INFLUENCE OF REYNOLDS AND PRANDTL NUMBERS ON THE EFFECTIVENESS OF INTENSIFICATION OF HEAT TRANSFER IN TUBES

E. K. Kalinin and S. A. Yarkho

Inzhenerno-Fizicheskii Zhurnal, Vol. 11, No. 4, pp. 426–431, 1966

UDC 536.25

The results are presented of an experimental investigation of intensification of heat transfer in tubes in the range of Re numbers $1.5 \cdot 10^3$–10^5 and of Pr numbers 0.7–50, in flows of gases, water, and a water-glycerin mixture. An analysis is made of the mechanism of heat transfer when the flow is rendered turbulent by artificial means.

The analysis [1] of numerous experimental papers on intensification of heat transfer indicates that the choice of type, size, and shape of artificial turbulence devices was made (with rare exceptions) without sufficient basis, and in the main without calculating the specific conditions in which it was proposed to use this or that type of heat-transfer surface. As regards attempts to analyze the mechanism of turbulent heat transfer in conditions of artificially induced turbulence, to set up a physical model of the phenomenon and to describe it analytically, no results have yet been published as far as we know. The reasons for this, apart from the objective difficulties (heat transfer in conditions with vortex and separated flow), are principally that the majority of authors use the hydrodynamic analogy method to analyze the thermal processes, and this method is evidently quite unsuitable in conditions of separated flow. The work of Nunner is typical [2], it being faultless in the experimental aspects, but not standing up to criticism on the theoretical side. The model of the phenomenon put forward by Nunner and its mathematical description give results which are not only quite contrary to physical sense, but conflict with the author's own experimental results. For example, according to his calculations, as Re and Pr increase, the increase of heat transfer compared with a smooth tube should be curtailed, i.e., the intensification should be impaired, which is at variance both with a whole series of experimental facts and with the physical essence of the phenomenon of heat transfer in conditions of artificially induced turbulence.

With the object of an experimental study of the influence of Re and Pr on the heat transfer intensification effect, as well as of broadening our ideas concerning the mechanism of heat transfer in conditions of artificially induced flow turbulence, we carried out investigations over a wide range of these parameters, both in a smooth tube and in tubes with different dimensions and distributions of artifical turbulence generators.

The investigation was conducted with hot gases, water, and a water-glycerin mixture, at q_W = const = 10^3–$2 \cdot 10^5$ W/m^2, in stainless steel tubes of diameter 10.6/9.6 mm; the length of the heated section was 110 tube diameters. The wall temperature was measured with chromel-alumel thermocouples, 20 to each tube, in conjunction with a R-2/1 potentiometer.

Artificial turbulence generators were created on the inside wall of the tubes by applying rolls to the tubes at periodic intervals and took the form of annular diaphragms of dimension $d/D = (D - 2k)/D = 0.985$–0.875, where D = 9.6 mm, and k is the height of the diaphragm. The pitch varied from 0.5 to 4 tube diameters.

Figure 1 shows the dependence of the mean heat transfer coefficient along the tube on the Reynolds number for a smooth tube and tubes with turbulence generators of different heights and 0.5 tube diameter pitch. The scatter of the experimental points does not exceeds ±10%.

These data are presented in Fig. 2. in the form of the dependence of the quantity Nu/Nu_s, which describes the heat transfer intensification effect, on Reynolds number. A broken line indicates the boundary value of the Reynolds number, Re*, above which the increase of heat transfer, in comparison with the smooth tube, remains constant, independent of Re. It follows from these graphs that the Reynolds number has a strong influence on heat transfer intensification, varying in different regions. We may discern three regions of Reynolds number: 1) the region $Re < Re_{cr}$, where there is no intensification of heat transfer in the tubes with artifical turbulence generators, the value of the critical Reynolds number decreasing here with increased height of the generators, and with $d/D = 0.875$ $Re_{cr} \approx 1600$; 2) the region $Re_{cr} < Re < Re^*$, where there is a sharp increase of the effect with increase of Reynolds number; 3) the region $Re > Re^*$, where $Nu/Nu_s > 1$ = const. For large Re numbers there may also be a region in which the effect falls with increase of Re.

Figure 3 shows the results of an investigation of the effect of Pr number of heat transfer in a smooth tube and in tubes with artificial turbulence generators. The tests were conducted in the region of Re close to the boundary value Re* and above it. It follows from the graphs that in the range of Pr examined, the latter influenced the heat transfer, both in the tubes with generators and in the smooth tube. Some difference, an increase of intensification with increase of Pr, occurs in the weakly developed turbulence region, as is seen from Fig. 1.

Figure 4 shows the influence of the height of the generators on the heat transfer intensification effect for a constant pitch of diaphragms of 0.5 tube diameter and various Re numbers. The graph clearly illustrates the fact that in the region of developed flow turbulence, increase of generator height is accompanied by increased heat transfer only up to a definite limit, and that beyond this limit Nu/Nu_s = const, and independent of the height of the diaphragm, while ξ/ξ_s increases strongly with further increase of height. This limiting height depends mostly on the Re number, and is noticeably larger in the region of weakly developed turbulence.

DISCUSSION AND ANALYSIS OF RESULTS

The fact that for subcritical values of Re the artificial generators do not promote enhanced heat transfer is explained by the fact that in these conditions not only do the generators not create vortex regions but the fluid between them stagnates. This creates an additional thermal resistance of the slow-moving sublayer of fluid. In our tests this circumstance appeared to be the cause of the noticeable decrease of heat transfer coefficient in the tubes with generators in comparison with the smooth tube. Typically, the greatest influence of the stagnant zones occurred in the initial sections of heat supply, i.e., just where the thickness of the stagnant zone is commensurate with that of the thermal boundary layer. At a large distance from the beginning of the heated section ($x/D > 100$), practically no decrease of heat transfer was observed (it is possible that here there was the additional influence of free convection to disturb the stagnant zones).

Thus, for $Re < Re_{cr}$, the heat transfer in the tubes with generators is less than in the smooth tube, but the nature of its dependence on Re and Pr practically coincides with that of the smooth tube.

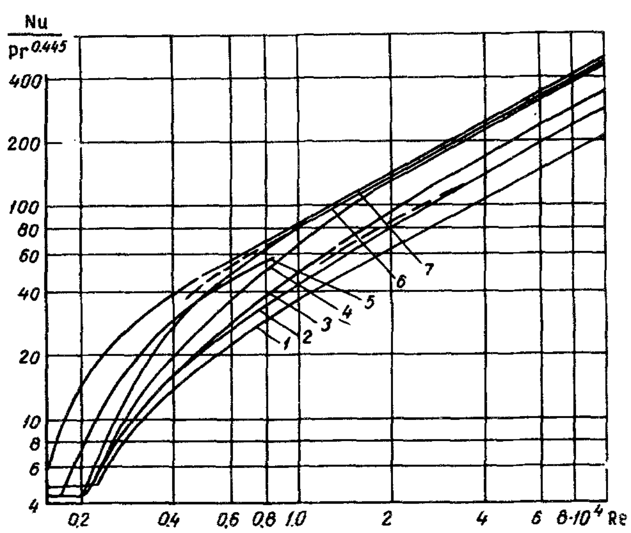

Fig. 1. Influence of Re number on the mean heat transfer coefficient in the tubes: 1) smooth; 2) d/D = = 0.983; 3) 0.966; 4) 0.943; 5) 0.912 (pitch 1.0); 6) 0.92; 7) 0.875. The solid lines are for water, and the broken lines for the water-glycerin mixture, d/D = 0.92 and 0.966.

In the region $Re_{cr} < Re < Re_{cr.s}$, i.e., at Re values greater than critical but below critical for the smooth tube, we find the largest increase of intensification of Nu/Nu_s with increase of Re. It should be especially stressed that the forcing of transition, i.e., decrease of the value of the critical Re number in the rolled tubes, is not typical for all diaphragm heights. In our test in the tubes with diaphragms $d/D > 0.92$, the critical Re number remained constant, though somewhat smaller than the Re_{cr} for the smooth tube. But with heights $d/D \leq 0.92$ there occurred a sharp fall of the critical Re number, which is, generally speaking, in conformity with the physical model of transition on a rough surface proposed by Dryden [3]. This is the case for a tube with $d/D = 0.875$, $Re_{cr} = 1580$. Decrease of Re_{cr} occurs because of early loss of stability, since the artificial generators are centers at which additional disturbances originate.

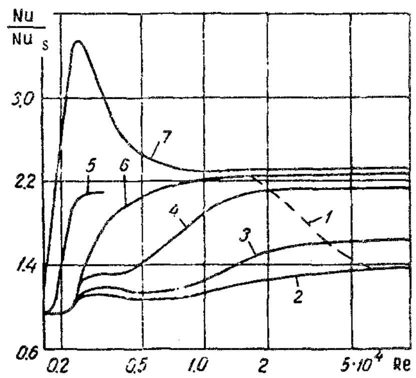

Fig. 2. Influence of Re number on the effectiveness of heat transfer intensification: 1) boundary of Re* values; 2–7) see Fig. 1.

Therefore, the sharp increase of the intensification effect in this region is explained by the earlier transition to turbulent flow, whereas the viscous regime is preserved in the smooth tube. The influence of Prandtl number in this quite small range of Re numbers was not investigated specially.

The range $Re_{cr} < Re < Re^*$ is characterized by the fact that at these Reynolds numbers the artificial turbulence generators actively affect the flow, making the layers of fluid near the wall turbulent. In the weakly developed turbulence region the best heat transfer intensification effect is obtained with generators of comparatively large height, i.e., those which promote earlier transition and make thick layers near the wall turbulent.

With further increase of Reynolds number, the turbulent transfer in the flow core increases so much that the thermal resistance is concentrated almost entirely in the thin wall layers of fluid, where the turbulent heat transfer coefficient ε_q decreases sharply, and the heat flux is a maximum. The thickness of these layers becomes commensurate with the height of the generators. It is not difficult to show that the Re* values correspond exactly to the thickness of the layers of fluid in which 99% of the total temperature head between the wall and the tube axis is developed. The Re* boundary shown in Fig. 2 was obtained by calculation. From the energy equation written for a smooth tube (the distribution of ε_T/ν was taken from the experimental data of Sleicher [4]) we found the dependence of the size of the thermal wall layer with the temperature drop $\Delta t = 0.99\Delta t_{max}$ on the Reynolds and Prandtl numbers. Equating this value to the heights of the diaphragms, we obtain

$$Re^* = 3150/(1 - d/D)^{1.14} Pr^{0.57}. \quad (1)$$

It follows from Fig. 2 that (1) corresponds satisfactorily to the Reynolds number values up to which the intensification effect occurs. In other words, until the heights of the generators become appreciably less than the thickness of the fluid layers which have the main thermal resistance, an increase in Re and Pr is accompanied by increase in the intensification effect.

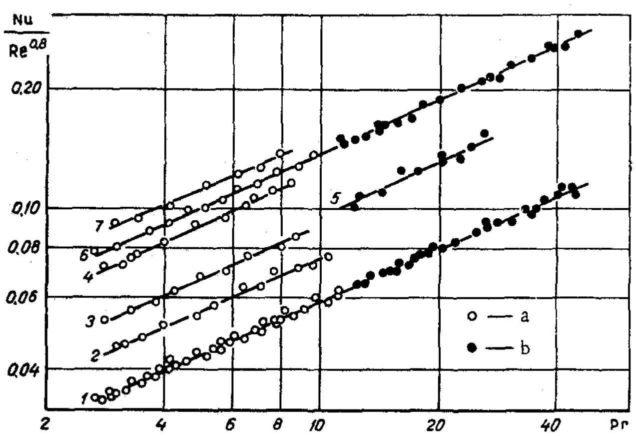

Fig. 3. Dependence of mean heat transfer coefficient on Pr number in tube: 1) smooth; 2) 2/ /D = 0.983; 3) 0.966; 4) 0.943; 5) 0.946 (pitch 1.0); 6) 0.92; 7) 0.875; a) water; b) water-glycerin mixture.

Some decrease in the intensification effect Nu/Nu_s at $Re > Re_{cr.s}$ (Fig. 2) occurs for natural reasons: in the smooth tube the heat transfer begins to increase sharply, and therefore the relative effect decreases.

It should be stressed that, according to Eq. (1), the Prandtl number, has an effect, though less than that of Re, on the thickness of the wall layer in which 99% of the total temperature head is developed. However, in the range Pr = 0.7–10 in gases and water, practically no influence of Pr on Re was observed. Similar conclusions were reached in [5,6]. At the same time, in going from water to the water-glycerin mixture, we find satisfactory agreement between the experimental data and those calculated from Eq. (1) regarding the influence of Pr on Re*. This is clearly illustrated in Fig. 1, from which it may be seen that in the water-glycerin mixture, the Nu/Nu_s = const point is reached at smaller values of Reynolds number, which also causes an increase in the intensification effect with increase of Pr in the region of comparatively undeveloped turbulence.

The region Re > Re* is characterized by the absence of an influence of Re and Pr on the heat transfer intensification effect, i.e., Nu/Nu_s = const. In these conditions an increase of Re and Pr leads to propagation of the artificial turbulence into the layer of fluid where 99% and even more of the available temperature head exists. Then the natural turbulent conduction in the diaphragm height zone also increases so much that the contribution of the artificial turbulent conductivity is not altered. Hence it is apparent that for Re > Re* we do not get an additional increase of heat transfer with increase of Re and Pr, and the nature of the influence of these parameters on the heat transfer becomes the same as in a smooth tube, but with $Nu/Nu_s > 1$.

The influence of the generator height on the heat transfer intensification effect, as we have verified, turns out to be varied in this wide range of Reynolds number. While high generators give the best effect in the weakly developed turbulence region, the use of high generators in developed turbulent flow is not favorable, since the increase of turbulent heat transfer at a large distance from the wall, where it is any way much greater than the molecular, though the heat flux is small (it is known that the heat flux varies over the tube radius according to a law that is close to linear), does not give an appreciable intensification of heat transfer, but is accompanied by a very large increase in the hydraulic losses. The original increase of Nu/Nu_s with increase of height of the diaphragms in the region Re > Re* is due to the fact that the strength of the vortices, and therefore, of the artificial turbulence of the wall region, increases. But this process is limited, the additional turbulent fluctuations are formed at the upper boundary of the vortex, but the vortex moves away from the wall with increasing generator height, which compensates the increase of intensification and leads to saturation of the $Nu/Nu_s = f(d/D)$ curves, other conditions being equal (Fig. 4).

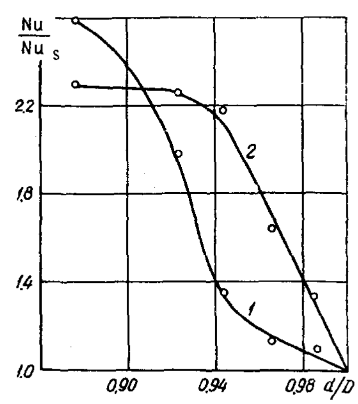

Fig. 4. Influence of turbulence generator height on the effectiveness of heat transfer intensification at Re values of 4000 (1), and 40 000 (2).

REFERENCES

1. A. I. Mikhailov, V. V. Borisov, and E. K. Kalinin, Closed-Cycle Gas-Turbine Equipment [in Russian], Izd. AN SSSR, 1962.

2. W. Nunner, VDI-Forschung, 22, 455, 1956.

3. H. L. Dryden, Turbulent Flow and Heat Transfer, ed. by C. C. Lin, Princeton, 1959.

4. C. A. Sleicher, Paper Soc. Mech. Eng. 57HT-9, 1957.

5. R. S. Hastrup, Jet Propulsion, 28, no. 4, 1958.

6. D. F. Dipprey, Intern. J. Heat and Mass Transfer, 6, no. 5, 1963.

21 June 1966

Moscow Aviation Institute