NOTATION

W, water equivalent; t, temperature, °K; z, spatial coordinate, m; α_1, α_2, coefficients of heat transfer from inner and outer surfaces of the inner tube, $W/m^2 \cdot K$; α_3, α_4, coefficients of heat transfer from inner and outer surfaces of the outer tube, $W/m^2 \cdot K$; p_k, length of arc of contact between spiralled inner tube and outer tube in the channel cross section, m; r_k, contact linear heat-transfer resistance, m·K/W; λ_1, λ_2, thermal conductivities of inner and outer tubes, W/m·K; T_1, T_2, inlet temperatures of flows of heating and heated media; d_1, d_2, inside and outside diameters of the inner tube, m; d_3, d_4, inside and outside diameters of the outer tube, m; L, length of the heat exchanger, m; G, mass flow rate, kg/sec. Indices used with W, T, and G: 1, heating medium; 2, heated medium; ', medium in inner tube; ", medium in space between tubes.

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CALCULATING THE EXTERNAL HEAT EXCHANGE AND AERODYNAMIC DRAG OF PLATE-FINNED AIR COOLERS

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Results are presented from an experimental study of heat exchange and pressure loss in plate-finned air coolers with different spacings of the fins.

In practice, mainly two types of air coolers and air condensers are used: those equipped with staggered bundles of tubes with circular or spiral fins, and those equipped with corridor bundles of tubes with plate fins. The hydrodynamics and heat exchange of apparatus of the first type have been studied fairly thoroughly, and recommendations have been made for calculating their thermal and aerodynamic performance characteristics [1, 2]. For apparatus of the second type, until now there has not been adopted a single, proven method of design. This is because, as was noted in [1], their surfaces are not similar. Different investigators [3-6, et al.] have obtained a large number of formulas for calculating heat exchange and friction loss which are suitable only for air coolers of the designs in question.

The use of special methods to analyze test data makes it hard to compare the results of different studies, as is indicated by the surveys in [1, 2]. The use of different linear dimensions as the determining dimensions in similitude criteria as well as the use of different dimensionless simplexes characterizing the geometric similarity of the heat-exchange surfaces, on the one hand, does not permit generalization of the available empirical data and, on the other hand, leads to a significant discrepancy between results obtained under identical test conditions.

In connection with this, we have attempted to refine the laws of external heat exchange and aerodynamic drag for plate-finned air coolers on the basis of studies of models with different fin spacings. We used a 400 × 400 mm wind tunnel to study air coolers with tubes with an outside diameter of 25 mm placed in a corridor arrangement. The tubes were made up of sections 170 mm long having two lengthwise (in the direction of the air flow) and five

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transverse rows of tubes spaced 70 and 76 mm apart, respectively. The number of lengthwise rows of tubes was varied from 8 to 18. The solid flat fins were made of steel 0.4 mm thick. We tested models of air coolers with a fin spacing of 8, 11, 13.4, 17.5, and 20 mm. Here, the degree of finning of the heat-exchanger surface was varied from 11 to 26. The tested equipment was cooled with ammonia and heated with hot vapors of the coolant. The prescribed temperature of the surface was ensured by the use of an independent cooling unit and was kept constant during the tests. Air was supplied by a centrifugal fan with a dc electric drive which made it possible to smoothly change the speed. Air flow rate was determined by means of precalibrated nozzles. Air velocity in the channels of the cooler was monitored with tubes for measuring dynamic pressure. The mass velocity of the air flow as varied during the tests from 3 to 13 $kg/m^2 \cdot sec$, while the Reynolds number was varied from 3000 to 28,000. The air temperature at the inlet was kept at a value near 30°C. The temperature of the air, the walls of the tubes, and the fins were measured with thermocouples. During the tests the mean temperature head changed within the range 8-18°C. The testing unit and experimental method were detailed in [7].

We selected the method for analyzing the test data on the basis of the following considerations. Since the rate of external heat transfer depends on the aerodynamic conditions under which air flows in the slit-like channels of the air cooler, we chose the hydraulic diameter of the channel formed by two adjacent fins as the linear dimension in the similitude criterion. It was taken into account that different geometric factors pertaining to the heat-exchange surface affect the hydrodynamics and heat transfer. Among these factors are: the outside diameter of the tubes and the transverse spacing between tubes, the thickness and spacing of the fins, and the degree of finning [1, 3, 4, 6, 8]. We also took into account the observation [8] that geometric similitude in a cross section can be characterized by the s_1d^{-1} and s_fd^{-1}. On the basis of the above, we chose the following simplex as the geometric similitude parameter, taking into account the design features of different air coolers:

$$\sigma = 2\delta_f s_f s_1 \beta (d^2 d_h)^{-1}.$$

Calculations showed that parameter σ ranges from 1 to 0.1 for all of the existing plate-finned air coolers with a fin spacing from 3 to 40 mm. Here, the quantities entering into this parameter changed within the following ranges: $\delta_f d_h^{-1}$ from 0.0055 to 0.05; $s_f d^{-1}$ from 0.12 to 4.25; $s_1 d^{-1}$ from 1.67 to 4.1; β from 4.6 to 39.

The thermal load on the air coolers was determined from the heat balances between the air and coolant. The mean temperature head was calculated from the differences in temperature between the mean temperature of the outside surface and the air at the inlet and outlet of the apparatus. The temperature of the outside surface of the air coolers was determined as the mean temperature of the outside walls of the tubes and fins. The mean fin temperature was determined as the arithmetic mean of the readings of thermocouples embedded at five stations along the fins from the base to the top. By analogy with [3], the heat-transfer coefficients were calculated for the total outside surface of the coolers from its mean temperature. Here, we considered not only the fin efficiency factor, but also the heat-transfer resistance of the tube—fin contact [1, 2]:

$$\alpha_u = \alpha_{cr} E_u, \tag{1}$$

where, according to [2]:

$$\alpha_{cr} = [\alpha_f E_f (\beta - 1) + \alpha_t] \beta^{-1}, \tag{2}$$

and in accordance with [1]

$$E_u = E_f \chi + (1 - E_f \chi) \beta^{-1}. \tag{3}$$

Direct determination of α_u is usually precluded by the awkward calculation of the fin efficiency factor E_f. If we now calculate heat transfer from the temperature of the tube wall, then on the basis of (1)-(3) the value of E_f is determined easily. Calculations showed that the fin efficiency factors entering into (1) range from 0.91 to 0.6 for the above ranges of temperature head and Reynolds number. Meanwhile, the higher values of the parameter σ correspond to higher values of the efficiency factor E_f with Re = const. The results agree within ±20% with the data in [1, 3, 7] with corresponding values of σ and Re.

Figure 1 shows results of studies of external heat transfer in coolers with different geometries of the finned surface. It is apparent from the dependences shown in this figure

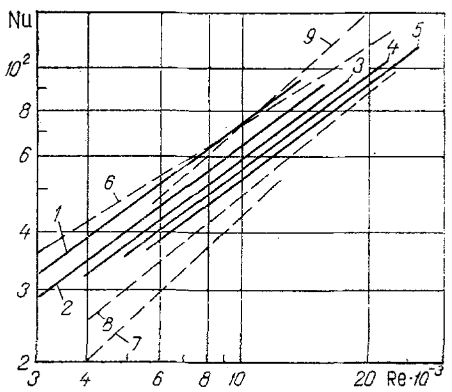

Fig. 1

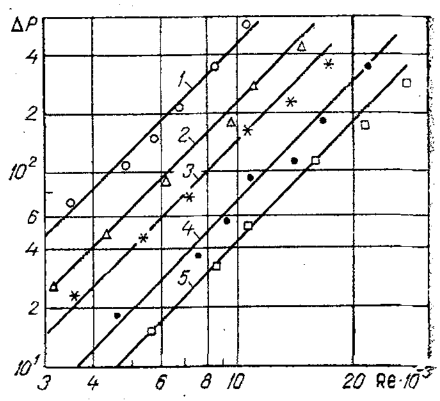

Fig. 2

Fig. 1. The functions Nu = f(Re) for finned air coolers with s_f = 8-20 mm: 1) σ = 0.645; 2) 0.5; 3) 0.423; 4) 0.33; 5) 0.278; 6) 0.237; 7) 0.504; 8) 0.21; 9) 0.62 (1-5 — our data; 6, 7 — [3]; 8 — [5]; 9 — [6]).

Fig. 2. Pressure losses in finned air coolers: 1) σ = 0.645; 2) 0.5; 3) 0.423; 4) 0.33; 5) 0.278. ΔP, Pa.

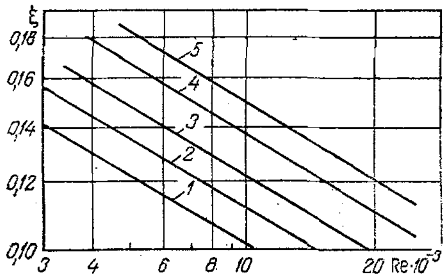

Fig. 3. Values of the friction coefficients. Same notation as in Fig. 2.

that the heat-transfer rate increases with an increase in the parameter σ, which in turn increases with a decrease in fin spacing. The latter is true because a reduction in fin spacing is accompanied by a reduction in the hydraulic diameter of the channel and an increase in the degree of finning of the heat-exchange surface. It is also apparent from Fig. 1 that the results presented in this article agree quite well with the data in [5, 6], recalculated by the proposed method, and disagree with the data in [3] — in which an increase in heat transfer with an increase in fin spacing was noted. The findings in [3] contradict the physical nature of the phenomenon in question and are not borne out by experience [5-7, 9]. The following criterial equation was obtained to calculate heat transfer in coolers from the air side:

$$\mathrm{Nu} = 0.113\sigma^{0.4}\mathrm{Re}^{0.72}, \tag{4}$$

which is valid in the range 3000 < Re < 40,000.

It is apparent from the empirical dependences for aerodynamic drag shown in Fig. 2 that a decrease in the parameter σ (an increase in fin spacing) is accompanied by a decrease in friction loss in the coolers. To calculate aerodynamic losses in air coolers with finned surfaces of different geometries, a large number of formulas have been obtained [1, 2, 4-6, etc.] in which the pressure losses are proportional to the mass velocity of the air to the power of 1.7 to 1.8 and depend on several different simplexes. It is thus recommended that the pressure loss be calculated from the traditional formula

$$\Delta P = 0.5\xi L d_h^{-1}\rho w^2. \tag{5}$$

For the air coolers investigated, the relative length Ld_h^{-1} ranged from 26 to 103.

The dependence of the friction coefficient ξ on the Reynolds number, shown in Fig. 3, indicates that the parameter σ has the reverse effect on the unit resistance. The dependences shown in Fig. 3 are approximated by the relation

$$\xi = 1.075\,\sigma^{-0.46}\,\mathrm{Re}^{-0.28}, \tag{6}$$

which corresponds in form to similar relations for slit channels and is valid for the same range of Reynolds numbers Re as Eq. (2). The results of calculations of pressure loss by Eqs. (3) and (4) in [2, 5] nearly coincide with our results. The results in [4] are too high by an average of 20%, which is probably due to the presence of projections on the plate air coolers. On the basis of this, the constant in Eq. (6) should evidently be given a value of 1.3 for fins with a non-smooth surface.

NOTATION

δ_f and s_f, fin thickness and spacing; d and s_1, outside diameter of tubes and transverse spacing between tubes; d_h, hydraulic diameter of the channels of the air cooler between adjacent fins; β, degree of finning; w, velocity of the air flow; ρ and ν, density and kinematic viscosity of the air at the mean temperature; α_u, coefficient of heat transfer from the outside surface of the air coolers at its mean temperature; α_{cr}, α_f, α_t, corrected coefficients of heat transfer from outside surface, from the surface of the fins, and from the surface of the tubes at the temperature of the tube wall; E_f, fin efficiency factor; χ, coefficient accounting for the method of fastening the fins; λ, thermal conductivity of the air; ξ, friction coefficient; ΔP, pressure drop; L, length of air cooler; $\mathrm{Nu} = \alpha_n d_h \lambda^{-1}$; $\mathrm{Re} = w d_h \nu^{-1}$.

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