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EXPERIMENTAL INVESTIGATION OF A NONSTANDARD LAYOUT OF A MULTIROW HORIZONTAL FINNED-TUBE BUNDLE WITH AN EXHAUST SHAFT

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In view of the fact that heat exchangers made of bimetallic finned tubes have found wide application in contemporary technology for the regime of natural convection, it became necessary to take into account the radiative component that may be appreciable in the total heat flux and that depends on the geometric parameters of the bundle and characteristics of the surface studied. It is suggested in the article to raise the heat power of a multirow finned bundle by applying a nonstandard layout of tubes by placing their upper row above the exhaust shaft, with the remaining ones under the shaft. From the results of experimental investigations and of a comparative analysis of heat transfer by radiation and convection of a multirow finned-tube heat transfer bundle with an exhaust shaft and different layouts of tubes, it is shown that the nonstandard layout of finned tubes of a multirow heat exchanging bundle makes it possible to increase the radiant heat flux by 1.4–1.5 times. In this case, the total heat transfer rate of the bundle with the use of a nonstandard layout is increased approximately by 5%. It is shown that it is worthwhile to optimize the layout of the bundle when the fraction of the radiant flux in the total heat flow exceeds 10%.

Keywords: *finned tube, exhaust shaft, zone method of calculation of a radiant heat flux, natural convection of air.*

Introduction. Heat exchanging bundles consisting of finned bimetallic tubes are applied in the sections of the apparatuses of air cooling of liquids and of condensation of vapors, in evaporators of cooling chambers, electric and water air heaters for heating ventilation air in systems of air heating and in utilizing the heat of low- and middle-temperature secondary energy resources. On using a heat exchanger in the regime of natural convection, a considerable part of heat is removed by radiation, which may amount to a substantial fraction in the total heat flux and depend on the geometric parameters of the bundle and on the characteristics of the surface studied [1, 2]. The system of equations that describes the set of the primary processes that compose radiative heat exchange of finned bundles is highly complex mathematically; therefore, calculations of radiant heat transfer are usually carried out proceeding from a number of simplifying premises with distortion of the real physical picture.

Calculation of Radiant Heat Flux by the Zone Method. In engineering practice, the radiant free-convection heat transfer of multirow finned tube bundles is calculated by the zone method [3], with isolation of two zones in the bundle (Fig. 1a): zone 2 is composed of the external halves of tubes of extreme transverse rows, and zone 3 is the remaining part of the bundle. Zone 1 is formed by the surrounding medium consisting of two planes bounding the bundle.

The formula of calculation of the radiant heat flux by the zone method in application to a bundle of finned tubes is written as follows:

$$Q_{\mathrm{rad}} = c_0 F \varepsilon_{\mathrm{ef}} \varphi_{\mathrm{t-m}} \frac{\Phi_{1-3} + \Phi_{2-3}(z-1)}{z} \left[\left(\frac{273 + t_{\mathrm{w}}}{100} \right)^4 - \left(\frac{273 + t_{\mathrm{ch}}}{100} \right)^4 \right], \tag{1}$$

where the emissivity of the blackbody is equal to $c_0 = 5.67\ \mathrm{W/(m^2{\cdot}K^4)}$. The surrounding medium was considered to be the ideal blackbody, since the surface area of the chamber greatly exceeded the mutual surface of the radiant heat transfer of the bundle. The formulas used for calculating the angular coefficients [1] do not take into account the influence of the longitudinal step of the tubes S_2 in the bundle, since according to [4] this influence is negligibly small and the angular coefficient of the space from the tube bundle to the medium depend virtually entirely on the transverse step S_1. The type of the layout of tubes in a bundle — in-line or staggered — also exerts a little influence on the angular coefficients. The

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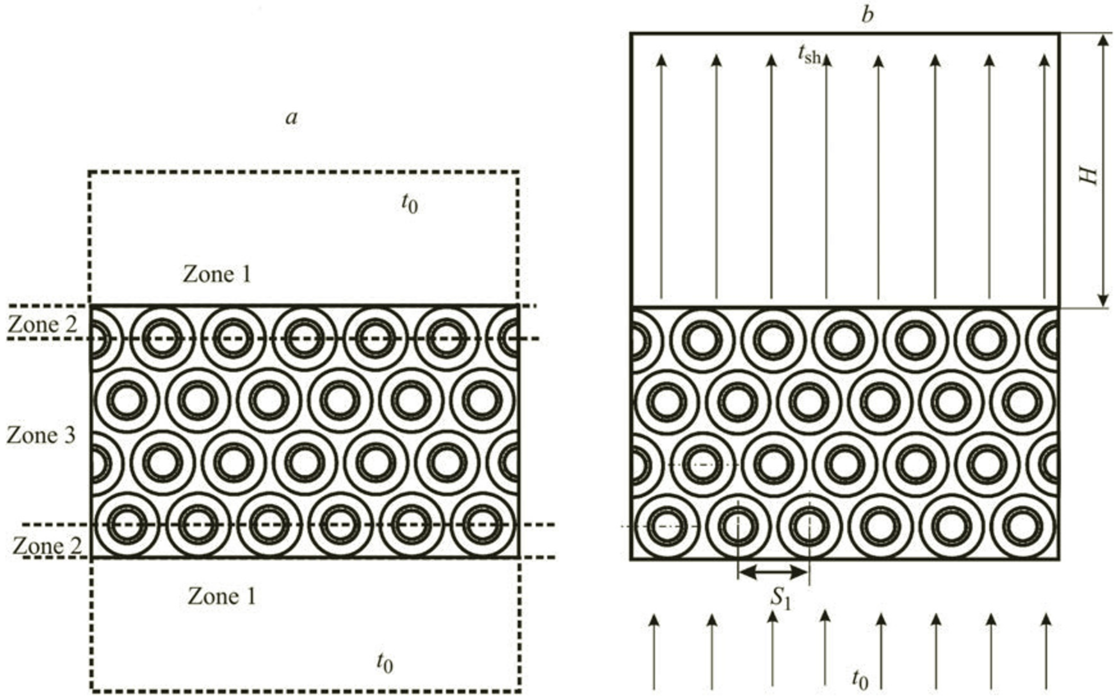

Fig. 1. Scheme of calculation of radiation of a tube bundle without a shaft (a) and with a shaft (b).

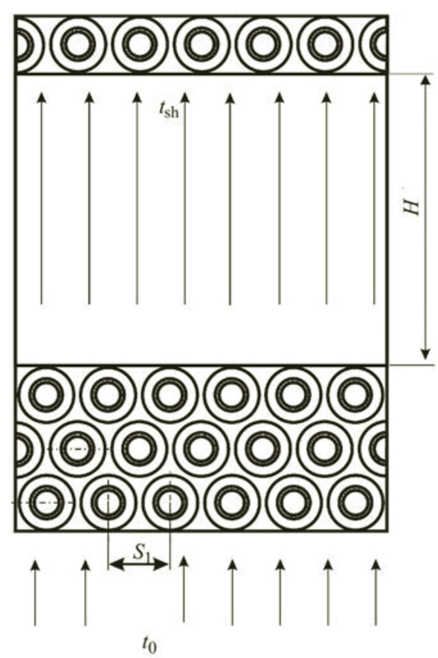

Fig. 2. Scheme of a four-row tube bundle with nonstandard layout.

reliability of this method of calculation was confirmed experimentally in works [5–7]. In four-row bundles with a tube-finning coefficient 16.8, thermal radiation may amount up to 25% in the total heat flux [4]. The main radiation (80–90%) is accomplished by the external halves of the tubes of the extreme transverse rows (Fig. 2).

Unfortunately, the region of temperatures of the surrounding air at which finned tubes can be used under the conditions of free convection is limited. But on being equipped with additional devices that intensify free convection, the

Fig. 3. Experimental setup with nonstandard layout of tubes in a bundle.

heat output of the heat exchanger may remain stable at higher surrounding air temperatures without using electric energy of the fan drive. One of such devices that does not consume electric energy is the exhaust shaft installed above the finned bundle for enhancing the draught of air [8–11].

On mounting the exhaust shaft above the bundle (Fig. 1b), the upper zone (1) becomes the zone of the exhaust shaft with the wall temperature t_{sh}. Assuming in the first approximation that the shaft is an ideal blackbody, we obtain a formula for calculating the radiant heat flux from the bundle of finned tubes with the shaft:

$$Q_{rad} = Q_{rad}^{0} + Q_{rad}^{sh} , \tag{2}$$

$$Q_{rad}^{0} = (0.5 + \gamma)c_0\varepsilon_{ef}F\varphi_{t-m}\,\frac{\Phi_{1-3} + \Phi_{2-3}(z-1)}{z}\left[\left(\frac{273 + t_w}{100}\right)^4 - \left(\frac{273 + t_{ch}}{100}\right)^4\right], \tag{3}$$

$$Q_{rad}^{sh} = (0.5 - \gamma)c_0\varepsilon_{ef}F\varphi_{t-m}\,\frac{\Phi_{1-3} + \Phi_{2-3}(z-1)}{z}\left[\left(\frac{273 + t_w}{100}\right)^4 - \left(\frac{273 + t_{sh}}{100}\right)^4\right], \tag{4}$$

where γ is the coefficient of emission into the shaft hole ($0 \le \gamma \le 0.5$), $\gamma = 0.5S_h/(S_{sh} + S_h)$, and S_h and S_{sh} are the areas of the hole and of the inner walls of the shaft, m^2. It is seen from formula (4) that with the shaft mounted above the bundle, the intensity of bundle radiation decreases (especially for the upper row), since $t_{sh} > t_{ch}$.

In compiling the heat balance it is necessary to take into account that the upper radiant flux that heats the shaft wall transfers the heat to the air in it, while the lower radiant flux is scattered in the surrounding medium:

$$Q_{con} + Q_{rad}^{sh} = c_p\rho V(t_{sh} - t_0) , \tag{5}$$

where the average isobaric heat capacity c_p and the air density ρ are determined at the temperature $0.5(t_0 + t_{sh})$. The thermal power of the bundle can be increased by applying a nonstandard layout — mounting of the upper row of tubes above the exhaust shaft, thus ensuring radiation of the external halves of tubes into the surrounding medium (Fig. 3). But this may lower the temperature in the shaft, which will lead to the lowering of the gravitational draught of the flow and, consequently, to a decrease in the convective heat transfer in the bundle.

Experimental Investigation. The goal of the work was experimental investigation of heat transfer by radiation and convection in a multirow heat-exchanging bundle of finned tubes with a draught shaft with a nonstandard layout of tubes, i.e., when the upper row of the bundle is located above the shaft and the remaining rows are located under the shaft. As an experimental sample, we selected a four-row staggered tube bundle with equilateral layout (intertube step S_1 = 70 mm), as-

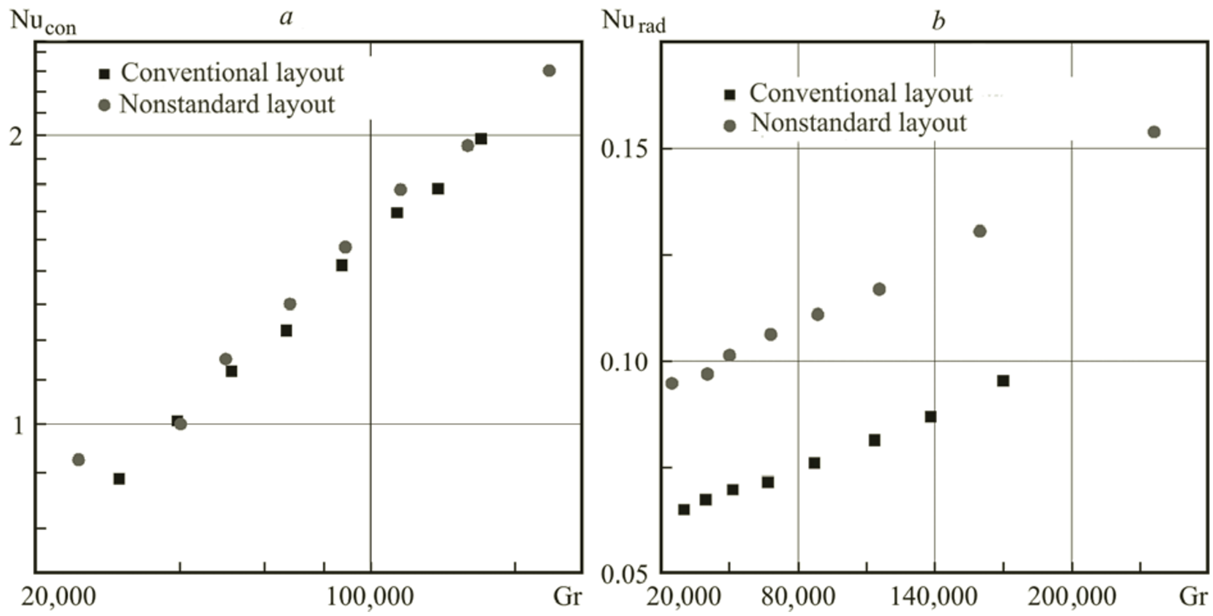

Fig. 4. Dependence of convective (a) and radiative (b) Nusselt numbers on the Grashof number for different types of layout of a four-row bundle.

sembled from finned tubes with the following parameters: diameter of screw finning $d = 0.0568$ m; diameter of the tube base $d_0 = 0.0264$ m; height, step, mean thickness of a fin $h = 0.0152$ m, $s = 0.0042$ m, and $\Delta = 0.00055$ m; heat releasing length of tube finning $l = 0.3$ m; the total length of finned tube $l_1 = 0.33$ m. The finning coefficient of a tube was $\varphi = 21$. The finning envelope was made of aluminum alloy AD1M, and the carrying tube was made from steel. The heat insulated draught shaft of height $H = 0.52$ m had internal rectangular section 0.3×0.42 m, equal to the overall size of the bundle.

The scheme of the experimental setup, its fitting with needed meters, and the method and order of carrying out experiments were elucidated in [12, 13]. To carry out a comparative analysis of the conventional and nonstandard layouts of the heat exchanging bundle, experimental investigations were carried out in two stages. In the first stage, the four-row bundle was installed under the shaft (Fig. 1a), whereas for creating identical conditions of aerodynamic resistance the shaft was covered by a cap with a hole equal to a compressed cross section of the bundle:

$$f_{\text{com}} = lnS_1\left[1-\left(d_0+\frac{2h\Delta}{s}\right)\Big/S_1\right], \tag{6}$$

where the quantity of tubes in one row of the bundle is equal to $n = 6$. In the second stage, a nonstandard layout of the tubes in the bundle was used, i.e., three rows were placed under the shaft and 1 row above it (Fig. 2).

The heat flux removed from the bundle to the air by convection is calculated from the equation

$$Q_{\text{con}} = W - Q_{\text{rad}} - Q_{\text{los}}\ . \tag{7}$$

Using the results of experimental investigations, we calculated average reduced coefficients of heat transfer by convection and radiation related to the total external surface of the tubes:

$$\alpha_{\text{con}} = \frac{Q_{\text{con}}}{(t_{\text{w}}-t_0)F}, \quad \alpha_{\text{rad}} = \frac{Q_{\text{rad}}}{(t_{\text{w}}-t_0)F}, \tag{8}$$

where the area of the heat releasing finned surface of tubes is equal to $F = nl\pi d_0\varphi$.

We processed the experimental data and presented them in a dimensionless form, i.e., the Nusselt number $\text{Nu}_{\text{con}} = \alpha_{\text{con}}d_0/\lambda$, $\text{Nu}_{\text{rad}} = \alpha_{\text{rad}}d_0/\lambda$, and in the Grashof number $\text{Gr} = \beta g d_0^3(t_{\text{w}}-t_0)/\nu^2$, where $\beta = 1/(t_0 + 273)$, $1/^{\circ}\text{C}$.

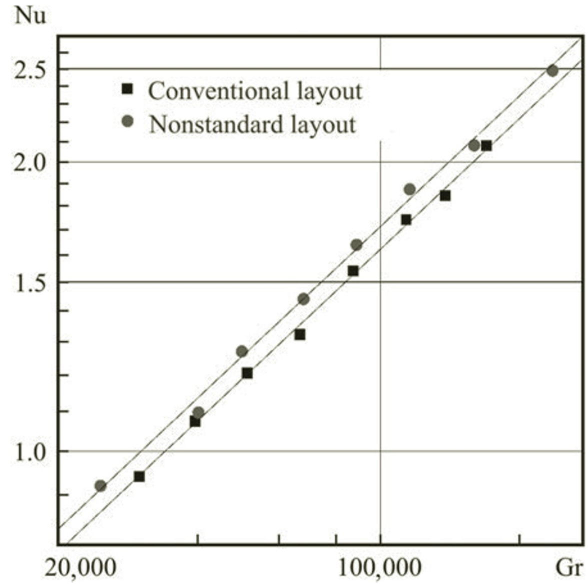

Fig. 5. Dependence of the general Nusselt numbers on the Grashof number for different types of layout of a four-row bundle.

The coefficients of heat conduction λ and kinematic viscosity ν were based on the surrounding medium temperature t_0 = 16–25°C. During experimental investigation of the finned bundle, the average wall temperature at the base of the fins was equal to t_w = 34–180°C.

Figure 4 presents experimental dependences of convective and radiant Nusselt numbers on the Grashof number for conventional and nonstandard layouts of a four-row bundle. As is seen from the figure, while the convective heat transfer is practically identical for conventional and nonstandard layouts, the radiant heat transfer is more intense (by 1.4–1.5 times) for the nonstandard layout. The radiant flux in this case amounts to 4–7% of the total heat flux for the conventional layout and to 6–10% for the nonstandard layout.

Figure 5 presents experimental dependences of the total Nusselt number $\mathrm{Nu} = \mathrm{Nu}_{con} + \mathrm{Nu}_{rad}$ on the Grashof number for conventional and nonstandard layouts of a four-row bundle.

As a result of the generalization of experimental data with deviation not exceeding 3%, we obtain the equation

$$\mathrm{Nu} = C\mathrm{Gr}^{0.45} , \tag{9}$$

where the proportionality factor C = 0.00913 for the conventional layout and C = 0.00964 for the nonstandard one.

Thus, an increase in the total heat transfer in the case of nonstandard layout of the bundle by about 5% is obtained.

Conclusions. Optimal (nonstandard) disposition of finned tubes of a multirow heat exchanging bundle allows one to increase the radiant heat flux 1.4–1.5 times. It is worthwhile to optimize the layout of a multirow heat-exchanging bundle when the fraction of the radiant flux in the total heat flow amounts to more than 10%.

NOTATION

c_0, blackbody radiation coefficient, c_0 = 5.67 W/(m^2·K^4); c_p, mean isobaric heat capacity of air, J/(kg·K); d, outer diameter of tube finning, m; d_0, finning diameter at the base, m; F, area of heat releasing finned surface of the bundle, m^2; f_{com}, compressed transverse section of the bundle, m^2; Gr, Grashof number; g, free fall acceleration, m/s^2; H, height of the draught shaft, m; h, height of tube fins, m; l, length of heat releasing finned part of the tube, m; l_1, total length of finned tube, m; Nu, Nusselt number; n, number of heated tubes in the first row of the bundle; Q_{los}, heat losses through the ends of tubes and current leads, W; Q_{con} and Q_{rad}, convective and radiant heat fluxes, W; S_1, transverse step of the disposal of tubes in a bundle, mm; s, step of the tube fins, m; t, determining temperature of air at the inlet into the bundle, °C; t_{ch}, temperature of the chamber surface, floor, and ceiling around the bundle, °C; t_w, temperature of the wall at the base of the fins, °C; t_{sh}, temperature of the walls of the draught shaft, °C; t_0, temperature of surrounding air in the chamber, °C; V, air flow rate in the bundle, m^3/s; W, heat power supplied to the bundle, W; z, number of rows in the bundle; α_{con} and α_{rad}, mean reduced convective and radiative heat transfer coefficients related to the total external surface, W/(m^2·K); β, coefficient of temperature expansion, K^{-1}; Δ, average thickness of a fin, m; ε_{ef}, effective emissivity of the "bundle–medium" system; λ, thermal conductivity coefficient of air, W/(m·K); ν, coefficient of kinematic viscosity of air, m^2/s; ρ, average density of air in the bundle, kg/m^3; Φ_{13} and Φ_{23}, resolving angular coefficients of radiation from the first zone to the third and from the second zone to the third; φ, coefficient of tube finning; φ_{t-m}, average emission view factors of a single tube with rounded fins to the surrounding medium with account for the geometric parameters of tube finning.

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