11. B. V. Dzyubenko, "Fluid friction in a swirling-flow heat exchanger," Inzh.-Fiz. Zh., 44, No. 3, 357-362 (1983).
12. P. A. Urbonas, "Experimental study of the fluid friction factor in a bundle of twisted tubes," in: Interuniversity Collection: Current Problems of Hydrodynamics and Heat Transfer in Powerplant Elements and Cryogenic Engineering [in Russian], No. 11, VZMI, Moscow (1982), pp. 78-82.
13. V. I. Subbotin, M. Kh. Ibragimov, P. A. Ushakov, et al., Hydraulics and Heat Transfer in Nuclear Power Plants [in Russian], Atomizdat, Moscow (1975).

FLUID FRICTION OF TUBES FITTED WITH HELICAL HEAT-TRANSFER ENHANCERS

A. I. Rzaev and L. L. Filatov

UDC 532.529.5

The results of an analysis of experimental data on the fluid friction of tubes with heat-transfer enhancers are given. An empirical equation is proposed for generalization of the experimental data.

A number of papers have been published to date on the enhancement of heat transfer in tubes fitted with various types of enhancers. The latter are frequently in the form of helical wire-coil inserts, single-start and multistart helical rib roughening (ridging), and multistart helical corrugations [1-6]. Along with the investigation of the thermal characteristics of such tubes, these papers often present data on the fluid friction, which are needed in order to assess the engineering efficiency of heat exchangers constructed from the given tubes.

Inasmuch as heat transfer associated with turbulent fluid flow in tubes with helical enhancers is of the greatest practical interest, data on the fluid friction are generally given either in the form of a graph of $\xi_{hx} = f(Re)$ for $Re = 3 \cdot 10^4 - 10^5$ [2] or in the form of an intricate dimensionless implicit function that generalizes only the specific data and is not suitable for practical engineering calculations [4].

In the present study, on the basis of an analysis of the available published experimental data, we attempt to derive a relation that is suitable for calculating fluid friction of tubes with helical enhancers and, if possible, will take into account the influence of the flow regime and the geometrical dimensions of the tube and enhancer, viz.: the tube diameter, the pitch of the helical insert or ridging, and the wire diameter or the ridge height.

We have selected for analysis the most complete friction data for 18 tubes with helical wire-coil inserts and helical ridging [1, 2]; these data were obtained for roughly the same range of Reynolds numbers. The principal dimensions of the tubes are summarized in Table 1.

Figure 1 shows the dependence of the variation of the relative friction factor ξ_{hx}/ξ_{sm}, determined for the analyzed tubes at $Re = 2 \cdot 10^4$, on the parameter d_{in}/S; the data are referred to the quantity $(h/S)^m$. The exponent m characterizing the influence of the parameter h/S has been evaluated [3] as $m = 0.23 Re^{0.07}$ and varied from 0.44 to 0.52 in the range of numbers $Re = 10^4 - 10^5$. A value m = 0.5 was taken in the first approximation. It is seen that the data in Fig. 1 are clustered with ± 17% scatter limits about an average line whose slope determines the degree of influence of the parameter d_{in}/S on the friction factor, i.e., $\xi_{hx} \sim (d_{in}/S)^n$, where n = 0.4.

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TABLE 1. Geometrical Dimensions of Tubes and Enhancers

Tube No.	d_{in}	S	h	k	d_{in}/S	h/S	Tube No.	d_{in}	S	h	k	d_{in}/S	h/S
	mm							mm					
Data of [1]							Data of [2]						
1	25	66	2	1	0,379	0,0303	9	22,6	8,7	0,36	1	2,6	0,0415
2	25	38	2	1	0,658	0,0526	10	22,7	6,6	0,35	1	3,46	0,0532
3	25	22	2	1	1,14	0,0909	11	14,5	9,8	0,43	6	1,48	0,0442
4	25	10	2	1	2,5	0,2	12	14,6	12,1	0,48	6	1,21	0,04
5	25	66	3	1	0,379	0,0455	13	14,5	8,1	0,42	6	1,78	0,0517
6	25	38	3	1	0,658	0,0789	14	17,4	9,9	0,43	6	1,75	0,0435
7	25	22	3	1	1,14	0,136	15	14,6	11,9	0,53	5	1,23	0,0441
8	25	10	3	1	2,5	0,3	16	14,6	7,1	0,41	6	2,05	0,0581
							17	20,7	8,4	0,52	6	2,46	0,0622
							18	14,6	12,1	0,32	5	1,15	0,0265

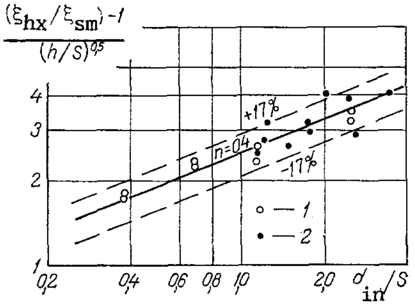

Fig. 1

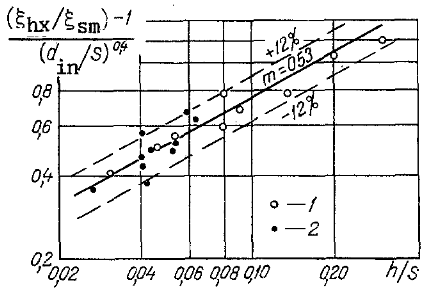

Fig. 2

Fig. 1. Variation of relative friction factor vs parameter d_{in}/S. 1) Helical wire-coil-inserted tubes [1]; 2) helically ridged tubes [2].

Fig. 2. Variation of relative friction factor vs parameter h/S. 1) Data of [1]; 2) [2].

It is evident from Fig. 2 that the refined value of the degree of influence of the parameter h/S on the friction factor ξ_{hx} is m = 0.53, and the analyzed data are clustered with ± 12% scatter limits about an average line that can be described by the equation

$$\xi_{hx}/\xi_{sm} = 1 + 2.54\,(d_{in}/S)^{0,4}\,(h/S)^{0,53}, \tag{1}$$

which ensures the passage to the limit $\xi_{hx} \to \xi_{sm}$ as $S \to \infty$ and $h \to 0$. Since $\xi_{sm} = 0.3164 \cdot Re^{-0.25}$ for turbulent flow, Eq. (1) for the determination of the friction factor of tubes with helical enhancers takes the final form

$$\xi_{hx} = 0.3164\,Re^{-0,25}\,[1 + 2,54\,(d_{in}/S)^{0,4}\,(h/S)^{0,53}]. \tag{2}$$

The analytical function is compared with the data of [1, 2] in Figs. 3a and 3b. The experimental data for tubes No. 17, 18, 4, and 5 (Table 1) were selected for the comparison. It is seen that the deviation of the experimental data from the calculations according to relation (2) does not exceed ± 5%.

In Fig. 3c the function (2) is compared with the data of [5] on the friction factor of a tube of diameter 25.4 mm roughened by ridges of height 0.25 mm with a slope $\alpha = 49°$ relative to the tube axis. The pitch of the helical ridging was determined from the inside diameter and slope angle according to the relation $S = d_{in} \cot \alpha$. It is evident from the figure that the analytical function agrees with the experimental data within ± 12% error limits.

It can thus be stated on the basis of the comparison results that the postulated equation (2) can be used in the ranges of the parameters $Re = 4 \cdot 10^4 - 10^5$, $d_{in}/S = 0.35-3.5$, and $h/S = 0.025-0.3$ to determine the fluid friction of tubes with the investigated types of heat-transfer enhancers within acceptable error limits.

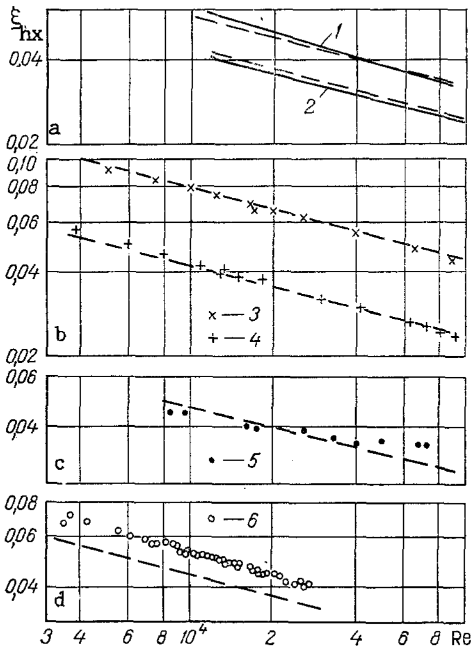

Fig. 3. Function $\xi_{hx} = f(Re)$. Comparison of experimental data with calculations according to Eq. (2) (dashed lines). a) Data of [2]: 1) tube No. 17; 2) No. 18. b) Data of [1]: 3) No. 4; 4) No. 5. c) Data of [5]: 5) tube with $d_{in} = 25.4$ mm, h = 0.25 mm, $\alpha = 49°$. d) Our data: 6) helically corrugated tube, $d_{in} = 12$ mm, h = 1 mm, S = 20.6 mm.

In Fig. 3d the function (2) is compared with data obtained by the present authors on the friction of tubes with a different type of enhancer in the form of multistart corrugations on the inner surface of the tube. It is seen that the experimental data are approximately 20% higher than the calculated values. This discrepancy can be explained as follows. The hydrodynamic pattern in tubes with helical corrugations in more complicated and involves the formation of vortex flows of the Taylor—Goertler type [7] in the fluid flowing in the corrugations. A scarcity of data makes it difficult to describe the friction factor quantitatively in tubes fitted with this type of enhancer.

NOTATION

d_{in}, inside diameter of tube; S, pitch of helical wire-coil inserts or ridging; h, height of ridge or wire diameter of coil insert; k, number of "starts"; ξ_{hx}, ξ_{sm}, fluid friction factors of tube with helical enhancer and of smooth tube; m, degree of influence of parameter h/S; n, degree of influence of parameter d_{in}/S; α, slope angle of ridging relative to tube axis; Re, Reynolds number.

LITERATURE CITED

1. R. Sethumadhavan and M. Rao Raja, "Turbulent flow heat transfer and fluid friction in helical-wire-coil-inserted tubes," Int. J. Heat Mass Transfer, 26, No. 2, 1833-1845 (1983).
2. J. G. Withers, "Tube-side heat transfer and pressure drop for tubes having helical internal ridging with turbulent-transitional flow of single-phase fluid. Part 2: Multiple-helix ridging," Heat Transfer Eng., 2, No. 2, 43-50 (1980).
3. E. M. Chizhevskaya, Yu. M. Brodov, P. G. Men', and V. A. Doroshenko, "Fluid friction in the flow of water in profiled 'twisted' tubes," Izv. Vyssh. Uchebn. Zaved., Energ., No. 10, 119-123 (1977).
4. S. Ganeshan and M. Rao Raja, "Studies on thermohydraulics of single- and multistart spirally corrugated tubes for water and time-independent power-law fluids," Int. Heat Mass Transfer, 25, No. 7, 1013-1022 (1982).

5. D. L. Gee and R. L. Webb, "Forced convection heat transfer in helically rib-roughened tubes," Int. J. Heat Mass Transfer, 23, No. 8, 1127-1136 (1980).
6. A. I. Rzaev, I. T. Alad'ev, and L. L. Filatov, "Fluid friction of single- and two-phase flows in tubes with helical corrugations," Teploenergetika, No. 10, 63-65 (1984).
7. V. K. Shchukin, Heat Transfer and Hydrodynamics of Internal Flows in Body-Force Fields [in Russian], Mashinostroenie, Moscow (1970).

PRESSURE LOSSES IN FINNED AIR COOLERS DURING FROST FORMATION

M. N. Chepurnoi and V. N. Lomakin

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The article presents the results of experimental investigations of the relative pressure losses in finned air coolers with different pitch of the finning.

When the operation of finned air coolers is accompanied by frost formation, the cross section for the passage of air continuously decreases, and in consequence the aerodynamic resistance increases and the performance of the fan and the air cooler decreases. Beginning at some instance after the onset of the deposition of frost, the heat-exchange surface of the apparatus operates with decreasing efficiency, and the expenditures on the transport of air continuously increase. When the regularities of the change of loss of pressure head during frost formation are known, a fan with the required characteristic can be chosen, and in addition it is possible to determine the optical regimes of operating the air coolers involved.

In the literature there are a limited number of investigations [1-4] shedding light on the change of the aerodynamic characteristics of air coolers upon frosting. The suggested methods of calculating pressure losses are not of a universal nature and cannot be extended to apparatus with a different geometry of the finning and different operating conditions. Ivanova [1] recommended that pressure losses be determined in the following way:

$$\Delta P = \Delta P_0 + A\tau^n,$$

where, in dependence on the geometry of the finning, the mass velocity, and the relative air humidity, more than thiry values for A and n are given. Yavnel' [2] presented the relative pressure losses in the form of dependences on the amount of frost settling on the surface of the air cooler, but there is no information on the method of determining this amount. On the other hand, the author recommends calculating the losses by formulas obtained without frost, and taking into account the reduction of the cross section for the passage of air. Calculations by this method are very difficult because for their realization it is required that the regularities of the growth of frost be known, and in addition, the cross section of the channels and the air speed in them have to be continuously recalculated. Lotz [3] correlated the aerodynamic losses with the density of the frost. Thus there does not exist so far a single method of calculating pressure losses in finned air coolers during frost formation that takes into account design features of the apparatuses as well as their operating conditions.

The present article represents an attempt at refining the regularities of the increase of pressure losses in finned air coolers with the object of generalizing them and of working out substantiated methods of calculation. In a wind tunnel 400 × 400 mm in size we investigated the operation of air coolers with finning pitches of 8, 11, 13.4, 17.5, and 20 mm. Frost formation in these air coolers had been investigated earlier [5]. The characteristics of the air coolers are presented in Table 1, and the experimental installation, the regimes and method of the experiments were described in [5, 6].

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