JOURNAL OF ENGINEERING PHYSICS

INVESTIGATION OF THE INTENSIFICATION OF HEAT TRANSFER IN A DENSE TUBE BUNDLE WASHED BY A LONGITUDINAL AIR FLOW

G. A. Dreitser and E. K. Kalinin

Inzhenerno-Fizicheskii Zhurnal, Vol. 15, No. 3, pp. 408–415, 1968

UDC 536.244

Experiments have been performed to investigate the intensification of heating and cooling obtained by periodically grooving the tubes of a staggered bundle with S/D = 1.2 washed by a longitudinal air flow in the range Re = $2 \cdot 10^3$–10^5.

Tubular heat exchangers with longitudinal flow through the intertube space are widely used in a number of branches of technology. To develop more compact exchangers it is necessary not only to intensify heat transfer in the intertube space by creating turbulence in various ways but also to use dense bundles of tubes (relative pitch S/D ≤ 1.2).

The intensification of heat transfer under these conditions was investigated in [1–5]. All these studies were concerned with staggered bundles with a large relative pitch, and the intensification of heat transfer was ensured by using external finning of variable height and pitch. As a rule, the two above-mentioned methods of reducing the size of tubular heat exchangers are mutually exclusive. The use of external finning increases the dimensions of the tubes and prevents the development of dense bundles, while the use of dense bundles interferes with the finning and makes it ineffective owing to the large increase in the hydraulic resistance of the bundle. Accordingly, a method is required that would not increase the outer diameter of the tubes, i.e., would permit the use of dense bundles. This requirement is satisfied by tubes with transverse knurled ring grooves. The depressions formed in the outer surface of the tubes create turbulence in the boundary layer and intensify heat treasfer. This method has the following principal advantages: it is suitable for dense tube bundles since it does not increase the outside diameter of the tubes; the diaphragms produced inside the tube by the forming process considerably intensify heat transfer inside the tube [6, 7]; the technology is relatively simple; the process is applicable at large specific heat fluxes; it does not require changes in the existing technology of tubular heat-exchanger assembly. The All-Union Scientific Research and Planning Design Institute of Metallurgical Machine Building has developed the mass production technology for tubes with ring grooves. This technology requires only standard metal-forming equipment.

We have investigated the intensification of heat transfer in longitudinal-flow, staggered tube bundles with S/D = 1.2 for several variants of the pitch and depth of the ring grooves.

The investigation was carried out on two experimental devices; test section No. 1, in which air was heated in the intertube space and test section No. 2, in which it was cooled.

Test section No. 1 (Fig. 1) consisted of 19 tubes of diameter 0.65 ± 0.01 mm and wall thickness D = 11 ± 0.01 mm assembled into a bundle by means of two brass tube plates. The distance between these plates was 1.5 m.

The air was heated by passing alternating current directly through the bundle. The tubes were capable of free displacement relative to the upper tube plate to compensate the difference in the thermal expansion of the housing and the tubes. The maximum nonuniformity of heat release along the tubes did not exceed 1.5–2%.

The heat transfer and resistance coefficients were measured on a predetermined stabilized-flow section of length l_0 = 800 mm. This section was 350 mm from the inlet ($l/d_e \cong 50$). Therefore, the data presented describe the mean heat transfer in the longitudinal-flow bundles or the local heat transfer at $l/d_e > 50$. In the heater we determined the mean heat-transfer coefficient for all the tubes of the bundle, which made it possible to disregard the nonuniformity of the heat release along the tubes and the distribution of heat-transfer agent over the cross section. The wall temperature of the heater tubes was measured in all 19 tubes at the beginning, middle, and end of the test section. The chromel-copel thermocouples were made of wire 0.2 mm in diameter and welded directly into the tube wall by argon-arc welding. The leads were led out inside the tubes through the ends of the heater. The flow temperature was measured at the inlet and outlet of the test section.

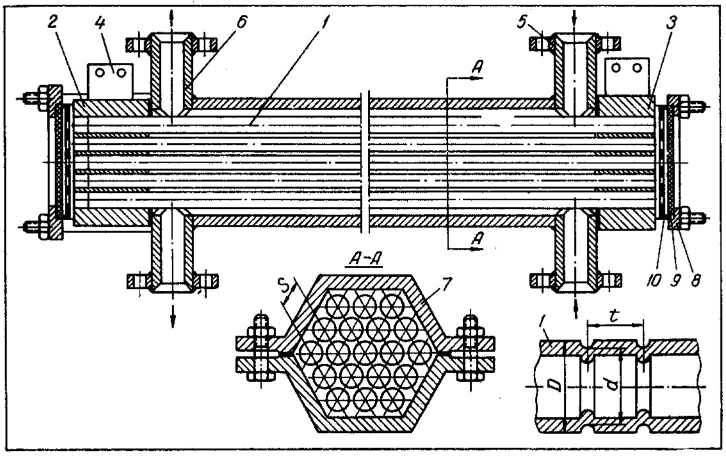

Fig. 1. Experimental heater (test section No. 1): 1) tube with ring grooves; 2) upper tube plate; 3) lower tube plate; 4) power terminal; 5) inlet connection; 6) outlet connection; 7) housing; 8) clamping flange; 9) cover; 10) fiberglass insulators for thermocouple leads.

The amount of heat released on the test section was determined from the change in the enthalpy of the air. It deviated from the measured electrical power by not more than 5–10%.

The experimental cooler (test section No. 2) was similar in construction to the heater. The cooling water was directed through the tubes in a counterflow. The thermocouples for measuring the tube wall temperature were located on the outside of the tubes and led out through a joint in the housing.

To measure the flow temperature in the intertube space, a moving longitudinal thermocouple was installed on the axis of the central cell. The hot junction of the thermocouples was obtained by butt welding chromel and copel wires. The leads were led out through openings in the tube plates.

In test section No. 1 the air flowed upward; in test section No. 2 it flowed downward. The d_e determined from the total wetted perimeter varied in the range 6.10–6.23 mm. The heater parameters varied within the following limits: $\bar{q}$ = 355–24 600 W/m^2; $\bar{t}_f$ = 62–140° C; $\bar{t}_w$ = 91–218° C; $\bar{T}_w/\bar{T}_f$ = 1.046–1.25; Reynolds number, determined with respect to d_e and t_f, $Re_f = 2 \cdot 10^3 – 8 \cdot 10^4$. In the cooler $\bar{q}$ = 400–49 000 W/m^2; $\bar{t}_f$ = 36.7–99.5° C; $\bar{t}_w$ = 10.8–46.8° C; $\bar{T}_w/\bar{T}_f$ = 0.88–0.99, $Re_f = 10^3 – 1.18 \cdot 10^5$.

For both test sections, the maximum error in determining the heat-transfer coefficients and resistance c coefficinets did not exceed 10÷15% and 10%, respectively.

We investigated the temperature profile around the perimeter of the tubes. No temperature nonuniformity around the perimeter was observed.

The investigation of heat-transfer intensification was preceded by a study of the heat transfer and hydraulic resistance of a bundle of smooth tubes [8, 9].

The correspondence of the experimental heat-transfer data obtained on test sections Nos. 1 and 2 by two different methods (discrepancy 2%) and the satisfactory agreement between these data and the results of other investigation in the turbulent region for staggered bundles with S/D = 1.2 [10, 11] indicate the correctness of the method selected and make it possible to employ it for investigating the intensification of heat transfer.

The intensification of heat transfer was investigated on six different tube bundles. In the heater we investigated bundles with d/D = 0.97 and 0.9 at t/D = 0.454 and with d/D = 0.90 at t/D = 0.909; and in the cooler we investigated bundles with d/D = 0.95 and 0.93 at t/D = 0.454 and with d/D = 0.93 at t/D = 0.909. The grooves in the tubes were about 1 mm wide.

In determining the equivalent diameter, the wetted perimeter, and the flow cross section of the bundles, we disregarded the presence of the grooves. The heat-transfer coefficient and the heat flux were related to the surface of the smooth tube.

In the laminar flow region grooving does not affect the heat transfer or hydraulic resistance (Fig. 2). As Re increases, the resistance and then the heat transfer of the grooved tubes become higher than those of the smooth tubes. At Re > Re_l the ratio Nu/Nu_{sm} (Nu_{sm} is determined at the same Re) begins to increase with increase in Re; however, beyond a certain number Re_2 the increase in heat transfer as compared with the smooth-tube bundle becomes stabilized and with further increase in Re there is no change in Nu/Nu_{sm}. Re_1 and Re_2 increase as the depth of the grooves decreases and are almost independent of the pitch. As Re increases, the difference in the resistance coefficients ξ and ξ_{sm} at first increases sharply, then more smoothly. Nu/Nu_{sm} and ξ/ξ_{sm} increase with increase in the depth (Fig. 3) and decrease in the pitch of the grooves.

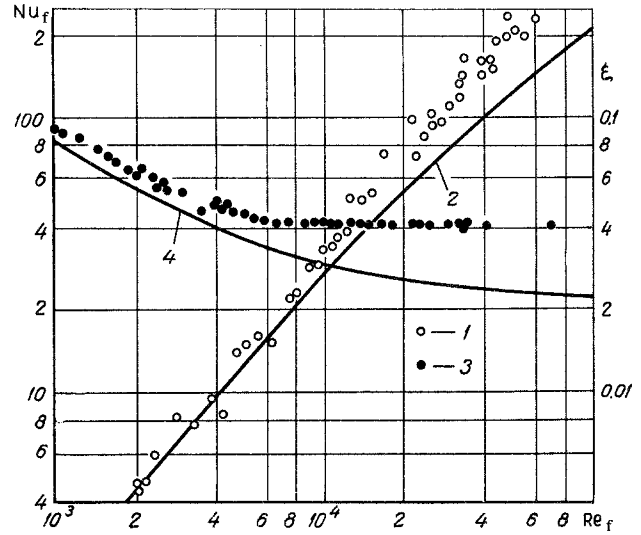

Fig. 2. Heat transfer and resistance coefficients in a bundle of tubes with S/D = 1.2: 1) heat transfer in a bundle with d/D = 0.90; t/D = 0.454; 2) heat transfer in a smooth-tube bundle; 3 and 4) resistance coefficients of a bundle with d/D = 0.90 and t/D = 0.454 and of a smooth-tube bundle, respectively.

As d/D decreases, the dependence of ξ on Re becomes flatter (Fig. 2), since the fraction of the pressure losses associated with the agitation of the flow in the grooves increases. These losses, equivalent to the local resistances, depend only slightly on Re, and, therefore, as they increase, the dependence of the total resistance coefficient on Re decreases.

The experimental data obtained made it possible to interpret the heat-transfer intensification mechanism as follows. In laminar flow, the grooves are filled with stagnant gas and do not affect the resistance and heat transfer. As Re increases, turbulent slugs appear in the wide portions of the cells forming the intertube space, and periodic bubbling develops in the grooves. These effects lead to the formation of a thin turbulent boundary layer and, hence, to an increase in heat transfer and hydraulic resistance. Since the turbulent slugs intensify heat transfer only on a small part of the tube perimeter adjacent to the wide parts of the cells and at the same time embrace the main core of the flow, the increase in hydraulic resistance leads the increase in heat transfer.

With further increase in Re, the thickness of the laminar sublayer decreases and stable eddies, whose strength continues to grow, are formed in the grooves. As Re increases, the eddies propagating along the grooves, gradually penetrate into the narrow parts of the intertube space and embrace the entire perimeter of the tube. In this case, the heat-transfer coefficient increases both because of the increase in the strength of the eddy formed in the groove and because of its propagation over the entire groove.

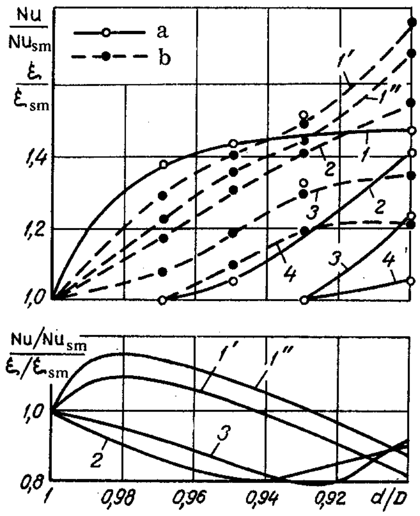

Fig. 3. The ratios Nu/Nu_{sm} ξ/ξ_{sm} and $(Nu/Nu_{sm})/(\xi/\xi_{sm})$ as functions of the depth of the grooving d/D at a pitch t/D > = 0.454: a) Nu/Nu_{sm}; b) ξ/ξ_{sm}; 1) Re = $= 4 \cdot 10^4 - 10^5$; 1') Re = 10^5; 1") Re = $4 \cdot 10^4$; 2) Re = $2 \cdot 10^4$; 3) Re = 10^4; 4) Re = $6 \cdot 10^3$.

At the upper edge of the eddy, where the velocity gradient is maximum, random perturbations of the flow grow into large fluctuations of the axial velocity component. Owing to the work done by the pressure fluctuations, the energy of the longitudinal fluctuations is transmitted to the transverse fluctuations. The large fluctuations are transported by the main stream along the walls, imparting their energy to the smaller fluctuations, until they are completely dissipated. The presence of these fluctuations at the wall also leads to an increase in turbulent heat conduction in the boundary layer and to intensified heat transfer.

As Re increases, the velocity profile become fuller, while the upper part of the eddy remains approximately at the same level. Therefore, even though the turbulent jet separating from the upper part of the eddy becomes more powerful, for the most part it does not enter the region where the principal temperature head is being consumed. Thus, the effect of heat transfer intensification due to the creation of turbulence ceases to grow. The hydraulic resistance of the bundle continues to increase, since the strength of the eddy is increasing.

The experimental data have been analyzed in relation to d_e and $\bar{t}_f$. For t/D = 0.454 and $0.9 \leq d/D \leq 0.97$ the Re numbers at which the Nu/Nu_{sm} begin to increase and become stabilized are expressed by the following relations:

$$Re_1 = \left(30\,\frac{d}{D} - 26.4\right)\cdot 10^4; \tag{1}$$

$$Re_2 = \left(16.8\,\frac{d}{D} - 12.1\right)\cdot 10^4. \tag{2}$$

For t/D = 0.454 and $0.9 \leq d/D \leq 1$ the experimental data on the heat transfer and hydraulic resistance are generalized by the relations:

for $Re \leqslant Re_1$

$$Nu/Nu_{sm} = 1; \tag{3}$$

for $Re_1 \leqslant Re \leqslant Re_2$

$$\frac{Nu}{Nu_{sm}} = 1 + 0.465\,\frac{\lg Re - \lg Re_1}{\lg Re_2 - \lg Re_1} \times$$

$$\times \left\{1 - \exp\left[-33.7\left(1 - \frac{d}{D}\right)\right]\right\}; \quad (4)$$

for $\mathrm{Re} > \mathrm{Re}_2$

$$\frac{\mathrm{Nu}}{\mathrm{Nu}_{sm}} = 1.465 - 0.465 \exp\left[-33,7\left(1 - \frac{d}{D}\right)\right]; \quad (5)$$

for $\mathrm{Re} \leqslant 3.1 \cdot 10^3$

$$\xi/\xi_{sm} = 1; \quad (6)$$

for

$$3.1 \cdot 10^3 < \mathrm{Re} < 2 \cdot 10^4$$

$$\xi/\xi_{sm} = 1 + 6.72(\lg \mathrm{Re} - 3.5) \times$$

$$\times \left(1 - \frac{d}{D}\right) - 0.035 \sin\left[20\left(\frac{d}{D} - 0.95\right)\pi\right]; \quad (7)$$

for $2 \cdot 10^4 \leqslant \mathrm{Re} < 10^5$

$$\xi/\xi_{sm} = 1 + 2.86(\lg \mathrm{Re} - 2.27)\left(1 - \frac{d}{D}\right) +$$

$$+ 0.09(\lg \mathrm{Re} - 4.3) \sin\left[20\left(\frac{d}{D} - 0.95\right)\pi\right]. \quad (8)$$

In (3)–(8), Nu_{sm} and ξ_{sm} were determined from [8,9]. For $\mathrm{Re} = 4 \cdot 10^4$ the ratios $\mathrm{Nu}/\mathrm{Nu}_{sm}$ and ξ/ξ_{sm} are 1.40 and 1.35, respectively, for $d/D = 0.95$, and 1.44 and 1.67 for $d/D = 0.9$ (at $t/D = 0.454$). As the pitch of the grooving increases, $\mathrm{Nu}/\mathrm{Nu}_{sm}$ and ξ/ξ_{sm} decrease. For $t/D = 0.909$ and $d/D = 0.90$ these ratios are equal to 1.3 and 1.23 at $\mathrm{Re} = 10^5$.

As d/D decreases, the effect of the temperature factor on the hydraulic resistance when the air is heated decreases, which is attributable to the fact that the nonisothermicity of the flow chiefly affects the viscosity component of the pressure losses, the relative magnitude of which decreases with the roughness. At $d/D = 0.9$ the correction which takes into account the effect of the temperature factor on the resistance coefficient $\xi/\xi_0 = 1$ (in the range of T_w/T_f investigated), under the condition $1 \geq d/D \geq 0.9$ and $t/D = 0.454$ is expressed by the relation

$$\frac{\xi}{\xi_0} = \left(\frac{\xi}{\xi_0}\right)_{sm} + 10\left(1 - \frac{d}{D}\right)\left[1 - \left(\frac{\xi}{\xi_0}\right)_{sm}\right], \quad (9)$$

where $(\xi/\xi)_{sm}$ is found from [8].

When the air cools, the effect of the temperature factor on the hydraulic resistance of bundles of grooved tubes is as insignificant as for a smooth-tube bundle. To estimate the effectiveness of the investigated method of heat-transfer intensification we made calculations for the tubular cooler ($S/D = 1.2$) of a closed-cycle gas turbine plant with a longitudinal gas flow in the intertube space. We examined coolers with smooth tubes and with the investigated variants of grooved tubes. The coolers were calculated for the same thermal capacity, heat-transfer agent flow rates and hydraulic resistance on the gas side. We also took into account the intensification of heat transfer inside the tubes on the water side. In accordance with [7], at $D/d = 0.97$ and $t/D = 0.454$ the heat transfer inside the tubes increases by a factor of 1.55 with an identical increase in resistance. With increase in the depth of the grooving, $\mathrm{Nu}/\mathrm{Nu}_{sm}$ and ξ/ξ_{sm} also increase; however, the increase in resistance leads the increase in heat transfer. Thus, for $d/D = 0.9$, the ratio $\mathrm{Nu}/\mathrm{Nu}_{sm} = 2.29$ and $\xi/\xi_{sm} = 6.5$.

In Fig. 4 the ratios of the volumes V/V_{sm}, lengths L/L_{sm}, and cross-sectional areas F/F_{sm} of the heat exchangers are presented as a function of d/D for $t/D = 0.454$. The volume of the cooler and, hence, its weight, first fall as the depth of the grooving increases, then grow somewhat. At $d/D = 0.93$ the volume of the cooler decreases as compared with that of the smooth-tube cooler by approximately one third. For grooving with parameters $d/D = 0.9$ and $t/D = 0.909$ the volume of the cooler is 80% of the volume of the smooth-tube equivalent.

Thus, the use of grooved tubes makes it possible to reduce the weight and volume of tubular heat exchangers by a factor of approximately 1.5 without affecting the hydraulic resistance. When the heat-transfer coefficient in the intertube space is much lower than that in the tubes, the optimum values in the range of d/D and t/D investigated are $d/D = 0.93–0.95$ and $t/D = 0.454$. Increasing the pitch of the grooving to $t/D = 0.909$ is undesirable, since to obtain a significant effect it is necessary to increase the depth of the grooving (at d/D = 0.90 Nu/Nu_{sm} = 1.3), and in this case we get a nonoptimum intensification of heat transfer inside the tube (Nu/Nu_{sm} = 1.5 at ξ/ξ_{sm} = 3.3 and Re = 10^5 [7].

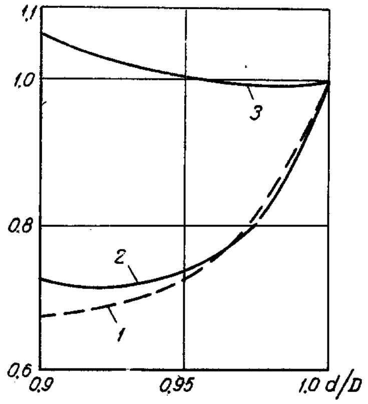

Fig. 4. Ratios of volumes, lengths, and cross-sectional areas as a function of grooving depth d/D (t/D = 0.454) for a tubular cooler with a longitudinal gas flow in the intertube space: 1) length ratio L/L_{sm} for coolers with grooved and smooth tubes; 2) volume ratio V/V_{sm}; 3) area ratio F/F_{sm}.

NOTATION

D is the outside diameter of tubes; d is the diameter of ring groove; d_e is the equivalent diameter of bundle; $\bar{q}$ is the mean heat flux; S is the pitch of tubes in bundle; t is the pitch of the ring grooves; $\bar{t}_f$ and $\bar{t}_w$ are the mean flow and wall temperatures; ξ is the resistance coefficient; $\bar{T}_w/\bar{T}_f$ is the temperature factor. Subscripts: sm denotes smooth-tube bundle; 0 represents isothermal flow.

REFERENCES

1. I. B. Danilov and V. B. Keilin, IFZh, 5, 1962.
2. N. N. Norkin and I. P. Chashchin, Teploénergetika, no. 6, 1962.
3. N. N. Norkin and I. P. Chashchin, Teploénergetika, no. 6, 1963.
4. N. Kattchee and W. V. Mackewich, Paper ASME, NHT-1, 1963.
5. E. K. Kalinin, G. A. Dreitser, and A. K. Kozlov, Izv. AN BSSR, ser. fiz.-tekhn. nauk, no. 2, 1966.
6. A. I. Mikhailov, V. V. Borisov, and E. K. Kalinin, Closed-Cycle Gas Turbine Plants [in Russian], izd. AN SSSR, 1962.
7. E. K. Kalinin and S. A. Yarkho, IFZh [Journal of Engineering Physics], 11, no. 4, 1966.
8. A. I. Mikhailov, E. K. Kalinin, and G. A. Dreitser, IFZh, 7 no. 11, 1964.
9. A. I. Mikhailov, E. K. Kalinin, and G. A. Greitser, IFZh [Journal of Engineering Physics], 10, no. 1. 1966.
10. E. V. Firsova, IFZh, 6, no. 5, 1963.
11. D. A. Dingee and J. W. Chastain, Reactor Heat Transfer Conference of 1956, AEC Report TID, 7529, 1957.

17 January 1968

Ordzhonikidze Aviation Institute, Moscow