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HEAT TRANSFER AND HYDRODYNAMICS OF AN ARRAY OF ROUND IMPINGING JETS WITH ONE-SIDED EXHAUST OF THE SPENT AIR

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Abstract – The paper presents the results of an experimental investigation of local heat transfer at the surface blown by an array of round impinging jets with the spent air exhaust on one side. The experiments were carried out in the following range of parameters: $\bar{f} = 0.006$–0.18; $h/d = 1$–10; $d = 1$–5 mm; $Re_d = 1.1 \times 10^3$–17×10^3. Applicability of the collector relationships to calculate air distribution over the length of perforation, the jet flow and stalling flow velocity ratio and the coefficients of the total hydraulic resistance of the jet arrays, is considered. The channel technique is suggested to correlate the data on heat transfer in an array of impinging jets with one-sided exhaust of air. The validity and sufficient universality of this technique has been also confirmed by recalculation of the reported data. With power losses for air pumping through the system taken into account, the relationship has been established between the optimum open surface of perforation $\bar{f}_{opt}$ and the relative spacing h/L. Empirical relationships are suggested for calculation of heat transfer in an array of impinging jets with flow exhaust on one side.

NOMENCLATURE

- A, universal parameter, equation (4);
- B, heat transfer surface width;
- d, diameter of holes in perforated plate;
- d_h, $\equiv 2h$, hydraulic diameter of the channel;
- f, function;
- $\bar{f}$, $\equiv f_o/f_1$, relative value of the open surface of perforation;
- $\dot{G}_i$, mass flow rate of air through an isolated portion of perforated plate;
- $\bar{G}_i$, $\equiv G_i/G_o$, relative mass flow rate;
- h, distance from the heat transfer surface to perforated plate;
- $\bar{h}$, $\equiv h/L$, relative distance;
- I, electric current passing through a heater;
- k, empirical factor in equation (6);
- L, length of the heat transfer surface in direction of the spent air exhaust;
- N_e, power expended for air pumping through an array of impinging jets with one-sided exhaust;
- Nu_i, $= \dfrac{\alpha_i d_h}{\lambda}$, Nusselt number;
- Nu_d, $= \dfrac{\alpha_i d}{\lambda}$, Nusselt number;
- P, pressure;
- Re_d, $= \dfrac{\rho v_{av} d}{\mu}$, Reynolds number;
- Re_i, $= \dfrac{\rho u_i d_h}{\mu}$, Reynolds numbers;
- s, spacing of jets (longitudinal and lateral);
- sh, ch, th, hyperbolic functions;
- T, temperature;
- q, specific heat flux;
- u_i, mass-average velocity in longitudinal direction (along the x axis);
- ΔU, voltage drop on a strip electric heater;
- v_i, velocity of jet exhaust;
- $\bar{v}_i$, $\equiv v_i/v_{av}$, relative velocity of jet exhaust;
- x, longitudinal coordinate (in the direction of spent air exhaust from the channel);
- $\bar{x}$, $\equiv x/L$, relative value of the longitudinal coordinate.

Greek symbols

- α_i, local heat transfer coefficient;
- $\tilde{\alpha}_i$, pseudolocal heat transfer coefficient;
- $\bar{\alpha}$, surface-average heat transfer coefficient;
- ε_i, $\equiv Nu_i/Nu_{i,\infty}$, correction factor;
- $\tilde{\varepsilon}_i$, $\equiv \tilde{Nu}_i/Nu_{i,\infty}$, correction factor;
- λ, heat conduction coefficient; friction coefficient in equation (5);
- μ, viscosity; discharge coefficient of a perforated plate;
- ρ, density;
- ζ_c, resistance coefficient of the channel, equation (5);
- ζ_2, coefficient of hydraulic resistance of the jet array with one-sided exhaust of spent air.

Subscripts

- i, number of a heater; number of a unit volume of the channel;
- o, refers to open surface of a perforated plate;
- 1, refers to heat transfer surface;
- 2, refers to the outlet section of the channel;

∞, refers to conditions of stabilized heat transfer in channel;
av, mass-average value;
opt, optimum value;
$loss$, heat flux losses;
$*$, full pressure in the flow.

1. INTRODUCTION

HEAT transfer in arrays of impinging jets determines the intensity of the working processes in a number of technological apparatus such as, for example, the systems of cooling assemblies of gas turbine blades and elements of electronic equipment, apparatus for drying or thermal treatment of different sheet materials, equipment for heating ingots and cooling metal in continuous casting, etc. In view of the great practical and scientific importance of the problem, the number of publications on the topic increases continuously despite the availability of voluminous literature on heat transfer in impinging jets (see the bibliographic lists in [1–3]).

A three-dimensional pattern of the flow in the neighbourhood of the barrier and formation of spatial vortical structures between jets make theoretical calculations of heat transfer in arrays of impinging jets impracticable. As a consequence, investigations in this specific field of convective heat transfer are carried out on the basis of the similarity theory and thermal simulation. Confining the discussion to only multi-row arrays of subsonic round impinging jets, it should be noted that the necessity of accounting for the perforation geometry and the stalling flow effect leads to an increase in the number of parameters which govern heat transfer in the above conditions. This fact, being attractive as far as the control of the heat transfer process and its practical application are concerned, greatly complicates the correlation of experimental data and the working out of recommendations for an optimum selection of the governing parameters. Because of the two-dimensionality of the heat transfer coefficients on a cooled (heated) surface it is imperative to distinguish and study, besides the local, also the pseudolocal, $\tilde{\alpha}_i$ (based on the elemental surfaces $\Delta x \cdot B$), and average, $\tilde{\alpha}$ (based on the heat transfer surface xB), heat transfer coefficients, or in mathematical formulation

$$\tilde{\alpha}_i = \frac{1}{B}\int_0^B \alpha_i \,\mathrm{d}y; \quad \tilde{\alpha} = \frac{1}{x}\int_0^x \tilde{\alpha}_i \,\mathrm{d}x.$$

Detailed studies of the local heat transfer distribution over the barrier were carried out by Gardon and Cobonpue in 1962 [4] and by Hollworth and Berry in 1978 [5] with the use of miniature foil heat flow transducers. In [4] an array of jets was produced by a series of nozzles, in [5] by a perforated plate. In both cases the spent air was ejected on all the four sides of the heat transfer surface. The results of these studies have shown that at large separations of jets ($s/d > 10$) the distribution of α_i over the jet incidence zone is virtually the same as in the case of a single impinging jet. The effect of the stalling flow of spent air was not detected owing to a small number of jets (4×4) used in the tests.

The need of miniature heat flow transducers and the necessity of further integration of α_i, both over the spacing and the direction of the flow escape, impede investigations along these lines, although there can hardly be a doubt that a knowledge of the local heat transfer laws would make it possible to devise more effectively the impinging jet systems for the required levels of heat transfer to be obtained on the surface. This is rather well confirmed by the work of Martin [2] on shaping a plane slot to improve uniform distribution of α_i in the direction of the flow escape.

Pseudolocal heat transfer in an array of round jets was considered by Huang, 1963 [6], Kercher and Tabakoff, 1971 [7], Tabakoff and Clevenger, 1972 [8] and Chance, 1974 [9]. Huang demonstrated the usefulness of the heat flux detector which operated on the principle of an additional wall, with the cylindrical (convex) surface being blown by an array of round heated jets. The distribution of $\tilde{\alpha}_i = f(x)$ was studied qualitatively depending on the magnitude of the perforated surface $\bar{f}$, on the expansion (contraction) area ratio of the outlet channel, and on variability of $\bar{f}$ over the rows of holes. In order to determine $\tilde{\alpha}_i$, Kercher and Tabakoff, Tabakoff and Clevenger, and Chance applied the electrocalorimetry method, with the heat transfer surface (plane or concave) being composed of 4–6 heaters so that from one to six rows of jets impinged on each of them. In their experiments the flow escaped only on one side. Both Kercher and Tabakoff [7] and Chance [9] have derived the correlations in which they took into account the negative effect on $\tilde{\alpha}_i$ of the spent air from the preceding rows of jets.

Tabakoff and Clevenger carried out a comparative study of the heat transfer intensity on a concave surface for the jets of different shapes. Ward *et al.* [10] derived the surface-averaged heat transfer coefficients by analog with mass transfer. The perforated plate had 4×4 rows of circular holes, i.e. the conditions nearly repeated those with one heater. For this reason, the data of [10] as well as the results of Freidman and Mueller [11] can be related to pseudolocal heat transfer coefficients.

Because of the requirements placed by the drying technology many research workers devoted their efforts to the study of the laws governing surface-average heat transfer in an array of round impinging jets. A detailed analysis of the experimental data and appropriate correlations obtained for different versions of the spent flow escape are described by Gardon and Cobonpue [4], Ott [12], Hilgeroth [13], Krötzsch [14], Smirnov [15], Rozenfeld [16], *et al.* Thus, the values of $\tilde{\alpha}$ calculated following recommendations of the above works for the conditions, say, $h/d = 3.93$; $s/d = 6.34$; $Re_d = 10^4$–10^5, coincide within $\pm 25\%$, but as regards the tendency of the effect of $\bar{f}$ on the strength of the average heat transfer and the selection of the optimum values of $\bar{f}_{opt}$ and $(h/d)_{opt}$, the discrepancies are significant. It was found in [6] that at $\bar{f}$ above 0.04, $\bar{\alpha}$ reaches the maximum and then diminishes, i.e. $\bar{f}_{opt}$ = 0.03–0.05 at h/d = 2–4. According to [11] the optimum array of jets is that with $\bar{f}_{opt}$ = 0.02–0.03 at h/d = 4–6. Finally, Martin in his work [2] obtained $\bar{f}_{opt}$ 0.015 at $h/d \simeq 5.4$. The said discrepancies are due to the complexity of the study of such a multi-parametric system as an array of impinging jets and to the absence of a unified and substantiated approach to correlation of experimental data.

The objective of the present research was to study simultaneously the hydraulic characteristics (the total hydraulic resistance coefficient of the system and air flow rate distribution over the perforated length) and the local heat transfer in an array of impinging jets with escape of the spent flow on one side. Based on the results obtained a new correlation technique is suggested, the applicability of which is confirmed by the data of other authors including those which up to now have been used only for illustration, as for example, the data of [6].

2. EXPERIMENTAL APPARATUS AND DATA TREATMENT TECHNIQUE

A schematic diagram of the experimental apparatus is shown in Fig. 1. Arrays of axisymmetric jets were formed by interchangeable perforated plates (1) 130 × 130 mm² in size with orthogonal arrangement of holes equally spaced in longitudinal and lateral directions. The plates were fastened to the botton of a movable receiver (2). Flow equalization was achieved by placing screens (4) upstream of the perforated surface.

All of the perforated plates were calibrated beforehand for determining the discharge coefficient μ. In the experiments, by using the results of readings of the static pressure P_i (9) and the total pressure P_o^* in the receiver, the distribution of the fluid flow rates over the perforated plate sections was determined from

$$\dot{G}_i = \mu\Delta f\sqrt{[2\rho(P_o^* - P_i)]}, \tag{1}$$

where μ was assumed constant over the entire plate. The total hydraulic resistance coefficient of the system was determined from

$$\zeta_2 = \frac{P_o^* - P_2}{(\rho u_2^2)/2} - 1. \tag{2}$$

In the study of the regularities in the local heat transfer the specific heat fluxes were determined by the method of electrocalorimetry. A Tufnol slab (3) with the heat transfer surface (6) was placed at the bottom of the container (5). The heat transfer surface was formed by 25 strip nickel heaters (0.05 × 5 × 150 mm) pasted onto the slab and spaced 0.5 mm apart in the direction lateral to the flow escape. The temperature at the center of each heater on the underside was measured by a 0.1 mm dia copper–constantan thermocouple and, generally, it did not exceed 350 K. The first and the last three heaters served as the guarding ones and were not taken into account in the correlation of experimental data. In all tests the cooling air from the compressor was fed into the system at the temperature close to the environmental values. In order to take into account the heat losses* into the surrounding medium due to Tufnol thermal conductivity, eight similar thermocouples were placed on the underside of the slab (3). This allowed the local heat transfer coefficient to be determined for each heater from

$$\alpha_i = \frac{25\Delta UI/f_1 - q_{loss}}{(T_i - T_0)} \tag{3}$$

in the neighbourhood of each point at which a thermocouple was placed, i.e. to obtain the distribution of α_i along the line lying in the plane which passes through the axes of the central row of holes for all perforated plates.

Figure 1 also shows the elements of the measuring circuit. Voltage pick-up probes (ΔU_i) and thermocouples (T_i) were connected through a switchboard to a digital voltmeter with a sensitivity of 0.001 mV. The results of measurements were registered on a digital-printing device. Semi-automatic readings of all probes and printing of the results were accomplished after the steady-state thermal conditions had been established in the system.

The geometrical characteristics of the arrays of impinging jets studied and the range of parameters within which the experiments were carried out are listed in Table 1.

The local heat transfer coefficients (along the line lying in the plane which passes through the axes of the central row of holes) were related to the difference between the wall temperature T_i and the air inlet temperature T_o, the latter governing the choice of the thermophysical properties of air. The longitudinal

* Radiation losses are not accounted for since the radiation coefficient for polished nickel is only 0.03 at $T \simeq 300$ K and q_r is not more than 0.5% of the dissipated power.

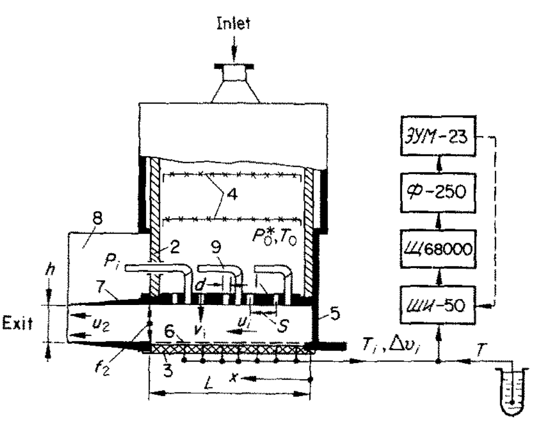

FIG. 1. Schematic diagram of experimental facility: 1, perforated plate; 2, receiver; 3, Tufnol slab; 4, smoothing screens; 5, container; 6, heat transfer surface (heaters); 7, visor; 8, side walls; 9, pressure sampling tubes.

Table 1. Basic characteristics of the array of impinging jets

No. of system (plates)	d, mm	Number of holes	s/d	$\bar{f}$	h/d	$Re_d \times 10^{-3}$	μ	A
2	1.0	441	6.0	0.0205	1.6–12	2.20–6.53	0.852	0.06–1.74
3	1.0	121	12.0	0.0056	1.6–12	4.73–9.0	0.654	0.04–0.30
4	1.5	121	8.0	0.0126	1.6–16	5.06–14.7	0.774	0.05–0.60
5	2.0	441	3.0	0.082	1.7–7.0	1.10–4.86	0.801	0.56–4.9
6	2.0	225	4.25	0.0418	1.6–12	3.44–9.21	0.809	0.16–1.44
7	2.0	121	6.0	0.0221	1.0–7.0	3.6–17.0	0.834	0.15–0.76
8	3.0	441	2.0	0.1846	1.0–10	4.54	0.765	0.54–6.25
9	3.0	121	4.0	0.0505	1.6–16	2.88–11.7	0.775	0.09–1.05
11	4.0	49	4.75	0.0364	1.0–7.0	5.25–23.8	0.757	0.05–0.29
12	5.0	49	3.9	0.0568	0.5–5.0	3.9–17.0	0.762	0.2–0.66

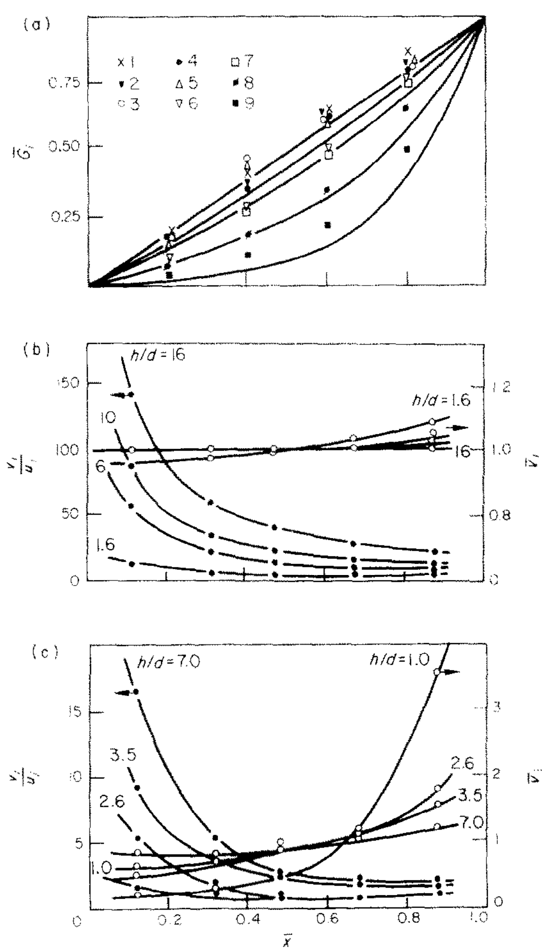

FIG. 2. Distribution of some hydrodynamic parameters of the flow over the channel length under perforated plate: (a) relative air flow rate; 1, at all h/d, plate No. 2; 2, the same, No. 3; 3, No. 4; 4, No. 7; 5, No. 11; 6, plate No. 5 at $h/d = 3.5$; $A = 1.16$; 7, the same, $h/d = 2.6$, $A = 1.61$; 8, $h/d = 1.6$, $A = 2.77$; 9, $h/d = 1.0$, $A = 4.9$. Solid curves, according to equation (7a) for respective A of plate No. 5; (b) relative velocity of jet exhaust (light points) and the jet and stalling flow velocity ratio (black points) for plate No. 4; $\bar{f} = 0.0126$; (c) the same for plate No. 5, $\bar{f} = 0.082$.

coordinate of the channel cross section was prescribed, as is usually the case, by the relative length of the channel, x/d_h, reckoning from the end-face wall.

3. AIR FLOW IN AN ARRAY OF IMPINGING JETS WITH SPENT AIR ESCAPE ON ONE SIDE

As follows from Fig. 2(a), in most of the jet arrays the distribution of air flow rate along the heat transfer surface is virtually uniform, i.e. $\bar{G}_i \sim \bar{x}$. For systems No. 5 and No. 8 a systematic deviation from this relation is observed with a greater rate of increase of $\bar{G}_i$ to the channel exit. The effect of the Reynolds number on the distribution of $\bar{G}_i(\bar{x})$ has proved to be insignificant. On the basis of the experimental results the normal exhaust velocity in jets has been calculated, $\bar{v}_i = v_i/v_{av}$, the typical distributions of which are presented in Fig. 2(b, c). The velocity of incidence is the lowest at the end-face wall ($\bar{x} = 0$) and increases in the direction toward the channel exit. These figures also show the axial behaviour of the velocity ratio v_i/u_i for some arrays of impinging jets studied. Obeying, in principle, the same attenuation law this ratio for system No. 4 [Fig. 2(b)] over the whole length is higher by an order of magnitude than for system No. 5 [Fig. 2 (c)].

The nature of distribution of the experimental data for $\bar{G}_i$ and $\bar{v}_i$ indicates that the air flow in an array of impinging jets with a one-sided flow escape is very close to the flow in intake collectors [18]. The universal parameter which fully governs the static pressure change and the velocities in the channel, is the following quantity

$$A = \mu \frac{f_o}{f_2} \sqrt{(1 + \zeta_c)}. \tag{4}$$

An exact determination of the channel resistance coefficient ζ_c is found to be difficult, since alongside the friction losses this coefficient takes into account the local vortical losses during merging of streams in the collector. Therefore, it was assumed for simplicity [17] that these hydraulic losses amount to half the friction

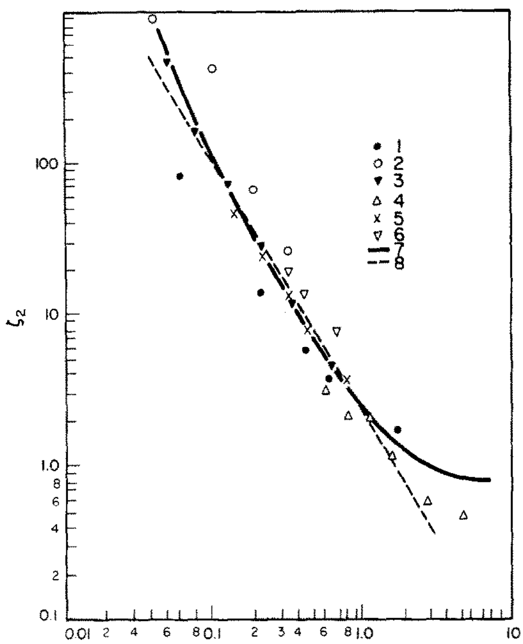

FIG. 3. Hydrodynamic resistance of an array of impinging jets with flow exhaust on one side: 1, plate No. 2; 2, No. 3; 3, No. 4; 4, No. 5; 5, No. 7; 6, No. 12; 7, according to equation (6); 8, according to equation (12).

losses in the same channel but at a constant (total) flow rate

$$\zeta_c \simeq 0.5\,\lambda \frac{L}{d_h}; \quad \lambda \simeq 0.3\,Re_2^{-0.25}. \tag{5}$$

In order to verify that the recommendations given in (17) can be used to calculate the array of impinging jets under study, the values of A have been determined from equation (4) for all plates and flow conditions. The data on the total hydraulic resistance coefficient ζ_2 of the arrays studied are presented in Fig. 3 as a function of the parameter A. Curve 7 corresponds to the expression

$$\zeta_2 = k\frac{1+\zeta_c}{\mathrm{th}^2 A} - 1, \tag{6}$$

in which the empirical coefficient $k = 1.5$ has been chosen from the condition of coincidence between the predicted and experimental values of ζ_2; the curve has been calculated for $\zeta_c = 0.3$.

The correlation of the experimental data which has been performed indicates that the remainder relations of [17] can also be applied to determine the flow rate distribution over the perforated length

$$\bar{G}_i = \bar{u}_i = \frac{\mathrm{sh}(A\bar{x})}{\mathrm{sh}\,A} \tag{7a}$$

and the exhaust velocity of jets

$$\bar{v}_i = A\frac{\mathrm{ch}(A\bar{x})}{\mathrm{sh}\,A}. \tag{7b}$$

The prediction based on equation (7a) for the conditions of the experiments carried out are given in Fig. 2(a) as solid lines.

The applicability of analogous relations to the analysis of the flow conditions in an array of two-dimensional impinging jets was also reported by Martin [2] who experimentally established the relationship between the degree of non-uniformity in the mass transfer coefficient distribution over the barrier surface and the relative cross-section area of the channel at the exit from the system.

4. HEAT TRANSFER IN AN ARRAY OF IMPINGING JETS

It is advisable to start the analysis of experimental data on heat transfer from distribution of Nu_d along the length $\bar{x}$ depending on the geometry of perforation, the ratio h/d and the number Re_d. Qualitatively, a change in Re_d in the range studied does not produce substantial changes in the distribution of $Nu_d(\bar{x})$ at h/d = const. for a fixed plate; the increase in $Nu_d|_{\bar{x}=\mathrm{const.}}$ is then proportional to Re_d^n, where $n = 0.6$–0.8 for different h/d.

It has been ascertained experimentally that despite a great variety of geometrical dimensions of holes, there are in the main, two types of the distribution of $Nu_d(\bar{x})$: at $s/d = 3.9$–12 the typical curves are those given in Fig. 4(a) and at $s/d = 1.5$–3.0, the typical curves are those given in Fig. 4(b).

In the first case, this is a wave-like distribution of the local heat transfer coefficients with an approximately constant mean value and the wave strength of 20–25%. Here, each jet has an individuality and it is only to the

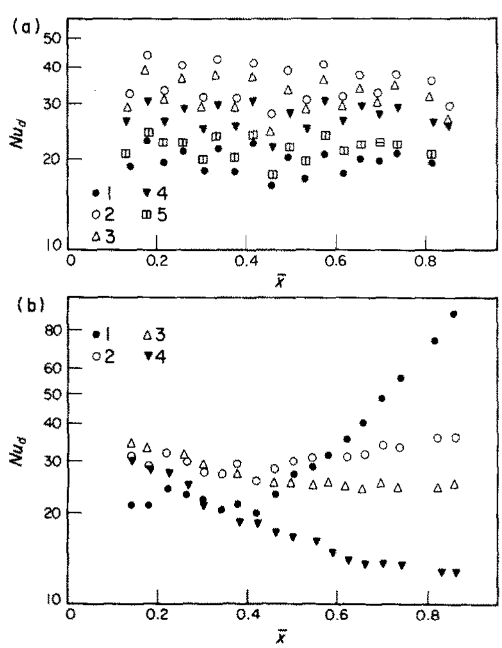

FIG. 4. Distribution of the local heat coefficients in an array of impinging jets: (a) plate No. 4, $Re_d = 8800$, $\bar{f} = 0.0126$ ($s/d = 8$): 1, $h/d = 1.6$; 2, 2.6; 3, 6.0; 4, 10; 5, 16; (b) plate No. 5, $Re_d = 3800$, $\bar{f} = 0.082$ ($s/d = 3$): 1, $h/d = 1.0$; 2, 2.6; 3, 3.5; 4, 7.0.

exit from the channel that the stalling flow makes the distribution of the heat transfer coefficients monotonous. In the second case, these are smooth curves the behaviour of which depends on h/d. The data of Figs. 2 and 4 considered simultaneously explain the distribution of the curves $Nu_d(\bar{x})$. Thus, a sharp increase in the flow rate, starting from $\bar{x} = 0.6$ at a low value of v_i/u_i, raises the curve $h/d = 1.0$ upward [Fig. 4(b)]. Conversely, virtually a uniform distribution of $\bar{v}_i$, but also at low ratios v_i/u_i, leads to the decrease of Nu_d to the exit from the channel [$h/d = 7$, Fig. 4(b)]. For plate 4 [Fig. 4(a)] high values of v_i/u_i at constant $\bar{v}_i$ over the channel length provide approximately constant values of Nu_d.

The analysis of the experimental data obtained has made it possible to develop a new correlation technique for the local heat transfer coefficients in arrays of impinging jets with the flow escape on one side [19]. This method is based on the concept of the correction factor $\varepsilon_i = Nu_i/Nu_{i,\infty}$ which takes into account enhancement of heat transfer over the channel portion considered relative to its minimal possible level. With such an approach, called the 'channel' technique, the surface cooled by an array of jets is considered as a wall of a plane channel with a distributed supply of the coolant. In these conditions the correction factor ε_i determines the difference of the experimental value of Nu_i from $Nu_{i,\infty} = 0.018\,Re_i^{0.8}$ (for air), where Re_i corresponds to the instantaneous flow rate $\dot{G}_i$ and the hydraulic diameter $d_h = 2h$ of the plane channel cross-section.

Figure 5 shows, as an example, the results of the use of the channel technique to correlate the local heat transfer coefficients for an array of impinging jets, system No. 3. The most significant advantage of the

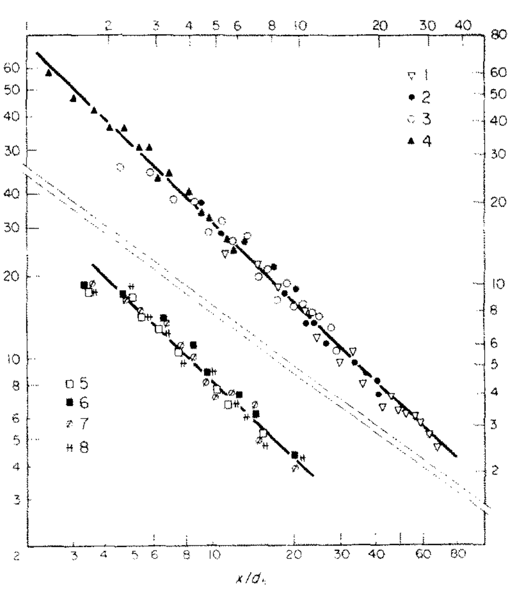

FIG. 5. Effect of the channel length on the correction factor ε_i for a system of largely spaced jets; $\bar{f} = 0.0056$; $Re_d = 7900$; 1, $h/d = 2.0$; 2, 3.2; 3, 5.0; 4, 12.0; $h/d = 3.2$; 5, $Re_d = 6250$; 6, 6640; 7, 9000; 8, 11 200.

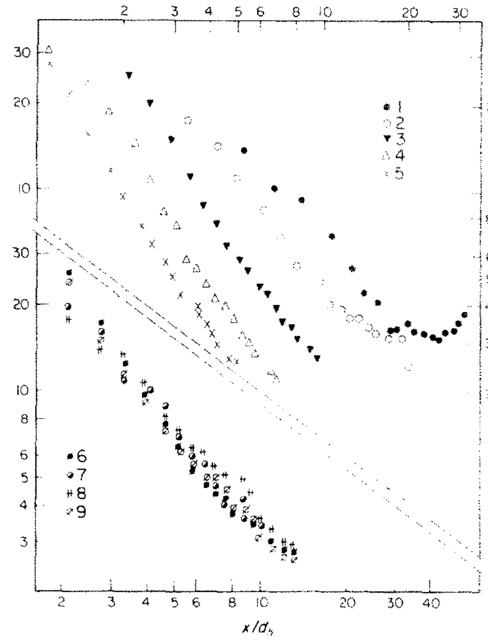

FIG. 6. Effect of the channel length on the correction factor ε_i for a system of closely spaced jets, $\bar{f} = 0.082$; $Re_d = 3800$: 1, $h/d = 1.0$; 2, 1.6; 3, 3.5; 4, 5.0; 5, 7.0 $h/d = 2.6$: 6, $Re_d = 1150$; 7, 1500; 8, 2060; 9, 3640.

method is, on the one hand, self-similarity of the correction factor ε_i, in the coordinates chosen, based on the Reynolds number Re_d and the parameter h/d and, on the other hand, an implicit account for the effect of the spent flow of the preceding rows of jets. In the traditional jet approach [7, 9] the effect of each of the above parameters is to be taken into account separately with the aid of a respective correction factor in the similarity equation.

Correlations similar to those given in Fig. 5 have also been obtained for the other jet arrays studied. However, when at $\bar{f} \lesssim 0.057$ the curves shift evenly, retaining self-similarity with respect to Re_d and h/d, then at $\bar{f} \gtrsim 0.082$ an ever increasing separation of the ε_i curves based on the parameter h/d was observed. Typical forms of correlations for $\bar{f} = 0.082$ (system No. 5) are presented in Fig. 6. It is evident that in an array of closely spaced jets ($s/d < 3$) the spent air from the preceding rows of jets decreases the downstream values of ε_i the greater the larger is the value of h/d. The quantitative aspect of interaction of the incident and the stalling flows under these conditions is illustrated by plots of the functions $v_i/u_i = f(\bar{x})$ given in Fig. 2. In particular, independent of h/d, the values of v_i/u_i for an array of system No. 5 are some 5–8 times less than those for system No. 4 with the open surface $\bar{f} = 0.0126$. The predominance of the jet effect appears to be responsible for the self-similarity of the quantity ε_i based on h/d; otherwise, the h/d ratio becomes the governing factor. However, an exact fixing of the boundary value of the open surface $\bar{f}$ is difficult due to smooth transition from one characteristic flow pattern to the other. The data of the present study allow the statement that at $\bar{f} \leqslant 0.05$ the jet effects are most pronounced, while at $\bar{f} = 0.08$–0.18 they are greatly reduced. Therefore, for each range of the open surface, different schemes have been developed for correlation of experimental data and respective similarity equations have been obtained for the local heat transfer in an array of impinging jets with flow escape on one side: at $\bar{f} = 0.006$–0.057 (arrays of largely spaced jets, Fig. 8):

$$\varepsilon_i = 10(x/d_h)^{-0.95}(\bar{f})^{-0.4}; \tag{8}$$

at $\bar{f} = 0.08$–0.18 (arrays of closely spaced jets, Fig. 7)

$$\varepsilon_i = 0.75\, A^{-0.38}(\bar{x})^{-1.34}(\bar{f})^{-0.4} \tag{9}$$

Comparison of the predicted [by equations (8), (9)] and experimental correction coefficients ε_i has shown that the scatter of the major number of points ($\sim 85\%$) does not exceed $\pm 15\%$, with the largest deviation being observed in the transitional, with respect to $\bar{f}$, zone. This allows the channel technique to be recommended for the use in practice to correlate the data on the local heat transfer in arrays of impinging jets. However, one should not forget that there are certain difficulties presented by correlation of the data on the laws governing heat transfer in the arrays which are characterized by at least five independent variables (G, $d, \bar{f}$, h/d, x/d_h). Therefore, some specific features of heat transfer, as for example, non-monotonicity in the distribution of α_i over the range $\bar{x} = 0.7$–1.0 for $h/d = 1.0$–2.6 in systems Nos. 5 and 8 (Fig. 7) are inexplicable even though the air flow rate distribution over the rows of jets and the velocity ratios of the mixing flows were taken into account when correlating the data by this technique.

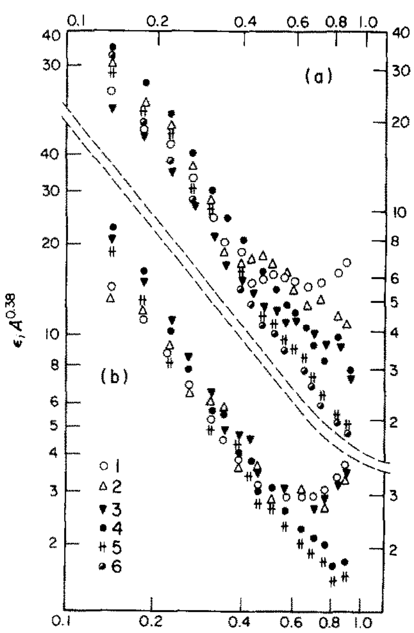

FIG. 7. Typical form of local heat transfer correlation for a system of closely spaced jets. (a) plate No. 8, $Re_d = 4540, \bar{f} = 0.185$; 1, $h/d = 1.0$; 2, 1.6; 3, 2.6; 4, 5.0; 5, 10. (b) plate No. 5, $Re_d = 3800, \bar{f} = 0.082$: 1, $h/d = 1.0$; 2, 1.6; 3, 2.6; 4, 3.5; 5, 5.0; 6, 7.0.

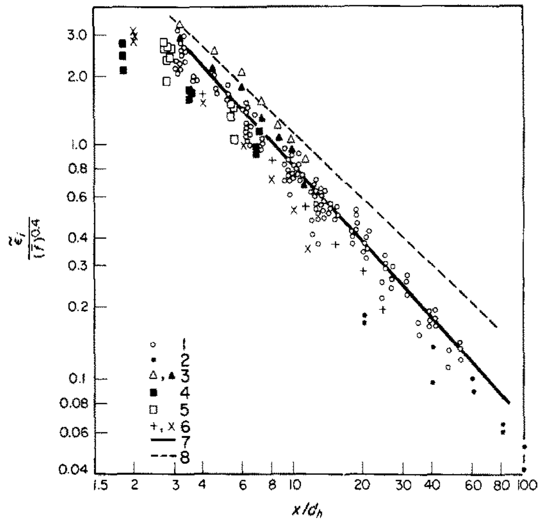

FIG. 8. Correlation by the channel technique of the reported data on pseudolocal heat transfer in an array of impinging jets with one-sided exhaust of flow. 1, $\bar{f} = 0.005$–0.08, Kercher and Tabakoff [7]; 2, $\bar{f} = 0.005$, Tabakoff and Clevenger [8]; 3, $\bar{f} = 0.04$ and 0.07, Huang [6]; 4, $\bar{f} = 0.022$, Word *et al.* [10]; 5, $\bar{f} = 0.0068$–0.28, Freidman and Mueller [11]; 6, $\bar{f} = 0.0276$, Chance [9]; 7, according to equation (10); 8, according to equation (8) for local heat transfer along the longitudinal axis of symmetry of the heat transfer surface.

5. COMPARISON WITH THE DATA OF OTHER AUTHORS

In order to evaluate the reliability of the channel technique for correlation of the data on heat transfer, the authors of the present paper have treated the reported data on the pseudolocal heat transfer coefficients $\tilde{\alpha}_i$ in arrays of impinging jets with a one-sided escape of the spent flow obtained by different methods.

The data of Kercher and Tabakoff [7] have been analyzed in the greatest detail. In making conversions it is assumed that air distribution over the perforated length is uniform ($A \leqslant 1.5$). Therefore, $Re_{d(i=1\text{-}4)} =$ const. and for the plane channel section between heaters 1 and 2 we may write

$$Re_{i=1} = Re_{d(i=1)} \frac{1.57\, \mathrm{d}\mu_o}{\mu_2} \simeq c_1 Re_d.$$

For the subsequent heaters $Re_i = iRe_{i=1}$, and then $Nu_{i,\infty}$ is determined. The effect of the stalling flow at $i \neq 1$ was taken into account by the following parameter

$$\frac{\dot{G}(x, i-1)}{\dot{G}(h, i)} \frac{h}{d} \approx \frac{Re_{(i-1)}}{2\, Re_{d(i)}}.$$

The values of the relation

$$\varepsilon_i(\bar{f})^{0.4} = \frac{Nu_i}{Nu_{i,\infty}}(\bar{f})^{0.4}$$

determined in such a way are plotted in Fig. 8 as a function of x_i/d_h where $x_i = i\Delta x$, Δx is the longitudinal dimension of the heat releasing surface of the heater. In all, more than 120 points have been recalculated taken from Figs. 6, 7, 9, 10, and 13 of [7]. As is seen, at x/d_h = 3–50 they are grouped together into a single relation.

Recalculation of the results obtained by Chance [9] for $\bar{f} = 0.0276$ and $h/d = 2$ and 4 was carried out in an analogous manner. Figure 8 also contains experimental data for cylindrical surfaces obtained by Huang (curve 5 of Fig. 11 from [6] for $\bar{f} = 0.04$, $h = 12.7$ mm, $d = 4$ mm and curve 5 of Fig. 12 for $\bar{f} = 0.07$, $h = 12.7$ mm, $d = 5.3$ mm), as well as the data obtained by Tabakoff and Clevenger [8] who presented no more than two distributions (for $Re_d = 5000$ and 10 000) of the pseudolocal heat transfer coefficients over five channel sections for the jet array of non-variable geometry: $d = 1$ mm, $h = 1$ mm, $s = 12.7$ mm, $\bar{f} = 0.005$, $L = 200$ mm, the channel width $B = 152.5$ mm. A rather important fact has been noted while converting the data of [8]: for $Re_{2(i=5)} = 10\,000$ and 20 000 (black points in Fig. 8) the experimental values of $Nu_{i=5}$ were more than two times lower than $Nu_{i=5,\infty} = 0.018\,Re_{i=5}^{0.8}$. This appears to indicate that, first, the channel technique allows evaluation of the validity of experimental data and, second, that for some unknown reasons the supplied power measured in [8] was underestimated. For all that, the regularity in the position of experimental points [8] in Fig. 8 extends the range as to x/d_h and thus once again justifies the use of the channel technique. Good agreement with the behaviour depicted in Fig. 8 is observed for the results of Freidman and Mueller [11] obtained at $\bar{f} = 0.0068$–0.0272 (except for the plate 0–2) and the data of Word *et al*, [10] obtained for one perforated plate at $\bar{f} = 0.022$ and $x/d_h = 1.17$–7. Because of the small experimental values of x/d_h, the average, rather than pseudolocal, heat transfer coefficients were considered in [10, 11]. Correlation of the results of different authors given in Fig. 8 yields the following empirical relationship for the pseudolocal heat transfer

$$\bar{\varepsilon}_i = 10(x/d_h)^{-1.1}(\bar{f})^{-0.4}. \quad (10)$$

6. OPTIMIZATION OF THE ARRAYS OF IMPINGING JETS

The proposed channel technique for the correlation of data on heat transfer in arrays of impinging jets allows rather a simple and vivid comparison between the systems of different geometries and determination of their optimal characteristics. Thus, if we ignore the pressure losses on pumping air and fix the following two parameters

$$\left.\begin{array}{l}(1)\ h = \text{const.}\quad \text{or}\ h/d = \text{const.}\\ (2)\ Re_2 = \dfrac{\rho u_2 d_h}{\mu_2} = c_2\dot{G} = \text{const}\end{array}\right\}, \quad (11)$$

we can obtain the functional relation between $\bar{\alpha}$ amd $\bar{f}$. With this approach, the advantage of the use of Re_2 as the governing parameter is obvious. The behaviour of the curves in Fig. 9, plotted for $h/d = 2.6$, indicates that for the comparison conditions mentioned the value of $\bar{f} \simeq 0.005$ is most efficient as regards the level of heat transfer. With the use of a traditional approach, when the Reynolds number is determined on the basis of the jet parameters, an explicit comparison of the efficiency of different systems is actually impossible.

Note that the values of $\bar{\alpha}$ given in Figs. 9 and 10 have been obtained in the present work by averaging α_i only along the longitudinal axis x; the data of [7, 10] are the coefficients of heat transfer averaged over the surface of one heater. It may there be seen that this does not effect the behaviour of the functions.

Let us consider now the problem of determining $\bar{f}_{opt}$ with account for power losses on air pumping through an array of jets. To this end, we approximate the data

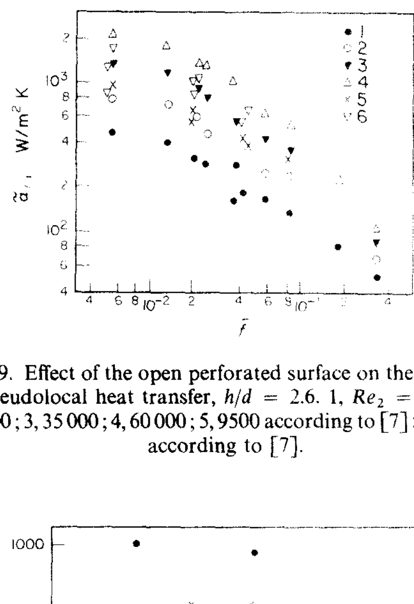

FIG. 9. Effect of the open perforated surface on the intensity of pseudolocal heat transfer, $h/d = 2.6$. 1, $Re_2 = 9500$; 2, 17 600; 3, 35 000; 4, 60 000; 5, 9500 according to [7]; 6, 17 500 according to [7].

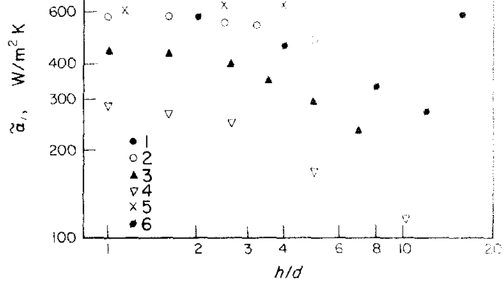

FIG. 10. Effect of the parameter h/d on the intensity of pseudolocal heat transfer. 1, $\bar{f} = 0.0126$ and $Re_2 = 28\,000$; 2, 0.0364 and 32 700; 3, 0.082 and 33 500; 4, 0.185 and 60 000; 5, 0.0195 and 9500 according to [7]; 6, 0.0218 and 8900 according to [10].

on the total resistance coefficient ζ_2 from Fig. 4 by the power relation (for $A \leqslant 3$)

$$\zeta_2 = \frac{\Delta P}{(\rho u_2^2)/2} - 1 = 2.3\,A^{-1.6} \tag{12}$$

and employ the condition (11). Replacing x by the entire length of the surface yields

$$N_e = V\Delta P = u_2 f_2 \frac{\rho u_2^2}{2}(1 + 2.3A^{-1.6}). \tag{13}$$

Introducing the notation

$$\bar{N}_e = \frac{N_e}{f_1}; \quad A = \mu\bar{f}/\bar{h}; \quad \zeta_c = 0,$$

we obtain

$$u_2 = \left[\frac{2\bar{N}_e}{\rho\bar{h}(\{1 + 2.3[\mu(\bar{f}/\bar{h})]^{-1.6}\}}\right]^{0.333}. \tag{14}$$

Then, having expressed $\tilde{\alpha}$ in terms of equations (10) and (14)

$$\frac{\tilde{\alpha}d_h}{\lambda} = 0.18\left(\frac{L}{2h}\right)^{-1.1}(\bar{f})^{-0.4}\left(\rho\frac{2h}{\mu_2}\right)^{0.8} \times \left[\frac{2\bar{N}_e}{\rho\bar{h}\{1 + 2.3[\mu(\bar{f}/\bar{h})]^{-1.6}\}}\right]^{0.267}$$

and having substituted all of the quantities which remain constant in the comparison into the LHS, we obtain

$$\tilde{\alpha}(\bar{f},\bar{h}) = \bar{h}^{0.733}(\bar{f})^{-0.4} \times \{1 + 2.3[\mu(\bar{f}/\bar{h})]^{-1.6}\}^{-0.267}. \tag{15}$$

For the condition $\partial\tilde{\alpha}/\partial\bar{f} = 0$ the following relation holds

$$\bar{f}_{opt} = \frac{0.187}{\mu}\bar{h}. \tag{16}$$

Thus, in contrast to Martin's recommendations [2] where $\bar{f}_{opt} = 0.0152$ for any h/d, the relationship has been established between the optimal value of the open surface of perforation and the relative height $\bar{h}$ expressed in fractions of the heat transfer surface length L. In the experiments carried out $\mu \simeq 0.8$, $\bar{h} = 0.03$–0.077, whence the values of $\bar{f}_{opt}$ range from 0.007 to 0.018.

It is interesting to note that equation (15) does not possess the condition at which $\partial\tilde{\alpha}/\partial\bar{h} = 0$, i.e. the intensity of heat release from the surface cannot be optimized on h/L. Figure 10 gives graphical presentation of the relationship $\tilde{\alpha}_i = f(h/d)$ which has been obtained empirically at fixed Re_2 and $\bar{f}$. It is seen that at decreasing h/d the strength of heat transfer increases monotonously within the h/d range studied. At $h/d = 1$–2 all of the curves are rather flattened. With increase in h/d, a more sharp decrease in heat transfer is observed for the arrays of jets with larger open surface of perforation ($\bar{f} > 0.08$). According to [7], the effect of this parameter on the levels of average heat transfer at $h/d = 1.2$–3.9 is less pronounced.

7. CONCLUSIONS

1. A qualitative and quantitative analysis has been carried out of the local heat transfer along the line lying in the plane of the axes of the central row of round impinging jets with the spent air escape on one side. The region of the values of $\bar{f}$ studied has been divided into two ranges and corresponding empirical relationships [equations (8) and (9)] have been obtained for each of these ranges for calculation of α_i.

2. The channel technique has been suggested for the correlation of data on heat transfer in an array of impinging jets with one-sided exhaust of the flow. The reliability and sufficient universality of the technique is confirmed by both the experimental data obtained in the present investigation and by recalculation of the reported data on pseudolocal heat transfer coefficients. That the latter be calculated, an empirical relationship (10) has been recommended. This relationship seems to be also valid for calculation of the average heat transfer over the surface of the length L under the conditions considered.

3. It is shown that the collector relationships can be used for calculation of the air flow rate distribution over the length of perforation, of the jet flow and stalling flow velocity ratio and of the total hydraulic resistance coefficient for the jet arrays considered.

4. As a consequence of the analysis of experimental data on the hydraulics and heat transfer of impinging jets, a relationship has been obtained between the optimal value of the open perforated surface $\bar{f}_{opt}$ and the distance $\bar{h}$ expressed in fractions of the entire heat transfer surface length; the optimal value of h/L in the studied range of $A \leqslant 3$ has not been discovered.

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HYDRODYNAMIQUE ET TRANSFERT THERMIQUE D'UNE RANGEE DE JETS CIRCULAIRES INCIDENTS AVEC ECHAPPEMENT D'AIR D'UN SEUL COTE

Résumé—On présente les résultats d'une étude expérimentale du transfert thermique local à la surface balayée par une rangée de jets incidents circulaires, avec sortie d'air sun un côté. Les domaines de variation des paramètres sont les suivants: $\bar{f} = 0,006$–$0,18$; $h/d = 1$–10; $d = 1$–5 mm; $Re_d = 1,1 \times 10^3$–17×10^3. On considère l'applicabilité au collecteur pour calculer la distribution d'air sur la longueur de perforation, les coefficients de résistance hydraulique totale de la rangée de jets. La technique du canal est suggérée pour unifier les données sur le transfert thermique pour une rangée de jets incidents avec sortie d'air d'un côté. La validité et la suffisante universalité de cette technique a été confirmée par le calcul en retour des résultats expérimentaux. Avec la prise en compte des pertes d'énergie pour la circulation forcée d'air, on établit la relation entre la surface optimale des perforations $\bar{f}_{opt}$ et l'espacement relatif h/L. Des relations empiriques sont suggérées pour le calcul du transfert thermique dans une rangée.

WÄRMEÜBERGANG UND STRÖMUNGSMECHANIK EINER ANORDNUNG VON RUNDEN AUFTREFFENDEN STRAHLEN BEI EINSEITIGEM AUSLASS DER ABLUFT

Zusammenfassung—Dieser Bericht liefert die Ergebnisse einer experimentellen Untersuchung des lokalen Wärmeübergangs an eine Oberfläche, die von einem Feld runder auftreffender Strahlen bei einseitigem Luftauslaß angeblasen wird. Die Experimente wurden innerhalb folgender Parameterbereiche durchgeführt: $\bar{f} = 0.006$–0.18; $h/d = 1$–10; $d = 1$–5 mm, $Re_d = 1.1 \times 10^3$–17×10^3. Die Anwendbarkeit der Beziehungen für das Sammelrohr zur Berechnung der Luftverteilung über die Länge der Perforation, des Strahlströmungs- und des Staupunktströmungs-Geschwindigkeitsverhältnisses und des gesamten hydraulischen Widerstandskoeffizienten des Systems wird betrachtet. Das Kanalverfahren wird vorgeschlagen, um die Werte das Wärmeübergangs zu korrelieren. Die Gültigkeit und genügende Vielseitigkeit wurde auch durch Nachrechnung der mitgeteilten Werte bestätigt. Unter der Berücksichtigung der Leistungsverluste, die durch die Förderung der Luft durch das System entstehen, wurde das Verhältnis zwischen der optimalen offenen Oberfläche der Perforation $\bar{f}_{opt}$ und der relativen Teilung h/L festgestellt. Für den Wärmeübergang werden empirische Beziehungen vorgeschlagen.

ТЕПЛООБМЕН И ГИДРОДИНАМИКА СИСТЕМ КРУГЛЫХ ИМПАКТНЫХ СТРУЙ С ОДНОСТОРОННИМ ВЫХОДОМ ОТРАБОТАВШЕГО ВОЗДУХА

Аннотация — Приведены результаты экспериментального исследования локального теплообмена на поверхности, обдуваемой системой круглых импактных струй с односторонним выходом отработанного воздуха. Опыты выполнены в диапазоне $\bar{f} = 0,006$–$0,18$; $h/d = 1$–10; $d = 1$–5 мм; $Re_d = 1,1 \times 10^3$–17×10^3. Показана применимость коллекторных соотношений для расчета распределения воздуха по длине перфорации, соотношения скоростей струйного и сносящего потоков и коэффициентов общего гидравлического сопротивления рассмотренных струйных систем. Предложена канальная методика обобщения данных по теплоотдаче в системе импактных струй с односторонним выходом воздуха. Надежность и достаточная универсальность этой методики подтверждена также пересчетом имеющихся в литературе данных. С учетом затраты мощности на прокачку воздуха через систему получена связь между оптимальной величиной открытой поверхности перфорации $\bar{f}_{opt}$ и относительным зазором h/L. Приведены эмпирические зависимости для расчета теплоотдачи в системе импактных струй с односторонним выходом потока.