# EFFECT OF GAS PHASE ON HEAT TRANSFER IN TURBULENT WATER FLOW IN A MODEL OF A COMPACT TRIANGULAR ROD BUNDLE

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Results are presented from an experimental study of the effect of a gas phase on temperature fields in a flow and a wall and on the mean heat-transfer coefficient in the flow of a water–gas mixture. It is shown that the effect is substantial.

Coolant flows in actual power plants are not perfectly uniform. Inert gas may be captured by liquid in a pump, gas (vapor) may enter a primary circuit from a secondary circuit, etc. It therefore becomes necessary to study heat transfer in such flows.

Several works [1-3, for example] have studied heat transfer in the turbulent flow of a mixture of a liquid and gas bubbles in pipes. These works have noted an increase in the heat-transfer coefficient $\alpha$ with the appearance of gas in a flow and an increase in the amount of gas in the flow. Meanwhile, the coefficient increases more substantially in the region of low volume-flow-rate gas contents $\varphi$ (to 10-15%). A further increase in  leads to a less substantial increase in $\alpha$.

The works [4-7] presented results of the most thorough studies of the mean temperature fields and heat-transfer coefficients for ascending and descending flows of a mixture of water and gas bubbles and an ascending flow of a mercury–gas–coolant mixture in the case of different Prandtl numbers and other thermophysical properties. The authors obtained basic statistical characteristics of the temperature pulsations in the flow of a gas–water mixture in a tube, as well as data on the local structure of a steady, turbulent, isothermal, two-component water–gas flow ascending and descending a vertical pipe. These studies somewhat clarify the question of the specific causes of the increase in heat-transfer coefficient at relatively low volume gas contents in tests both with water and with mercury. The increase in the coefficient is connected with additional agitation of the flow due to the relative motion of the phases under the influence of buoyant forces. An increase in the scale of turbulence in the two-component flow was also noted. The intensification of turbulent transport is manifest clearly in a distortion of the temperature field and a decrease in thermal gradients at all points of the flow as gas content increases. The appearance of the gas phase increases the heat-transfer coefficient to a lesser degree at higher liquid flow rates, since the additional agitation of the flow by the kinetic energy of the gas is less pronounced against the background of the liquid's own turbulence.

Although the mean gas content is the main parameter which determines the improvement in heat transfer, the mechanism by which the gas phase affects heat transfer is very complex and is connected with the distribution of the phase over the cross section, the size of the bubbles and the character of flow of the liquid about them, the form of the channel, the direction of the flow, and many other factors.

The literature does contain a study of heat transfer for two-component flows in complex channels [8]. The authors here obtained the first experimental data on the behavior of mean temperature fields and the intensity of temperature pulsations in a flowing water–gas mixture in a square channel.

There are several distinctive features of heat transfer in channels of complex cross section (see [9-11], for example). It is characterized by a considerably more complicated dependence on the determining parameters — the number of which is itself considerably greater than for flow in a circular tube. The criterional relation in the case of fuel elements with a single shell has the form

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TABLE 1. Determining Parameters of the Experiment ($\varphi_1 = 0$; $\varphi_i$, %)

| $Re \cdot 10^{-3}$ | $\bar{q} \cdot 10^{-3}$ W/m² | $\varphi_2$ | $\varphi_3$ | $\varphi_4$ | $\varphi_5$ | $\varphi_6$ | $\varphi_7$ | $\varphi_8$ | $\varphi_9$ |
|---|---|---|---|---|---|---|---|---|---|
| 60,0 | 12,2 | 0,62 | 1,23 | 2,64 | 4,92 | 7,57 | — | — | — |
| 30,0 | 12,2 | 1,02 | 2,03 | 4,97 | 9,3 | 14,96 | 19,98 | — | — |
| 10,8 | 16,2 | 1,4 | 2,0 | 2,97 | 4,67 | 7,0 | 9,93 | 14,6 | 20,0 |
| 10,5 | 8,35 | 1,4 | 2,0 | 2,97 | 4,67 | 6,6 | 9,93 | 14,6 | 20,3 |
| 5,4 | 6,3 | 2,9 | 3,38 | 4,76 | 7,4 | 13,8 | 19,35 | 24,2 | 29,6 |

$$Nu = \frac{\alpha d_h}{\lambda_f} = f\left(Re,\ Pr,\ x,\ \frac{\lambda_t}{\lambda_f},\ \frac{\lambda_w}{\lambda_f},\ \frac{r_1}{r_0}\right).$$

The tests conducted in the present work are a continuation of the experiments described in [11]. Thus, the parameters characterizing the regimes in the present investigation (thermal power input, liquid flow rate, coolant temperature at inlet and outlet) were given the same values as in studying heat transfer without a gas phase (Table 1). The construction of the experimental section was detailed in [11]. Figure 1 shows the channel cross section and the location of the embedded thermocouples. The gas phase was injected into the center of the flow at the inlet to the section after an agitating ring. The phase was injected through a single tube with an inside diameter of 4 mm.

We measured the temperature field in the wall of the test section and in the liquid flow at a distance of $l/d_h = 67$ from the beginning of steady-state heating with different liquid flow rates (Re) and different mean heat fluxes $\bar{q}$. The temperature fields were measured as a function of the volume-flow-rate gas content $\varphi$ and the water temperature at the test-section inlet and outlet.

The error of the temperature t was 0.1°C, while the errors of the water and gas flow rates were both 1%. The analysis of the test data amounted to calculation of the temperature fields and mean heat-transfer coefficients from the thermocouple readings. The temperature fields were represented in dimensionless form:

$$T_w = \frac{t_w - t_f^0}{\bar{q} r_0} \lambda_f; \quad T_f = \frac{t_f - t_f^0}{\bar{q} r_0} \lambda_f.$$

The mean temperature of the heat-transfer surface $\bar{t}_w$ was determined from the formula

$$\bar{t}_w = \bar{t}_{wb} - \Delta \bar{t}_{wb}. \tag{1}$$

Here $\bar{t}_{wb} = \frac{1}{\gamma_0} \int\limits_0^{\gamma_0} t_w d\gamma.$

The correction for the embedding of the thermocouples $\Delta\bar{t}_{wb}$ was found from the mean heat flux on the assumption that the thermal conductivity of the wall was independent of temperature:

$$\Delta \bar{t}_{wb} = \frac{1}{\gamma_0} \int\limits_0^{\gamma_0} \Delta t_{wb} d\gamma = \frac{1}{\gamma_0} \int\limits_0^{\gamma_0} \frac{q_w}{\lambda_w} \delta_b d\gamma = \frac{\delta_b}{\lambda_w} \bar{q}_w. \tag{2}$$

The assumption that $\lambda_w$ is independent of temperature in calculating $\Delta\bar{t}_{wb}$ is valid because the thermal conductivity of steel 1Kh18N10T changed only slightly with temperature under the conditions of the experiment.

The mean heat-transfer coefficient was calculated from the formula

$$\bar{\alpha} = \frac{\bar{q}}{\bar{t}_w - \bar{t}_f} \tag{3}$$

and $\bar{t}_f$ was found by linear interpolation of the section inlet and outlet temperatures of the liquid.

We took the hydraulic diameter $d_h$ as the determining parameter and $\bar{t}_f$ as the determining temperature in calculating the Reynolds and Nusselt numbers. We took the thermophysical properties of water at $\bar{t}_f$, i.e. we did not consider the effect of the gas phase and the viscosity and thermal conductivity of the flow.

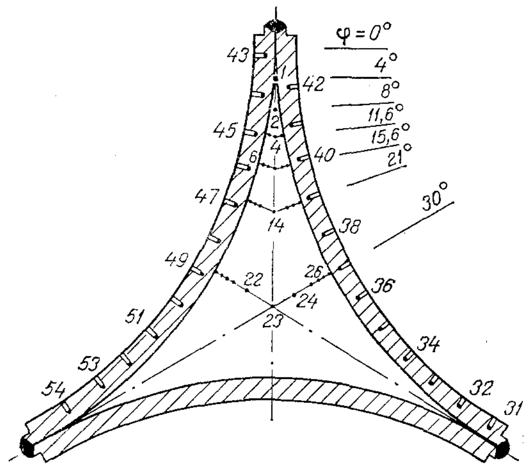


Fig. 1. Cross section of channel with location of thermocouples.

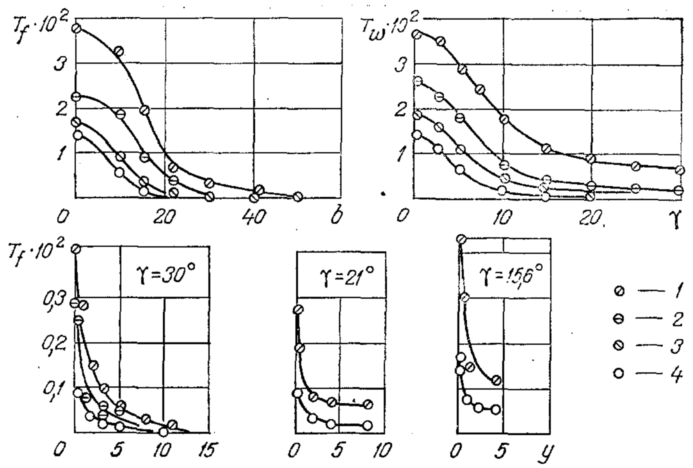


Fig. 2. Nonuniformity of dimensionless temperatures $T_f$ and $T_w$ with Re = $4 \cdot 10^3$, $\bar{q} = 6.2 \cdot 10^3$ in relation to gas content: 1) without gas; 2) $\varphi$ = 2.9%; 3) 9.9; 4) 24.2. b, y, mm; γ, deg.

The following fundamental law follows from comparison of the temperature fields obtained from measurement with identical water flow rates and heat fluxes but different gas contents (Fig. 2): the temperature gradients across the flow decrease in all directions with an increase in gas content in the ranges studied. This decrease is due to an increase in heat transfer by the flow in any direction of the measured cross section.

The increase in turbulent diffusion of heat in the flow is caused by the same mechanism as in a pipe [6, 7]. This mechanism probably amounts to the following: gas bubbles in the flow are moved upward relative to the water flow by buoyant forces and create additional eddies. The scale of these eddies is greater than the scale associated with the single-phase turbulence. They can therefore transfer a greater quantity of heat, having a relatively small reserve of kinetic energy. Thanks to the large scale of the motion, the effect of the individual bubbles is propagated over nearly the entire cross section of the flow.

As in a pipe, in the compact rod-bundle model the gas content affects the temperature field in the flow more at low Reynolds numbers than at high Re. This is because of the mechanism described above: given the same gas content, the amount of additional energy imparted to the flow by the relative motion of the phases on different backgrounds of "intrinsic" turbulence makes a different relative contribution at different Reynolds numbers.

In connection with an increase in the coefficients of turbulent thermal diffusion in the flow in different directions, the nonuniformity of the wall temperature decreases with an increase in the amount of gas in the flow (with a constant Re and constant heat flux)

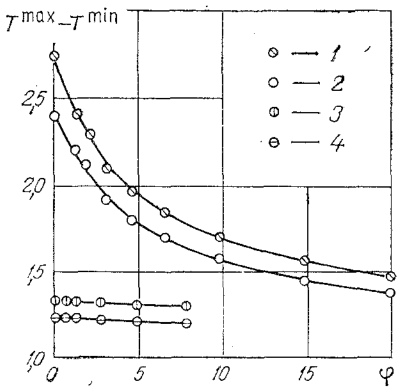


Fig. 3

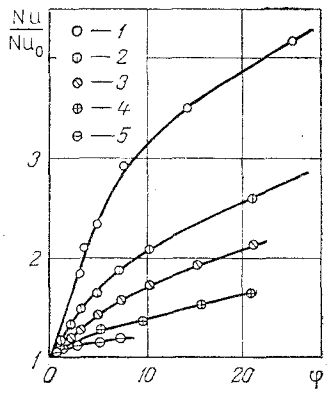


Fig. 4

Fig. 3. Change in $\Delta T_f$ and $\Delta T_w$ in relation to φ (%) with Re = $10.8 \cdot 10^3$ ($\bar{q} = 16.2 \cdot 10^3$) (1, 2) and Re = $6 \cdot 10^4$ ($\bar{q} = 12.2 \cdot 10^3$) (3, 4): 1, 3) $\Delta T_f$; 2, 4) $\Delta T_w$.

Fig. 4. Change in $Nu/Nu_0$ with: 1) Re = $5.4 \cdot 10^3$; $\bar{q}$ = $6.3 \cdot 10^3$; 2) $10.5 \cdot 10^3$ and $8.35 \cdot 10^3$, respectively; 3) $10.8 \cdot 10^3$ and $16.2 \cdot 10^3$; 4) $3.0 \cdot 10^4$ and $12.2 \cdot 10^3$; 5) $6.0 \cdot 10^4$ and $12.2 \cdot 10^3$.

(Fig. 2). This effect, more substantial at low Re, decreases with an increase in Re. It should be pointed out that the maximum nonuniformity of the wall temperature ($T_w^{max} - T_w^{min}$) decreases with an increase in the gas content mainly thanks to a decrease in the gradients in the angle range from 5 to 30° (see Fig. 2), i.e., the effect of the gas phase propagates fairly deeply into the corners of the channel. Only in the range from 0 to 5° in the corner zone do the wall temperature gradients about the perimeter remain nearly constant with a change in gas content.

Figure 3 shows the dependence of the nonuniformity of the wall temperature $\Delta T_w$ on the gas content at different Reynolds numbers. This effect weakens at $\varphi > 5-10\%$ approaching a constant value which is evidently determined by the nonuniformity in the corner zone of the channel ($\gamma$ = 0-5°).

An increase in the gas content of the flow increases the heat-transfer coefficient (Fig. 4), similar to the relation in a pipe. The effect of the gas phase may diminish at high pressures.

## NOTATION

x, relative spacing of rod grid; $r_0$, external radius of fuel-element shell; $r_1$, internal radius of fuel-element shell; $\gamma$, angle in degrees; $\varphi$, volume-flow-rate gas content, %; $\lambda_f$, $\lambda_w$, $\lambda_t$, thermal conductivities of coolant, shell, and fuel; $d_h$, hydraulic diameter of channel; $\alpha$, heat-transfer coefficient; $l$, length of heated section of channel; $t_f^0$, temperature of liquid at central point of channel cross section; $\bar{t}_{wb}$, mean wall temperature at site of thermocouple installation; $\bar{t}_f$, mean temperature of gas–liquid flow in the measured cross section; Re, Reynolds number; Pr, Prandtl number; Nu, Nusselt number; b, distance along bisector from point of contact of rods; $\delta_b$, depth of installation of thermocouples.

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# CERTAIN FEATURES OF TRANSVERSE FLOW ABOUT A CYLINDER WITH LONGITUDINAL RIBS

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It was experimentally established that the structure of the boundary-layer separation and the flow regime preceding separation can be influenced by installing longitudinal ribs on the surface over which flow is occurring.

It was shown earlier [1-4] that the free oscillation and vibration of structures in a liquid flow can be completely or partly damped by installing interceptors, transverse and oblique ribs, etc. on the surface about which flow is occurring. These devices change the character of separation of the boundary layer on the diffuser section of flow. The present work attempts to study the change in separated flow with the placement of lengthwise ribs on a wall, the ribs chosen so as to minimize the additional aerodynamic load on the structure.

The study was conducted in two wind tunnels with an open working part with nozzle diameters of 0.44 and 2.2 m. We used two circular cylindrical models with diameters $d_1$ = 44 mm and $d_2$ = 160 mm, respectively. Transverse flow over the models allowed us to clearly observe separation of the boundary layer from the initially hydraulically smooth surface. The smaller model No. 1 had a length $l$ = 634 mm, and its ends were located beyond the limits of the flow in the working part of the tunnel. The length of the larger model No. 2 was 1080 mm, and its ends were fitted with a ring 0.36 m in diameter. The longitudinal ribs were rings of thickness $s_1$ = 0.5 mm and height t = 0.5-9.0 mm on model No. 1 and thickness $s_1$ = 1 mm and height t = 4 mm on model No. 2. The distance between the rings $s_2$ was varied within the range 3-75 mm.

The tests involved determination of the distribution of static pressure $p(\varphi)$ over the model surface and the total pressure $p_t(y)$ in its wake, as well as the intensity of the velocity pulsations $\varepsilon(\varphi)$ in the boundary layer a fixed distance $y \approx 0.5$ mm from the wall of the large model at different cross sections along its length. The total pressures in the wake behind the model were measured in cross sections where the flow was equalized along the length and the static pressure became almost equal to the pressure $p_\infty$ in the unperturbed flow. The coefficients of profile drag of the cylinder $c_x$ were calculated by the impulse method from the measured total pressures in the wake and the pressure distribution $\bar{p}(y)$ on the surface of the model. In obtaining the measurements, we used a drainage system for the models, a rack of 23 pitot tubes, a constant-temperature hot-wire anemometer, and sliding traversing equipment.

The character of flow in the wake behind the cylinder was evaluated by installing a panel normal to the cylinder at different distances from it. The panel had silk threads

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