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MODERN PROBLEMS OF INTENSIFICATION OF HEAT EXCHANGE IN CHANNELS

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The current achievements in the field of intensification of heat exchange in tube heat-exchange apparatus are considered. Requirements for high-efficiency heat-transfer surfaces have been formulated. Problems of intensification of heat exchange in bundles of finned tubes and in heat-exchange apparatus under conditions of condensation and boiling of heat-transfer agents and under scaling conditions have been investigated. High-efficiency designs of heat-exchange apparatus are presented.

Introduction. Heat-exchange apparatus are used in aviation and space engineering, power engineering, the chemical, petroleum, and food industries, refrigeration and cryogenic engineering, heating, hot-water, and conditioning systems, and in different heat engines. As the power capacities and outputs increase, the mass and dimensions of heat-exchange apparatus in use become larger and larger. A great amount of alloyed and nonferrous metals are spent on their production.

The problem of decreasing the mass and dimensions of heat-exchange apparatus is currently central. The most promising way of solving this problem is intensification of heat exchange.

Experience in designing and operating different heat and mass-exchange apparatus has shown that the methods of intensification of heat exchange that have been developed by now provide a decrease in the dimensions and in the specific quantity of metal (mass) of these apparatus by a factor of 1.5–2 or more as compared to analogous commercial apparatus on condition that the heat power and the power for pumping of heat-transfer agents remain the same.

Investigations on the intensification of heat exchange are being carried out in different countries, and their pace is rapidly increasing. It should be noted that investigations carried out in the former USSR have made a great contribution to the solution of this problem, especially to the development of methods for intensification of heat exchange that can be realized in practice. It will suffice to remember the works of V. M. Antuf'ev, V. M. Buznik, G. I. Voronin, E. V. Dubrovskii, N. V. Zozulya, E. K. Kalinin, V. K. Migai, V. K. Shchukin, and many other scientists. And only the fact that the practical use of high-efficiency heat-exchange apparatus and savings in metal are of no interest to our industry can explain the poor introduction of domestic apparatus into the national economy. We should call attention to one more sad fact: since the main publications of our scientists on the intensification of heat exchange have been in Russian and the majority of foreign scientists do not know Russian, these works remain unknown till now to world science. As a result, in numerous foreign publications devoted to problems that were investigated earlier in the former USSR, analogous works published in Russian are not even mentioned. Of course, there can be a certain bias here, but still the main reason has been stated above.

By now, various methods of intensification of convective heat exchange have been proposed and investigated.

As applied to the flow of single-phase heat-transfer agents, the following methods are used: turbulizers of the flow on the surface; rough surfaces and surfaces developed due to finning; swirling of the flow by

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spiral fins, screw devices, and swirlers, installed at the inlet to the channel; addition of gas bubbles to a liquid flow and of solid particles or liquid drops to a gas flow; rotation or vibration of the heat-transfer surface; pulsations of the heat-transfer agent flow; action of electrostatic fields on the flow; suction of the flow from the boundary layer, and jet systems. The efficiency of these methods is dissimilar. In the best case, one is able to increase the heat transfer by a factor of 2–3, but with significantly different energy expenditures for different methods of intensification.

Under boiling conditions, intensification of heat exchange provides not only an increase in the heat transfer in the case of nucleate and film boiling, but also an increase in the maximum heat flux in the case of nucleate boiling and an increase in the minimum heat flow in the case of film boiling as well as an increase in the corresponding critical temperature heads, i.e., a shift of the boiling curve toward higher temperature heads and heat fluxes. It should be noted that the possibilities of intensification of heat exchange in the case of boiling are much higher than in the case of single-phase flows. For example, in the case of film boiling, one is able to increase the heat-exchange coefficient by a factor of 10 and the critical heat flux by a factor of more than 3. In the case of boiling, along with turbulizers, screw devices, and finning, the following methods are used for intensification of heat exchange in boiling: thin coatings of a low-conducting or a porous material are applied to the surface, nonisothermic fins are installed, and rough surfaces are used.

For intensification of heat exchange in the case of condensation, it is proposed to use turbulizers or fins, which break down the condensate film; nonwettable coatings; liquid stimulators for creation of dropwise condensation, and swirling of the flow or rotation of the heat-transfer surface.

Application of combined methods of intensification was found to be highly efficient in many cases: combination of turbulizers with finning of the surface or with swirling of the flow, and also use of screw devices in the case of flowing suspensions and turbulizers with low-conducting coatings in the case of boiling.

It should be noted that, in practice, in deciding on one method of intensification of heat exchange or another, one has to take into account not only the efficiency of the surface itself, but also its universality for different single-phase and two-phase heat-transfer agents, the ease of manufacture of the surface, the ease of assembly of the heat-exchange apparatus, the strength requirements, the fouling of the surface, the peculiarities of operation, etc. All these circumstances significantly complicate the choice of one method out of the large number of investigated methods of intensification.

1. Modern Designs of Tube Heat-Exchange Apparatus. The compactness of the apparatus is determined by the diameter of the tubes used and the allowable steps of their arrangement in the tube plate, which in turn is limited by the technology level achieved in the corresponding industries and by considerations of convenience of repair, cleaning, and operation. These circumstances as a whole determined the level of compactness of heat-exchange apparatus used in different fields of engineering.

At the present time in Russia, tubes with the following outside diameters are used in heat-exchange apparatus operating in different industries: D_{out} = 17–50 mm in the chemical industry, D_{out} = 16–50 mm in power engineering, D_{out} = 16 mm (minimum diameter) in heating and hot-water systems, D_{out} = 10–12 mm in shipbuilding, D_{out} = 6–8 mm in refrigeration and cryogenic engineering, and D_{out} = 3–4 mm in aviation and space engineering. The step of arrangement of tubes in the bundle is usually chosen from the range S_{out}/D_{out} = 1.27–1.5. In rare cases, for the most compact apparatus, S_{out}/D_{our} decreases to 1.2.

Heat-exchange apparatus differ in design insignificantly. They are subdivided into two types: apparatus with a longitudinal or transverse flow around the space between the tubes of the type "tube in tube" and apparatus with spiral tubes. To increase the heat-exchange efficiency in the space between the tubes, finning of the outer surface of the tubes is used: common plane fins for a bundle of round or oval tubes, rectangular transverse or longitudinal fins, round ribs, multipass spiral fins, and wire finning. Fins are made of copper, aluminum, or other highly heat-conducting materials and provide an increase in the heat-transfer surface from outside by a factor of up to 20. As a rule, they are smooth, i.e., the possibilities of increasing the heat transfer on them due to additional artificial turbulization of the flow are not used.

2. Method of Estimating the Efficiency of Intensification of Heat Exchange in Channels. It should be noted that in many investigations on intensification of heat exchange, the efficiency of the methods under study escaped the attention of researchers completely or the estimation methods used were unsuitable for practical purposes. As a rule, the authors of publications devoted to intensification of heat exchange compare the results obtained in the form of dependences between the ratios $\mathrm{Nu}/\mathrm{Nu}_{\mathrm{sm}}$ and ξ/ξ_{sm} or estimate the efficiency of the studied intensification method by the parameter $(\mathrm{Nu}/\mathrm{Nu}_{\mathrm{sm}})/(\xi/\xi_{\mathrm{sm}})$, naturally, focusing their attention on the results in which this parameter is more than unity.

We would like to call your attention to the fact that the possibility of obtaining an anticipating increase in the heat transfer relative to the increase in the hydraulic resistance, as compared to an analogous smooth channel, is of great scientific interest, but does not always lead to the highest efficiency of intensification of heat exchange. It is known [1] that it is easiest easy to estimate the efficiency of intensification of heat exchange if the heat-transfer volumes or the heat-transfer surfaces of two heat-exchange apparatus made of a surface with heat-exchange intensification and of a surface without it are compared for the same heat powers and the same powers spent on pumping of the heat-transfer agent (for the same flow rates of the heat-transfer agent, this means that the compared apparatus will have the same pressure losses). If the channels compared have the same diameters, if the existence of turbulizers in the channel is not taken into account in determining the heat-transfer surface and the rate of the flow in the channel, and if the heat-exchange coefficient in the channel under consideration is much smaller than on the other side of the heat exchanger, the ratios between the volumes of the apparatus compared in the case of turbulent flow of the heat-transfer agent will be as follows:

$$\frac{V}{V_{\mathrm{sm}}} = \frac{(\xi/\xi_{\mathrm{sm}})_{\mathrm{Re}}^{0.4}}{(\mathrm{Nu}/\mathrm{Nu}_{\mathrm{sm}})_{\mathrm{Re}}^{1.4}}, \tag{1}$$

where the ratios $(\xi/\xi_{\mathrm{sm}})_{\mathrm{Re}}$ and $(\mathrm{Nu}/\mathrm{Nu}_{\mathrm{sm}})_{\mathrm{Re}}$ are taken for the same Reynolds numbers, in this case, for the heat exchanger with intensification of heat exchange. As is seen from (1), the intensification is efficient if $(\xi/\xi_{\mathrm{sm}}) < (\mathrm{Nu}/\mathrm{Nu}_{\mathrm{sm}})^{3.5}$. The optimum of V/V_{sm} does not correspond to the optimum of the quantity $(\mathrm{Nu}/\mathrm{Nu}_{\mathrm{sm}})/(\xi/\xi_{\mathrm{sm}})$. With account for (1), we will consider different methods of intensification of heat exchange.

3. Choice of a Rotational Method of Intensification of Heat Exchange for the Case of Gas and Liquid Flows in Tubes. Abundant evidence shows that of all the known methods of intensification of heat exchange in tubes, artificial turbulization of a flow by ring diaphragms and continuous swirling of a flow have received primary attention as efficient and technologically realizable methods.

As an example of the efficient artificial turbulization of a flow, we consider the method developed at the Moscow State Aviation Institute for tube heat-exchange apparatus [1].

The essence of the method is as follows. Periodically arranged annular grooves are made by rolling on the outer surface of the tube (Fig. 1).

In this case, ring diaphragms with a smooth configuration are formed on the inner surface of the tubes. The ring diaphragms and the grooves create turbulence in the near-wall layer of the flow and provide intensification of heat exchange inside and outside the tubes, which makes it possible to use these tubes in close tube bundles and does not change the existing technology of assembly of heat-exchange apparatus.

The technology of knurled tubes which has been developed at the All-Union Scientific-Research Institute of Metallurgical Machine Building is simple and allows the use of standard equipment; the cost of the knurling does not exceed several percent of the cost of the tubes. A device mounted on a lathe provides a

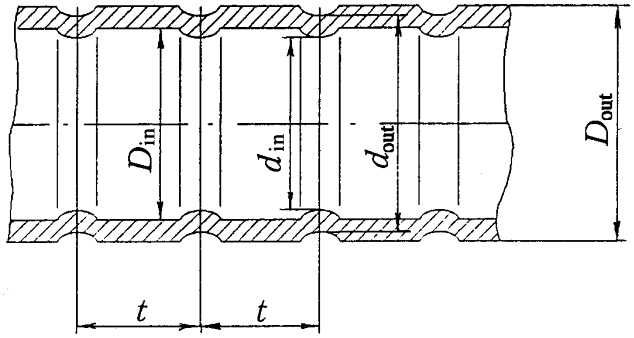

Fig. 1. Tube with ring turbulizers.

production rate of knurling of up to 1–2 m/min. A specialized mill provides the knurling of tubes with a production rate of up to 9 m/min.

The developed tubes with ring turbulizers can be used for apparatus operating with gases and liquids and under conditions of boiling and condensation of heat-transfer agents, i.e., they possess the universality necessary for practical use. Moreover, these tubes are characterized by a lowered fouling. Hence, tubes with ring turbulizers satisfy all the requirements necessary for their widespread practical use.

It should be noted that it is precisely these tubes in which the law of change in the heat transfer on the walls of the channels with discrete turbulization of the flow in the case of forced convection has been revealed for the first time and recognized as a discovery. This law implies that for a certain arrangement of turbulizers whose dimensions fall within a certain range, the increase in the heat transfer is larger than the increase in the hydraulic resistance as compared to an analogous smooth channel [2]. The use of the ratio $(Nu/Nu_{sm}) > (\xi/\xi_{sm})$ that is realized in practice makes it possible to decrease not only the volume of a heat-exchange apparatus but also its cross-sectional area for the given values of the heat power and hydraulic resistance of the apparatus.

In the case of an air flow in these tubes, the maximum values of the intensification of heat exchange $Nu/Nu_{sm} = 2.65$, 2.82, and 3.12 were obtained, respectively, for $Re = 10^4$, 10^5, and $4 \cdot 10^5$. In the opinion of Migai [3], these data are close to the limiting values of the intensification of heat exchange in tubes due to turbulization of the flow calculated by him, $Nu/Nu_{sm} = 4.06$ and 3.62 for $Pr = 0.7$ and $Re = 10^4$ and 10^5, respectively, i.e., the possibilities of further increase in the intensification of heat exchange of using this method are limited.

The use of this method of intensification of heat exchange makes it possible to decrease the volume of a heat-exchange apparatus by a factor of 1.5–2 for constant values of the heat power and the power for pumping of the heat-transfer agents. In the transition region of flow of the heat-transfer agent, the intensification effect is even higher and makes it possible to decrease the volume of the apparatus by a factor of up to 2.5.

The design of a heat-exchange tube developed earlier [1] has been improved significantly by application of turbulizers with a smooth configuration. In knurling of ring diaphragms, different shapes of turbulizers and the tube surfaces can be obtained, depending on the technological regimes, the material, and the thickness of the wall. Since the heat transfer in a tube changes insignificantly and the resistance decreases by 25–40% when the shape of the turbulizers is changed on condition that their height and pitch remain unchanged, the use of turbulizers with a smooth configuration makes it possible to increase significantly the efficiency of heat exchangers with these tubes.

Let us now consider the available data on intensification of heat exchange in tubes due to a continuous swirl of a flow, which can be provided by twisted tapes or screw inserts positioned along the whole length of the tube (Fig. 2). In contrast to a local swirl, this swirl is technologically simpler and provides a larger increase in the average heat transfer, since the degree of swirling of the flow decreases along the

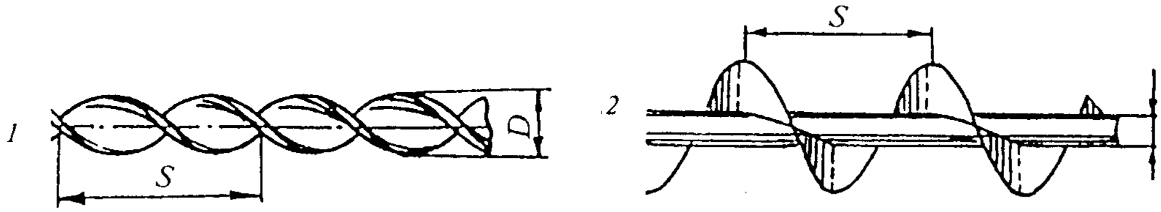

Fig. 2. Screw inserts: 1) twisted tape; 2) screw.

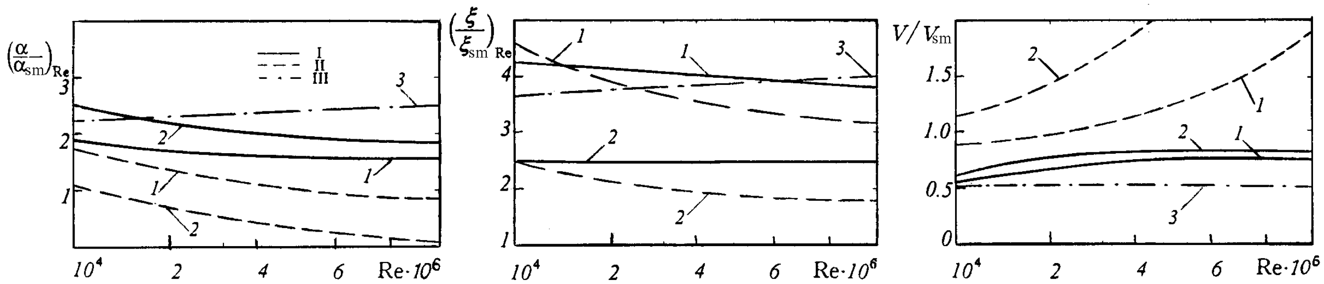

Fig. 3. Influence of the Reynolds number on the increase in the heat-exchange coefficients $(\alpha/\alpha_{sm})_{Re}$ and in the hydraulic resistance $(\xi/\xi_{sm})_{Re}$ and on the ratio between the volumes of the heat-exchange apparatus V/V_{sm} (I, swirling of the flow by a twisted tape; II, swirling of the flow by a screw; III, ring diaphragms): 1) S/D = 4; 2) 10; 3) ring diaphragms with d/D = 0.94 and t/D = 0.25.

length of the channel. However, the hydraulic resistance increases in this case because of additional pressure losses by friction on the surface of the tape or of the screw.

V. K. Shchukin [4] has obtained corresponding dependences that accumulate a large body of experimental data of different researchers. However, their use for analysis of the swirl efficiency is complicated by the fact that in them the velocity of the flow in the tubes with inserts was determined with regard to the blocking of the tube by inserts, and the equivalent diameter of the tube with an insert was taken for the determining size. With the use of the data from [4], formulas for $(\xi/\xi_{sm})_{Re}$ and $(\alpha/\alpha_{sm})_{Re}$, which are handy for practical calculations and allow one to easily determine the increase in the heat-exchange coefficients and in the hydraulic resistance as compared to a tube free of inserts at a given geometry of the inserts and a constant flow rate of the heat-transfer agent and, consequently, to find the efficiency of intensification of heat exchange, have been obtained in [5]. The use of these dependences allowed practical workers to answer the following question of interest to them: How will the arrangement of screw inserts inside tubes increase the hydraulic resistance and the heat flux, the flow rate and all other things being equal?

Some results are presented in Fig. 3, where the dependences of $(\alpha/\alpha_{sm})_{Re}$, $(\xi/\xi_{sm})_{Re}$, and V/V_{sm} on the Reynolds number are shown (the latter was obtained under the conditions described in Sec. 2).

For Re = 10^4, a twisted tape gives α/α_{sm} = 2.34–1.8 and ξ/ξ_{sm} = 4.05–2.5, which makes it possible to obtain a decreased volume of the apparatus V/V_{sm} = 0.53–0.64. With increase in Re, the efficiency of the tape inserts decreases significantly: α/α_{sm} = 1.88–1.49 and ξ/ξ_{sm} = 5.55–1.65 for Re = 10^5, which gives V/V_{sm} = 0.822–0.70. It should be noted that for none of the Re values and twist pitches of the tape has it been possible to obtain $\alpha/\alpha_{sm} \geq \xi/\xi_{sm}$, i.e., the anticipating increase in the heat-transfer coefficient as compared to the increase in the hydraulic resistance.

In Fig. 3, the results obtained are compared to the above-described data [1] for ring diaphragms. The use of ring turbulizers provides a means for a stable increase in the heat transfer by a factor of 2.3–2.43 in the range of values of Re = 10^4–10^5, characteristic for heat-exchange apparatus, with an increase in the hydraulic resistance by a factor of 3.8–4.15, which makes it possible to obtain V/V_{sm} = 0.52–0.5 or a decrease in the volume of the apparatus by a factor of 1.95–2 (the anticipating increase in the heat transfer in such tubes is attained for large values of d/D = 0.97–0.98; V/V_{sm} = 0.5–0.6 in this case). Hence, the efficiency of intensification of heat exchange performed with the use of tape inserts for Re = 10^4 is somewhat lower, and for Re = 10^5 it is markedly lower than the efficiency of intensification of heat exchange performed with the use of ring turbulizers.

As is seen from Fig. 3, the efficiency of screw inserts is much lower than the efficiency of tape inserts. Even with minimum values of d_0/D = 0.33 and δ/D = 0.05, for twist pitches $S/D \cong 4$–12, we obtain α/α_{sm} = 1.75–1.16 and ξ/ξ_{sm} = 4.74–2.64 when Re $\cong 10^4$ and α/α_{sm} = 0.88–0.58 and ξ/ξ_{sm} = 3.4–1.38 when Re $\cong 10^4$. In this case, V/V_{sm} = 0.84–1.19 for Re = 10^4 and V/V_{sm} = 1.9–2.67 for Re = 10^5.

Thus, a small increase in the efficiency ($V/V_{sm} < 1$) can be obtained only for S/D = 4 and Re = 10^4. With increase in Re and S/D, the value of $V/V_{sm} > 1$, i.e., the use of screw inserts gives a negative result since it deteriorates the parameters of heat-exchange apparatus. Screw inserts with large relative diameters of the screw core d_0/D and large thicknesses of the screw fins δ/D were found to be even less efficient.

Unfortunately, Shchukin [4] and other researchers of the heat exchange in a swirling flow did not attempt to explain the increase in heat transfer in the experiments with screws as compared to a tube free of screws ($\alpha/\alpha_{sm} < 1$), and this result has escaped their attention completely.

All the above-mentioned data for screw inserts have been obtained in the case where these inserts were tight against the inner walls of the tubes. If an annular gap appears between the inserts and the tube, the efficiency of intensification of heat exchange decreases markedly [4]. Other methods of swirl (spiral channels, swirl of the flow at the inlet to the channel, twisted tubes, spiral wire inserts, spiral or longitudinal fins inside the tubes) are less efficient than the methods considered above. Such methods as the creation of pulsations in the flow using special pulsers installed at the inlet and the use of rough surfaces are also less efficient.

4. Efficiency of Some New Methods of Intensification of Heat Exchange in Channels. It should be noted that the fundamental theoretical and experimental investigations on the intensification of heat exchange in channels, which were performed in the 1950–1970s, in particular, the investigation of the turbulent structure of the flow in channels with different devices for intensification of heat exchange, enable the majority of specialists engaged in problems of intensification of heat exchange to choose the investigation methods and concrete efficient methods of intensification correctly. As a result, in recent years, the number of low-efficiency methods used in investigations decreased and the level of investigations increased.

However, there are exceptions. As an example, we consider [6], in which the intensification of heat exchange in a channel with a triangular cross section and rounded corners and inserts creating turbulence was investigated as applied to a channel for cooling of gas-turbine blades. An insert with bulges and cavities with a spherical surface, alternating in the longitudinal direction, did not give an increase in the heat transfer on the walls of the channel. An insert with alternating cuts whose edges were flanged in opposite directions increased the heat transfer by 35% with an increase of 3–4 times in the hydraulic resistance. It is easy to foresee such a result, since these inserts were arranged in the core of the flow and had no influence on the turbulization of the near-wall layer that has the main influence on the intensity of heat exchange. Unfortunately, weakly influencing the heat transfer, these inserts significantly increase the resistance of the channel. It is well known [7] that in channels of such type, efficient intensification is attained in the presence of transverse bulges and grooves on their side walls. In this case, the value of Nu/Nu_{sm} increased by a factor of 2.6 with an increase of up to 6 times in the value of ξ/ξ_{sm}; what is more, there is a region where $Nu/Nu_{sm} > \xi/\xi_{sm}$ up to $Nu/Nu_{sm} = \xi/\xi_{sm} = 2.1$.

In recent years, a great number of investigations on the intensification of heat exchange on surfaces with spherical cavities (hollows) were performed. We think that the given method of improvement of heat exchange is one of the ways of artificial turbulization of the flow near the wall. We set off it from other methods, since this problem was actively discussed in the last few years at many conferences and seminars.

The essence of the above-mentioned method of intensification of heat exchange, which has been proposed by a group of workers of the I. V. Kurchatov Institute of Nuclear Power, headed by G. I. Kiknadze, is as follows. Spherical cavities – hollows – are arranged in a certain order on the heat-transfer surface. It is the opinion of Kiknadze et al. [8] that the flow around such surfaces brings about the formation of self-organizing dynamic vortex structures flowing out of the hollows in the form of jets evolving in a wide range of flow velocities and conditions. Vortex jets have a column-like shape. They continuously draw the medium from the hollows, the near-wall layer, and the surface around these hollows and carry it to the core of the main flow. Kiknadze et al. think that these vortices are similar to a tornado; therefore, they call them self-organizing tornado-like vortices or structures. In the opinion of Kiknadze et al., generation of such vortices in gas and liquid flows is accompanied by an increase in the rate of heat- and mass-exchange processes as compared to their rate on a smooth surface in the case of a nonanticipating or a lagging increase in the hydraulic resistance. Moreover, Kiknadze et al. argue that in the case of smooth by-passes of spherical bundles, for a number of their arrangements and flow conditions, the pressure losses by pumping of a gas or a liquid can be decreased as compared to a smooth surface, all other conditions of the flow being the same.

The developers of this method believed that it is possible to increase the heat-exchange coefficient in gases and liquids in rectilinear channels by a factor of 2–3 with a lagging increase in the hydraulic resistance or with its constant value, to increase the critical heat loads in systems cooled by water by a factor of 1.5, to realize cooling regimes in which the temperature of the heat-transfer surface remains constant despite the 1.5–2-fold increase in the heat flux from the surface for the same pressure, temperature, and flow rate of the heat-transfer agent, and to decrease the deposit of impurities from the flow of the heat-transfer agent on the heat-transfer surface in flow of the heat-transfer agent. This assurance was stated again in [9]. Such sensational promises generated great interest among scientists and, about 12–15 years ago, many specialists in heat exchange began to investigate this phenomenon. Investigations were carried out at the Institute of Applied Physics of the Academy of Sciences of the USSR, the N. É. Bauman Moscow State Technical University, the Scientific-Research Design Institute of Power Engineering, the Scientific-Production Association "Trud," the Central Institute of Nuclear Engineering, the All-Union Heat-Engineering Institute, the Moscow Aviation Technical Institute, the Kazan Aviation Institute, the Scientific-Production Association of the Central Boiler and Turbine Institute, and a number of other organizations: an unprecedented case in domestic science; previously, as a rule, specialists in heat-exchange intensification preferred to study their own ideas. The investigations carried out were, of course, useful and made it possible to remove a touch of sensationalism from this problem.

Unfortunately, the experiments carried out recently [10, 11] have shown more modest results. In [10], the intensification of heat exchange in planar channels of width 30 mm and height 1.2 and 3 mm was investigated. Cavities were made on one of the channel surfaces as well as on both surfaces. The diameter of the hollows was d_{hol} = 3 mm and their depth was h = 0.15 and 0.3 mm. The ratio of the surface occupied by a hollow to the total heat-transfer surface was f = 0.2–0.78 and the Reynolds numbers were Re = $6{\cdot}10^3$–$2{\cdot}10^5$. For maximum values of h and f, an increase in the heat transfer by a factor of 2–1.7 is accompanied by an increase of 2–3 times in the resistance.

In [11], the intensification of heat exchange in a ring channel with an outside diameter of 14.33 mm and an inside diameter of 9.78 mm was investigated. On the inner tube, hollows with d_{hol} = 2.2 mm, h = 0.5 mm, and h/d_{hol} = 0.11 were made. The experiments were carried out in unilateral heating of the air and Re = $9{\cdot}10^3$–$9{\cdot}10^4$. In the case of unilateral heat supply, with increase in the temperature factor, the heat transfer decreases sharply, and for its value of 1.15–1.5 the increase in the heat transfer is $\mathrm{Nu}/\mathrm{Nu}_{\mathrm{sm}} \cong$ 1.1–1.2, while ξ/ξ_{sm} = 1.35–1.4.

It should be noted that the results of E. F. Kuznetsov, which are close to the above-mentioned results, were demonstrated by Migai [12] already in 1988. The experiments were performed for a flow inside a tube in triangular and planar channels with hollows on two surfaces as well as in a tube with bulges inside it.

Values of the intensification of heat exchange of up to $Nu/Nu_{sm} = 1.7–1.8$ were observed for a planar channel with large $h/H = 0.35–0.4$. In this case, $\xi/\xi_{sm} = 1.5–2.1$, and there is a point where $Nu/Nu_{sm} = \xi/\xi_{sm} = 1.8$. It is apparent that if h/H decreases, the efficiency deteriorates significantly.

Thus, the presented experimental data show that the above-mentioned method allows one, at best, to obtain results usual for the methods of intensification of heat exchange due to artificial turbulization of the flow, i.e., $Nu/Nu_{sm} = 1.5–2$ is accompanied, as a rule, by a close or a larger increase in ξ/ξ_{sm}. The statements of Kiknadze et al. [9] that the heat transfer on the surfaces with hollows can be increased without increase in the resistance were not confirmed by the experiments carried out.

5. Intensification of Heat Exchange in Finned-Tube Bundles. The problem of intensification of heat exchange in finned-tube bundles in transverse flow is as urgent as before. Attempts to increase the heat exchange of the fins by cutting them and by special arrangement of tubes with cut fins did not provide a high efficiency (the heat transfer, at best, increases by 36%).

It is apparent that efficient methods of intensification of heat exchange in this case and in the case of flow of a heat-transfer agent in channels can be developed only on the basis of a detailed investigation of the pattern of the flow in the space between the tubes. Some results of such investigations are presented in [13], where a number of interesting features have been revealed by way of visualization of the flow and measurement of the distributions of the static pressures and the intensities of heat exchange on the surface of a fin. It has been established that near the roots of the fins secondary recirculation flows exist which determine the character of heat transfer on the surface of a finned cylinder and lead to a marked three-dimensionality in its wake.

Based on the revealed laws, Pis'mennyi et al. [13] have proposed a method of intensification of heat exchange in banks of transversely finned tubes by way of convergent bending of the fins, which makes it possible to increase the heat transfer by 77% with a similar increase in the hydraulic resistance for optimum values of the angle and the degree of bending of the fins and the optimum characteristics of the tube bundle. The bent back sections of the fins reduce the flow in the stern part of the finned tube, directing intense secondary flows to the space behind the carrying cylinder. As a result, large sections of the stern surface are engaged in active heat exchange.

6. Intensification of Heat Exchange in the Case of Boiling. The above-mentioned methods of intensification of heat exchange in boiling and the results obtained have been considered in [1, 7, 14, 15]. We focus on the possibilities of improving the heat exchange in the case of boiling, first of all, for molten metals and other liquids, which do not provide a reliable wettability of heat-transfer surfaces due to different physical actions on the transition surface region [14]. The attractiveness of these methods lies mainly in the fact that their realization does not require changes in the design of steam generators.

The shape and size of the roughness of the heat-transfer surface determine the possibility of nucleation of a vapor bubble with critical radius R_{cr}. In analytical determination of stable radii of these nuclei, the fact that they must have a configuration that would enable them to fit into the cavities of the rough surface is taken into account. The rate of nucleation, i.e., the rate of formation of the critical nucleus, is determined by the changes in this configuration. The stable radius of the nucleus is

$$R_n = R_{cr} = \beta \frac{2\sigma v'' T_s}{T_w - T_s} . \tag{2}$$

It has been established based on the data on inhomogeneities of real surfaces and the microstructure of the interphase boundaries that the size of vapor-phase nuclei can range from $R_{n\,min}$ determined by Eq. (2) to $R_n = R_m$. Since the superheated liquid microlayer arises first in the cavities of the surface microstructure, such a real model of nucleation accounts for the three-dimensional structure of the phase interface. Within the limits of the model under consideration, I. Z. Kopp [14] has determined the region of possible realization of nucleate boiling in the coordinates p–R_m (pressure–cavity-mouth radius, which is equal to the radius of a stable nucleus). For optimization of the vapor-generation conditions, the parameters entering into Eq. (2) can be varied (change in p or R_m).

The investigations carried out have shown that the problem of intensification of heat exchange and optimization of the structure of the heat-transfer surface in the case of boiling must be investigated and solved with consideration for the concrete conditions of all the processes.

7. Intensification of Heat Exchange in Condensation. It seems likely that of all the investigated methods of intensification of heat exchange in the case of condensation, the best results are given by the above-mentioned tubes with ring turbulizers (see Fig. 1). As has been shown in [1], in the case of condensation of a vapor on the outer surface of horizontal tubes, the heat-exchange coefficient increases by a factor of 1.8–2.65; the deeper the grooves, the smaller their step, and the smaller the radius of curvature of the prominent part of the tubes, the larger its value. In the case of condensation of a vapor on the outer surface of vertical tubes, the intensification is lower: it is 1.3–1.5 for a stationary vapor and 1.9–2.8 for a moving vapor.

Since the heat transfer inside these tubes is also significantly intensified (by a factor of 2–2.5 in the experiments under consideration), their employment makes it possible to decrease the volume of condensers as a whole by a factor of 1.5–2, which is significantly better as compared to other methods. For example, the use of longitudinally knurled tubes or tubes with wire longitudinal finning makes it possible, at best, to increase the heat-exchange coefficient by a factor of 1.4–1.6, and the use of profile twisted tubes makes it possible to increase the heat-exchange coefficient by only 15%; the hydraulic resistance increases by 40–60% in this case.

It should be noted that in the case of condensation of a vapor on vertical tubes with annular grooves, the heat-exchange coefficient can significantly be increased due to the inclination of the tubes by 3–5^o or to the inclined arrangement of the ring diaphragms.

8. Scaling in Tubes with Turbulizers. The problem of decreasing the contaminations on heat-transfer surfaces is considered as very urgent. The use of cooling water containing salts determining its hardness leads to a contamination of the heat-transfer surfaces by these salts, which precipitate when the temperature of the cooling water increases. Interest in the study of the possibility of decreasing the scaling on heat-transfer surfaces using artificial turbulization has grown in the last few years. Tubes with ring turbulizers were found to be very efficient, which was confirmed by special experiments in which the outer and inner surfaces of tubes were in a flow of water of increased hardness (Fig. 1) with different parameters of turbulization [16]. The water velocity changed from 0.1 to 1.5 m/sec and its temperature from 50 to 90^{o}C; the duration of the experiments was up to 360 h. As a result of these experiments, the dependences of the thermal resistance of the scaling layer R_f outside and inside the tubes on the parameters of the turbulizers, the water velocity, and the time have been obtained.

Turbulizers decrease the scaling on both surfaces of the tubes by a factor of 3–5, and the time dependence of R_f is asymptotic in character; after 100–150 h the value of R_f becomes constant.

The higher the height of the diaphragms or the depth of the grooves and the smaller the step of their arrangement in tubes with turbulizers, the smaller the scaling in the tubes. It should be noted that in tubes with diaphragms, in which water of increased hardness (of up to 20 mg-eq/liter) flowed, the heat-exchange coefficient decreased by no more than 10% and the hydraulic resistance practically did not change after 100 h of operation. In a smooth tube, the heat-exchange coefficient decreased by 30% and the resistance increased by 25% in this time. Therefore, the efficiency of tubes with turbulizers increased in the presence of scaling. Whereas in the absence of scaling the heat-exchange coefficient increases by a factor of 1–2.5, in the presence of it this coefficient increases by a factor of 3.5–5 after 300 h of operation.

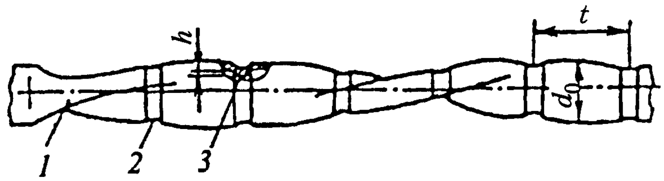

Fig. 4. Twisted tube with an oval profile and transverse grooves: 1) tube; 2) grooves; 3) diaphragms.

The experiments carried out have shown that since the scaling outside and inside the tubes with turbulizers is much smaller than that in smooth tubes, the use of these tubes makes it possible to provide a stable operation of heat-exchange apparatus without special measures for cleaning their surfaces.

9. High-Efficiency Heat-Exchange Apparatus. The experimental data presented in [1, 7, 15] have been confirmed by the testing of commercial heat-exchange apparatus. In apparatus with single-phase media, the heat power increased by 60–70%, and in condensers it increased by a factor of up to 2.2.

In [17, 18], the results of testing of new water-water heaters of nuclear power plants are presented. The use of tubes with ring turbulizers made it possible to increase the heat-exchange coefficient by a factor of up to 1.62. The use of twisted tubes increased the heat-exchange coefficient by a factor of 1.5–1.85. The greatest effect has been obtained with twisted tubes with transverse grooves knurled on their outer surface and bulges inside them (Fig. 4). These tubes made it possible to increase the heat exchange coefficient by a factor of up to 2.5 in a wide range of variation of the ratio between the heat-exchange coefficients outside and inside the tubes. The heat-exchange coefficient increases by a factor of 2.4–2.5 inside these tubes and by 30–50% outside them in the case of longitudinal flow around the bundles.

Conclusions. Unfortunately, because of the limited size of the article we cannot dwell on other fairly efficient methods of intensification of heat exchange (jet systems, porous and low-temperature coatings) and methods of intensification of heat exchange in plate heat exchangers.

However, the results presented show that there are well-investigated and technologically simply realizable fairly efficient methods of intensification of heat exchange in channels of heat-exchange apparatus. Realization of these methods makes it possible to decrease significantly the specific quantity of metal of modern heat-exchange apparatus and increase their operating reliability.

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NOTATION

D and D_{out}, inside and outside diameters of the tube; d, diameter of the ring diaphragms; d_{out}, diameter of the annular grooves; d_0, diameter of the screw core; d_{eq}, equivalent diameter of the channel; d_{hol}, diameter of the hollow; f, ratio of the surface of the hollows to the total heat-transfer surface; h, depth of the hollow; Nu, Nusselt number; p, pressure; R, radius; R_f, thermal resistance of the scaling layer; r, heat of evaporation; Re, Reynolds number; S, twist pitch; S_{out}, step of arrangement of tubes in the bundle; T_s and T_w, saturation temperature and temperature of the wall; t, step of arrangement of diaphragms in the tube; V, volume of the heat-exchange apparatus; v'', specific volume of the vapor; α, heat-exchange coefficient; β, coefficient; σ, surface tension. Subscripts: sm, smooth; n, nucleus; cr, critical; m, mouth; min, minimum; f, fouling; out, outside.

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