STUDY OF HEAT AND MASS TRANSFER BETWEEN A HEATED CHANNEL AND AN UPPER MIXING CHAMBER

S. D. Malkin, V. B. Khabenskii,
Yu. A. Migrov, and V. K. Efimov

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Results are presented from experimental studies and criterional equations are obtained for heat and mass transfer between a vertical heated channel and an upper mixing chamber in the case of unstable stratification of the density of the coolant and low rates of forced motion.

At low rates of forced motion of the coolant in the first loop in nuclear power plants and IRT-type research reactors in certain transient and emergency regimes, heat removal from the active zone may take place not only as a result of forced convection, but also by mass transfer between the channel and the upper mixing chamber of the reactor.

The cause of this mass transfer is unstable stratification of density within the channel and chamber, i.e., the coolant in the chamber is cooler than the coolant in the channel. At low rates of forced motion, this creates favorable conditions for the development of natural convection. The presence of mass transfer between the channel and the upper chamber is important from the point of view of the reliability of cooling of the active zone of the reactor, since it makes it possible to remove heat even in the case of the cessation of forced circulation through the active zone or individual channels and, thus, to significantly damp an increase in the temperature of the coolant and fuel elements. As an example, we can cite several regimes in which unstable stratification of density occurs at low rates (or in the absence) of forced motion:

1) regime of heating of a water-moderated, water-cooled power reactor (WWPR) under pressure with natural circulation from the cold state;

2) regime of emergency cooling of the WWPR under pressure with natural circulation, when, due to nonsynchronous reduction of power and flow rate, the coolant in the upper chamber may cool and the coolant flow rate may decrease at certain moments of time;

3) regime of emergency disconnection of IRT-type research reactors in which forced circulation is reduced in a controlled manner under normal conditions. In this case, after stoppage of the pumps, the interaction of gravitational and inertial forces may lead to a cessation or change in the direction of circulation in the active zone or part of the channels.

Despite the importance and urgency of these problems, they have been studied very little. Here, greatest attention has been given to thermal processes in dead-end vertical chambers, referred to as thermosiphons in [1-3].

In most studies made of thermosiphons, the problem posed was that of determining the heat transfer rate in such systems, i.e., obtaining the dependence of the Nusselt number on the determining parameters. The investigators generally have examined short dead-end channels with a relative height $l/d < 15$. Experimental study of the character of fluid flow in such channels has shown that laminated flow occurs, whereby the cold liquid descends in the center and the hot liquid rises next to the channel wall, forming a clearly distinguishable boundary layer. Theoretical solution in a boundary-layer approximation and analysis of the equations presented in [3, 4] shows that the determining similitude criteria for these processes are the criterion Ra and parametric criteria of geometric similitude (l/d for a pipe). In [4], in particular, it was shown that the complex (d/l)Ra can be used to generalize experimental data on heat transfer in thermosiphons.

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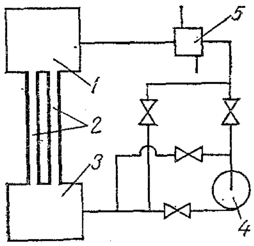

Fig. 1

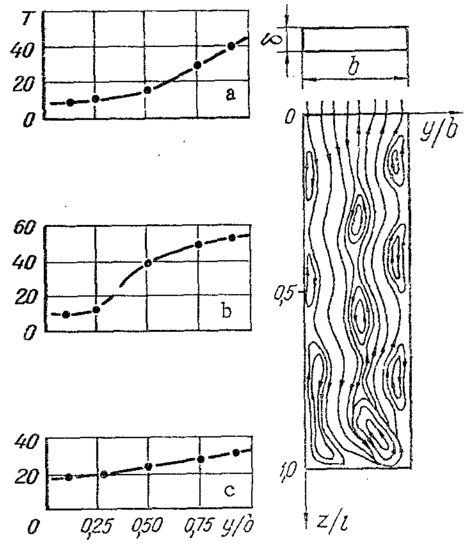

Fig. 2

Fig. 1. Diagram of thermophysical stand: 1, 3) upper and lower mixing chamber; 2) slit channels of working section; 4) circulating pump; 5) heat exchanger.

Fig. 2. Distribution of coolant temperatures over the width of the slit channel at different stations along the channel ($W_f = 0$, $t_{uc} = 2.5°C$): a) $z/l = 0.06$; b) 0.25; c) 0.45, T, °C.

Under reactor conditions, the relative height of process channels in the active zone is generally an order greater than the relative height of the thermosiphons used and amounts to about 100-200. Thus, boundary-layer conditions are realized in such channels only on the inlet section, comprising a small part of the total height. This is the first circumstance which prevents the use of recommendations in the literature on thermosiphons for reactor conditions. The second circumstance, which makes it necessary to conduct special experimental studies, is that nearly all of the recommendations in the literature pertain to pipes, and it is doubtful that they could be applied to the narrow slit-type channels typical of research reactors or to the channels with fuel-element bundles used in nuclear power plants.

To determine the quantitative characteristics of mass transfer between channels of the active zone and the upper chamber in the absence and at low rates of forced circulation, we conducted experimental studies on a thermophysical stand (Fig. 1). The working section was two parallel slit channels (channel length $l = 1000$ mm, width $b = 70$ mm, slit gap $\delta = 3$ mm) joined by upper and lower mixing chambers. The channels have independent electrical heating. Systems of such stands provided for any small (including zero) values of forced flow rate with both descending and ascending circulation of the coolant. The temperature of the liquid in the upper mixing chamber in each experiment was kept constant, which ensured unstable stratification of the density of the liquid between the chamber and the heated channel for any direction of flow. The system of empirical measurements allowed us to fix the rate of forced circulation of the coolant in the loop, the flow rate in one of the parallel channels, the temperature of the liquid in the mixing chambers, and the temperature field over the height and width of one of the channels.

In the first stage of the experiment, we studied the structure of the flow in the slit channel in the absence of forced circulation in the channel and with low rates of forced flow of coolant from top to bottom. Figure 2 shows a typical temperature field over the channel width and height and the flow structure as determined from visual observation. It is apparent from the figure that the character of the flow in the narrow slit channels is quite different from the character of flow in short pipes. In the slit channel, the flow is asymmetrical to the degree that there is downward flow in one part of the channel width and upward flow in the other part. The ascending and descending flows of coolant in the slits are wavelike over the channel height, with the formation of eddies between the two oppositely directed flows.

The completed study showed that the character of coolant motion which exists in the presence of mass exchange between the channel and the upper chamber ensures an almost con-

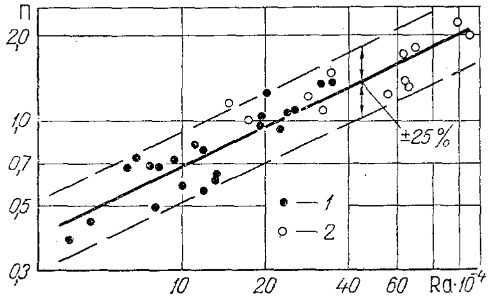

Fig. 3. Dependence of dimensionless temperature of liquid in a heated channel on Ra in the absence of forced motion in the channel: 1) slit channel with l/d_{eh} = 173 and q = const; 2) slit channel with l/d_{eh} = 50 and t_w = const; $\Pi \equiv \frac{q d_{eh}}{\lambda_l (t_c - t_{uc})}$.

stant value over the channel height for the average temperature of the coolant across the channel (t_c). Knowledge of this temperature is necessary to determine the dynamic head and the flow rate through the channel in regimes with natural circulation of coolant, as well as to calculate the temperature of the wall of the fuel elements.

Let us first examine the case when there is no forced flow through the heated channel and the thermal state of the coolant in the channel is completely determined by mass exchange between the channel and the upper chamber. We will assume that in this case there is a certain effective exchange flow rate of coolant between the channel and upper chamber which ensures removal of heat from the channel given a certain difference between the characteristic temperature of the coolant in the channel t_c and the temperature in the upper chamber t_{uc}. By analogy with forced flow, we can determine the effective exchange flow rate from a relation which is convenient in dynamic calculations:

$$G^0_{exc} = \frac{Q}{c_p (t_c - t_{uc})}. \tag{1}$$

To determine the characteristic temperature of the coolant in the channel (t_c), we conducted experimental studies on the stand by a special method which consists essentially of the following.

Heat is supplied to one of the parallel channels of the working section. In the absence of forced motion of the coolant in the loop, this supplying of heat leads to natural circulation with an ascending flow in the heated channel and a descending flow in the unheated channel. We then connect the pump and gradually (in small steps) increase the rate of forced circulation in the loop, with the coolant flowing downward. Here, upward flow is retained in the heated channel in a certain range of liquid flow rates, while downward flow of coolant is established in the unheated channel with a rate equal to the sum of the rates of forced flow in the loop and the rate of ascending flow in the heated channel. An increase in the flow rate in the loop* with a fixed thermal load leads to a gradual decrease in the rate of ascending flow of the liquid in the heated channel down to a zero rate of flow and subsequent reversal of circulation in it.

Since the parallel channels are coupled hydraulically, then for the conditions of absence of coolant flow in the heated channel we write:

$$\lambda_v \frac{l}{d_{eu}} \frac{G_f^2}{2\rho_u F_u^2} = gl(\rho_u - \rho_h). \tag{2}$$

We can use Eq. (2) to determine the mean density of the coolant in the heated channel in the absence of forced motion in same and, thus, the effective temperature of the liquid

*A steady-state flow rate was reached at each stage in the experiment, after which we recorded the parameters measured.

(t_c) in the channel. The latter is in this case completely determined by the rate of mass transfer between the channel and the upper mixing chamber. The exchange flow rate is determined from Eq. (1) with allowance for the known capacity of the channel and the temperature of the coolant in the chamber.

Thus, to determine the rate of mass transfer between the heated channel and the upper mixing chamber by the above method, it is necessary to record parameters of the system corresponding to a change in the flow direction in this channel from ascending to descending. In our experiment, the transition to descending flow in the channel was determined from the sharp change in the readings of thermocouples installed at its inlet and outlet.

To generalize experimental data on the basis of analysis of the process dimensions and equations in [3, 4], it is easy to obtain the dimensionless determining parameter $\Pi = \frac{q\, d_{eh}}{\lambda_l (t_c - t_{uc})}$ and the dimensionless determining parameters (for a slit channel) Π_1 = Ra; $\Pi_2 = l/d_{eh}$; $\Pi_3 = b/d_{eh}$.

In [5] in a study of the stability of convection motion of a single-phase liquid, it was shown that for closed slit channels with $l/d_{eh} \gg 1$, $b/d_{eh} \gg 1$ the effect of the parameters Π_2 and Π_3 decays, and the thermal-hydraulic characteristics of the channel in the region of instability are determined by the parameter Π_1. Thus, it can be suggested that under the conditions examined, with $\Pi_2 \gg 1$ and $\Pi_3 \gg 1$ and a distinct wave motion of the coolant, a universal relation of the following type may exist

$$\Pi = f(\Pi_1). \tag{3}$$

An analysis of the empirical data in the coordinates (Π, Π_1) is presented in Fig. 3. Most of the experiments were conducted on a slit channel with l/d_{eh} = 173 with a uniform heat flow. Also plotted in Fig. 3 is empirical data obtained on a slit channel with l/d_{eh} = 50 with a constant wall temperature. All of the empirical points, with a certain scatter due to measurement error, can be approximated by the relation

$$\frac{q d_{eh}}{\lambda_l (t_h - t_{uc})} = 3.5 \cdot 10^{-3}\, \mathrm{Ra}^{0.46}. \tag{4}$$

Given a certain temperature in the upper chamber and a certain heat flux on the heating surface of the channel, Eq. (4) makes it possible to determine the weighted mean temperature of the coolant in the channel t_c, while Eq. (1) permits determination of the exchange flow rate G^0_{exc}. We use $t = 0.5(t_c + t_{uc})$ as the determining temperature in the criterion Ra when determining the thermophysical parameters. Thus, the temperature should be calculated from Eq. (4) iteratively, since the thermophysical parameters in Ra depend on t_c. A limited number of experiments was conducted on fluted slit channels simulating the through section of a rod bundle. The tests showed that the structure and stability of the flow in such channels are governed by the same laws as in smooth slit channels. This gives us a basis for concluding that Eq. (4) can be used for rod bundles, although additional tests are required for a final conclusion.

Forced motion of the coolant in principle decreases the rate of mass transfer between the heated channel and the upper mixing chamber. Analysis of the test data shows that a linear dependence of the exchange flow rate on the rate of forced circulation can be used as a first approximation. Here, mass exchange between the channel and chamber ceases when the rate of forced motion of coolant becomes equal to the exchange flow rate obtained for conditions of the absence of forced motion. The exchange mass flow rate can be approximately determined by considering that the area of the cross section for an exchange flow entering from the chamber into the channel and leaving from the channel into the chamber is equal to

$$(\rho W)^0_{exc} = \frac{2G^0_{exc}}{F_h}. \tag{5}$$

The exchange flow rates determined in the tests agreed with the theoretical values obtained from (5). In the range of the parameters in which the tests were conducted on the thermophysical stand, the values of Re^0_{exc} changed within the range 600-1000. At these Reynolds numbers of forced motion, we see a transition to a stable flow pattern with a coolant temperature distribution over the channel height corresponding to a unidimensional description of the process (i.e., with the equality $Re^0_{exc} = Re_f$, mass transfer between the channel and upper chamber ceases). Thus, the exchange flow rate in the presence of forced motion can be determined in accordance with the following relation:

$$G_{exc}=\begin{cases} G^0_{exc}\left(1-\dfrac{Re_f}{Re^0_{exc}}\right) & \text{at } \dfrac{Re_f}{Re^0_{exc}}\leqslant 1; \\ 0 & \text{at } \dfrac{Re_f}{Re^0_{exc}}>1. \end{cases} \tag{6}$$

Thus, Eqs. (1), (4), and (6) make it possible to determine the thermal state of the channel in steady-state and transient regimes with allowance for heat and mass transfer with the upper mixing chamber in the case of unstable stratification of the density of the coolant in the chamber and channel and in the absence or at low rates of forced motion.

NOTATION

b, l, δ, width, length, and gap of slit channel; z, coordinate; d_{eh} and d_{eu}, equivalent diameter of heated and unheated channels; λ_l, μ_l, and ν_l, thermal conductivity and absolute and kinematic viscosities; c_p, specific heat of the liquid; ρ_h and ρ_u, mean density of the liquid in the heated and unheated channels; q, heat flux; t_c, weighted mean temperature of liquid in heated channel; t_{uc}, temperature of liquid in upper chamber; λ_u, drag of unheated channel; G_f, forced flow rate of liquid in loop; F_u, area of cross section of unheated channel; W_f, rate of forced motion; $Re_f = \dfrac{|W_f| d_{eh}}{\nu_l}$, $Re^0_{exc} = \dfrac{(\rho W)^0_{exc} d_{eh}}{\mu_l}$, Reynolds number; $Ra = \dfrac{g\beta c_p \rho_l d^4_{eh} q}{\lambda^2_l \nu_l}$, Rayleigh number.

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