CONVECTIVE HEAT TRANSFER IN EXTERNAL FLOWS

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Research on convective heat transfer in single-phase external flows, carried out at the Institute of Physical and Technical Problems of Energetics of the Academy of Sciences of the Lithuanian SSR over a broad interval of Reynolds and Prandtl numbers, is described and the principal results are presented. Attention is focused on questions of heat transfer intensification.

Thirty years have passed since the founding in Kaunas of the Institute of Physical and Technical Problems of Energetics of the Academy of Sciences of the Lithuanian SSR. During this period the Institute has grown into a slarge scientific-research center in which problems relating to the development of power engineering and the design of power plants and various heat exchangers are solved by means of a multidisciplinary approach. The Institute carries out comprehensive thermophysical research, concentrating on the study of convective heat transfer and the properties and strength of heat-resistant materials.

The work of the Institute in the field of single-phase convective heat transfer in the Prandtl number interval from 0.7 to 5000 is well known, as are its investigations into high-temperature gas flows.

This article presents only the most important of the results on convective heat transfer in external flows obtained at the Institute in recent years. A full and detailed account of the work of the Institute in this field can be found in the author's monograph [1].

Heat transfer in heat exchangers is mainly the result of the heat-transfer agent flowing over a solid wall on which a laminar or turbulent boundary layer is formed. The results of research in the area of laminar flow are summed up in [2]. In recent years interest has centered on the characteristics of the turbulent boundary layer. Even the first investigations into the heat-transfer processes on a flat plate in air, transformer oil, and water flows in Reynolds and Prandtl number intervals of 10^4-10^7 and 0.7-380, respectively, yielded important results [3]. Relations were obtained for calculating the heat transfer on a smooth plate, both fully heated and with an initial unheated section. It was found that the effective viscosity and hence the diffusion rate are much higher for turbulent than for laminar flow, which also affects the heat transfer. Thus, for a laminar boundary layer the surface heat-transfer coefficient is proportional to the velocity to the power 0.5, and for a turbulent boundary layer to the power 0.8. The Prandtl number exponent increases from 0.33 for the laminar to 0.43 for the turbulent boundary layer. During heat transfer the variation of the temperature of the fluid over the thickness of the boundary layer is accompanied by changes in its physical properties, which determine the deformation of the velocity and temperature profiles and hence the heat-transfer coefficient, since the intensity of heat transfer varies with the direction of the heat flux. The mean temperature of the approach flow is taken as the characteristic temperature, and then in order to take the temperature variation of the physical properties of the fluid into account the ratio $\mathrm{Pr}/\mathrm{Pr}_w$ is introduced into the similarity relation:

$$\mathrm{Nu} = c\,\mathrm{Re}^m\,\mathrm{Pr}^n\,(\mathrm{Pr}/\mathrm{Pr}_w)^p\,. \tag{1}$$

In the dimensionless numbers the physical properties are taken with respect to the flow temperature, and in Pr_w with respect to the wall temperature. As a result, for calculating the heat transfer on a plate from similarity equation (1), the exponent p was found to be equal to 0.25 for heating of the fluid and 0.17 for cooling. For a turbulent boundary layer on the plate, in relation (1) $m = 0.8$, $n = 0.43$, and $c = 0.037$.

The turbulent boundary layer is characterized by a sharp transition from the zone with predominantly molecular transport to the zone with predominantly molar transport. Starting from the one-dimensional heat flux equation, we have

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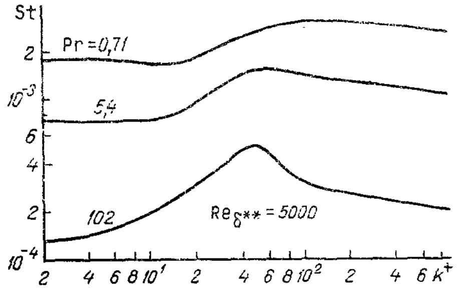

Fig. 1. Heat transfer as a function of the roughness parameter for various Pr.

$$\left(\frac{1}{\Pr}+\frac{1}{\Pr_t}\frac{\varepsilon_\tau}{\nu}\right)\frac{d\vartheta^+}{dy^+}=1, \tag{2}$$

where $\vartheta^+ = \vartheta/\vartheta^*$; $\vartheta_* = q_w/\rho c_p u_*$; $Pr_t = \varepsilon_\tau/\varepsilon_q$.

It follows from Eq. (2) that as Pr increases, so does the relative influence of turbulent transfer near the wall, since when Re = idem, with increase in Pr the second term in Eq. (2), being a small quantity in the viscous sublayer, may become tens of times greater than the first, which expresses the molecular transport.

According to our calculations [4], in an air flow the viscous sublayer corresponds to 25% of the total temperature difference Δt between the temperatures of the wall and the approach flow, and in a flow of transformer oil with Pr = 55 to 90%. In an air flow most of the thermal resistance is concentrated in the transition zone of the turbulent boundary layer, whereas in liquids with Pt $\gg$ 1 it is concentrated in the viscous sublayer. Thus, in order to obtain the required heat-transfer intensification in gases it is necessary to create turbulence throughout the wall zone, and in liquids with a large value of Pr, essentially in the viscous sublayer. Among the methods of intensifying convective heat transfer, one of the most important is to create artificial roughness on the wall surface. Thus, the greater Pr, the lower the height of the roughness elements at which a significant increase in heat transfer can be achieved. It should be kept in mind that in liquid flows the greatest heat-transfer efficiency is obtained in the partially developed roughness regime.

In order to characterize the roughness elements, we introduce the dimensionless roughness height:

$$k^+ = \frac{ku_*}{\nu}. \tag{3}$$

The heat-transfer data for a rough plate [5] indicate that the effect of the parameter k^+ varies with the Pr number. In an air flow, as k^+ increases a uniform increase in heat transfer is observed (Fig. 1). In a transformer oil flow (Pr = 102) the maximum increase in the heat-transfer coefficient corresponds to partially developed roughness, i.e., $k^+ < 50$. Usually, a further increase in k^+ leads to a fall in heat transfer, although there are possibilities of considerably broadening the regime of maximum heat transfer.

If the roughness elements are densely packed, then stagnant zones develop between them, and the height k may not be the controlling geometric characteristic.

The laws of heat-transfer intensification follow from an analysis of the processes associated with flow past a single roughness element [6]. Maximum heat transfer is always observed in the zone of attachment of the separated flow. It directly reflects the mean heat-transfer level not only for a single element but also for a system. It can therefore serve as a guide to its possible enhancement. The weak dependence of the optimum distribution of two-dimensional obstacles in the interval (8-12) k is due to the fact that under highly turbulent flow conditions the length of the separation zone tends towards a certain minimum value, and the heat transfer towards a maximum.

Since when Pr = 1 maximum heat transfer takes place in the fully developed roughness regime, the problem of reducing the drag of the rough surfaces is an extremely important one. At the IPTPE this problem has been investigated by A. Pyadishyus. The object was to regulate the turbulence generation mechanism and create large-scale motions in the flow. This was achieved by using streamlined roughness elements and placing two-dimensional obstacles at an angle to the flow. In both cases the process is accompanied by the development of signs of three-dimensionality in the separation zone. This leads to a series of favorable

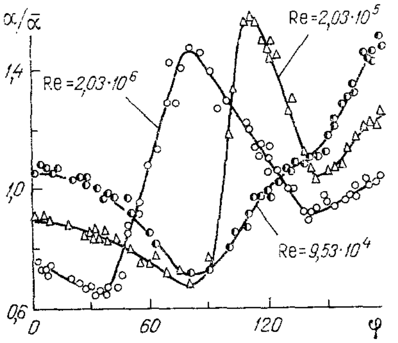

Fig. 2

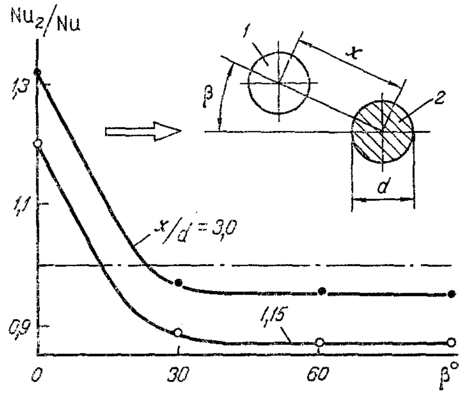

Fig. 3

Fig. 2. Local relative heat transfer for a cylinder at various Re.

Fig. 3. Effect of front cylinder on the relative heat transfer at a second, rear cylinder.

effects involving a reduction in the drag, an increase in the optimum spacing between obstacles, and a broadening of the interval of regimes in which the heat transfer remains at the maximum level. The practical realization of this flow situation is most effective at low Reynolds numbers. In the case of developed turbulent flow the most obvious criterion of enhanced heat-transfer efficiency is the efficiency of the smooth surface, when the expression for the heat transfer-friction analogy is valid. In the transition flow regime when Pr = 100 for the rough and smooth surfaces the ratio $(St/St_0)/(c_f/c_{f_0}) = 1.6$.

Tubes are important elements of heat-transfer surfaces. Therefore, the heat-transfer problems associated with flow in and over tubes are of particular interest. A significant increase in flow velocity and Reynolds number has an important influence on the nature of the flow, causing a qualitative change in heat transfer. This is clearly apparent from our data [7] on the local heat transfer around the perimeter of a tube in a cross flow of water (Fig. 2). At low Reynolds numbers ($Re = 9.5\cdot10^4$) the heat transfer has a maximum at the front of the cylinder; however, as the laminar boundary layer develops and its thickness increases, the heat transfer falls. After separation of the laminar boundary layer ($\phi \simeq 80°$) in the vortex zone the heat transfer gradually increases. In the critical flow regime ($Re = 2\cdot10^5$) at $\phi = 80°$ a separating bubble is formed; this then develops in the turbulent boundary layer and breaks away at $\phi \approx 140°$.

In the supercritical flow regime ($Re = 2\cdot10^6$) the first heat-transfer minimum corresponds to direct transition from a laminar to a turbulent boundary layer ($\phi = 40°$), and the second to separation of the turbulent boundary layer. As the Re and the turbulence of the external flow increase, the first heat-transfer minimum on the cylinder is displaced toward the forward stagnation point and the region covered by the laminar boundary layer contracts. It was previously assumed that in the supercritical regime the laminar boundary layer covers 50% of the surface; however, research at the IPTPE has shown that the true figure is only 15-20%.

For practical purposes it is important to make an accurate determination of the average heat transfer. At the Institute both the local and the average heat transfer for a tube in cross flows of different fluids have been investigated over a wide range of Prandtl (0.7-1000) and Reynolds ($1\text{-}4\cdot10^6$) numbers. The effect of turbulence, roughness and other factors on the nature of the flow and the heat transfer was taken into account. All these data and the formulas derived from them are extensively used in practical applications and are summarized in [8].

In order to calculate the average heat transfer for a tube in a cross flow we use the similarity equation (1), taking the temperature of the flow, gas or liquid, as the characteristic temperature, and as the controlling parameters the diameter of the tube and the flow velocity in the narrowest cross section. According to our experiments [1, 8], at low Re values the exponent m = 0.4; as Re becomes greater, it gradually increases, reaching 0.8 or more in the supercritical regime, while the exponent of Pr n = 0.37 and reaches 0.40 only in the supercritical regime. For heating the exponent of the ratio Pr/Pr_w p = 0.25, while for cooling p = 0.20; however, if the temperature head is not very great, it is possible to take p = 0.25.

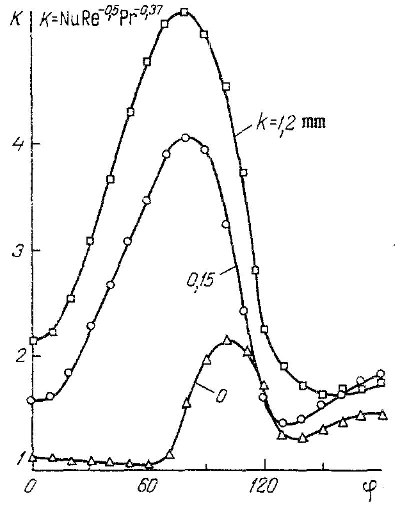

Fig. 4

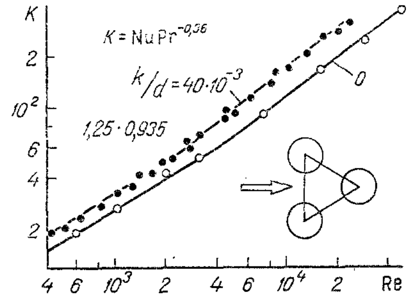

Fig. 5

Fig. 4. Effect of roughness on the local heat transfer for a cylinder in a cross flow with Re = 10^6 and Tu = 7%. ϕ in degrees.

Fig. 5. Mean heat transfer for a rough-surfaced tube in a staggered bundle.

As functions of Re, the quantities c and m for an individual tube take the following values:

$\mathrm{Re} = 1 - 40$,	$c = 0{,}76$,	$m = 0{,}4$,
$\mathrm{Re} = 40 - 1 \cdot 10^3$,	$c = 0{,}52$,	$m = 0{,}5$,
$\mathrm{Re} = 10^3 - 2 \cdot 10^5$,	$c = 0{,}26$,	$m = 0{,}6$,
$\mathrm{Re} = 2 \cdot 10^5 - 10^7$,	$c = 0{,}023$,	$m = 0{,}8$.

As the turbulence of the approach stream increases, there is a significant increase in the average heat transfer, the growth being approximately proportional to $\mathrm{Tu}^{0.15}$.

In practice, the turbulence in heat exchangers is higher. This is due to the effect of various factors. For example, in shell-and-tube heat exchangers bundles of tubes are used and the turbulence in their interior depends on the arrangement of the tubes; turbulence is also generated by baffles of various types.

The tubes forming the bundles usually have a staggered or in-line arrangement. In the interior rows of the individual bundles the flow turbulence may reach 40-60%. Accordingly, as compared with the first row, the heat transfer increases by up to 140-180%, the increase being somewhat greater in the case of staggered tube bundles. At the same time, the nature of the flow over the tubes changes because of the constricted space between tubes. The exponent of Pr will therefore be closer to 0.36. The exponent of Re varies in the same way as for a single tube, but in the subcritical flow regime for in-line bundles it will be somewhat higher and equal to 0.63-0.65. For staggered tube bundles in a cross flow the heat transfer rate also depends on the relative transverse ($a = s_1/d$) and longitudinal ($b = s_2/d$) pitches.

In order to calculate the heat transfer we use Eq. (1), where on the interval Re = 1-$2 \cdot 10^6$ for an interior row of a staggered tube bundle the exponent m varies from 0.4 to 0.8 and the constant c from 1.04 to 0.036; for an in-line tube arrangement m varies similarly, while the constant c varies from 0.9 to 0.033.

The methods of calculating the heat transfer and drag for tubes in cross flows developed at the IPTPE [1, 9-11] are widely known. They have been incorporated in a number of heat exchanger design standards and in numerous handbooks.

At present, the work of the Institute is mainly focused on the solution of problems of heat transfer intensification and rational tube bundle geometry. In heat exchangers it is common to encounter systems of two tubes oriented at various angles β in a cross flow and with various distances x between them. Research carried out on such systems by Zhyugzhda and

Dauetas [12] shows that in the subcritical and critical flow regimes, as distinct from the hydrodynamic characteristics, the average heat transfer of the first cylinder depends little on the angle β.

At small values of β the heat transfer for the second cylinder is higher than that for an individual cylinder, when β = 15-20° it approaches the latter, and with further increase in β falls below it.

As may be seen from Fig. 3, the heat transfer rate for the second cylinder depends on the angle β and on x/d, the relative cylinder spacing.

The flow and heat transfer in bundles with a cross arrangement of the tubes have a number of special characteristics. Here, the variability of the heat transfer as a function of the angle of rotation is sharply expressed. At the Institute in-line bundles of this type in an air flow have been investigated by Makaryavichyus and co-workers. As models they used bundles of tubes 6 mm in diameter in a cross flow.

A significant dependence of the heat transfer on the angle of rotation γ is observed when the longitudinal pitch is small: b = 1.16. Even a small rotation γ = 7.5° leads to an appreciable increase in the heat transfer. However, increasing γ further to 90° does not give such a significant effect.

More and more use is being made of rough tubes in order to intensify heat transfer. The Institute has carried out extensive research in this area in recent years. The process of heat transfer on rough tubes is similar to that examined above in connection with rough plates. Moreover, as the height of the roughness elements on the tubes increases, the transition from a laminar to a turbulent boundary layer takes place at lower values of Re, and the average heat transfer increases.

As may be seen from Fig. 4, in individual zones of the tube the local rough-surface (as compared with smooth) heat transfer increases by a factor of 2-3. Depending on the roughness, there is a sharp change in the distribution of the hydrodynamic flow regimes over the surface. The average heat transfer dynamics show that the transition to the critical flow regime depends significantly on the roughness and with increase in the latter is displaced towards the region of small Reynolds numbers [8]. The drag and heat transfer of various rough bundles in gas and transformer oil flows have also been investigated for various roughness-to-tube diameter ratios (k/d).

According to Fig. 5, the average heat transfer is 40% higher for rough bundles of staggered tubes than for smooth bundles [13].

In designing gas–liquid heat exchangers special attention must be paid to the heat flux balance. In air flows an increase in the heat flux can be achieved by increasing the heat transfer surface by using various fins. High fins can usefully be slotted. As research has shown, the optimum slotting angle relative to the tube axis is 45°. This further intensifies the motion at the base of the interfin space.

In practice it is usual to employ finning with values of ε up to ε = 20 (where $\varepsilon = F/F_0$ is the finning ratio, F is the total area of the finned tube, and F_0 is the area of the tube without the finning). Finning with ε > 20 is used in special-purpose heat exchangers. Cases in which ε = 1-10 have not yet received much attention. Data on the local and average heat transfer and drag characteristics of finned tube bundles are presented in [1, 10, 11, 14] for a broad range of Re and Pr and typical finning parameters when ε < 10.

The effective height and pitch of the fins depend on the thickness of the boundary layer. Essentially, the height of the fins must be greater than the thickness of the boundary layer, since the active surface area should be maximized. Finned surfaces are employed as an effective intensification device mainly in gas flows. However, they are being increasingly used in liquid flows also. In fact, an important topical problem is the development of efficient finned surfaces for liquid flows. Since in liquid flows the boundary layer is much thinner, the fins should be lower than in gas flows. In gas flows the boundary layer is thicker and the effective fin height may reach 25 mm, whereas in liquids, according to our research [11], the fin height should be 1.5 mm. In a gas flow the heat transfer coefficient is higher for smooth than for finned tubes, but the compactness of finned tube bundles is much greater. Accordingly, the specific heat transfer per cubic meter of heat exchanger volume is higher for finned than for smooth tube bundles. At the same time, as the compactness of the finned surfaces increases, the main problem becomes that of improving their heat transfer coefficient.

In recent years the Institute has developed various modifications of heat exchangers consisting of tubes spirally profiled outside and inside with different pitches [15]. In these heat exchangers the heat transfer is doubled with a corresponding increase by a factor of 2.7 in the fluid drag. When these tubes are substituted for round ones, the overall dimensions of the apparatus can be reduced by a factor of 1.7 with no increase in power consumption.

Coiled tubes of oval profile have circular ends for mounting in tube disks. In the heat exchanger thay are arranged in contact along the maximum dimension of the oval. When the heat-transfer media circulate through the tubes and the tube space they are given a spiral swirl. The flow is most complex in the intertube space of the exchanger which can be conventionally regarded as a system of alternating interconnected helical and through channels. In this system the turbulence is generated by the fixed wall and the friction between layers of fluid traveling at different velocities. The flow in a bundle of coiled tubes is also affected by the secondary circulation due to the action of the centrifugal forces resulting from the flow of the heat transfer agent in the helical channels.

The question of unsteady heat and mass transfer in bundles of coiled tubes has also been examined, since in a number of cases the transient processes associated with a change in the operating regime or the starting or stopping of heat exchange apparatus may be of decisive importance in relation to their trouble-free operation, as well as for the development of automatic control systems. A model of the flow of a homogenized medium has been developed for unsteady heat and mass transfer processes. Methods have been proposed for generalizing the experimental data on unsteady heat and mass transfer in bundles of coiled tubes with various types of unsteadiness: sharp and smooth variation of the thermal load when the apparatus is started or stopped or switched from one operating regime to another, and variation of the heat-transfer agent flow rate [16].

In conclusion, it should be noted that the heat-transfer processes in single-phase flows have been thoroughly studied and attention should now be focused on the development of improved designs of compact heat exchangers. In this connection, at the IPTPE considerable resources are being concentrated on developing and introducing new methods of intensifying heat transfer and on seeking optimum variants of compact heat exchangers.

NOTATION

c_f, coefficient of friction; d, tube diameter, m; F, area, m^2; k, height of the roughness elements, m; k^+, dimensionless roughness height; s_1 and s_2, transverse and longitudinal pitches, m; u, velocity, m/sec; u_*, dynamic velocity, m/sec; α, heat-transfer coefficient, W/(m^2·K); δ, thickness of the hydrodynamic boundary layer, m; δ^{**}, momentum loss thickness, m; ε_τ, coefficient of eddy viscosity, m^2/sec; ε_q, turbulent thermal diffusivity, m^2/sec; ϑ, temperature head, °C; ϑ_*, characteristic temperature; ϑ^+, dimensionless temperature; ν, coefficient of kinematic viscosity, m^2/sec; ρ, density, kg/m^3; τ, shear stress, N/m^2; ϕ, angle around the tube, deg; Nu, Nusselt number; Re, Reynolds number; Pr, Prandtl number; St, Stanton number. Subscripts: 0, smooth surface; w, wall.

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EVOLUTION OF A TURBULENT BURST

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A complete solution of the problem of symmetric turbulent burst decay in a quiescent fluid is obtained using the semiempirical theory of turbulence.

The problem considered is of interest from many points of view. In homogeneous shear flows near a wall the intersection and self-intersection of eddies cause isolated, sharply defined turbulent bursts which completely determine the development of turbulence in the flow. This was convincingly demonstrated by the classical experiments of Kline's group at Stanford [1]. Moreover, turbulence in a stratified (with respect to density in a gravity field) ocean is patchy or "insular" [2]. The occurrence of patches of turbulence is associated with turbulent bursts resulting from the breaking of internal waves or local shear flow instability and subsequent mixing.

Generally speaking, the patches of turbulence associated with a burst are initially asymmetric. However, they rapidly acquire a symmetrical shape and, accordingly, the problem of the evolution of symmetrical turbulent patches is of fundamental importance. This problem is examined in the present article which, in addition to summarizing recent research, presents a number of new results. As always, most interest attaches to the intermediate-asymptotic stage of evolution of the burst, when the size of the turbulent patch is much greater than the initial value. In this stage the evolution of the patch obeys self-similar laws. Here, however, the self-similarity is nonclassical, so-called incomplete self-similarity (self-similarity of the second kind, scaling). Time enters into the self-similar variables to a power determined by the solution of the nonlinear eigenvalue problem. The solution of the problem is obtained within the framework of two variants of theory of turbulence of the Kolmogorov type [3, 4]: the classical (b, ℓ) variant and the (b, ε) variant [5-7]. It is an important advantage of the burst problem that, because of its symmetry, at the boundaries the turbulent energy fluxes, energy dissipation rates, etc., are equal to zero, so that there is no need to use additional (often very dubious) hypotheses to determine them. The results are similar, so that the solution obtained is also of interest from the standpoint of testing the Kolmogorov semiempirical theory for essentially unsteady flows.

1. Formulation of the Problem

It is proposed to consider (Fig. 1a) the evolution of a sudden burst of turbulence in a homogeneous quiescent fluid. The simplest symmetrical burst shape is a statistically uniform horizontal layer. Accordingly, we will investigate the decay in an unbounded quiescent

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