EXPERIMENTAL STUDY OF HEAT TRANSFER IN NARROW SLIT CHANNELS

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UDC 536.27

The procedures and results of an experimental study of heat transfer in narrow slit channels are described.

One of the ways of increasing heat transfer to a cross flow of coolant through pipe clusters is to decrease the distance between the pipes. As the heat transfer is increased, the hydraulic resistance can be decreased by decreasing the coolant velocity and shortening the length of passages through the interpipe space [1]. Since shortening of these passage lengths results in a decrease in the number of rows of pipes, this ultimately leads to apparatus with single rows of pipes. The space between these horizontal pipes consists of slit channels bounded by their side surfaces and end supports [2]. The heat-transfer surfaces of these shapes have very small equivalent diameter (from 0.5 to 1.0 mm) and short lengths (from 10 to 50 mm) along the direction of coolant flow. For channels of such dimensions high heat flux from the transfer surface can be obtained even with laminar flow conditions. The best known theoretical studies of steady state heat transfer and hydraulic resistance in such apparatus are those of Stephan [3] and Schiller [4]. These studies however assume smooth channels and uniform velocity at entry. In practice these assumptions may not be applicable. Experimental data [5-7] for heat transfer and hydraulic resistance lie outside the above ranges of dimensions. A further experimental study was therefore desirable. Transient conditions were chosen as the most suitable for determining the heat transfer coefficient. The measurements were further simplified by maintaining the temperature constant at any time for $Bi < 0.1$. To achieve this the specimens were made of copper and limited in length to 20 mm. The main body 1, of the apparatus (Fig. 1) was made of Textolite and its internal surfaces were thermally insulated 2. The specimens 3, mounted inside the body consisted of two copper parallel pipes between which distance pieces were placed. The weight of the specimens was about 500 g. Chromel–Alumel thermocouples 4, were positioned at a depth of 10 mm inside the specimens and sealed into them. The specimens were heated by removable heater 5. Their shape in the apparatus body was controlled by the screws 7 compressing them against rubber pads 6. The position of the thermocouples (in the direction perpendicular to the plane Fig. 1) was 40 mm from the end. The working fluid was air passed to the apparatus through the connection 8, and to the slit channels between the experimental specimens thereby cooling them. This air was exhausted to the ambient atmosphere. The temperature drop θ between the specimens and the air was measured by the thermocouple 4 connected differentially with the thermocouple in the entry flow into the apparatus. A recording KP-59 potentiometer was used for measurements.* As previously known [8] determination of the heat-transfer coefficient by the regulative method requires an experimental determination of the rate of change of temperature of the specimen for given flow condition over its heat-transfer surface. The cooling rate is given by

$$m = -\frac{1}{\theta} \cdot \frac{d\theta}{d\tau}. \tag{1}$$

If the specimen temperature does not change while being cooled (the first type of regulation method) then the cooling rate can be associated with the average heat-transfer coefficient as follows

* The air flow was metered by a throttle controlled from a sonic jet of short longitudinal section (a small hole in a foil diaphragm).

Translated from Inzhenerno-Fizicheskii Zhurnal, Vol. 20, No. 4, pp. 706-710, April, 1971. Original article submitted May 12, 1970.

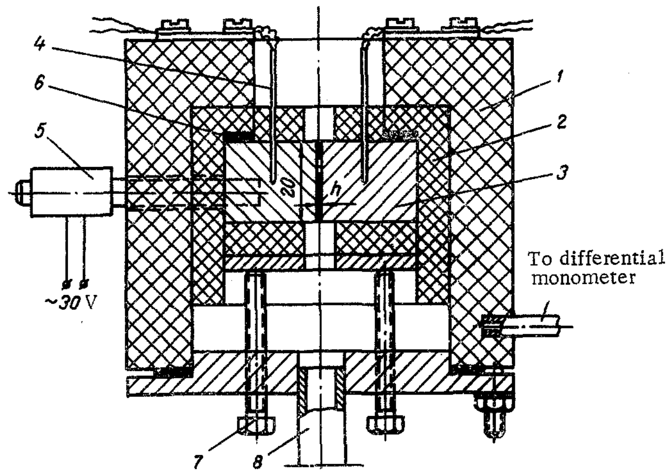

Fig. 1. Experimental setup: 1) body; 2) heat insulation; 3) experimental sample; 4) thermocouple; 5) removable heater; 6) packing layer; 7) press screw; 8) supplying nozzle.

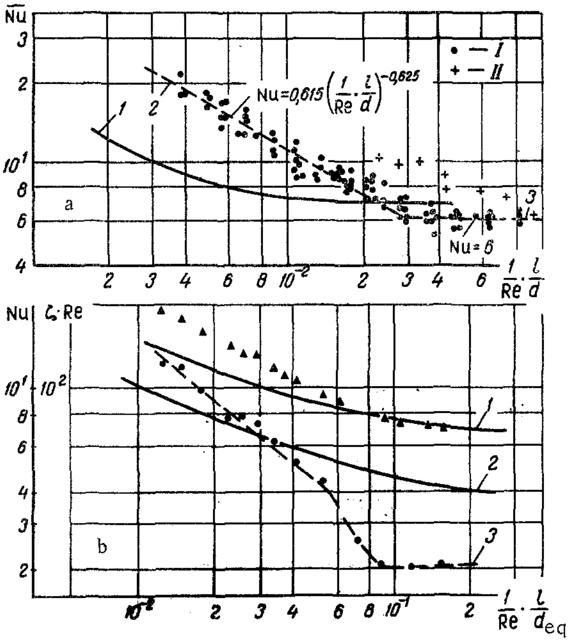

Fig. 2. Generalization of experimental data for surfaces Nos. 1-4 (a) and of experimental data for surface No. 5 (b). For a: I) surfaces Nos. 1-3; II) surface No. 4 (h = 0.2 mm). For b: 1) ξ Re; 2, 3) Nu.

$$m = \psi \frac{\alpha F}{C} \tag{2}$$

(at Bi < 0.1, ψ = 1). In the present study this relationship was not applicable because of appreciable temperature change during flow through the channel. It was therefore necessary to obtain the relationship between the average heat-transfer coefficient and the cooling rate of the specimen. Increasing the temperature of the fluid flowing through the channel gives

$$\delta t = \frac{C}{W} \cdot \frac{d\theta}{d\tau}. \tag{3}$$

Hence it is possible to obtain a relationship between the heat transfer and the cooling rate

$$\varepsilon = \frac{\delta t}{\theta} = \frac{C}{W} \cdot \frac{d\theta}{\theta d\tau} = \frac{mC}{W}. \tag{4}$$

The equation is rated for any heat emission along the channel and allows the relationship between the average heat-transfer coefficient and the cooling rate to be obtained. In the present study the wall temperature along the channel was uniform at any time and hence

$$\varepsilon = 1 - \exp\left(-\frac{\alpha F}{W}\right). \tag{5}$$

Substitution of (4) into (5) and manipulation gives

$$\mathrm{Nu} = 0.575\, \mathrm{Re}\, \mathrm{Pr} \lg \frac{1}{1-\varepsilon}. \tag{6}$$

The heat transfer in the experiment can be divided into two parts, the first is the heat flow to the gas and the second is the heat to the surroundings. The temperatures of both the ambient air and the air flow through the specimen were similar and therefore both heat flows had the same temperature driving force θ

$$q_0 = m_0 C\theta, \quad q_n = m_n C\theta.$$

Since $q_g = W\delta t$ it follows that

$$m_0 C\theta = W\delta t + m_n C\theta,$$

whence

$$\varepsilon = \frac{\delta t}{\theta} = \frac{m_0 C}{W} - \frac{m_n C}{W} = \frac{C}{W}(m_0 - m_n) = \frac{Cm}{W}. \tag{7}$$

Therefore the cooling rate of the specimen by heat transfer m to the air flow is equal to the difference between the total cooling rate m_0 and the heat loss rate m_n to the surroundings. The values of both of these were found by the usual methods [8]. Slit channels of 0.1 to 0.5 mm width were investigated and their lengths varied from 20 to 100 mm. The entry to these channels was sharp edged. The possibilities were also studied of increasing the heat transfer by roughening the channel walls both to promote turbulent transfer and to increase the heat-transfer surface.

Five types of surface were tested in the experimental programs: type 1 – mirror finish surface; type 2 – surface sanded to a 7 μ roughness; type 3 – surface as in 2 but with a 15 μ roughness; type 4 – surface with roughening ribs of 8 μ running perpendicular to the flow across the whole width of the channel; and type 5 – surface with broken rib formed by a three angled cut. The height of these ribs was 0.5 mm and their wavelength was 0.1 mm.

The results of the thermal and hydrodynamic investigations are given in Fig. 2a and b. Figure 2a gives experimental results for the channels with surface types 1–4. Line 1 shows the theoretical values [3] for Pr = 0.7. In range of length $(1/\mathrm{Re})(l/d) = 4 \cdot 10^{-3}$–$3 \cdot 10^{-2}$ the data is approximately correlated by the relationship (line 2)

$$\mathrm{Nu} = 0.615\left(\frac{1}{\mathrm{Re}} \cdot \frac{l}{d}\right)^{-0.625}. \tag{8}$$

When $(1/\mathrm{Re})(l/d) > 3 \cdot 10^{-2}$ the heat transfer obeys a constant relationship Nu = 6. An exception to this occurred for the channel of 0.2 mm width and type 4 surface.

Hence the heat transfer in the present study of channels differs considerably from the theoretical analysis. This can be accounted for by the effect of the sharp entry shape in forcing the flow back to the walls for laminar conditions and in generating turbulence from high velocities. In the range $(1/\mathrm{Re})(l/d) = 4 \cdot 10^{-3}$–1 the experimental data for the hydrodynamic resistance of surface types 1–4 approximates

$$\bar{\zeta}\mathrm{Re} = 96 + \frac{0.85}{\frac{1}{\mathrm{Re}} \cdot \frac{l}{d}}, \tag{9}$$

which agrees with well-known theoretical analysis [4]. Figure 2b gives experimental data for the type 5 ribbed surface. For low velocities the heat transfer was not as good as that for circular channels (line 2 [3]). This is due to the formation of stagnant zones in the recesses behind the ribs. The whole of the surface begins to contribute substantially to heat transfer for $(1/\mathrm{Re})(l/d) \leq 3 \cdot 10^{-2}$. It should be noted that the entry and flow conditions have little effect on the hydrodynamic resistance although they do influence heat transfer.

Line 1 on Fig. 2b is in agreement with the theoretical analysis of the resistance in the entry of circular channels.

NOTATION

Re is the Reynolds number;
Pr is the Prandtl number;
Bi is the Biot number;
Nu is the Nusselt number;
l is the length of the channel;
θ is the temperature difference;
m is the rate of sample cooling due to heat transfer with heat-transfer agent;
m_n is the rate of sample cooling due to heat losses;
m_0 is the total rate of sample cooling;

τ is the time;
α is the heat-transfer coefficient;
F is the heat-transfer surface;
C is the heat capacity of sample;
ψ is the nonuniformity coefficient of temperature;
δt is the increment of heat-transfer agent temperature;
W is the water equivalent of heat-transfer agent;
ε is the heat-transfer efficiency;
q_0 is the total heat flux from sample;
q_g is the heat flux to gas;
q_n are the heat losses to surroundings.

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