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SELECTION OF ENERGETICALLY EFFICIENT METHOD OF SITUATING FINNED TUBES IN THE HEAT-EXCHANGE SECTION OF AN AIR COOLER

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A computational and analytical study of the energy efficiency of the configuration of finned tubes in the heat exchange section of an air cooler is carried out by the Antuf'ev method for different types of configuration of finned tubes in the heat-exchange section of an air cooler. In the investigation the range of variation of the air speed in the compressed frontal part of the section was in the range 2.87–25.66 m/s and of the tube finning coefficient, in the range 5.1–27.4 (which corresponds to the range of variation of the parameters of the air coolers employed). In the range of variation of the consumption of power N_0 by the fan motor typical of the high-speed operating modes of air coolers with a checkerboard configuration in the section bundle, the average thermal efficiency of finned tubes in the section bundle is 25% higher than the unstaggered configuration with N_0 = idem. Thus, in the sections of air coolers, it is best to use a checkerboard configuration of the finned tubes, which makes it possible to achieve high thermal efficiency of the finned tubes along with energy and resource conservation.

Keywords: air coolers, bimetallic finned tube, spiral rolled aluminum fin, checkerboard tube bundle, convective heat exchange, aerodynamic resistance.

Petroleum refining, petrochemical, and chemical enterprises are the principal users of air coolers of water and other fluids with the greatest reservoir capacities.

The use of air coolers to cool circulating water in the production cycle produces an up to 20-fold reduction in water consumption of enterprises per ton of refined oil [1].

The heat-exchange section is one of the basic structural elements of an air cooler. In a heat-exchange section removal of heat from the cooled product is accomplished by forced air convection. The heat-exchange section is a typical liquid-gas heat exchanger and, in view of the low thermophysical properties of air (as a cooling agent), is distinguished by significant overall dimensions and weight, which is typical of the air cooler as a whole [2].

The heat-exchange section consists of bimetallic finned tubes of round cross-section with spiral round ribs. After being assembled into a tubular bundle, the bimetallic finned tubes are streamlined externally by a single forced flow of air travelling perpendicular to the longitudinal axes of the tubes. Either a checkerboard (staggered) configuration of the bimetallic finned tubes (Fig. 1a) or an unstaggered (in-line) configuration of the tubes (cf. Fig. 1b) is possible in the bundle of a section.

The objective of the present study is to determine an effective arrangement (configuration) of the finned tubes in a bundle for different velocity regimes of the motion of air in an air cooler.

The configuration characteristics of the bundle comprise transverse S_1 and longitudinal S_2 steps in the configuration of the tubes and the number of transverse series of tubes z. The compactness coefficient Π (m^2/m^3), which characterizes the density of the configuration of the tubes, is a complex parameter of the bundle (which takes into account the geometric characteristics of the configuration of the tubes and the geometry of the finning).

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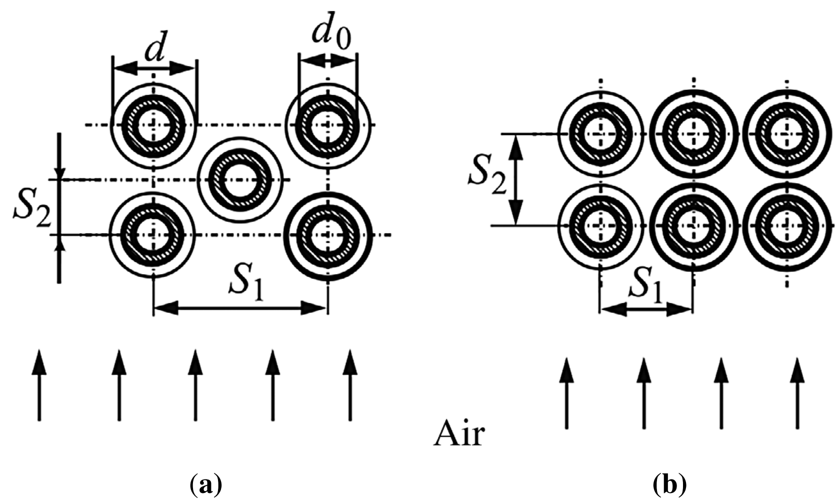

Fig. 1. Schematic diagram of comparable bundles of finned tubes: (a) — checkerboard configuration, (b) — unstaggered configurtion.

For a bundle of tubes with round spiral fins [2],

$$\Pi = \pi d_0 \varphi/(S_1 S_2), \tag{1}$$

where d_0 is the diameter of the tube at the base of a fin, m, and φ the coefficient of finning of a tube computed from the formula

$$\varphi = 1 + \frac{2h}{sd_0}(d_0 + h + \Delta), \tag{2}$$

where h, s, and Δ is, respectively, the height, step, and average thickness of a fin.

Let us compare the configurational characteristics of the checkerboard and unstaggered configurations. The minimal value of the transverse step is the same for the checkerboard and unstaggered bundles and is given by $S_{1\,\min}^{\mathrm{ch}} = S_{1\,\min}^{\mathrm{un}} = d$ (outer diameter of fin). The minimal value of the longitudinal step in an unstaggered bundle $S_{2\,\min}^{\mathrm{un}} = d$, but in a checkerboard bundle $S_{2\,\min}^{\mathrm{ch}} < d$ (cf. Fig. 1a). Consequently, $S_{2\,\min}^{\mathrm{ch}} < S_{2\,\min}^{\mathrm{un}}$, i.e., with identical geometry of finning of a tube $\Pi^{\mathrm{ch}} < \Pi^{\mathrm{un}}$, while the volume of a checkerboard bundle is less than the volume of an unstaggered bundle (with $z = \mathrm{idem}$).

However, in this type of comparison the thermal $\mathrm{Nu} = f(\mathrm{Re})$ and aerodynamic $\mathrm{Eu} = f(\mathrm{Re})$ characteristics of the bundles are not taken into account ($\mathrm{Nu} = \alpha d_0/\lambda$ is the Nusselt number and $\mathrm{Re} = wd_0/\nu$ — Reynolds number, $\mathrm{Eu} = \Delta P/(\rho w^2)$ — Euler number, α — relative heat emission from the direction of the air, W/(m^2·K), λ — thermal conductivity factor of air, W/(m·K), w — speed of air in compressed (narrow) transverse section of bundle, m/s, ν — coefficient of kinematic viscosity of air, m^2·s, ΔP — drop in static pressure of air in bundle, Pa, and ρ — density of air, kg/m^3).

No comprehensive investigations of the thermo-aerodynamic characteristics of checkerboard and unstaggered bundles in a transverse flow of air as a function of the geometric parameters of bimetallic finned tubes embedded in the structure of the heat-exchange section were undertaken in the course of the development and management of the production of air coolers [1]; moreover, the data of one study [3] serves to attest to high thermal (energy) efficiency of checkerboard bundles as compared to unstaggered bundles. Therefore, a checkerboard configuration of finned tubes (in domestically produced air coolers [1, 2]) and in air coolers produced by foreign manufacturers) was adopted a priori in the bundles of the heat-exchange sections of air coolers.

A significant volume of experimental data related to heat emission and hydraulic resistance (in an air flow) of checkerboard and unstaggered bundles composed of tubes of different magnitudes (over a broad range characteristic of standardized general-purpose air coolers), geometric dimensions of finning and coefficients of finning, number of transverse series, and air speed in the compressed section of the bundle has now been accumulated. Realization of the objective of the present study is now possible with the use of the available file of data, for which purpose a well-known technique [3] of comparison of convective heat-exchange surfaces of tubes or bundles of tubes relative to thermal (energy) efficiency was employed. This technique is characterized by a dependence of the intensity of the relative heat emission α (W/(m$^2\cdot$K)) on the specific expenditure of power N_0 (W/(m^2) used to pump heat carrier (air is the heat carrier for an air cooler) through the heat-exchange section). The energy efficiency is estimated quantitatively (in the case of identical power expenditures $N_0 = \text{idem}$) by the coefficient of thermal efficiency

$$\psi_i = a^{\text{ch}}/a^{\text{un}}, \tag{3}$$

where α^{ch} and α^{un} are, respectively, the relative coefficient of heat emission of checkerboard bundles and relative coefficient of heat emission of unstaggered bundles (adopted as the standard).

The specific power expenditures in forced motion of air are calculated from the formula [2]

$$N_0 = 0.318\psi' \frac{\text{Eu}_0}{\varphi} \rho w^3, \tag{4}$$

where $\psi' = \left(\frac{S_1}{d} - 1\right) + \frac{2h}{d_0}\left(\frac{S_1}{d} - \frac{\Delta}{s}\right)$ is a coefficient that expresses the geometric dimensions of finning of a tube and the configurational characteristics of a bundle, w — air speed in compressed frontal section of bundle, m/s, ρ — density of air, kg/m^3, and $\text{Eu}_0 = \text{Eu}/z$ — Euler number scaled on the basis of a single transverse series of a bundle of tubes.

Both the method of total thermal simulation and the method of local (approximate) thermal simulation are used to study heat emission of bundles of finned tubes in an air flow (Here it is assumed that the thermal conditions in the method of local simulation of the model (bundle) are completely similar to the thermal conditions of a full-scale sample, i.e., the heat-exchange section) where the thermal conditions are observed for only a single tube (calorimeter) of the bundle in the case of local simulation. The magnitudes of heat emission determined by local simulation (particularly in the case of dense (tight) configurations of tubes) are overstated by a magnitude of up to 30–35% [4] by comparison with the results of total simulation. Moreover, the actual strengths of the intensity of heat emission are also distorted, which may lead to incorrect conclusions concerning the energy efficiency of bundles of tubes. Therefore, all the computed values of α^{ch} and α^{un} were adjusted to identical conditions (i.e., to the data of total simulation) through the introduction of corrections to the method of simulation of convective heat exchange (for checkerboard bundles based on the data of [3], and for unstaggered bundles, based on the data of [4]).

The similitude equations presented in [4–11] were used to calculate the values of the coefficient of relative heat emission α and Euler numbers Eu for two values of the air speed w_1 and w_2 in the range of working velocity regimes of air coolers. The physical properties of air in the Re, Nu, and Eu numbers in the similitude equations corresponded to a temperature of 50 °C.

Table 1

Bundle	Type of Bundle	Step of Tube, mm		Dimensions of Finning, mm					φ	Π, m^2/m^3	Speed of Air, m/s		Source
		S_1	S_2	d	d_0	h	s	Δ			w_1	w_2	
1	ch	54.4	64	50	32	9	6	1.3	5.1	147	5.61	22.44	[4]
2	un												
3	ch	58	50.4	56	28	14	3	0.75	15.2	457	3.85	19.23	[5]
4	un		58							397			
5	ch	39	39	39	20	9.5	4	0.8	8.3	343	5.39	17.96	[6]
6	un												
7	ch	50	56	48.6	29	9.8	3.2	0.4	9.3	302	3.72	12.39	[7]
8	un												
9	ch	70	60.6	55.85	25.85	15	2.56	0.75	19.9	380	4.17	20.78	[8]
10	un												
11	ch	59	51.1	55.28	25	15.14	2.53	0.325	20.37	530	2.87	14.36	[9]
12	un		59							459			[10]
13	ch	103.5	84	69	21	24	4	1.25	27.4	208	5.12	25.65	[11]
14	un	110.3	85.1							191			

Remark: ch — checkerboard bundle, un — unstaggered bundle.

The configurational characteristics and geometric dimensions of round finning of tubes as well as additional data necessary (according to the present procedure for performing calculations) for construction of the thermal efficiency curves $\alpha = f(N_0)$ of comparable checkerboard and unstaggered bundles with identical standard size of the tube and material of the fins are presented in the accompanying table.

The values of the relative coefficient of heat emission from the direction of the air, $W/(m^2 \cdot K)$, were determined in the experimental investigation of bundles 1–14 (cf. Table 1):

$$\alpha = Q/[F(t_{at} - t)], \tag{5}$$

where Q is the heat flow transmitted by convection to a transverse flow of air travelling outside the tube, W, F — area of heat-transmission surface of finned tubes, m^2, t_{at} — average temperature of surface of tubes at base of fins, °C, and t — average temperature of air flow in the bundle, °C.

The following unexpected results are self-evident from an analysis of the calculations for bundles 1, 2, 3, and 4 (Fig. 2).

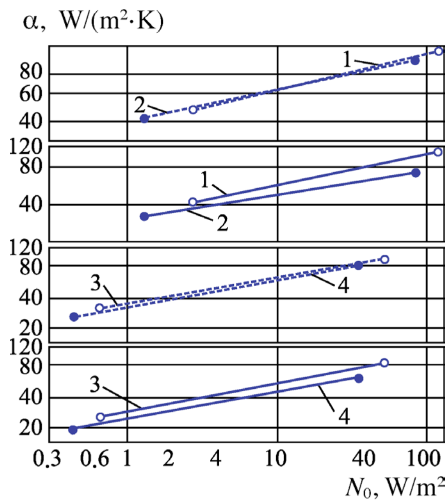

Fig. 2. Thermal efficiency curves of bundles with the use of different methods of simulation of heat emission: --- — local simulation, —— — total simulation, ○ — computed values for checkerboard bundles, ● — computed values for unstaggered bundles, 1–4 — notation of bundles (cf. Table 1).

The lines of thermal efficiency $\alpha = f(N_0)$ constructed from data of local (approximate) simulation of heat emission (cf. Fig. 2, dashed lines) of bundles 1, 2 and 3, 4 coincide over the entire range of variation of the expenditures of power N_0 in the fan drive (when used to pump out air). Consequently, the characteristics of the thermal cleanness of the checkerboard and unstaggered bundles are identical, i.e., equivalent in terms of intensity of heat emission. However, this is in contradiction to the well-known law obtained in total thermal simulation. That is, the coefficient of heat emission of checkerboard bundles of finned tubes must exceed the values of the coefficient of heat emission for unstaggered bundles [2, 4]. This contradiction is eliminated where the thermal efficiency of bundles 1, 2, and 3, 4 are calculated from data of heat emission for total thermal simulation (cf. Fig. 2, continuous lines). That is, the thermal efficiency curves of checkerboard bundles 1, 3 are higher than the analogous curves for unstaggered bundles 2, 4. For example, with $N_0 = 10$ W/m^2 the coefficient of thermal efficiency of checkerboard bundle 1 (relative to the efficiency of an unstaggered bundle) $\psi_1 = \alpha_1/\alpha_2 = 1.20$, while for bundle 3, $\psi_3 = \alpha_3/\alpha_4 = 1.23$. The intensity of heat emission of checkerboard bundles is 20–30% higher than for the unstaggered bundles, i.e., the power consumed in the fan drive is spent more efficiently.

From these data there follows the basic conclusion that use of equations of local thermal simulation is not permissible in the case of comparative estimation of the energy efficiency of bundles with tight configuration of finned tubes (for which the values of the relative steps are close to 1: $\sigma_1 = S_1/d \approx 1.03$–$1.12$, $\sigma_2 = S_2/d \approx 1$, which is typical of the heat-exchange sections of air coolers [2]), since the results of calculations found by means of these equations are quantitatively unreliable.

Graphs of the thermal efficiency of bundles 5–14 were constructed over a broad range of variation of the geometric dimensions of the ribs and tubes and the configurational characteristics of the bundles based on the foregoing results (with the use of equations of total thermal simulation), but with approximately different values of steps S_1 and S_2 in each standard sized group (Fig. 3).

For all the checkerboard bundles the thermal efficiency curves were higher than the analogous curves for the unstaggered bundles (cf. Fig. 3), i.e., the heightened intensity of heat emission in the case of a checkerboard configuration of finned tubes in the lattice of the heat-exchange section as compared to an unstaggered configuration of tubes is an inherent property of this configuration. Quantitative values of the thermal efficiency coefficient (for example, with $N_0 = 10$ W/m^2) of checkerboard bundles 5, 7, 9, 11, and 13 are calculated as

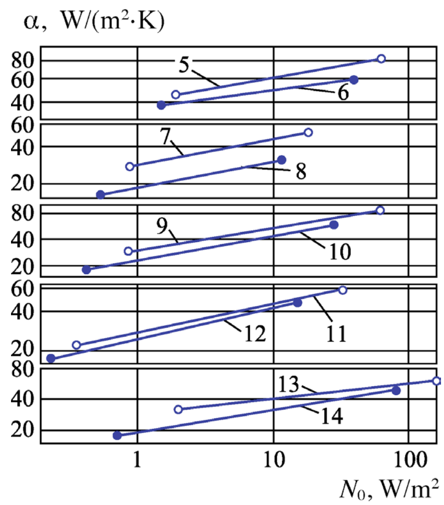

Fig. 3. Thermal efficiency curves of bundles in the case of total thermal simulation: ○ — computed values for checkerboard bundles, ● — computed values for unstaggered bundles, 5–14 — notation of bundles (cf. Table 1).

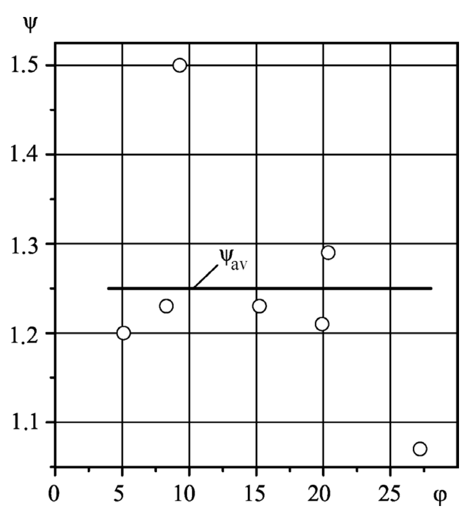

Fig. 4. Graphical diagram of values of coefficient of thermal efficiency of checkerboard bundles in the case $N_0 = 10$ W/m^2: ○ — computed values.

$\psi_5 = \alpha_5/\alpha_6 = 1.23$, $\psi_7 = \alpha_7/\alpha_8 = 1.50$, $\psi_9 = \alpha_9/\alpha_{10} = 1.21$, $\psi_{11} = \alpha_{11}/\alpha_{12} = 1.07$, and $\psi_{13} = \alpha_{13}/\alpha_{14} = 1.29$, where α_5–α_{14} correspond to points of intersection of lines N_0 = idem with the efficiency lines $\alpha = f(N_0)$ in Fig. 3.

Consequently, the coefficient of heat emission of a checkerboard bundle proves to be 1.07–1.50 times greater than the coefficient of heat emission of an unstaggered bundle for different standard sizes of finning (here the efficiency coefficient of an unstaggered bundle is provisionally adopted as unity, though with the use of the comparison technique presented in [2] it may be adopted as the standard technique, that is, the base technique for any type of bundle).

A particular form of the dependence of the efficiency coefficient on the finning parameters (Fig. 4) was not identified in the latter study, though an elevated thermal efficiency of the checkerboard configuration of finned tubes in a bundle and its high configurational and structural parameters were demonstrated.

The average exceedance of the thermal efficiency of a checkerboard bundle over the thermal efficiency of an unstaggered bundle (in the range $\varphi = 5.1–27.4$) is found to be $\psi_{av} = \Sigma\psi_i/n_i = 1.25$ ($n_i = 7$ is the number of checkerboard bundles; cf. Table 1).

The results of a comparative theoretical thermal calculation (with the use of the technique of [2]) of two 2AVG-7 type air coolers used to cool natural gas (with pressure of 7.5 MPa) from 75 °C to 45 °C are presented in [12]. The thermal capacity of the device is 3629 kW with temperature of the cooling air at the entry 30 °C. The fan unit comprises two Tornado T-50-4 fans with rotational speed of the electric motor of the fan 4.2 s^{-1} (250 min^{-1}). The heat-exchange section in both air coolers is produced from the same bimetallic finned tubes with rolled aluminum ribs, the parameters of the finned tubes corresponding to bundle 9 (cf. Table 1). The support tube is produced from carbon steel with outer diameter 25 mm and wall thickness 2 mm. In the first version the bimetallic finned tubes are arranged in a checkerboard configuration, while in the second version, in an unstaggered configuration (the two steps S_1 and S_2 are identical and correspond to the data of Table 1 for bundles 9 and 10). The results of the calculation were as follows: heat flow is removed with the use of a checkerboard bundle of bimetallic finned tubes with number of series of tubes $z = 6$, while with the use of the unstaggered configuration with $z = 7$ tubes. With the use of the unstaggered configuration the power consumption of the fan is $35.86/33.45 = 1.072$ times higher and the metal consumption of the bundle $7/6 = 1.7$ times higher (in fact, the metal consumption of the bundle will be even greater due to the increased dimensions of the tube plates).

The results that were found confirm the reliability and dependability of the present investigation of the thermal efficiency of checkerboard bundles.

Thus, in designing or modernizing air coolers it is best to use the checkerboard configuration of bimetallic finned tubes in the heat-exchange sections.

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