).
7. V. G. Nevolin, Zh. Tekh. Fiz., 51, No. 8, 1758-1760 (1981).
8. P. L. Kapitsa, Zh. Éksp. Teor. Fiz., 21, No. 5, 588-594 (1951).
9. V. A. Briskman, Dokl. Akad. Nauk SSSR, 226, No. 5, 1085-1087 (1976).

SURFACE FLOW WITH DISCRETE SINKS

A. V. Shchukin and R. S. Agachev

UDC 621.438:542.46:532.546

Measurements have been made on the coefficient of friction, the pressure, and the turbulence at the surface involving discrete sinks provided by a transverse slot and a hole.

Channel inlet sections are frequently inclined to the surfaces in tangential cooling-air supply to convective film cooling systems for gas turbines (Fig. 1a) or else may be perpendicular to the latter (Fig. 1b). In the first case, the pressure loss in the supply can be calculated from data for the inlet sections of wind tunnels [1] and for aerospace vehicle surfaces [2], but in the second, the scope is more restricted, as there have been only isolated studies in the literature on the overall characteristics [3] or an analysis based on one-dimensional theory [4], where formulas were derived for hole flow coefficients. Pressure coefficients behind holes have been derived [5] and the flows have been visualized when the diameter of such a hole is comparable with the width of the supply channel. Boundary-layer theory applied for vanishing viscosity [6] has given the friction function, but it applies only for distributed porous sinks.

Here we examine flow around a surface having a transverse slot or hole through which the air is partly removed (Fig. 1c). We consider conditions corresponding to inlet channels in gas-turbine cooling systems, where the sink parameter $m_s < 1$.

The experiments were performed with a single slot and single hole, on the assumption that the effects from preceding and subsequent holes or slots would be absent. The channel height H was also taken to be substantially larger than the slot width s for hole diameter d.

A. N. Tupolev Kazan Aviation Institute. Translated from Inzhenerno-fizicheskii Zhurnal, Vol. 61, No. 3, pp. 378-382, September, 1991. Original article submitted October 16, 1990.

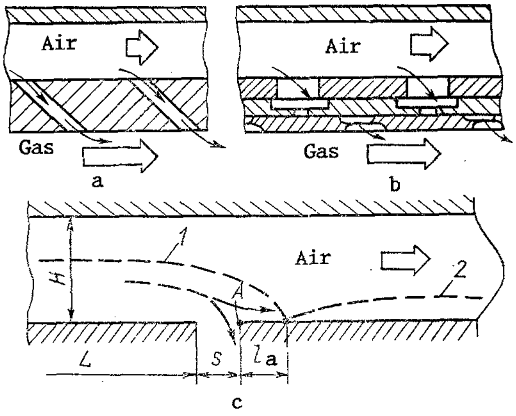

Fig. 1. Schemes for discrete sinks with inclined (a) and perpendicular (b) channels and flow pattern (c) in a channel containing a single transverse slot: 1) projection of the separation surface between the potential flow and the zone perturbed by the sink; 2) schematic edge of new boundary layer.

The [5] tests and the [4] analyses indicate that such a sink (Fig. 1c) will result in point A being a turbulence generator. We have visualized the flow in this system by means of water.

The working parameter ranges were $m_s = 0\text{-}0.21$ and Reynolds number $Re_s = (0.3\text{-}2)\cdot 10^4$.

The flow attachment zone beyond point A (Fig. 1c) was only slightly dependent on m_s and Re_s. Under these conditions, the relative extent of that zone was 1.0 ± 0.5. The spread in l_a/s with identical working parameters indicates that the position of that zone boundary is nonstationary.

Experiments by the electrohydrodynamic simulation method [7] confirmed that l_a/s did not differ by more than 20% from that obtained by visualization. The vortex structure behind the slot occurred not only with $H \approx s$, as found in [5], but also for $H \gg s$.

The gas-dynamic studies were made with a rectangular cross section system having width 80 mm and height 40 mm. At the inlet there was a grid providing the initial turbulence level $Tu_0 \approx 4\%$, and that Tu corresponds to the turbulence in inlet channels for air cooling systems [8], with the turbulence determined from the longitudinal component of the velocity pulsations. We give below data for the maximum degrees of turbulence occurring for the Tu distribution over the height of the channel.

We varied the working parameters over the following ranges: $Re_m = (1.7\text{-}15)\cdot 10^4$; $Re_s = (5.5\text{-}10)\cdot 10^4$; $m_s = 0\text{-}0.21$; T_a^* was 300 ± 5 K. The boundary-layer parameters on the part of the plate 500 mm from the inlet with $w_\infty = 56$ m/sec were $\delta = 5.6\cdot 10^{-3}$ m; $\delta^* = 32.4\cdot 10^{-5}$ m; $\delta^{**} = 25.4\cdot 10^{-5}$ m; $H = 1.28$.

Tests were done with the wall containing the slot replaced by an impermeable one, which gave the standard velocity profiles, in which n in

$$w/w_\infty = (y/\delta)^{1/n} \tag{1}$$

was 8.

We calculated the coefficients of friction for a smooth plate from the measured profiles by means of the Ludwig–Tilman [9] formula, the values being compared with the standard friction law [6]. We also performed calibration experiments for the thermoanemometer.

Figure 2 shows the measured turbulence behind the transverse slot for air with various m_s. Figure 2a shows that the turbulence falls as x/s increases for any m_s in a nearly exponential fashion. Tu in the first measurement section was 5-8%, while in the last it was almost equal to the value without the sink. As m_s increased, so did the turbulence for x/s = idem, and the effect could be traced out to x/s = 4-5. After that, Tu stabilized and was unaffected by m_s.

The maximal turbulence occurs in the attachment zone, whose length was (1 ± 0.5)s under these conditions. The far edge of the slot is the source of the elevated turbulence

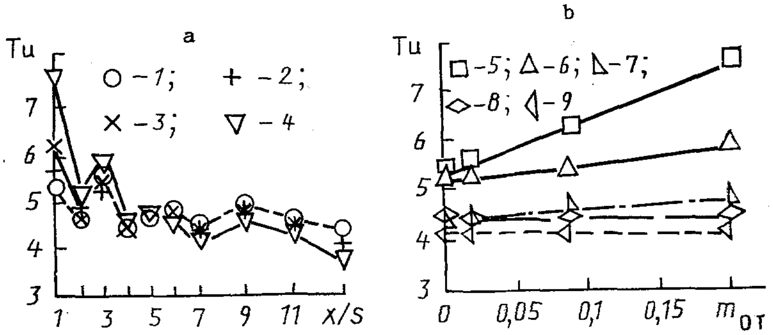

Fig. 2. Turbulence (%) in sections behind a slot acting as sink (a) and as a function of the sink parameter m_s (b): 1) m_s = 0; 2) 0.027; 3) 0.082; 4) 0.21; 5) x/s = 1; 6) 3; 7) 5; 8) 6; 9) 13.

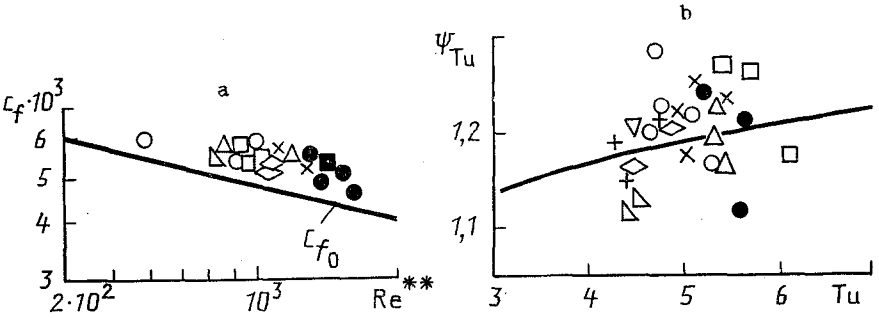

Fig. 3. Coefficient of friction after a sink slot behind the attachment zone: a) comparison with standard friction law; b) comparison with Eq. (2); the points correspond to the various working parameters and the measurements in reference sections.

(point A in Fig. 1c) together with the attachment zone beyond the slot. A similar conclusion can be drawn from the visualization data in [5].

Figure 3a shows that the coefficient of friction behind a transverse slot exceeds the standard value by about 20% for $1 < x/s < 13$.

Figure 3b shows that the relative frictional coefficient $(c_f/c_{f_0})_{Re^{**}}$ found in experiment with a plate containing a sink can be described satisfactorily by [6, 9, 10]

$$\Psi_{Tu} = (c_f/c_{f_0})_{Re^{**}} = 1 + 0{,}25\,\mathrm{th}\,(0{,}2\,\mathrm{Tu}). \qquad (2)$$

In this case, $\Psi_{Tu} \approx 1.2$, so these tests on turbulent boundary layers beyond the attachment zones following sinks (slots) imply that the surface frictional coefficient is described by (2).

Figure 2 shows that a sink slot produces turbulence even when $m_s = 0$, which is confirmed by conclusions, such as in [11], on increased heat transfer at a perforated surface even though no air passes through it.

The next series was done with a circular hole in the same ranges. The hole diameter was d = 10 mm. Figure 4 shows that the pressure coefficient begins to alter at about 1.5d before the leading edge.

The more rapid reduction in $\bar{p} = p_i/p_1$ as the hole is approached indicates flow acceleration before entry into the drainage channel. Beyond the hole, the pressure coefficient takes a maximal value and then falls. The higher m_s, the more $\bar{p}$ differs from the value without the sink. In the range $x/d \approx 1.5$-2.5, $\bar{p}$ stabilizes and becomes independent of the longitudinal coordinate.

The visualization results for the attachment zone beyond the hole showed that the width is about 2.5d for the parameter range used.

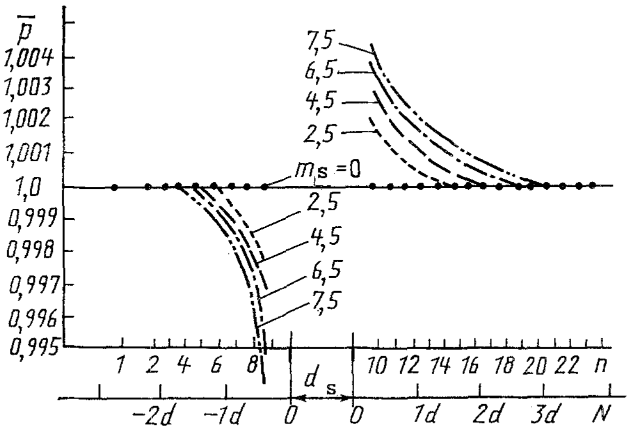

Fig. 4. Distributions for pressure coefficient $\bar{p}$ in the region of a sink hole with variable m_s (N is the number of hole diameters and n the number of tapping points).

NOTATION

H) inlet channel height; d and s) hole diameter and slot width; w_∞) speed of cooling air in main channel; ℓ_a) width of attachment zone after slot; δ, δ^*, δ^{**}) boundary-layer, displacement, and momentum-loss thicknesses; c_f, c_{f_0}) local surface frictional coefficient and coefficient of friction for a smooth plate (standard friction law); p_1) local static pressure in unperturbed flow before sink; p_i) local static pressure; Tu degree of turbulence; Re_m, Re_s) Reynolds numbers characterizing flow conditions in main channel and in sink hole; $m_s = (\rho w)_s/(\rho w)_m$) sink parameter.

LITERATURE CITED

1. I. L. Povkh, Aerodynamic Experiments in Engineering [in Russian], Leningrad (1974).
2. N. F. Krasnov and V. N. Koshevoi, Control and Stabilization in Aerodynamics [in Russian], Moscow (1978).
3. V. N. Sakharov and A. T. Trishkin, "Bladed machines and jet equipment," Tr. TsIAM, No. 1157, Issue 7, 198-205 (1985).
4. E. N. Bogomolov, Working Processes in Cooled Gas Turbine Engines with Perforated Blades [in Russian], Moscow (1987).
5. J. P. Fohr, J. Mech. Appl., 2, No. 2, 121-152 (1978).
6. S. S. Kutateladze and A. I. Leont'ev, Heat Transfer and Friction in a Turbulent Boundary Layer [in Russian], Moscow (1985).
7. L. A. Kazdoba, Electrical Simulation of Heat and Mass Transfer Phenomena [in Russian], Moscow (1972).
8. I. T. Shvets and E. P. Dyban, Air Cooling of Gas Turbine Components [in Russian], Kiev (1974).
9. A. A. Zhukauskas, Convective Transfer in Heat Exchanges [in Russian], Moscow (1982).
10. E. P. Dyban and É. Ya. Épik, Heat and Mass Transfer and Hydrodynamics in Turbulent Flows [in Russian], Kiev (1985).
11. R. D. Lander and R. W. Fich, AIAA Pap., No. 9, 1-9 (1972).