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EXPERIMENTAL STUDY OF NONSTEADY HEAT CONVECTION IN A PLANE CHANNEL

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UDC 536.24

Results are presented from an experimental study of the effect of thermal and hydrodynamic transience on convective heat transfer in a short plane channel.

Nonsteady thermal processes play a significant and in several cases determining role in modern power plants and manufacturing facilities. To reliably calculate these processes, it is necessary to know the laws governing nonsteady heat convection for different laws of change in the boundary conditions and for different channel cross sections. Several studies [1, 2] have examined processes in circular tubes in detail. Here we perform a special investigation of nonsteady heat transfer in plane channels.

As is known, nonsteady problems of convective heat transfer should be examined as coupled problems [2]. However, such problems cannot be solved theoretically due to the lack of data on the distribution of the turbulent coefficient of momentum transfer and turbulent heat-transfer coefficient across the channel.

Experimental study of coupled problems makes sense only to check specific solutions, since it is impracticable to generalize such experiments due to the effect of additional parameters. Thus, engineering methods of calculation are presently constructed with the use of a unidimensional description of processes in the heat carrier. Here, a local heat-transfer coefficient is employed

$$\alpha(x,\ \tau) = \frac{q_w(x,\ \tau)}{T_w(x,\ \tau) - T_b(x,\ \tau)}, \tag{1}$$

and it is dependent on the nonsteady boundary conditions.

It has been shown in several theoretical and experimental studies of nonsteady heat transfer that the heat-transfer coefficients found under variable thermal loads and heat-carrier flow rates may differ appreciably from the coefficients calculated from quasisteady relations [1, 2]. With a turbulent flow regime, the effect of thermal and hydrodynamic transience on the relationship between the nonsteady and quasisteady heat-transfer coefficients is expressed in the form of the equation

$$K = \frac{\mathrm{Nu}}{\mathrm{Nu}_s} = f\left(K^*_{Tg},\ K_G,\ \mathrm{Re},\ \frac{T_w}{T_b},\ \frac{x}{d_\mathrm{e}}\right), \tag{2}$$

where

$$K^*_{Tg} = \frac{\partial T_w}{\partial \tau}\,\frac{1}{T_w}\sqrt{\frac{\lambda_b F}{c_p g G}} \tag{3}$$

and

$$K_G = \frac{\partial G}{\partial \tau}\,\frac{d_\mathrm{e}^2}{G \nu_b} \tag{4}$$

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are parameters proposed in [1] as criteria of the effect of transience on the structure of turbulent flow and heat transfer.

The working section of the experimental unit was a horizontal plane channel 300 mm long with a 40 × 9 mm cross section formed by a thick-walled steel frame. The channel was subdivided into two symmetrical channels by a planar electric heater. The heater consisted of two steel sheets 40 mm wide and 0.2 mm thick. The plates were kept from moving vertically by being fastened in textolite guides located in the wide walls of the frame. The plates were placed in tension by a spring to compensate for strains during heating. The plate was soldered to a copper busbar in the inlet part of the channel. A gap of 3.6 mm was left between the plate and busbar for the thermoelectrodes of Chromel–Copel thermocouples and a glass-cloth liner. The thermocouples were soldered 20 mm apart to the inside of each plate. They were also installed in the unheated walls of the frame to measure the air temperature in the channel inlet and outlet. The presence of an electric-heater packet of two plates located symmetrically in the flow in the middle of the channel allowed us to obtain two identical plane channels 2.5 mm high (d_e = 4.7 mm) with unidirectional heating. One advantage of this scheme is the absence of any need to allow for the escape of heat from the heated wall of the channel or to install compensating heaters. The results obtained are also more reliable. Thus, the design of the experimental unit made it possible to study heat transfer in a unidirectionally heated plane channel with simultaneous stabilization of the velocity and temperature fields.

The unit allowed us to reproduce nonsteady thermal processes both with variable heat release in the channel wall and with a variable rate of flow of the heat carrier. The rate of heat release was changed by connecting or disconnecting a voltage source to the electrically heated plates with a constant rate of air flow in the channel.

In the experiments with a variable rate of flow of the heat carrier, it was necessary to isolate the effect of hydrodynamic transience. This in turn required that the temperature of the heated wall be kept constant while the air flow rate was changed. To do this, we used a device which changed the voltage drop on the plate with the closure and opening of the electropneumatic valve which regulated air flow rate [3].

During the experiments we measured the following quantities: the temperature of the inside surface of the heated plate – in 11 sections on the long axis of the plate at distances from the beginning of heating x/d_e = 2.13; 6.4; 10.6; 14.9; 1915; 23.4; 27.6; 31.9; 36.2; 39.8; 43.6; the temperature of the unheated wall of the channel (at the same distances from the beginning of heating) at depths of 3 and 10 mm from the surface of the frame exposed to the flow of heat carrier; the temperature of the air in the channel inlet and outlet; pressure in the channel inlet; the voltage drop and current on the heated plates. In the experiments with variable heat release, air flow rate was measured by a diaphragm. In the experiments with variable flow rate, it was measured by a ring with a post-critical pressure gradient [4]. Loop oscillographs of the N010-M type were used as the recording devices.

The main parameters were measured in the experiments in the following ranges: volumetric heat release $q_V = 0\text{-}4.5 \cdot 10^8$ W/m³; heat flux $q_W = 0\text{-}1.8 \cdot 10^5$ W/m²; temperature of the heated wall T_{W1} = 278-540°K; air temperature in the outlet T_b = 278-310°K; temperature factor T_{W1}/T_b = 1=1.75; total air flow rate $G = (8.5\text{-}68.5) \cdot 10^{-3}$ kg/sec; Reynolds number determined from the equivalent diameter $Re = (1.0\text{-}8.5) \cdot 10^4$; rate of change in wall temperature $|\partial T_{W1}/\partial \tau|$ = 0-70 K/sec; rate of changed in flow rate $|\partial G/\partial \tau| = (4\text{-}20) \cdot 10^{-3}$ kg/sec².

The experimental data was analyzed by the following method. Heat flux on the heat-transfer surface was determined from the heat balance equation for the heat-releasing plate:

$$q_{w1}(x, \tau) = q_{v1}(x, \tau)\,\delta_1 - q_{\text{absb}1}(x, \tau) - q_{\text{rad}}(x, \tau). \tag{5}$$

Since Bi << 1 under the conditions of the experiment, then

$$q_{\text{absb}1}(x, \tau) = (\rho c)_w \delta_1 \left(\frac{\partial T_{\text{u}1}}{\partial \tau}\right), \tag{6}$$

while the temperature of the surface of the heated plate exposed to the flow of heat carrier can be determined from the heat balance equation for this surface [1]:

$$T_{w1}(x, \tau) = T_{\text{u}1}(x, \tau) - \frac{\delta_1^2}{2\lambda_w}\left[q_{v1}(x, \tau) - (\rho c)_w \left(\frac{\partial T_{\text{u}1}}{\partial \tau}\right)\right], \tag{7}$$

where $T_{u1}(x, \tau)$ and $q_{v1}(x, \tau)$ are determined during the experiment.

The radiant heat flux was calculated from the formula

$$q_{rad}(x, \tau) = \varepsilon_{cr} c_s \left\{ \left[\frac{T_{w1}(x, \tau)}{100} \right]^4 - \left[\frac{T_{w2}(x, \tau)}{100} \right]^4 \right\}; \quad (8)$$

and the value of ε_{cr} for the heated plate was determined in special calibration tests and was determined to be equal to 0.4.

In comparing the heat balance in the test channel (in contrast to tubes and channels with a bidirectional heat supply), it is necessary to take into account heat transfer on the unheated wall. This heat transfer can be considered if we know the thermal radiation from the heated plate to the unheated plate and the quantity of heat used to heat this wall. Thus, the heat flux in the heat carrier

$$q_b(x, \tau) = q_{w1}(x, \tau) + q_{w2}(x, \tau), \quad (9)$$

where

$$q_{w2} = q_{rad} - q_{absb2}. \quad (10)$$

Considering that the temperature profiles in the wall change over the thickness in accordance with a law which is close to linear, we can determine the heat absorbed by the wall from the formula

$$q_{absb2}(x, \tau) = (\rho c)_{w2} \delta_2 \frac{\left[\left(\frac{\partial T_{w2}}{\partial \tau} \right) + \left(\frac{\partial T_{w2}}{\partial \tau} \right) \right]}{2}. \quad (11)$$

We can use the known temperature of the heat carrier at the inlet T_{b0} and the change in heat flux $q_b(x, \tau)$ over the length and time to calculate the change in the mean-mass temperature of the flow over the length:

$$T_b(x, \tau) = T_{b0}(\tau) + \frac{b}{G(\tau) c_p} \int_0^x q_b(x, \tau)\, dx. \quad (12)$$

Then Eq. (1) is used to calculate values of the heat-transfer coefficient $\alpha_1(x, \tau)$ and the Nusselt number Nu. Values of Nu_s were determined from formulas obtained from analysis of experiments under steady-state conditions on the given working section:

with $x/d_e \leq 30$

$$Nu_s = 0.025 \left(\frac{x}{d_e} \right)^{-0.19} Re^{0.8}, \quad (13)$$

with $x/d_e > 30$

$$Nu_s = 0.013 Re^{0.8}. \quad (14)$$

These formulas agree well with well-known data for a planar tube with unidirectional heating [5]. The nonsteady and steady values of the heat-transfer coefficient were compared in the relation

$$K = \frac{Nu(x, \tau)}{Nu_s(x, \tau)} \Big/ \frac{Nu_\infty(x)}{Nu_{\infty s}(x)}. \quad (15)$$

With a sudden increase in the thermal load, the temperature of the wall increases more rapidly, the greater the relative distance from the inlet x/d_e (Fig. 1). Here, the heat-transfer coefficient is higher with an increase in wall temperature than it is under steady-state cnditions ($K > 1$). The difference of K from 1 is greater, $\partial T_{w1}/\partial \tau$.

With a decrease in heat release on the plate, the drop in T_{w1} occurs more rapidly as x/d_e increases (Fig. 1). Here, $K < 1$, i.e., heat transfer is less than under steady conditions. Meanwhile, the difference of K from 1 increases with an increase in $|\partial T_w/\partial \tau|$. Values of K for the above experimental conditions changed within the range (1-4) with an increase in the thermal load and changed within the range (0.7-1) with a decrease in the thermal load.

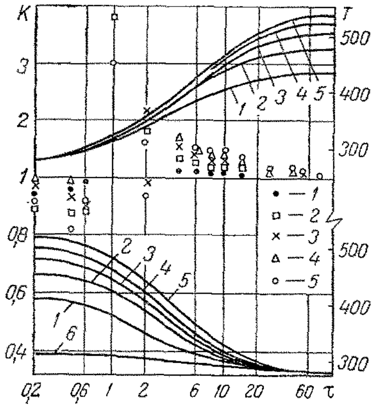

Fig. 1. Dependence of the parameter K, the temperature of the heated wall T_{w1}, and the temperature of the heat carrier T_b on time with an increase (K > 1) and a decrease (K < 1) in the thermal load for Re = (2-2.5)·10^4, T_{w1}/T_b = 1-1.66; 1-5) T_{w1} (lines) and K (points)at x/d_e = 10.6; 19.1; 27.0; 36.2; 44.6; b) T_b, T_{w1}, T_b, K; τ, sec.

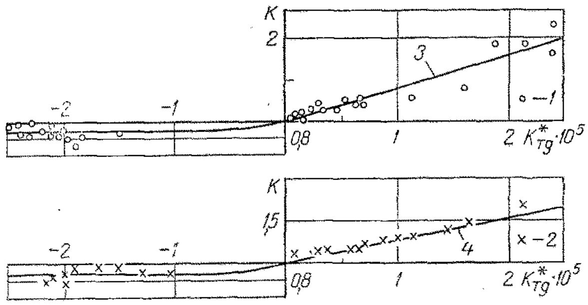

Fig. 2. Effect of the number Re on the dependence of K on the parameter K^*_{Tg} for x/d_e = 19; 1, 2) experimental data for Re = (2-2.5)·10^4 and (8-9)·10^4; 3, 4) generalizing relations for Re = 2.25·10^4 and Re = 8.5·10^4, respectively.

In generalizing the test data, it was necessary to determine the dependence of K on K^*_{Tg}, Re, T_w/T_b, x/d_e. Since the changes in these parameters during the tests were related to each other, the analysis was performed as follows. All of the values of K and K^*_{Tg} obtained for each x/d_e as a result of analysis of the test data were distributed according to the ranges of Re and T_{w1}/T_b to which they belonged: for Re – (1.0-1.5)·10^4; (1.5–2)·10^4; (2.5–3)·10^4 – (8–8.5)·10^4; but T_w/T_b – (1–1.1); (1.1–1.2)–(1.7–1.8). We determined the dependence of K on K^*_{Tg} for each range. This dependence was referred to mean values of Re and T_{w1}/T_b for each range of their variation.

Figure 2 shows average dependences of K and K^*_{Tg} with different Re. With an increase in the load, K is greater, the greater K^*_{Tg}. With a decrease in the load, K is smaller, the greater $|K^*_{Tg}|$. With an increase in Re, the effects of transience weaken both at $\partial T_{w1}/\partial\tau > 0$ and at $\partial T_{w1}/\partial\tau < 0$. We did not detect any effect of the temperature factor on K in our analysis. At $K^*_{Tg} > 0$, the effect of transience increases with an increase in x/d_e.

In the investigated range of variation of the parameters of the nonsteady processes, the results of the experiments are generalized by the following relations:

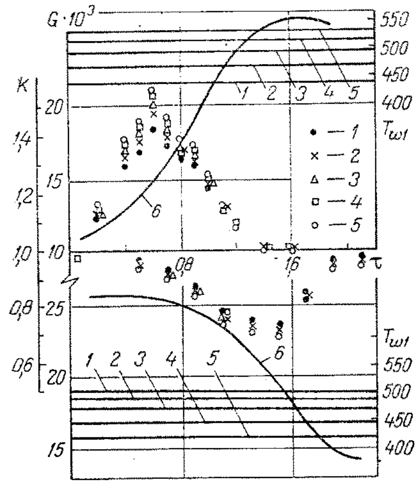

Fig. 3. Dependence of G, T_{w1}, T_b, and the parameter K on time with acceleration ($K_G > 1$) and deceleration ($K_G < 1$) of the flow of heat carrier for $T_{w1}/T_b = 1.6$; 1-5) T_{w1} (lines) and K (points) at x/d_e = 10.6; 19.1; 27.7; 36.2; 44.0; 6) $G \cdot 10^3$ kg/sec.

for $0.22 \cdot 10^{-4} > K^*_{Tg} > 0$

$$K = 1 + \left(\frac{460}{\mathrm{Re}^{0.51}} + \frac{8.13}{\mathrm{Re}^{0.46}} \frac{x}{d_e}\right) K^*_{Tg} \cdot 10^4, \quad (16)$$

for $0 > K^*_{Tg} > -0.22 \cdot 10^{-4}$

$$K = \exp(0.813 K^*_{Tg} \cdot 10^4). \quad (17)$$

Thus, the completed experiments show that with a change in wall temperature, nonsteady heat transfer can be described by means of a heat-transfer coefficient dependent both on the dimensionless parameters Re and x/d_e and on the nonsteady boundary conditions or the parameters K^*_{Tg}. With a change in the thermal load or wall temperature T_{w1}, heat transfer is influenced mainly by deformation of the temperature profile. This deformation is accounted for by the derivative $\partial T_w/\partial \tau$ and is determined by the parameter K^*_{Tg}. The effect of the number Re on heat transfer is indirectly considered by means of the parameter K^*_{Tg}, which for identical $\partial T_w/\partial \tau$ and T_w decreases with an increase in Re. At $K^*_{Tg} < 0$, this effect is taken into account so accurately that it is not necessary to additionally account for the effect of Re. At $K^*_{Tg} > 0$, the effect of Re is weaker and is accounted for by Eq. (16). Thus, the effect of thermal transience on heat transfer decreases with both an increase and a decrease in the thermal load. More accurately, an increase or decrease in thermal load is accompanied by a decrease in the relative change in turbulence due to nonsteady thermal action, since the level of turbulence increases with an increase in Re.

Figure 3 shows results of analysis of experiments with a variable rate of flow of the heat carrier. Acceleration of the flow leads to an increase in the heat-transfer coefficient compared to the quasisteady case (K = 1.0-2.0). With deceleration of the flow, heat transfer is below the quasisteady value (K = 1.0-0.6). The value of K approaches 1 in both cases as flow rate stabilizes.

The test data obtained with a change in flow rate was generalized in the same manner as with a variable thermal load. The graphs in Fig. 4 reflect the effect of Re on the dependence of K on K_G with fixed x/d_e. It can be seen from the figure that an increase in $|K_G|$ leads to an increase in the difference between nonsteady heat transfer and quasisteady heat transfer.

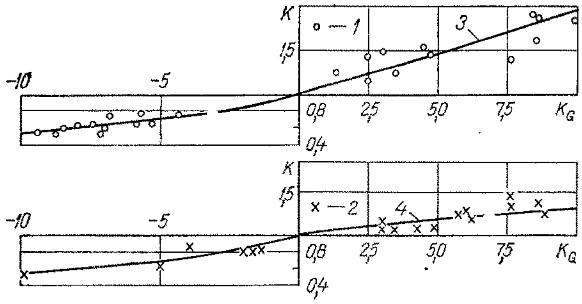

Fig. 4. Effect of the number Re on the dependence of K on the parameter K_G for $x/d_e = 19$; 1, 2) experimental data for $Re = (1\text{-}1.5)\cdot10^4$ and $Re = (3\text{-}4)\cdot10^4$; 3, 4) generalizing relations for $Re = 1.25\cdot10^4$ and $Re = 3.5\cdot10^4$.

Meanwhile, at $K_G > 0$ and $K > 1$, this dependence becomes weaker with increase in Re. At $Re > 4.5\cdot10^4$, hydrodynamic transience has no effect on heat transfer according to our observations. Analysis of similar curves with different values of x/d_e and Re allowed us to obtain the following relations for heat transfer in the case of nonsteady flow rate (for the above-indicated parameters of the process):

for $12 > K_G > 0$

$$K = 1 + \left\{8 + 44\exp\left[\mathrm{Re}\cdot10^{-4}(0.71 - 0.296\mathrm{Re}\cdot10^{-4})\right] + (1.4 - 0.36\mathrm{Re}\cdot10^{-4})\frac{x}{d_e}\right\} K_G \cdot 10^{-3}, \tag{18}$$

for $-12 < K_G < 0$

$$K = \exp\left(\frac{0.065K_G}{\mathrm{Re}\cdot10^{-4}}\right). \tag{19}$$

Equations (16-19) generalize the experimental data with a scatter of no more than 15-20%.

If we compare data on the effect of thermal and hydrodynamic transience in a plane channel with similar effects in a circular tube [2], we see that they agree qualitatively. However, the effect of transience is stronger in the plane channel.

The results of the present study indicate a need to allow for the effect of nonsteady boundary conditions on the heat-transfer coefficient in the calculation of nonsteady thermal processes in plane channels.

The results can be used to develop engineering methods of calculating nonsteady processes in heat exchangers, including plane channels. Similar to [6], they also make it possible to determine the range of applicability of quasisteady methods with a specified permissible accuracy of averaging of the heat-transfer coefficient.

NOTATION

α, heat-transfer coefficient; x, coordinate along the channel; τ, time; q, heat flux; T, temperature; λ, thermal conductivity; F, cross-sectional area of the channel; c_p, specific heat; $g = 9.8\ g/sec^2$; G, mass flow rate; d_e, equivalent diameter of the channel; ν, kinematic viscosity; Re, Reynolds number; Bi, Biot number; δ, wall thickness; $(\rho c)_w$, volumetric specific heat of the wall; ε_{cr}, corrected emissivity; c_s, radiation coefficient of a blackbody; q_{rad}, radiant heat flux; q_{absb}, heat flux absorbed by the wall; b, channel width; Nu, Nusselt number obtained from analysis of the experiment; Nu_s, Nusselt number calculated from the quasisteady relation; Nu_∞, Nusselt number for the given x/d_e at the end of the nonsteady process ($\tau \to \infty$); $Nu_{\infty s}$, quasisteady value of the Nusselt number for the given x/d_e and $\tau \to \infty$; K^*_{Tg}, parameter of thermal transience; K_G, parameter of hydrodynamic transience; K, quantity characterizing the difference between the nonsteady heat-transfer coefficient and the quasisteady value. Indices: w, for the surface exposed to the flow; b, for the heat carrier; u, for the unexposed surface; 1, heated wall; 2, unheated wall; 0, inlet; s, quasisteady relation.

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EXPERIMENTAL STUDY OF REGIMES OF LOW-VELOCITY MOTION OF A SUBHEATED LIQUID IN HORIZONTAL ANNULAR CHANNELS WITH A HEAT-TRANSFER CRISIS

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The range of existence of different regimes of flow of a two-phase mixture during initiation of a heat-transfer crisis is determined.

The use of gravitational—evaporative cooling systems with natural circulation of the coolant is promising for certain thermally loaded elements of radioelectronic equipment having annular channels. Such systems make it possible to circulate the coolant at velocities up to 0.6 m/sec at pressures of 0.2-1.5 MPa in the cooling system.

The lack of reliable recommendations on calculating the critical heat flows in the low-pressure region makes experimental study of such processes important. A significant volume of expermental data has been accumulated on the heat-transfer crisis during boiling for systems with vertical heated channels at high coolant pressures (more than 10 MPa [1, 2]). Since the conditions of cooling of the heating surface are different for horizontal and vertical flows at low circulation rates, it is not possible to use the available data on q_{cr} for vertical flows.

As is known [3, 4], the critical heat flux depends mainly on the regime of motion of the two-phase mixture in the channel. Regime diagrams are used to identify different structures. The domestic and foreign literature contains different modifications of such diagrams for vertical and horizontal channels. However, most of these diagrams have been constructed for adiabatic flows. Thermal nonequilibrium connected with the flow of subheated liquid near the heated surface has a certain effect on features of the motion of a two-phase mixture. This is manifested most clearly at specific heat fluxes close to the critical fluxes. There is no data in the literature on the mechanism of occurrence of the crisis in horizontal channels under the conditions typical of gravitational-evaporative cooling systems. It is therefore interesting to study the conditions of occurrence of the heat-transfer crisis in horizontal channels at low pressures ($\rho'/\rho'' \approx 10^3$).

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