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STUDY OF FREE-CONVECTIVE HEAT EXCHANGE OF AIR-COOLABLE FINNED TUBE BUNDLES INTENSIFIED BY EXHAUST SHAFT

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The results of experimental studies of free-convective heat transfer of single-, double- and four-row bundles of finned tubes without and with an exhaust shaft are presented. It is shown that the inter-tube spacing significantly affects the energy efficiency of the bundle. The optimum inter-tube spacing is 61 mm for single-row, about 70 mm for double-row, and over 70 mm for four-row bundles. With increase of the number of rows in the bundle the specific capacity of the bundle increases, but the rate of its increase decreases. The specific heat capacity of a single-row bundle with an inter-tube spacing of 58 mm is higher than with an inter-tube spacing of 64 mm. The specific heat capacity of two- and four-row bundles is less at low exhaust shaft heights, but it is more when the relative shaft height is more than 0.14 and 0.26, respectively. The heat transfer of the bundles can be increased (more than threefold) using optimal ratio of the outlet section of the shaft and the flow capacity of the tube.

Keywords: air-coolable devices, bimetallic finned tube, staggered tube bundle, free-convective heat transfer, exhaust shaft.

Air-coolable heat exchangers (ACH) are in high demand in petrochemical, gas, and food industries, electronic engineering, refrigeration, etc. because of their eco-friendliness and operation without cooling water [1].

For ACH operation it is important to ensure steady flow of the cooling air without electric power consumption for driving the fans. The range of temperatures of the surrounding air at which ACH could be used under free convection conditions is limited. However, high heat efficiency of the heat exchanger in a wide range of temperatures of the surrounding air can be ensured by installing an exhaust shaft above the ACH.

Aside from the types and parameters of the heat exchanger tubes, dimensions and design parameters of the ACH, flow section, and exhaust shaft height also markedly affect the energy efficiency of a system consisting of an ACH and an exhaust shaft. The results of the experimental and numerical studies of the ACH built of finned tubes with an exhaust shaft are furnished in [2–7].

The goal of this work was experimental study and generalization of the data on convective heat transfer from horizontal bundles of finned tubes to vertical air flows generated by an exhaust shaft of controllable height.

Single-, two- and four-row of bundles of horizontally placed finned tubes (six tubes in a row, i.e. $n = 6$) with spirally rolled aluminum fins (inter-tube spacings $S_1 = 58, 64, \text{and } 70$ mm) were studied.

Two- and four-row bundles consisted of even-spaced staggered tubes (six tubes in odd row and five tubes and two half-tubes in even row). The tubes are installed in holes of 4-mm thick plywood boards and the ends of the tubes are shielded by fluoroplastic plugs. The following are the geometric dimensions of the bimetallic finned tube (mm): diameter of finning $d = 56.8$, diameter of tube at the base $d_0 = 26.4$, fin height $h = 15.2$, fin spacing $s = 2.43$, average fin thickness $\Delta = 0.55$, length of the finned section of the tube $l = 300$. The finning factor $\varphi = 21$.

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To eliminate the influence of air circulation in the premises on the experimental results during the studies, the tube bundle was placed in a $0.8 \times 0.8 \times 1.0$ m chamber closed along the perimeter and open at the top and bottom for free air movement.

An exhaust shaft with a rectangular base that turns through a reducer into a cylindrical tube of diameter $d_{sh} = 0.105$ m is installed above the experimental tube bundles for intensifying air flow motion. The shaft height (controllable) $H = 0.52$, 1.16, 1.48, and 2.12 m. To reduce heat losses, the exterior of the shaft is insulated with a 0.02–0.03-m thick layer of mineral fiber.

The heat transfer was studied by the full modeling method. The finned tubes were heated by inserted thermoelectric heaters. The central tube in the bundle was a calorimeter (design of the tube-calorimeter and its temperature sensors are described in [8]).

The design of the experimental unit with its measuring instruments, the method of study and the order of execution of the experiments are described in [9]. In the experiments, the temperature of the tube-calorimeter surface at the base of the fins (arithmetic-mean temperature as per thermocouple readings) varied in the range T_{amt} = 30–195 °C, the temperature of the air in the chamber t_0 = 17–27 °C, the mean temperature in the shaft t_{sh} = 22–125 °C, and the electric power supplied to the calorimeter W = 10–250 W.

The average reduced convective coefficient of heat transfer referred to the full external surface of the tube, W/(m^2·K):

$$\alpha_c = \frac{Q_c}{(t_{amt} - t_0)\,F}, \tag{1}$$

where Q_c is the convective heat flow, W, $F = \pi l d_0 \varphi$ is the area of the heat transferring finned surface of the tube, m^2.

The heat flow Q_c (W) transferred from the tube to the air by convection is calculated by the equation

$$Q_c = W - Q_r - Q_l, \tag{2}$$

where W is electric power supplied to the calorimeter, W, Q_r is the heat flux removed from the tube by radiation, W [10], Q_l are the heat losses through the tube ends and the current conductors, W [11].

The experimental results are presented as dependencies of the Nusselt number on the Raleigh number:

$$\mathrm{Nu} = \alpha_c d_0 / \lambda, \tag{3}$$

$$\mathrm{Ra} = \frac{g\beta d_0^{\,3}(t_{amt} - t_0)}{\nu a}, \tag{4}$$

where λ is the coefficient of thermal conductivity, W/(m·K), ν is the coefficient of kinematic viscosity, m^2/sec a is the coefficient of thermal diffusivity, m^2/sec, g is the free fall acceleration, m/sec^2, β is the coefficient of thermal expansion, K^{-1}.

The Nusselt and Raleigh numbers as well as the thermophysical properties of the air were determined at the temperature of the surrounding air t_0, °C.

The relationships $\mathrm{Nu} = f(\mathrm{Ra})$ for single- and four-row tube bundles with inter-tube spacing S_1 = 58, 64, and 70 mm without an exhaust shaft (free convection) and with an exhaust shaft of height H = 0.52–2.12 m are shown in Fig. 1.

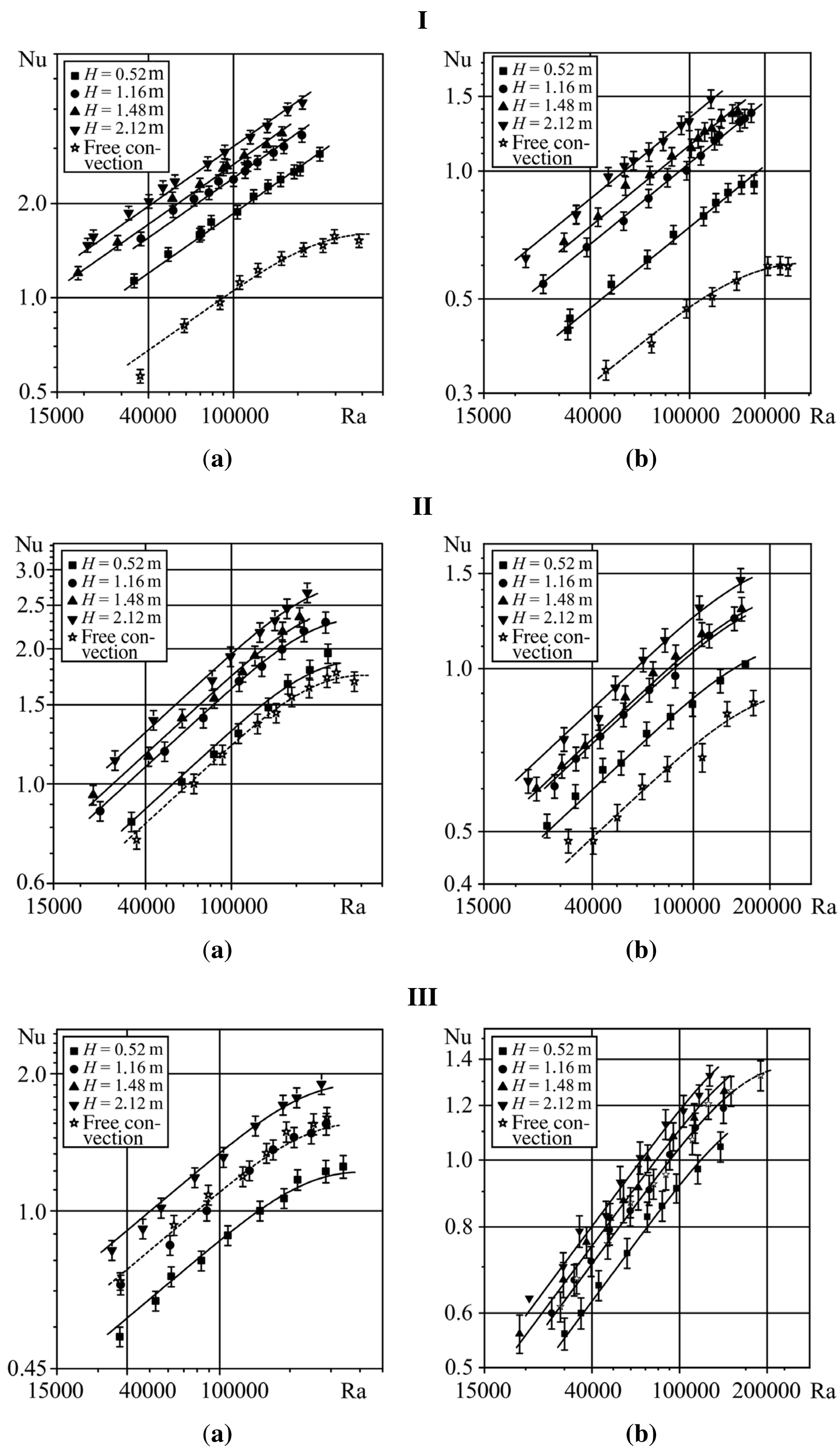

Fig. 1. Relationships of Nusselt number Nu with Raleigh number Ra for various exhaust shaft heights H for single-row (a) and four-row (b) finned tube bundles with inter-tube spacing: I — $S_1 = 58$ mm, II — $S_1 = 64$ mm, III — $S_1 = 70$ mm.

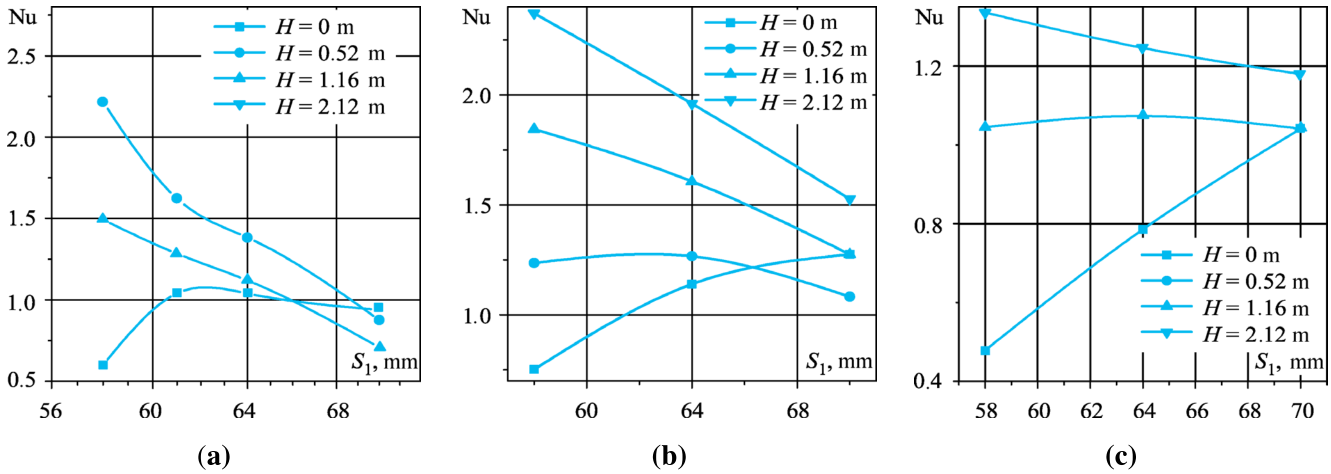

Fig. 2. Relationships of Nusselt number with transverse tube spacing S_1 at various shaft heights H for single-row (a), two-row (b), and four-row (c) finned tube bundles at constant Raleigh number $\mathrm{Ra} = 10^5$.

It is evident from Fig. 1 that, after installation of a shaft, heat transfer of the tube bundles with inter-tube spacings $S_1 = 58$ and 64 mm increases markedly (twofold and more) regardless of the number of rows of tubes in the bundle.

After installation of a shaft with height $H = 0.52$ m, heat transfer of bundles with inter-tube spacing $S_1 =$ 70 mm decreased 1.27 times for single-row and 1.13 times for four-row bundles, i.e., more the number of rows in the bundle, the less the decrease of heat transfer. For a single-row bundle, this fact is explained [3] by low resistance of the bundle in comparison with the resistance of the shaft.

Evidently, resistance of the bundle increases with increase of rows, but it does not exceed the resistance of the shaft.

The relationships of Nusselt number with the transverse spacing S_1 at various shaft heights H for single-row (a), two-row staggered (b), and four-row staggered (c) finned tube bundles at constant Raleigh number $\mathrm{Ra} = 10^5$ are shown in Fig. 2.

Under free convections conditions (without an exhaust shaft) increase of inter-tube spacing facilitates increase of convective heat transfer of the bundle up to a certain limit. The optimum inter-tube spacing (S_1) is 61 mm for single-, about 70 mm for two-row, and over 70 mm for four-row finned bundles. If an exhaust shaft is installed above the bundle, an increase in inter-tube spacing (decrease in aerodynamic resistance of the bundle) leads to a decrease in heat transfer, and fewer the rows, greater the influence of spacing. Heat transfer decreases twofold and more with increase of inter-tube spacing from 58 mm to 70 mm.

For the four-row bundle with an exhaust shaft of height $H = 1.16$ m, change in inter-tube spacing has a negligible effect on heat transfer, and at $H = 0.52$ m, heat transfer increases by 1.24 times with increase of the inter-tube spacing.

For generalization and analysis of heat transfer processes, precise determination of heat transfer coefficients is essential. For this purpose, criterial equations with physically validated relative dimensions of the investigated object and the determining temperature of the experimental medium (air) are used.

For an exhaust shaft mounted above a heat exchanger tube bundle, the following dimensionless parameters are proposed: $\chi_{sh} = f_{hole}/f_{com}$ is the ratio of narrowing of the outlet section of the shaft f_{hole} to the compressed section of the bundle f_{com}; $H_e = H/d_e$ is the equivalent shaft height (ratio of actual shaft height H to equivalent diameter d_e of the bundle).

Use of the parameter H/d_e is not physically validated because the air flow velocity through a bundle increases with increase of the height H and the equivalent diameter d_e, i.e., the dependence of the coefficient of heat transfer on the parameter H/d_e is ambiguous.

To get dimensionless physically validated parameter of the *bundle–shaft* system, let us use the equation for air flow in the exhaust shaft [12]:

$$\frac{Hg\theta}{w_\mathrm{hole}^2} = \mathrm{Eu_b}\frac{\rho_\mathrm{sh}}{\rho_\mathrm{b}}\chi_\mathrm{sh}^2 + \mathrm{Eu_{fr}} + \mathrm{Eu_{vort}} - \frac{\psi}{2}\theta, \tag{5}$$

where $\theta = (\rho_0 - \rho_\mathrm{sh})/\rho_\mathrm{sh}$ is the relative difference of air densities, ρ_0, ρ_sh, ρ_b are the average density of air in the environment, in the shaft, and in the bundle, respectively, kg/m^3, w_hole is the air velocity in the hole of the outlet section of the shaft, m/sec, $\mathrm{Eu_b}$ is the Euler number of the tube bundle, $\mathrm{Eu_{fr}} = \lambda_\mathrm{fr}(H - h_\mathrm{dif})/2d_\mathrm{sh}$ is the Euler number of friction of the air flow in the shaft, $\mathrm{Eu_{vort}} = 5.5(\chi_\mathrm{hole} - 0.03)/(\chi_\mathrm{hole} + 0.03)$ is the Euler number of vortex of the flow in the shaft, $\psi = 1.3$ is the coefficient of formation of a heated air region above the shaft, h_dif is the height of the diffuser above the bundle, λ_fr is the coefficient hydraulic friction, $\chi_\mathrm{hole} = f_\mathrm{hole}/f_\mathrm{front}$ is the ratio of the area of internal cross section of the shaft $\left(f_\mathrm{hole} = \pi d_\mathrm{sh}^2/4\right)$ to the area of the front section of bundle $(f_\mathrm{front} = S_1 nl)$.

The equation for the air flow velocity in the hole of the outlet section of the shaft was obtained by using flow rate and heat transfer equations:

$$\begin{aligned} w_\mathrm{hole} &= \frac{G_v}{f_\mathrm{hole}} = \frac{Q^\mathrm{c} + Q^\mathrm{r}_\mathrm{sh}}{f_\mathrm{front}\chi_\mathrm{hole}c_p(t_\mathrm{sh} - t_0)} = \frac{\alpha_\mathrm{c+r}F(t_\mathrm{amt} - t_0)}{f_\mathrm{front}\chi_\mathrm{hole}c_p(t_\mathrm{sh} - t_0)} \\ &= \frac{\alpha_\mathrm{c+r}d_0}{\lambda}\cdot\frac{\lambda}{c_p d_0}\cdot\frac{F}{f_\mathrm{front}\chi_\mathrm{hole}}\cdot\frac{t_\mathrm{amt} - t_0}{t_\mathrm{sh} - t_0} \\ &= \mathrm{Nu_{c+r}}\cdot\frac{a}{d_0}\cdot\frac{F}{f_\mathrm{front}\chi_\mathrm{hole}}\cdot\frac{1}{\theta_t}, \end{aligned} \tag{6}$$

where G_v is the volume rate of air flow through the shaft, m^3/sec, Q^c is the heat transferred from the bundle to the air by convection, W, Q^r_sh is the heat transferred from the bundle to the shaft by radiation, W [10], $\alpha_\mathrm{c+r}$ is the coefficient of heat transfer from the finned surface of the bundle to the shaft by radiation and to the air by convection, W/(m^2·K), c_p is the isobaric volume heat capacity of the air, J/(m^3·K), $a = \lambda/c_p$ is the coefficient of thermal diffusivity of the air, m^2/sec, $\theta_t = (t_\mathrm{sh} - t_0)/(t_\mathrm{amt} - t_0)$ is the relative temperature difference in the bundle.

Then

$$\begin{aligned} \frac{Hg\theta}{w_\mathrm{hole}^2} &= \frac{Hw_\mathrm{hole}\theta}{w_\mathrm{hole}^2\tau}\cdot\chi_\mathrm{sh}\cdot\frac{\rho_\mathrm{sh}}{\rho_\mathrm{b}} = \frac{H}{w_\mathrm{hole}\tau}\cdot\chi_\mathrm{sh}\cdot\frac{\rho_\mathrm{sh}}{\rho_\mathrm{b}}\cdot\theta \\ &= \frac{d_0^2}{\mathrm{Nu_{c+r}}a\tau}\cdot\frac{Hf_\mathrm{front}\chi_\mathrm{hole}\chi_\mathrm{sh}}{d_0 F}\cdot\theta\cdot\theta_t\cdot\frac{\rho_\mathrm{sh}}{\rho_\mathrm{b}} \end{aligned}$$

$$= \frac{\mathrm{Pr}}{\mathrm{Nu}_{c+r}\mathrm{Fo}} \cdot \frac{Hf_{\mathrm{front}}\chi_{\mathrm{hole}}\chi_{\mathrm{sh}}}{d_0 F} \cdot \theta \cdot \theta_t \cdot \frac{\rho_{\mathrm{sh}}}{\rho_{\mathrm{b}}}, \tag{7}$$

where $g = w_{\mathrm{b}}/\tau = (w_{\mathrm{hole}}/\tau) \cdot \chi_{\mathrm{sh}} \cdot (\rho_{\mathrm{sh}}/\rho_{\mathrm{b}})$, τ is the typical duration of acceleration of the mobile surrounding air to the velocity in the compressed section of the bundle, sec, $\mathrm{Fo} = \nu\tau/d_0^2$ is the dynamic Fourier number, $\mathrm{Pr} = \nu/a$ is the Prandtl number.

The ratio of the heat exchange area F and the area of the front section f_{front} of the bundle

$$\frac{F}{f_{\mathrm{front}}} = \frac{nz\varphi \cdot \pi d_0 l}{nS_1 l} = \frac{z\varphi \cdot \pi d_0}{S_1}. \tag{8}$$

As a result,

$$\frac{Hg\theta}{w_{\mathrm{hole}}^2} = \frac{\mathrm{Pr}}{\mathrm{Nu}_{c+r}\mathrm{Fo}} \cdot \frac{H\chi\chi_{\mathrm{sh}}^2 S_1}{d_0 z\varphi \cdot \pi d_0} \cdot \theta \cdot \theta_t \cdot \frac{\rho_{\mathrm{sh}}}{\rho_{\mathrm{b}}}, \tag{9}$$

where $\chi_{\mathrm{hole}} = \chi\chi_{\mathrm{sh}}$, $\chi = f_{\mathrm{com}}/f_{\mathrm{front}}$ is the coefficient of bundle compression (ratio of the area of the compressed section to the area of the front section of the bundle), $z = 1, 2, 4$ is the number of rows of tubes in the bundle, $f_{\mathrm{com}} = l \cdot n \cdot (S_1 - d_0 - 2h\Delta/s)$ is the area of the compressed section of the bundle, m^2.

Thus, a final equation that shows the correlation of hydrodynamics and heat exchange in the *bundle–shaft* system was derived:

$$H_{\mathrm{rel}} K \cdot \theta \cdot \theta_t = \mathrm{Nu}_{c+r}\mathrm{Fo}\,\mathrm{Pr}^{-1}\,\mathrm{Eu}_{\text{b-sh}}, \tag{10}$$

where

$$\mathrm{Eu}_{\text{b-sh}} = \mathrm{Eu}_{\mathrm{b}} + \left(\mathrm{Eu}_{\mathrm{fr}} + \mathrm{Eu}_{\mathrm{vort}} - \frac{\psi}{2}\theta\right) \Big/ \left(\frac{\rho_{\mathrm{sh}}}{\rho_{\mathrm{b}}}\chi_{\mathrm{sh}}^2\right)$$

is the Euler number of the *bundle–shaft* system which relates to the velocity in the bundle and depends on the coefficients of hydraulic resistance in the tube bundle and the shaft, $H_{\mathrm{rel}} = H/(d_0\varphi\pi)$ is the relative height of the shaft, $\pi d_0 \varphi$ is the length of the circumference of the flat tube with a surface area equal to the surface area of the finned tube, m, $K = S_1\chi/(zd_0)$ is the coefficient of flow capacity of the bundle.

Tube bundles with different numbers of rows cannot be correctly compared in terms of Nusselt number, i.e., without consideration of the difference in the heat exchange surface.

Therefore, the dependencies of the specific thermal capacity of the bundle $\mathrm{Nu}/K = \mathrm{Nu} \cdot zd_0/(S_1\chi)$ on the relative shaft height H_{rel} for single-row $(z = 1)$, two-row $(z = 2)$, and four-row $(z = 4)$ finned tube bundles with inter-tube spacing $S_1 = 58, 64$, and 70 mm at constant Raleigh number $\mathrm{Ra} = 10^5$ are shown in Fig. 3.

As evident from Fig. 3b, after installation of an exhaust shaft, the specific heat capacity of bundles with an inter-tube spacing $S_1 = 70$ mm initially decreases (because of high aerodynamic resistance of the shaft compared to the resistance of the bundles), but later increases with the increase in height of the shaft H_{rel} irrespective of the number rows of tubes, and the values of the heat capacity of bundles with $S_1 = 70$ mm are lower than those of the bundles with smaller spacings (in the entire range of exhaust shaft heights).

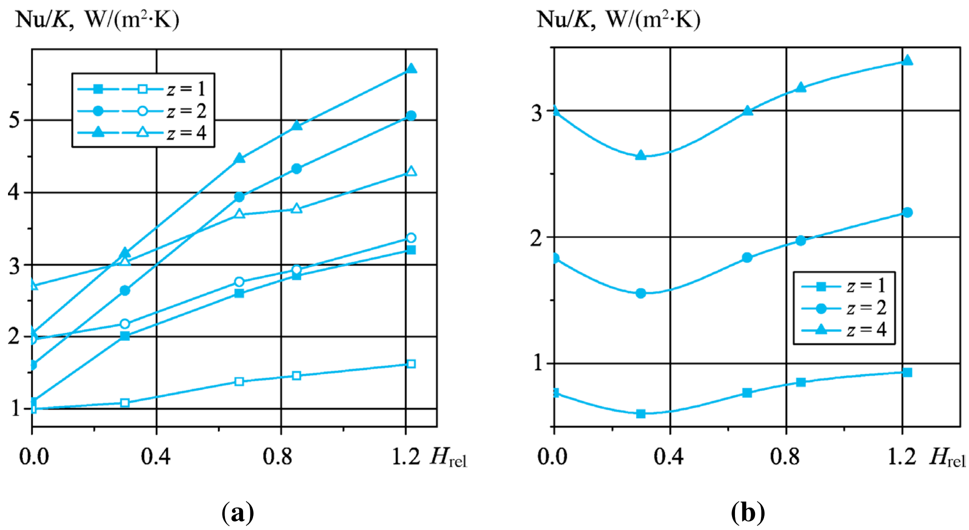

Fig. 3. Dependencies of specific thermal capacity Nu/K on relative shaft height H_{rel} for single-row (z = 1), two-row (z = 2), and four-row (z = 4) finned tube bundles with inter-tube spacing S_1 = 58 mm (dark signs) and 64 mm (light signs) (a) and S_1 = 70 mm (b) at constant Raleigh number $\mathrm{Ra} = 10^5$.

The specific heat capacity of a single-row bundle with a spacing S_1 = 58 mm is higher than that of a bundle with S_1 = 64 mm. The specific heat capacity of two- and four-row bundles is less than that of single-row bundles at lower exhaust shaft heights, but is higher at relative shaft height H_{rel} of more than 0.14 (at $z = 2$) and more than 0.26 (at $z = 4$).

The specific heat capacity of a bundle increases in proportion to the increase in the number of rows of tubes in the entire range of inter-tube spacings S_1.

The change in Nu number for multi-row bundles is expressed by the equations (discrepancy ±11%):

$$\frac{\mathrm{Nu}_{z=2}}{\mathrm{Nu}_{z=1}} = 1.04\frac{S_1}{d_0} - 1.52,$$
$$\frac{\mathrm{Nu}_{z=4}}{\mathrm{Nu}_{z=1}} = 1.22\frac{S_1}{d_0} - 2.26. \qquad (11)$$

CONCLUSIONS

In the experimental studies of single-, two-, and four-row bundles of finned tubes without and with an exhaust shaft, it was found that inter-tube spacing markedly affects the energy efficiency of the tube bundle and an optimum inter-tube spacing corresponds to each exhaust shaft height level.

The specific heat capacity increases with increase in the number of rows of tubes in the bundle, but the rate of increase of specific capacity diminishes.

The heat transfer of the bundle can be increased considerably (three times and more) by increasing the shaft height, but only at an optimum ratio of the outlet section of the shaft and the coefficient of the flow capacity of the bundle.

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