DESIGN OF COOLING CHANNELS WITH IMPROVED THERMOHYDRAULIC CHARACTERISTICS

G. N. Dul'nev, V. A. Korablev, and A. V. Sharkov UDC 536.242

A method is proposed for designing the profile of a slot channel so as to ensure a uniform temperature field in the cooled object.

In the design of cooling systems for equipment it is often necessary to ensure that the surface of the cooled element remain as nearly isothermal as possible with minimum expenditure of energy on pumping the cooling fluid. In widely used cooling channels with the fluid moving through the gap between the surface of the cooled element and the jacket surrounding it one finds, however, several factors which make solution of this problem very difficult. The major such factors are changes in the temperature of the fluid caused by heat transfer from the channel walls, changes in the coefficient of convective heat transfer within the thermally initial channel zone, and variation of the internal heat source over the length of the cooled element.

A uniform temperature field at the surface of an element can be obtained by matching the local coefficients of heat transfer from element surface to surrounding fluid. The magnitude of these coefficients depends on the flow rate and the thermophysical properties of the fluid, on the channel profile, the direction of the thermal flux, and the temperature field on the surface of the cooled element.

We will consider the steady-state temperature field of a plate-type body, a temperature field varying only along the x-coordinate in the direction of the fluid flow. The equation describing this temperature field is

$$\frac{\partial^2 T(x)}{\partial x^2} - \frac{\alpha(x)\Pi}{\lambda_s S}[T(x) - T_F(x)] + \frac{q(x)}{\lambda_s S} = 0. \tag{1}$$

The element surface will be isothermal when the condition

$$\frac{\partial T(x)}{\partial x} = 0. \tag{2}$$

Then the coefficient of convective heat transfer at the element surface must satisfy the equation

$$\alpha(x) = \frac{q(x)}{\Pi[T - T_F(x)]}. \tag{3}$$

With the heat-transfer coefficient referred to the difference between surface temperature T and fluid temperature at the channel entrance T_{in}, we have

$$\alpha^*(x) = \frac{q(x)}{\Pi(T - T_{in})}. \tag{4}$$

One of the most important methods of affecting the coefficient of convective heat transfer is varying the velocity of the fluid v(x), which depends on the local cross section S(x) and on the fluid flow rate G_V

$$v(x) = G_V/S(x). \tag{5}$$

We will assume that the thickness of boundary layers in a variable-section slot channel without flow separation from the walls depends on the distance x and on the gap width $\delta(x)$ in the same manner as in a flat channel. Then for calculating the heat-transfer coefficients one can use the expressions applicable to a flat channel. The validity of this assumption has been confirmed by the results of two studies [1, 2].

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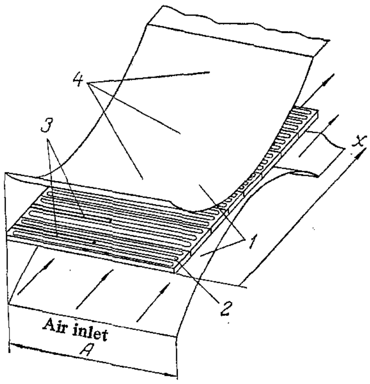

Fig. 1. Active segment of experimental channel.

We will use the relations for fluid flow: laminar

$$N_{Nu_d}^* = 0.46\left(N_{Re_d}\frac{d}{x}\right)^{0.5} N_{Pr}^{0.33}, \tag{6}$$

when $x/N_{Re_d}\, d < 0.1$ [3] and turbulent

$$N_{Nu_x}^* = 0.0289 N_{Re_x}^{0.8} N_{Pr}^{0.4} \tag{7}$$

when $2 \cdot 10^5 < N_{Re_x} < 10^7$ [4]. The values of N_{Nu^*} and N_{Re} in expressions (6) and (7) are referred to the difference between fluid temperature at the channel entrance and local temperature at a channel wall.

Expressions (6) and (7) yield for the heat-transfer coefficients

$$\alpha^*(x) = 0.46\lambda\left(\frac{v(x)}{\nu x}\right)^{0.5} N_{Pr}^{0.33}, \tag{8}$$

$$\alpha^*(x) = 0.0289\,\frac{\lambda}{x}\left(\frac{v(x)\,x}{\nu}\right)^{0.8} N_{Pr}^{0.4} \tag{9}$$

corresponding to laminar flow and to turbulent flow, respectively.

An expression for the local velocity in a channel whose hot plate is streamlined by a fluid on both sides (Fig. 1) will be derived from expression (5), with the area of the channel cross section represented in the form

$$S(x) = 2A\delta(x) = \Pi\delta(x). \tag{10}$$

Inserting expression (5) into expressions (8) and (9), with relation (10) taken into account, yields

$$\alpha^*(x) = 0.46\lambda\left(\frac{G_V}{\delta(x)\,\Pi\nu x}\right)^{0.5} N_{Pr}^{0.33} \tag{11}$$

for laminar flow and

$$\alpha^*(x) = 0.0289\,\frac{\lambda}{x}\left(\frac{G_V}{\delta(x)\,\nu\Pi}\right)^{0.8} N_{Pr}^{0.4} \tag{12}$$

for turbulent flow. From expressions (4), (11), and (12) one can obtain a formula for calculating $\delta(x)$ and thus designing the profile of a cooling channel on the basis of the fluid flow rate, the distribution of the internal heat source, the geometrical dimensions of the cooled object, the thermophysical properties of the fluid, and the required thermostabilization temperature

$$\delta(x) = 0.212\,\frac{G_V\Pi}{\nu x}\left[\frac{\lambda(T - T_{in})}{q(x)}\right]^2 N_{Pr}^{0.67} \tag{13}$$

for laminar flow or

$$\delta(x) = 0.0119\,\frac{G_V}{\nu}\left(\frac{\Pi}{x}\right)^{0.25}\left[\frac{\lambda(T - T_{in})}{q(x)}\right] N_{Pr}^{0.5} \tag{14}$$

for turbulent flow.

With air used as the cooling fluid, assuming an inlet temperature of $\sim 300°K$, expressions (13) and (14) become

$$\delta(x) = 7.46\,\frac{G_V \Pi}{x}\left[\frac{T - T_{in}}{q(x)}\right], \quad (15)$$

$$\delta(x) = 6.87\,G_V\left(\frac{\Pi}{x}\right)^{0.25}\left[\frac{T - T_{in}}{q(x)}\right]. \quad (16)$$

A channel profile designed according to these expressions will ensure attainment of an isothermal surface of the cooled element.

For verification of the applicability of the design relations (6) and (7) to the design of profiles of variable-section slot channels, there was constructed an experimental test stand (Fig. 1) consisting essentially of a variable-section channel formed by two plates 1 which had been bent into a specific profile with a 200 × 135 mm large heater plate 2 placed between them. The gap $\delta(x)$ between the heater plate and the channel walls was measured over the 2-11-mm range. The heater plate 2 had been split into eight separate segments so that electric power could be supplied to and measured in each. At the center of each plate segment was installed a copper–constantan thermocouple 3. The temperature of a channel wall 1 was monitored by copper–constantan thermocouples 4. The electric power $W_{el,i}$ supplied to each segment was regulated so as to raise the entire segment to the same temperature. The thermal energy released by each segment of the heater plate was carried away by air pumped through the channel and was transmitted by radiation from the channel walls.

The experimental heat-transfer coefficient was calculated according to the expression $\alpha^*(x) = (W_{el,i} - Q_{L,i})/F(T - T_{in})$.

An analysis of measurement errors revealed that heat leakage along the thermocouples, the current leads, and the plate fasteners had been negligible. The loss of thermal flux depended principally on radiative heat transfer from heater segment to channel wall and could be calculated according to the relation $Q_{L,i} = \sigma\epsilon_S F(T^4 - T^4_{w,i})$. Under the given experimental conditions this loss was small and did not exceed 15% of $W_{el,i}$ in the worst case.

As an example, in Fig. 2 are shown the results of three experiments with different air flow rates. The dots represent values of local heat transfer coefficients based on measurement and the curves represent profiles of the heat transfer coefficient based on calculations according to relation (8). The graph in Fig. 2 indicates that the trend of variation of experimental data along the channel corresponds to the proposed design relations.

The results of measurements and of calculations according to relation (6) are shown in Fig. 3. A comparison of calculated and experimental data reveals satisfactory agreement between them. A noticeable systematic difference appears only within the region of the narrowest channel segment, which can be attributed to essentially the following causes: the

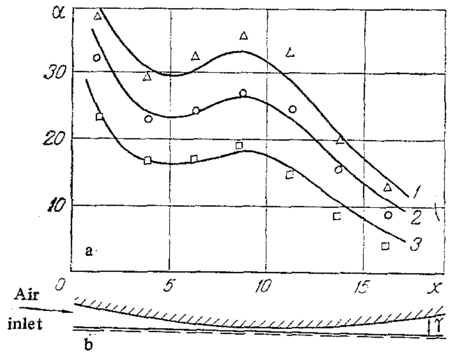

Fig. 2. Dependence of the intensity of convective heat transfer on the air flow rate and on the channel profile: (a) values of local heat-transfer coefficients with $N_{Re,d}$ = 1) 3490, 2) 2164, 3) 1082; (b) profile of experimental channel with heater plate underneath; α(W/m²·K), x(10⁻² m).

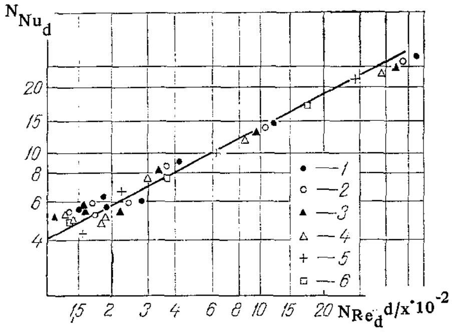

Fig. 3

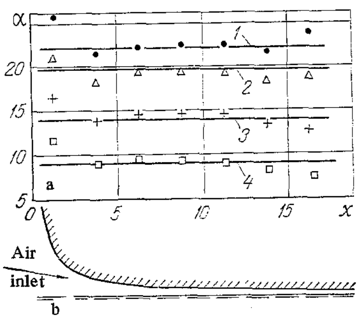

Fig. 4

Fig. 3. Results of experimental study of variable-section channel shown in Fig. 2b; N_{Re_d} = 1) 3490, 2) 3250, 3) 2970, 4) 2580, 5) 2164, 6) 1082; solid curve represents calculations according to relation (6).

Fig. 4. Dependence of the intensity of heat transfer on the air flow rate and on the channel profile, calculated according to relation (13) for q(x) = const: (a) values of local heat-transfer coefficients with N_{Re_d} = 1) 2750, 2) 2230, 3) 1870, 4) 435; curves represent calculations according to relation (11) with the same values of $N_{Re,d}$; (b) channel profile calculated according to relation (13) for q(x) = const, with the heater plate underneath; α ($W/m^2 \cdot K$), x (10^{-2} m).

experimental points deviating most lie near the limit of the applicability range for relation (6); already small deviations of the gap width δ(x) from the nominal one within the narrowest channel segment do appreciably affect the stream velocity and thus the heat-transfer coefficient.

For experimental verification of the method of designing a variable-section channel according to relation (13), there was designed and built a channel ensuring a uniform wall temperature at a uniform thermal flux density over the length of the heater plate. These experiments were performed with various air flow rates and various thermal flux densities. The results are shown in Fig. 4a. A satisfactory agreement between results of measurements and calculations (difference not exceeding 10%) is noted here within the channel region where the divergence angle γ (Fig. 2) between the two channel walls does not exceed 40°. This is not accidental, since it has been demonstrated [2] that relation (7) ceases to be valid for $\gamma > 40°$.

NOTATION

x, longitudinal coordinate, m; T(x), element temperature, °K; $T_F(x)$, the fluid temperature, °K; T, temperature of the isothermal surface, °K; T_{in}, fluid temperature at the channel entrance, °K; $T_{w,i}$, fluid temperature at a channel wall, °K; Π, wetted perimeter of the element, m; S, cross-sectional area of the cooled element, m^2; S(x), cross-sectional area of the cooling channel, m^2; G_V, volume flow rate of the fluid, m^3/sec; $W_{el,i}$, electric power supplied to a segment of the element, W; $Q_{L,i}$, loss of thermal flux, W; F, area of the heat emitting surface of an element segment, m^2; q(x), density of internal heat sources in the element, W/m; v(x), fluid velocity, m/sec; d, hydraulic diameter of the channel, m; α(x) and α*(x), coefficients of convective heat transfer referred to the local element temperature and to the local fluid temperature in the channel or to the fluid temperature at the channel entrance respectively, $W/m^2 \cdot K$; λ_S, thermal conductivity of the element material, W/m·K; λ, thermal conductivity of the cooling fluid, W/m·K; ϵ_S, referred emissivity of the channel wall material; ν, kinematic viscosity of the fluid, m^2/sec; α(x), gap width in a slot channel, m; σ, Stefan–Boltzmann constant, $W/m^2 \cdot K^4$; N_{Re}, Reynolds number; N_{Nu_d}* and N_{Nu_x}*, Nusselt numbers referred, respectively, to d and x as the characteristic dimension; and N_{Pr}, Prandtl number.

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HEAT TRANSFER IN ENCLOSURES WITH PROGRAMMED HEAT RELEASE

V. K. Aver'yanov and S. I. Bykov **UDC 536.21**

The problems of calculating the indoor air temperature, heat consumption, and room temperatures with arbitrary thermal perturbations are examined.

In estimating the efficiency of programmed heat release and optimizing it, it is necessary to calculate the thermal conditions indoors taking into account nonstationary heat-transfer processes. Including a large number of nonstationary thermal perturbations permits improving the temperature–humidity conditions indoors and decreasing the amount of heat used [1].

The difficulty of introducing thermal conditions operationally with the help of automated control systems increases considerably in the setup being examined. This is related to the fact that the known solutions [2, 3], obtained for harmonic, jumplike changes in $q(\tau)$, $t_0(\tau)$ and other actions, are either not accurate enough or greatly complicate the operational estimate of parameters, since in order to use the solutions, it is necessary to expand the heat use function $q(\tau)$, external temperature $t_0(\tau)$, and other functions in a Fourier series in each subsequent calculation. In this connection, it is useful to examine the solution of the problem of heat transfer indoors using splines [4, 5], which give a simpler algorithm for operational control of the temperature conditions.

The system of equations that describes heat transfer indoors can be represented in the following form: for the heat balance indoors in accordance with [1]

$$VC_p \frac{dt_i(\tau)}{d\tau} = -F_o k_o (t_i(\tau) - t_o(\tau)) - C_i(\tau)(t_i(\tau) - t_o(\tau)) + q(\tau) - \sum_j F_{oj}\alpha_{oij}(t_i(\tau) - t_{ocj}(\tau)) - \sum_j F_{ii}\alpha_{ij}(t_i(\tau) - t_{icj}(\tau)),$$

for heat transfer in the external walls [2, 6]

$$\frac{\partial T_{oj}(x,\tau)}{\partial \tau} = a_{oj}\frac{\partial^2 T_{oj}(x,\tau)}{\partial x^2}, \quad 0 < x < l_{oj}, \tag{2}$$

$$-\lambda_{oj}\frac{\partial T_{oj}(0,\tau)}{\partial x} = \alpha_o^j (t_i(\tau) - T_{oj}(0,\tau)), \tag{3}$$

$$\lambda_{oj}\frac{\partial T_{oj}(l_{oj},\tau)}{\partial x} = \alpha_{oj}(t_o(\tau) - T_{oj}(l_{oj},\tau)), \tag{4}$$

$$T_{oj}(x,0) = T_{oj}(x). \tag{5}$$

An analogous system of equations is valid for the interior walls:

$$\frac{\partial T_{ij}(x,\tau)}{\partial \tau} = a_{ij}\frac{\partial^2 T_{ij}(x,\tau)}{\partial x^2}, \quad 0 < x < \frac{l_{ij}}{2}, \tag{6}$$

$$\lambda_{ij}\frac{\partial T_{ij}\left(\frac{l_{ij}}{2},\tau\right)}{\partial x} = \alpha_{ij}\left(t_i(\tau) - T_{ij}\left(\frac{l_{ij}}{2},\tau\right)\right), \tag{7}$$

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