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IN-LINE AND CROSS-FLOW HELICAL TUBE HEAT EXCHANGERS

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Abstract—The paper presents the results of an experimental study of heat transfer and hydraulic resistance for the bundles of helical tubes in longitudinal and cross flow. The effectiveness of using these tubes in heat exchange equipment is analyzed. It is shown that helical tubes permit an appreciable increase in heat transfer and a substantial reduction of the heat exchanger dimensions.

NOMENCLATURE

- d, maximum size of tube profile;
- d_e, equivalent diameter;
- d_c, characteristic dimension for a tube bundle in cross flow;
- F, tube surface;
- f, cross-sectional area of tube;
- l, tube length;
- N, number of tubes in heat exchanger;
- r, radial coordinate reckoned from tube axis;
- s, lead of a helix;
- s_2, longitudinal spacing of tubes in a bundle;
- Δp, pressure losses;
- T_w, wall temperature;
- T_f, flow temperature;
- V, volume of heat exchanger;
- V_t, volume of tubes in a bundle;
- u_m, axial velocity in flow core;
- u_t, tangential velocity;
- u_r, radial velocity;
- u, mean-flow speed;
- u', longitudinal velocity fluctuation;
- v', w', transverse velocity fluctuations;
- y', coordinate reckoned from the heat exchanger shell wall in lateral direction;
- y_o, inner layer thickness at the shell wall;
- z, number of tubes in longitudinal direction;
- α, heat transfer coefficient;
- Π, tube perimeter;
- ψ, heat agent-based voidage of a bundle;
- ξ, hydraulic resistance coefficient;
- ρ, density;
- μ, dynamic viscosity;
- ε, effective turbulence intensity in a bundle;
- Fr_M, modified Froude number ($s^2/d_e \cdot d$);
- Nu, Re, Pr, Nusselt, Reynolds and Prandtl numbers.

Subscripts

- f, flow;
- w, wall;
- t, circular tube;
- m, maximum;
- av, averaged over bundle cells;
- st, straight;
- 1, on the inside of a tube;
- 2, on the outside of a tube.

INTRODUCTION

AT THE present time no branch of technology can be cited that would not use heat exchange equipment and devices where heat is transferred between agents flowing in channels of different geometries. Therefore, the problem of reducing their overall dimensions and mass, metal content and cost is a very topical one which can be solved by enhancing heat transfer in heat exchanger channels.

Among other means, the enhancement of heat transfer can be achieved by twisting the flow in channels between bundles of helical tubes of oval profile in longitudinal or cross flow [1]. This also intensifies the heat transfer of an agent flowing inside of the tubes [1].

A heat exchanger with a longitudinal flow past a bundle of helical tubes is shown diagramatically in Fig. 1. A specific feature of this apparatus is that helical tubes of oval profile are aligned so that they touch each other at the points of the maximum dimension of the oval [2]. The round ends of these tubes are secured in tube sheets.

A schematic diagram of the heat exchanger with cross flow past a bundle of helical tubes is given in Fig. 2. A specific feature of this apparatus is that helical tubes of oval profile are spaced in each transverse row so that they form slit channels along the tube bundle length with the maximum width equal to half the difference between the maximum and minimum dimensions of the oval, and touch only the tubes of the neighbouring rows [3].

The efficiency of these heat exchange apparatus can be estimated using the results of studies of heat transfer and hydraulic resistance of a heat transfer agent flowing inside the tubes and intertubular space [4, 5] as well as the data on turbulent flow structure and characteristics of transport in the space between helical tubes of bundles in longitudinal flow [6, 7].

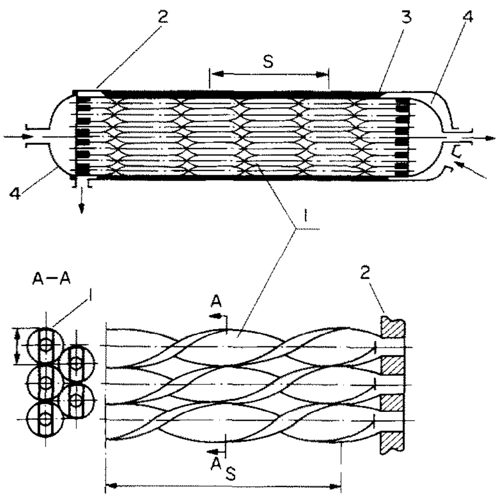

Fig. 1. Schematic diagram of a heat exchanger with longitudinal flow past a bundle of helical tubes: 1, tube; 2, tube sheets; 3, shell; 4, collectors for supply and discharge of heat exchanging media.

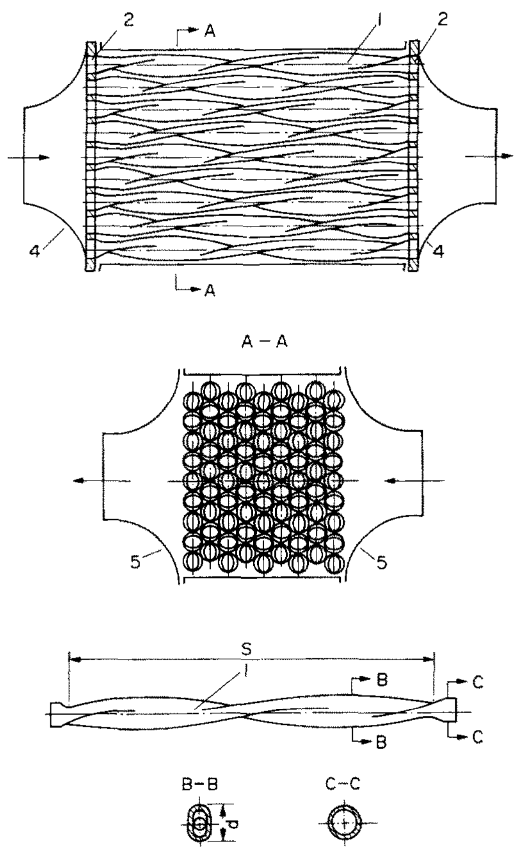

Fig. 2. Schematic diagram of a heat exchanger with a cross flow past a bundle of helical tubes: 1, tube; 2, tube sheets; 3, shell; 4, 5 collectors for supply and discharge of heat exchanging media.

The investigations carried out have shown that experimental data on heat transfer and hydraulic resistance in the space between helical tubes in longitudinal flow can be correlated by using, besides the Reynolds number

$$Re_{d_\mathcal{J}} = \frac{u d_e \rho_f}{\mu_f}, \tag{1}$$

the modified Froude number (for geometrically dissimilar tube bundles)

$$Fr_M = \frac{s^2}{d_e \cdot d}, \tag{2}$$

which characterizes the effect of centrifugal forces on the flow. Then the experimental data on heat transfer and hydraulic resistance coefficient at $Fr = 232$–2440 over the stretch of a stabilized turbulent flow can be described by the following relationships in dimensionless terms [1, 5]:

$$Nu_{d_\mathcal{J}} = 0.023\, Re_{d_\mathcal{J}}^{0.8}\, Pr_f^{0.4} \left[1 + \frac{3.6}{Fr_M^{0.357}}\right] \times \left(\frac{T_w}{T_f}\right)^{-0.55} \tag{3}$$

$$\xi = \frac{0.3164}{Re_{d_\mathcal{J}}^{0.25}} \left[1 + \frac{3.6}{Fr_M^{0.357}}\right]. \tag{4}$$

Starting at a certain value of $Fr_M (\approx 100)$, a substantial increase in the Nu number and still greater increase in the coefficient ξ [4] are observed. These results are given in Fig. 3. Figure 3 also presents the data on heat transfer for a transitional flow region at $Re = 3.10^3$, where heat transfer is even stronger than in the turbulent flow region. In the transitional flow region, the following relationship is valid [5]:

$$Nu_{d_\mathcal{J}} = 83.5\, Fr_M^{-1.2}\, Re_{d_\mathcal{J}}^{0.212 Fr_M^{0.194}} \times Pr_f^{0.4} \left(\frac{T_w}{T_f}\right)^{-0.55} \left[1 + \frac{3.6}{Fr_M^{0.357}}\right]. \tag{5}$$

The values of Nu and ξ in Fig. 3 for bundles of helical tubes are related to Nu_t and ξ_t which are determined by the relationships for circular tubes [8]:

$$Nu_t = 0.023\, Re_{d_\mathcal{J}}^{0.8}\, Pr_f^{0.4} \left(\frac{T_w}{T_f}\right)^{-0.55}, \tag{6}$$

$$\xi_t = \frac{0.3164}{Re_{d_\mathcal{J}}^{0.25}}. \tag{7}$$

As is seen from Fig. 3, Nu/Nu_t increases with a decrease of the Re number in the transitional flow region, while the ratio ξ/ξ_t remains practically intact up to $Fr_M \approx 100$. Thus, for bundles of helical tubes with $Fr_M = 232$, $Nu/Nu_t = \xi/\xi_t = 1.5$ at $Re = 10^4$, while at $Re = 3.10^3$, $Nu/Nu_t = 1.75$ and $\xi/\xi_t = 1.5$. Hence in a longitudinal flow past bundles of helical tubes there are regions (as to the Reynolds numbers) where heat transfer increases more rapidly than the hydraulic resistance, as compared with a circular tube.

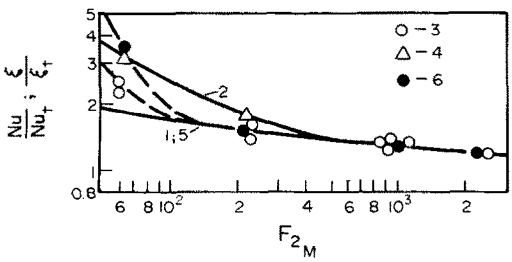

FIG. 3. The effect of Fr_M on the Nusselt number and hydraulic resistance coefficient for a longitudinal flow past a bundle of helical tubes related to similar parameters of straight tubes: 1, equation (3) at $Re = 10^4$; 2, equation (5) at $Re = 3 \cdot 10^3$; 3, experimental data on heat transfer at $Re = 10^4$; 4, the same at $Re = 3 \cdot 10^3$; 5, equation (4); 6, experimental data on the coefficient ξ.

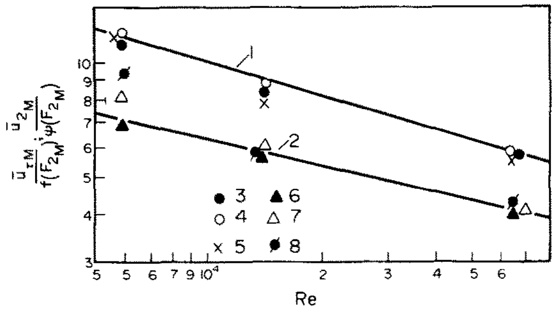

FIG. 4. Relative maximum values of tangential and radial velocities as functions of the Re and Fr_M numbers: 1, equation (9); 2, equation (10); 3, 6, experimental data for $Fr_M = 178$; 4, 7, experimental data for $Fr_M = 296$; 5, 8, experimental data for $Fr_M = 1187$.

The nature of heat transfer enhancement in the case of longitudinal flow past bundles of helical tubes has been elucidated by studying the structures of the flow and its transport properties. Distributions of the total velocity vector were determined with the aid of a pressure pipe, while distributions of the averaged velocities and of its longitudinal oscillating component, with the aid of a hot-wire anemometer [1, 6]. The distributions of the total velocity vector and its longitudinal component were found to obey the 1/7-power law, provided a certain local thickness of the wall layer is introduced, with the velocity profile being fuller at any point of the tube perimeter than in an equivalent circular channel. This indicates that flow twisting expands the flow core area and all the more markedly the smaller the number Fr_M. The tangential velocity within the external portion of the wall layer near the tube is distributed following the quasi-rigid rotation law

$$u_\tau r^{-1} = \text{const}, \tag{8}$$

while its distribution in the flow core depends on interaction of "vorticity filaments" the role of which is played by helical tubes. The radial velocity for the direction passing through the axes of neighbouring tubes and the cell they form is directed toward the tube wall in the region of the oval maximum size. On the windward side of the tube profile, the radial velocity is directed from the tube wall to the flow core. This type of motion leads to a continuous exchange by portions of liquid in the tube bundle cross-section and is one of the reasons for heat transfer enhancement in a bundle of helical tubes. Note that as the Re and Fr_M numbers decrease, the intensity of eddy motion in a tube bundle increases [6]. The maximum tangential and radial velocities related to the longitudinal one are determined, depending on the Re and Fr_M numbers, by the following empirical formulae (Fig. 4):

$$\bar{u}_{\tau m} = \left(\frac{u_\tau}{u_m}\right)_{av} = \frac{1.24}{Re^{0.275}}\left(1 + \frac{23.3}{Fr_M} + \frac{31700}{Fr_M^2}\right), \tag{9}$$

$$\bar{u}_{rm} = \left(\frac{u_{rm}}{u_m}\right)_{av} = \frac{0.44}{Re^{0.213}}\left(1 + \frac{95.7}{Fr_M} + \frac{18100}{Fr_M^2}\right). \tag{10}$$

The study of the longitudinal oscillating velocity in a bundle of helical tubes made it possible to discover another reason for heat transfer intensification, i.e. an additional flow agitation [6] caused by velocity gradient at the tube wall and velocity gradient in the flow core due to the contacts between neighbouring tubes. The flow past the places where the tubes come in contact also causes flow burbling. Therefore, the oscillating velocity in the flow core has a sinusoidal distribution along the tube spacing length [6]:

$$\frac{\sqrt{u'^2}}{u_m} = 0.075 + 0.025 \sin 2\pi \frac{y - y_0}{d} \tag{11}$$

with a period equal to the tubular grid pitch (d).

The intensity of turbulence in a bundle of helical tubes, on the average over the cross-section, can be described as a function of Re and Fr_M by

$$\left(\frac{\sqrt{u'^2}}{u_m}\right)_{av} = \frac{7.2}{Re^{0.155 + 40.57 Fr_M^{-1} + 1700 Fr_M^{-2}}} \times \left[1 + \frac{Fr_M - 178}{7.5(19.5 - 0.135 Fr_M)}\right] \tag{12}$$

at $Fr_M = 178 - 1187$ and $Re = 6 \cdot 10^3 - 1.1 \cdot 10^5$. It follows from (12) that the smaller the numbers Re and Fr_M, the stronger the turbulence. This, in particular, explains a substantial increase of heat transfer exactly in the transitional flow region and in tube bundles with a small helix lead.

The dependence (12) is given in Fig. 5 where it is compared with the data of [7] on effective turbulence intensity measured by the method of diffusion from a point source. The effective turbulence intensity is determined on condition that

$$\varepsilon = \frac{\sqrt{u'^2}}{u} = \frac{\sqrt{v'^2}}{u} = \frac{\sqrt{w'^2}}{u} \tag{13}$$

differs from the relative longitudinal oscillating velocity $(\sqrt{u'^2}/u_m)_{av}$ by being virtually independent of the Re number and associated with the Fr_M number within the range $Fr_M = 314$–1530 through the formula

$$\varepsilon = 0.044(1 + 8.1\, Fr_M^{-0.278}). \tag{14}$$

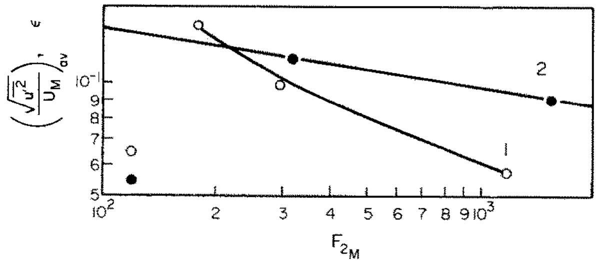

FIG. 5. Dependence of the effective turbulence intensity and relative longitudinal oscillating velocity upon Fr_M at $Re \approx 10^4$: 1, equation (12); 2, equation (14); 3, experimental data on oscillating velocity; 4, experimental data on ε [7].

With an increasing Fr number ε decreases (Fig. 5) more slowly than $(\sqrt{u'^2}/u_m)_{av}$. This is attributed to the account, besides turbulent diffusion, of the organized convective transport along helical tube channels and secondary flow circulation when determining ε. Within $Fr_M = 178$–296 at $Re \approx 10^4$ the values of $(\sqrt{u'^2}/u_m)_{av}$ are closer to the values of ε determined from equation (14), i.e. with a decreasing Fr_M the fraction contributed to ε by turbulent diffusion increases.

The results of the study of heat conduction and hydraulic resistance in bundles with a cross flow past helical tubes of oval profile carried out at $s/d = 12.2$ and $Re = 10^3$–$3 \cdot 10^4$ are given in [3]. A packed bundle of 10×10 tube rows was installed in a 180×133 mm channel. The length of the tubes amounted to 1.25 of the helix lead. The method of local modelling was resorted to in experiments with one tube in the sixth row of the bundle being heated. Electrical calorimetric measurements were employed. All the experiments were run under steady state conditions. For processing the data the flow velocity $u = u_o/\psi$ was taken as the reference one, where u_o is the upstream velocity, ψ is the heat agent-based voidage of a bundle ($\psi = 1 - \Sigma V_t/V$).

The characteristic dimension was taken to be

$$d_c = 4\frac{\psi}{1-\psi}\frac{V_t}{F_t}\frac{\Pi_t/2}{s_2}, \qquad (15)$$

where V_t is the tube volume, F_t the tube surface, Π_t the tube perimeter, s_2 the longitudinal pitch of the bundle. By substituting into (15) $V_t = f_t \cdot l$, $F_t = \Pi_t \cdot l$, where f_t is the cross-sectional area of the tube, l is the tube length, we obtain

$$d_c = 4\frac{\psi}{1-\psi}\cdot\frac{f_t}{2s_2}. \qquad (16)$$

The resistance was determined by measuring the pressure drop over the bundle and heat transfer, by measuring upstream temperature, tube surface temperature and heat generation over the length of the helix lead.

The surface temperature was measured at 16 cross-sections over the length of the helix lead and at 4 generatrices at each cross-section. The mean surface temperature was determined as the arithmetic mean of all the measurements under conditions studied. Radiative heat transfer was accounted for in determination of convective heat generation although the ratio between the surface temperatures of the tube studied and of neighbouring tubes did not exceed 1.1.

The results of the study of heat transfer and hydraulic resistance of a bundle of helical tubes in a cross-flow are presented in Fig. 6. The heat transfer of the bundle over the whole range of Re numbers studied, $Re = 10^3 - 3 \cdot 10^4$, is described by

$$Nu = 0.823\,Re^{0.57}, \qquad (17)$$

while the hydraulic resistance coefficient

$$\xi = \frac{\Delta p}{z\frac{\rho u^2}{2}}, \qquad (18)$$

(where Δp is the pressure drop over the bundle, z is the number of tube rows) does not depend on Re (Fig. 6) within $Re = 10^3$–$5 \cdot 10^3$ and is equal to $\xi = 3.94$. At $Re > 5 \cdot 10^3$, the coefficient ξ decreases with an increasing Re according to

$$\xi = 17.3\,Re^{-0.175} \qquad (19)$$

In equations (17) and (19), the mean flow temperature is taken as the reference one.

The results of the study of heat conduction and

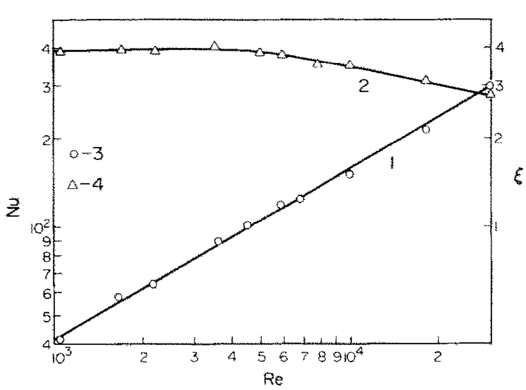

FIG. 6. The effect of Re on the Nusselt number and hydraulic resistance coefficient for a heat exchanger with a cross flow past a bundle of helical tubes: 1, equation (17); 2, equation (19); 3, experimental data on heat transfer; 4, experimental data on ξ.

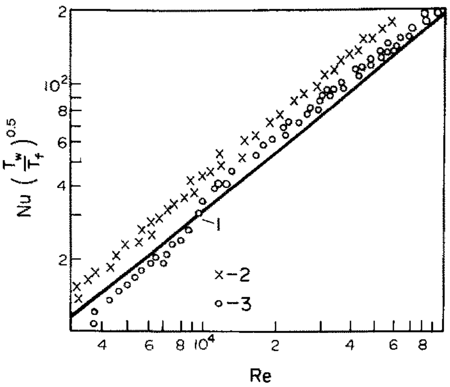

FIG. 7. Heat transfer for a flow inside of helical tubes: 1, straight tube; 2, $s/d = 6.21$; 3, $s/d = 16.69$.

hydraulic resistance of a heat transfer agent flowing inside of shaped tubes are presented in detail in [1] and are described by the following relationships (Figs. 7, 8):

$$Nu = 0.019\, Re^{0.8}\,[1 + 0.547/(s/d)^{0.83}], \quad (20)$$

$$\xi = 0.316\,[1 + 3.27\,(s/d)^{-0.87}]\, Re^{-0.25} \quad (21)$$

for $s/d = 6.2$–16.7; $Re = 6 \cdot 10^3$–10^5; $T_w/T_f = 1$–1.55. The numbers Re and Nu in (20) and (21) are based on the equivalent diameter and mean flow temperature.

As is seen from Figs. 7 and 8, flow twisting substantially intensifies heat transfer inside of helical tubes at $s/d = 6.2$. A 1.4-fold increase in heat transfer is accompanied by about a 1.7-fold increase of the hydraulic resistance as compared to a straight smooth tube.

The efficiency of the heat-exchange apparatus suggested has been estimated by the method described in detail in [9, 10] and which basically is as follows. The heat exchange devices suggested were compared with those made of circular tubes but having the same spacing of tubes in a bundle and the same perimeter Π. Comparison is carried out at the same flow rates of heat transfer agents, heat and pumping powers. It is assumed that the heat transfer coefficient on the side compared is lower than that on the other side.

Heat powers of the exchangers compared were equal to $Q = \alpha \Delta t \Pi l N$ and $Q_{st} = \alpha_{st} \Delta t_{st} \Pi_{st} l_{st} N_{st}$, where α is the heat transfer coefficient, Δt temperature difference, l the tube length, N the number of tubes in a heat exchanger. The temperature differences in both heat exchangers are the same ($\Delta t = \Delta t_{st}$), while the pressure losses equal to

$$\Delta p = \frac{l}{d_e} \cdot \frac{\rho u^2}{2}$$

and

$$\Delta p_{st} = \xi_{st} \frac{l_{st}}{d_{est}} \cdot \frac{\rho_{st} u_{st}^2}{2},$$

in which u is the mean heat transfer agent velocity, ρ density, ξ the hydraulic resistance coefficient, d_e the equivalent diameter, with $\rho = \rho_{st}$. For straight channels $Nu_{st} = c_1 Re^n$ and $\xi_{st} = c_2 Re^m$. For heat exchangers with helical tubes an increase in heat transfer and hydraulic resistance is accounted for by the ratios $(Nu/Nu_{st})_{Re}$ and $(\xi/\xi_{st})_{Re}$, which are the functions of Re for the geometry of the channels considered. Therefore

$$\frac{\alpha}{\alpha_{st}} = \left(\frac{Nu}{Nu_{st}}\right)_{Re} \left(\frac{Re}{Re_{st}}\right)^n \frac{d_{est}}{d_e}, \quad (22)$$

$$\frac{\xi}{\xi_{st}} = \left(\frac{\xi}{\xi_{st}}\right)_{Re} \left(\frac{Re}{Re_{st}}\right)^m, \quad (23)$$

where the subscript "Re" means that the ratios Nu/Nu_{st} and ξ/ξ_{st} are taken at the same Reynolds numbers in the heat exchangers with straight and helical tubes and which is equal in the case considered to the Reynolds number in the latter heat exchanger.

The above relations allow one to obtain the ratios between the number of tubes, lengths and volumes of the exchangers compared

$$\frac{N}{N_{st}} = \left[\frac{(\xi/\xi_{st})_{Re}}{(Nu/Nu_{st})_{Re}(d_e/d_{est})^2}\right]^{1/3-n+m}, \quad (24)$$

$$\frac{l}{l_{st}} = \left[\frac{(d_e/d_{est})^{5-3n+m}}{(Nu/Nu_{st})_{Re}^{2+m}\,(\xi/\xi_{st})_{Re}^{1-n}}\right]^{1/3-n+m}, \quad (25)$$

$$\frac{V}{V_{st}} = \left[\frac{(\xi/\xi_{st})^n (d_{est})^{3-3n+m}}{(Nu/Nu_{st})^{3+m}}\right]^{1/3-n+m} \quad (26)$$

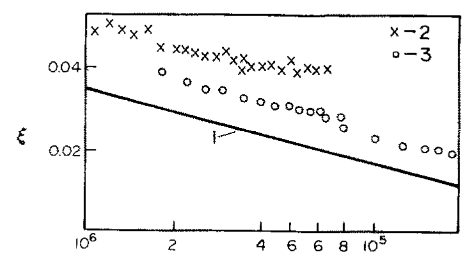

FIG. 8. Hydraulic resistance for a flow inside of helical tubes: 1, straight tube; 2, $s/d = 6.21$; 3, $s/d = 16.69$.

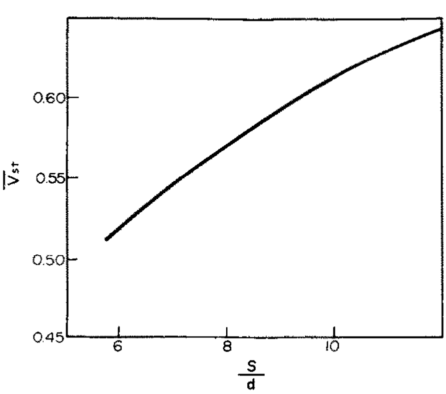

FIG. 9. Dependence of the volume ratio of heat exchangers V/V_{st} upon longitudinal flow in intertubular space on the helix lead at $\alpha_2 \ll \alpha_1$.

For a turbulent flow $n = 0.8$ and $m = -0.2$ and then

$$\frac{V}{V_{st}} = \frac{(\xi/\xi_{st})^{0.4}(d_{est})^{0.2}}{(Nu/Nu_{st})^{1.4}}. \tag{27}$$

Figure 9 shows the dependence of the relative volumes of heat exchange apparatus, V/V_{st}, on s/d for the case, when the heat transfer coefficients outside of the tubes are much smaller than those within. As seen the helical tubes allow a 1.5–2-fold reduction of the heat exchanger volume. On the other hand, if the heat transfer coefficient outside of the tubes is much higher than that inside, the helical tubes make it possible to reduce the heat exchanger volume by 20–25%.

Once the ratios $(V/V_{st})_1$ and $(V/V_{st})_2$ for the both sides of a heat exchanger are known, then, by neglecting thermal resistance of tube walls, one can obtain for a general case the ratio of volumes of heat exchangers compared

$$\frac{V}{V_{st}} = \left(\frac{V}{V_{st}}\right)_1 \left[\frac{1 + \alpha_1/\alpha_2}{1 + \dfrac{(V/V_{st})_1}{(V/V_{st})_2} \cdot \dfrac{\alpha_1}{\alpha_2}}\right] \tag{28}$$

where α_1 and α_2 are the heat transfer coefficients for the inner and outer surfaces of helical tubes and

$$\frac{\alpha_1}{\alpha_2} = \frac{\alpha_1}{\alpha_{1\,st}} \cdot \frac{\alpha_{2\,st}}{\alpha_2} \cdot \frac{\alpha_{1\,st}}{\alpha_{2\,st}} = \left(\frac{Nu}{Nu_{st}}\right)_1 \left(\frac{Nu_{st}}{Nu}\right)_2 \cdot \frac{\alpha_{1\,st}}{\alpha_{2\,st}}. \tag{29}$$

Since the ratios $(Nu/Nu_{st})_1$ and $(Nu/Nu_{st})_2$ are known for the surfaces compared, then, by assigning the ratio of the heat transfer coefficients in a heat exchanger with straight tubes $\alpha_{1\,st}/\alpha_{2\,st}$, one can obtain the value of V/V_{st} which is intermediate between $(V/V_{st})_1$ and $(V/V_{st})_2$.

Comparison between the heat exchangers with a cross flow in the intertubular space [10] is based on the use of the relations

$$\Delta p = \xi z \frac{\rho u^2}{2} \quad \text{and} \quad \Delta p_{st} = \xi_{st} z_{st} \frac{\rho_{st} u_{st}^2}{2}, \tag{30}$$

(where u is the reference velocity, ξ is the resistance coefficient, z the number of tubes in the lateral direction) and of equation (15) for the characteristic dimension d_c. Then

$$\frac{\alpha}{\alpha_{st}} = \left(\frac{Nu}{Nu_{st}}\right)_{Re} \left(\frac{Re}{Re_{st}}\right)^n \frac{d_{cst}}{d_c}, \tag{31}$$

while ξ/ξ_{st} is determined by equation (23).

The ratio of volumes of heat exchangers for $\alpha \gg \alpha_2$ is

$$\frac{V}{V_{st}} = \left\{\frac{(\xi/\xi_{st})^n_{Re}\,(\psi/\psi_{st})^{3+m-2n}\,[(1-\psi)/(1-\psi_{st})]^{2m+2n+mn}}{(Nu/Nu_{st})^{m+3}}\right\}^{1/3+m-n} \tag{32}$$

where ψ is the heat agent-based voidage of a bundle.

The results of calculation of V/V_{st} from equation (32) are presented in Fig. 10. This figure also contains the curves Nu/Nu_{st} and ξ/ξ_{st} obtained at different Re's by comparing equations (17) and (19) with the data of [11] for bundles of straight tubes. As is seen from this figure the replacement of straight tubes by helical ones

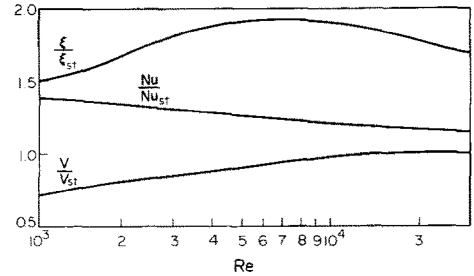

FIG. 10. Dependence of the volume ratio of heat exchangers V/V_{st} upon cross flow in intertubular space on the Reynolds number in this space at $\alpha_2 \ll \alpha_1$.

leads to a 1.2–1.4-fold in increase in heat transfer with a 1.5–1.8-fold increase in hydraulic resistance which allows a reduction in the heat exchanger volume of up to 30%.

CONCLUDING REMARKS

(1) The schemes of heat exchangers are suggested which allow a substantial increase in the heat transfer rate due to flow twisting in the channels of complex geometry.

(2) Experimental relationships are derived for calculation of heat transfer and hydraulic resistance coefficients for heat transfer agent flow inside and outside of the tubes of in-line and cross flow heat exchangers with bundles of helical tubes of oval profile. These relationships have been employed to estimate the efficiency of the apparatus suggested.

(3) It is shown that the use of helical tubes in heat exchangers of the designs considered increases the heat transfer rate and makes it possible to manufacture more compact heat exchangers as compared to those with straight tubes, both for longitudinal and cross flow past tube bundles.

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ECHANGEURS DE CHALEUR A TUBES HELICOÏDAUX EN LIGNE OU A COURANT CROISE

Résumé—On présente les résultats d'une étude expérimentale sur le transfert thermique et la perte de charge pour des faisceaux de tubes hélicoïdaux dans un écoulement longitudinal ou croisé. L'efficacité de ces tubes dans un échangeur de chaleur est analysée. On montre que les tubes hélicoïdaux permettent un accroissement appréciable du transfert thermique et une réduction sensible des dimensions de l'échangeur.

SPIRALROHRWÄRMEAUSTAUSCHER MIT LÄNGS- UND KREUZSTROMFÜHRUNG

Zusammenfassung—In dem Aufsatz werden die Ergebnisse einer experimentellen Studie zum Wärmeübergang und Druckverlust von Spiralrohrbündeln bei Längs- und Kreuzstromführung mitgeteilt. Die Effektivität bei der Verwendung dieser Rohre in Wärmeaustauschern wird untersucht. Es zeigt sich, daß Spiralrohre eine erhebliche Zunahme des Wärmeübergangs und eine wesentliche Verringerung der Abmessungen der Wärmeaustauscher ermöglichen.

ТЕПЛООБМЕННИКИ С ПРОДОЛЬНЫМ И ПОПЕРЕЧНЫМ ОБТЕКАНИЕМ ПУЧКОВ ВИНТООБРАЗНО ЗАКРУЧЕННЫХ ТРУБ

Аннотация — Представлены результаты экспериментального исследования теплообмена и гидравлического сопротивления пучков винтообразно закрученных труб с продольным и поперечным обтеканием. Проведен анализ эффективности использования данных труб в теплообменных аппаратах. Показано, что витые трубы позволяют существенно интенсифицировать теплообмен и уменьшить размеры теплообменных аппаратов.