coordinate; x_2, vertical coordinate; λ, Taylor microscale; ν, kinematic viscosity coefficient; U_* and ℓ_*, characteristic velocity and length scales defined by (2). Subscripts: 0 denotes the centerline value, ∞ the free-stream value, and ( )' the root-mean-square value.

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INVESTIGATION OF LOCAL HEAT TRANSFER ASSOCIATED WITH THE EVAPORATION OF A LIQUID FILM ON A HORIZONTAL RIBBED TUBE

Yu. V. Putilin, V. G. Rifert,
V. L. Podbereznyi, and P. A. Barabash

UDC 536.423.1

The heat-transfer enhancement effect of longitudinal ribs on a horizontal tube is discussed on the basis of experimental data on the local heat-transfer coefficients.

In [1] measurements of the mean heat-transfer coefficients for film flow over horizontal, longitudinally profiled tubes demonstrated the possibility of a 1.5-2-fold intensification of the heat transfer as compared with a smooth tube.

In order to investigate the mechanism of the heat-transfer process associated with film flow over a profiled tube, we made an experimental determination of the local heat-transfer coefficients. The experiments were carried out on an apparatus previously used to investigate the heat transfer associated with the evaporation of a film preheated to the saturation point on a smooth tube [2].

The experimental tubes had an outside diameter of 38 mm. As in [2], for measuring the mean wall temperature we used resistance microthermometers, arranged as shown in Fig. 1.

The microthermometers are conventionally represented by dots numbered 1-10. Into each profiled test piece we fitted 10 microthermometers in two groups of five, each of which

Kiev Polytechnic Institute.

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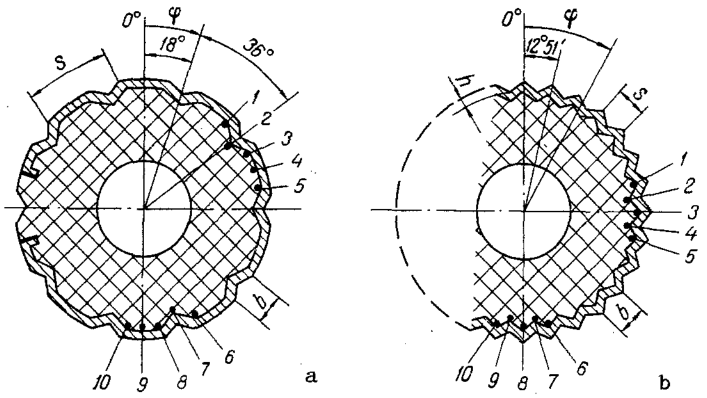


Fig. 1. Arrangement of thermocouples in the experimental ribbed tubes: a) type I, S = 11.9 mm; b) type II, S = 4.25 mm.

measured the surface temperature of two adjacent profile elements (triangular grooves) ahead of the groove, in the groove itself, and between grooves. By rotating the experimental tube about its axis it was possible to change the position of these profile elements in terms of the angular coordinate ϕ and determine the local values of the surface temperature around the perimeter of the tube.

Progressive rotation of the experimental tube through an angle corresponding to the intergroove pitch made it possible to measure the surface temperature in various positions relative to ϕ and to duplicate the measurements using the microthermometers of the other group.

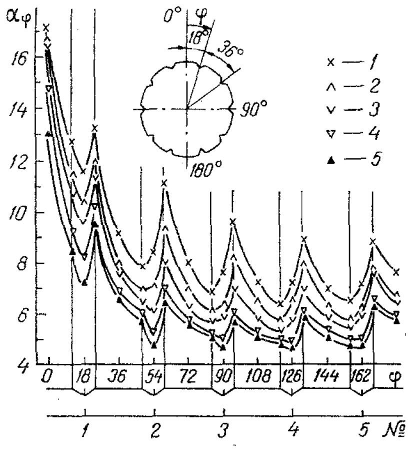
The test sections were profiled with triangular grooves of width b = 3 mm and depth h = 1 mm parallel to the generator. The pitch S = 4.25 and 11.9 mm. The experiments were carried out in the film evaporation regime (t_i = 100°C) at a constant heat flux density q = 10 kW/m^2, i.e., under the same conditions as those under which the smooth-walled tube was tested.

At this value of q, as with film flow over a smooth tube, there was no visible vapor bubble formation in the film, which indicated the predominance of the removal of heat of evaporation from the film surface.

On the axis of abscissas, in addition to the angular coordinate ϕ, we have shown the disposition of the grooves around the perimeter of the tube and the numbering of the grooves along the flow path of the film (1-5). As for a smooth tube [3], the maximum values of α_ϕ were obtained at the top of the tube (the greater the higher the density) and were practically the same as the corresponding values of the heat-transfer coefficients for a smooth tube [3]. On the interval from ϕ = 0° to the first groove along the flow path the heat-transfer coefficients decrease in the same way as the α_ϕ for a smooth tube.

Then the nature of the local heat-transfer coefficient distribution changes sharply as compared with a smooth tube.

At the longitudinal grooves the heat-transfer coefficient increases strongly from a minimum (ahead of or in the groove) to a maximum (at the downstream edge of the groove), and on the graphs of the function $\alpha_\phi = f(\phi)$ for all flow densities peaks corresponding to these α_ϕ maxima are formed. On the smooth parts of the tube surface between grooves the heat transfer falls back again. When the film passes from a smooth section into a groove, depending on its position on the perimeter of the tube (angular coordinate ϕ) and the flow density, the heat-transfer coefficient either continues to fall to a minimum in the bottom of the groove or begins to increase directly from the leading edge of the groove. In this connection, a certain regularity may be observed. Thus, at the first groove near the top of the tube (ϕ = 18°), for all flow densities Γ = 0.4-0.12 kg/(m·sec) the heat-transfer coefficients reach a minimum in the bottom of the groove, which is evidently associated with the fact that this groove is in the most favorable position for the formation of stagnant zones. In the second groove at the two greatest values of the flow density the heat-transfer coefficients begin to increase directly from the leading edge of the groove, where the minimum values of α_ϕ are recorded. In grooves 3 and 4 the $\alpha_\phi = f(\phi)$ dependence also behaves in this



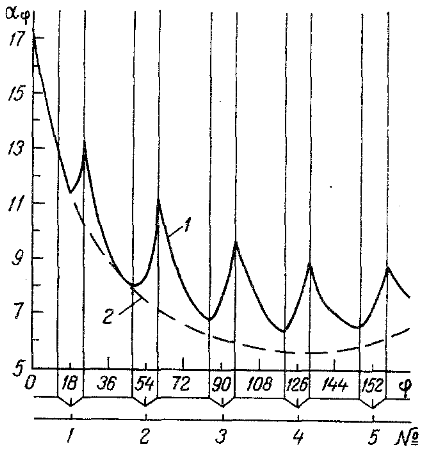


Fig. 2 Fig. 3

Fig. 2. Local heat transfer associated with the evaporation of a liquid film from a tube with the profile shown in Fig. 1a: 1) $\Gamma = 0.4$; 2) 0.32; 3) 0.25; 4) 0.16; 5) 0.12. α_ϕ, kW/(m²·K); ϕ, deg.

Fig. 3. Local heat transfer associated with the evaporation of a liquid film from a profiled (see Fig. 1a) 1 and a smooth tube 2.

way for $\Gamma = 0.25$ kg/(m·sec). In the last (fifth) groove a slight fall in α_ϕ in the bottom of the groove is observed only when $\Gamma = 0.12$ kg/(m·sec), while for all the other values of Γ the minimum of the heat-transfer coefficient is recorded at the leading edge.

Thus, the longitudinal grooves divide the perimeter of the tube into sections on which the heat-transfer coefficients vary from a maximum at the trailing edge to a minimum in or ahead of the next groove. This variation of the heat transfer was obtained for all values of Γ on the interval 0.12-0.4 kg/(m·sec), the maximum and minimum values of the heat-transfer coefficients on each successive section along the flow path of the film being smaller than the corresponding values of α_ϕ on the preceding section, except at the back of the tube ($\phi = 162°$).

This heat-transfer behavior is possible, in our view, as a result of the destruction of the growing hydrodynamic and thermal boundary layer by the grooves owing to the development of a transverse velocity component at the point where the film crosses the groove.

To permit a comparison of the variation of the local heat-transfer coefficients around the perimeters of smooth and longitudinally profiled (type I) tubes, in Fig. 3 we have plotted the $\alpha_\phi = f(\phi)$ graphs for both tubes when $\Gamma = 0.4$ kg/(m·sec). From this figure it is easy to estimate the heat transfer enhancement obtainable with a profiled tube.

Clearly, for the profiled tube on the part of the perimeter extending from $\phi = 0°$ to the first groove the heat-transfer coefficient practically coincides with the values of α_ϕ for a smooth tube. Then along the flow path there is a sharp increase in the heat-transfer coefficient at each triangular groove and a further decrease in the value of α_ϕ on the smooth part of the surface (between grooves). Thus, for the profiled tube the $\alpha_\phi = f(\phi)$ graph is an alternation of maximum and minimum values of the heat-transfer coefficient, as distinct from the corresponding graph for a smooth tube, which reflects a monotonic variation of the heat transfer around the perimeter. The minimum values of α_ϕ on the profiled tube differ only very slightly from the smooth tube values (at corresponding points on the perimeter). The maxima of α_ϕ grow smaller with each successive groove and are much less than the value at the point where the liquid meets the tube ($\phi = 0°$). Thus, the general variation of the heat-transfer coefficient as the film flows over the surface of the profiled tube remains the same as for a smooth tube, i.e., the coefficient decreases along the flow path, with a certain increase at the back of the tube. The only difference is the appearance on the heat-transfer surface of sections with a relatively high (as compared with the smooth tube) value of the heat-transfer coefficient corresponding to the coordinates of the grooves.

Since the increase in the heat-transfer coefficient at each groove (see Fig. 2) is the greater the greater the flow density, the heat-transfer enhancement obtainable by profiling the tube also increases with the flow density. Since on a profiled tube of type I the zones

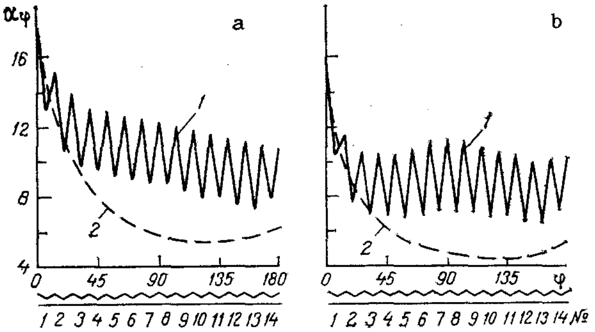


Fig. 4. Local heat transfer associated with the evaporation of a liquid film on a tube with a profile of the type shown in Fig. 1b (a – Γ = 0.4; b – Γ = 0.16): 1) profiled, 2) smooth surface.

of enhanced heat transfer are relatively few, the degree of heat-transfer enhancement obtained with this tube is low.

The maximum degree of heat-transfer enhancement, expressed as the ratio of the mean heat-transfer coefficients over the perimeters of the profiled and smooth-walled tubes $\bar{\alpha}_\phi/\bar{\alpha}_{gl}$ (obtained by numerical integration using the trapozoidal rule), is found to correspond to Γ = 0.4 kg/(m·sec) and is equal to 1.18. For lower values of Γ the enhancement is even less (1.05 when Γ = 0.12 kg/(m·sec)).

To improve the heat-transfer enhancement it is necessary to reduce the distance between grooves, thus eliminating the smooth parts of the surface on which the boundary-layer thickness increases.

In Fig. 4, in the form of graphs of the function $\alpha_\phi = f(\phi)$ for Γ = 0.4 and 0.16 kg/(m·sec), we have reproduced the experimental local heat-transfer data for a profiled tube of type II having 28 triangular grooves with a pitch of 4.26 mm (the dimensions of the grooves are the same as for the tube of type I). Along the axis of abscissas, in addition to the angular coordinate ϕ, we have indicated the arrangement of the grooves around the tube perimeter and their numbering along the flow path (1-14).

The liquid flowed onto the tube (at ϕ = 0°) midway between two grooves, i.e., along one of the surface ribs.

It is clear from Fig. 4 that the variation of the local heat-transfer coefficients over the perimeter of the type II tube differs sharply from the graphs of $\alpha_\phi = f(\phi)$ obtained for the type I and smooth tubes. It takes the form of a continuous sinusoidal curve with α_ϕ minima in the bottoms and maxima at the edges of the longitudinal grooves. Except for the initial zone, from the top of the tube to groove 1, where the heat transfer falls off sharply, over most of the perimeter both the maxima and the minima of α_ϕ vary only slightly along the flow path.

Thus, whereas when Γ = 0.4 kg/(m·sec) the maximum and minimum values of α_ϕ gradually decrease (by a factor of approximately 1.3), when Γ = 0.16 kg/(m·sec) they remain practically constant over the perimeter of the tube, increasing somewhat near ϕ = 90°. The difference between the maximum and minimum values of the local heat-transfer coefficients also remains approximately constant over the entire perimeter of the tube, except for the initial zone, and amounts to $\Delta\alpha_\phi$ = 3.2-3.5 kW/m²·K.

Since the initial zone, in which the local heat transfer falls sharply, occupies only 7% of the heat-transfer surface, its contribution to the mean heat-transfer coefficient is only slight. On the other 93% of the surface of the profiled tube (type II), the values of the heat-transfer coefficients in the grooves and on the ridges are either practically constant (when Γ = 0.16 kg/(m·sec)) or decrease only slightly (by a factor of 1.3 when Γ = 0.4 kg/(m·sec)) over the flow path. At the same time, on the smooth and type I tubes the decrease in heat transfer with increase in the flow path extends over almost the entire perimeter of the tube (except for the back) and is concentrated (factor of 2-3) within the interval from ϕ = 0 to 60°.

The minimum values of the heat-transfer coefficients obtained for type II tubes exceed, as distinct from the minimum values of α_ϕ for the type I tube, the heat-transfer coefficient for the smooth-walled tube (at corresponding angular coordinates around the perimeter of the tube).

The investigations showed that both the nature of the variation of the local heat transfer around the perimeter of a longitudinally profiled tube of type II and the absolute values differ considerably from those for smooth tubes and profiled tubes of type I.

Calculations of the heat-transfer coefficients α_ϕ averaged over the perimeter of profiled tubes of type II, carried out by numerical integration, indicate that for these tubes the degree of heat-transfer enhancement, as compared with the smooth-walled type, is considerably greater than for tubes of type I. In fact, $\alpha_\phi/\alpha_{gl} = 1.44$ for $\Gamma = 0.4$ kg/(m·sec) and $\alpha_\phi/\alpha_{gl} = 1.52$ for $\Gamma = 0.16$ kg/(m·sec).

Thus, the conclusion previously drawn from the results of experiments on a longitudinally profiled tube having 10 triangular grooves in its outer surface (type I) that in order to enhance substantially the heat transfer to a flowing film, as compared with a smooth tube, it is necessary to reduce the distance between grooves has been experimentally confirmed, together with the physical model of the process of heat-transfer enhancement on a profiled surface based on the continuous destruction of the hydrodynamic and thermal boundary layers.

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INVESTIGATION OF DROPLET DEPOSITION ON A PLATE WITH TWO-PHASE FLOW SEPARATION AT THE LEADING EDGE

O. M. Khmara and S. V. Ryzhkov

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The shape of the leading edge of the plate and the Stokes and Reynolds numbers are shown to have a decisive influence on the droplet deposition rate.

The protection of the operators of manual pneumatic tools from the effects of the finely dispersed water and oil aerosols present in spent compressed air is a topical problem. One method of solution may be to intensify the deposition of the aerosols on the surfaces of the noise suppressor elements, from which the deposited moisture is carried away in the form of large drops resulting from the breakup of the liquid film in the exhaust. These drops settle out before they can reach the operator's breathing zone.

In the presence of a separationless turbulent two-phase flow the deposition of aerosol particles from 1 to 20 μm in diameter is chiefly determined by the turbulent-inertial mechanism [1]. When the two-phase flow separates, a considerable increase (by tens of times) in the rate of deposition of finely dispersed liquid aerosols, as compared with separationless steady flow, is observed near the attachment zone [2].

So far little has been published on the factors determining the particle deposition rate in the entrance zones of plates and channels. The influence of edge vortex effects on heat and mass transfer has been more carefully investigated. Thus, in [3] it was shown that in the case of flow separation on the flat leading edge of a plate the heat transfer coefficient reaches a maximum at $x/H = 8$, i.e., at the flow attachment point. The distance from the

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