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Numerical Study of Flow and Heat Transfer in a Single-Row Bundle of Horizontal Finned Tubes Under Conditions of Air Thermogravitational Convection

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Abstract. The results of numerical modeling of airflow and heat transfer in a horizontally oriented single-row bundle consisting of six finned heated tubes under thermogravitational conditions are presented. The study is performed for the bundles with a compact tube arrangement and varying fin pitch. The cases without and with the rectangular exhaust shafts of different heights installed above the bundle have been investigated. For the most compact bundle configuration in the absence of a shaft in the considered range of moderate values of the Grashof number (up to 5.5×10^5), a good agreement was obtained between the calculated integral Nusselt number at the tube surface and experimental data taken from the literature. The combination of the fin pitch (spatial arrangement on the tubes) and the shaft height determined in computations allowed to obtain the optimal bundle design with the highest intensity of heat removal.

Keywords: air-cooling, heat exchangers, tube bundles, finned tubes, fin pitch, exhaust shaft, thermogravitational convection, numerical simulation.

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Introduction

Air-cooled heat exchangers that operate in a passive mode, devoid of forcing mechanisms, under natural convection circumstances are extensively utilized in practice, ensuring process

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dependability and cost-effectiveness [1]. The most commonly used heat exchangers consist of tube bundles with external fins.

The intensity of heat transfer from the tube surfaces is influenced to a large extent by thermal conditions and the geometric characteristics of the bundle, including its design as well as the shape and positioning of the fins. The efficiency of heat exchangers can be noticeably enhanced by placing an exhaust shaft above it, which leads to strengthening the flow adjacent to the tube surfaces. The challenge of choosing the best finning configuration and tube arrangement in bundles is always relevant due to the large range of technologies and operational conditions.

In recent decades, the long-dominant analytical approaches have been largely replaced by experimental and numerical studies on the problem of horizontally oriented finned tubes in the free convection mode using up-to-date methods and tools [1]. The related research has been mainly focused on single tubes with different geometric configurations and thermal conditions.

A number of experimental studies were aimed at investigations of the influence of the diameter of carrying tubes and fins, the number of fins, fin spacings and shapes, and the operating Rayleigh number on integral heat transfer rate (see, for example, [2, 3]). Based on the measurement results, generalized correlations have been proposed for calculating the mean Nusselt number depending on the geometric parameters and the Rayleigh number. Only a few experiments have studied features of local heat transfer in the inter-fin gaps with varying fin diameter, shape and spacing [4, 5].

A series of thorough experimental studies were devoted to tube bundles consisting of industrial finned tubes located in a free space (for example, [6–8] and links in them). The influence of the temperature difference between the tube base and ambient air, the longitudinal and transversal distances between tubes, the number of rows in the bundles, and the finning factor on the mean Nusselt number was analyzed.

The results of relatively recent experimental investigations for tube bundles with different numbers of rows and with an exhaust shaft are presented, in particular, in [9, 10]; the effects of the Rayleigh number, inter-tube and inter-fin spacing, the finning factor and shaft height on the mean Nusselt number and some local heat transfer characteristics were explored.

Numerical studies of free-convective heat transfer from a single horizontal finned tube have become relatively widespread since the 2010s. The calculations were based mainly on simplified spatial settings and stationary conditions; the values of the regime criteria and the geometric parameters of the tubes and fins were varied in order to discover optimal design with the best heat transfer properties [11–13].

The work [14] contains some key results of unsteady computations of laminar free convective flow and heat transfer in a single-row tube bundle, performed at a modern level on fairly detailed grids with varying transversal tube pitch and the temperature difference between the carrying tube and ambient air, performed under conditions close to the experiments [8]. In addition to comparison with the experiments on the mean Nusselt number, special attention has been paid to local features of the velocity and temperature fields and their influence on the integral characteristics.

The results of a numerical study of the influence of an exhaust shaft on the flow structure, local and integral heat exchange during unsteady thermogravitational airflow through a double-row bundle of finned tubes are given in [15]. Relevant experimental data on the values of the mean Nusselt number have been obtained in [10], showing that the installation of the shaft entails an increase in the intensity of heat removal up to three times.

Thus, numerical studies that systematically model the flow and heat transfer in tube bundles with an emphasis on subtle local effects with a joint assessment of the influence on processes of such parameters as the inter-fin distance, thermal conditions, and the presence of an exhaust shaft of different heights above the bundle are very limited.

The present paper is focused on investigations of thermogravitational airflow near a horizontal single-row tube bundle consisting of six heated finned tubes at moderate values of the Grashof number (up to 5.5×10^5). Numerical modeling of the airflow and heat transfer is carried out for the cases with different geometrical configurations of the tube bundles (various fin pitches) and different heights of a rectangular exhaust shaft that was installed above the bundles. For the case without a shaft and with the most compact configuration of the bundles, the comparison with experimental data [8] on integral heat transfer rate in the considered range of the Grashof number values is performed.

1. Problem formulation and computational aspects

1.1. Geometrical model

The computational domain for the problem examined is presented in Fig. 1a,b. The single-row tube bundle consists of six horizontally oriented identical tubes. It is assumed that the bundle is unlimited in the axial direction, and the flow is periodic in the z-direction with the fin pitch. Additionally, the flow is considered symmetrical with respect to the middle plane of the inter-fin gap.

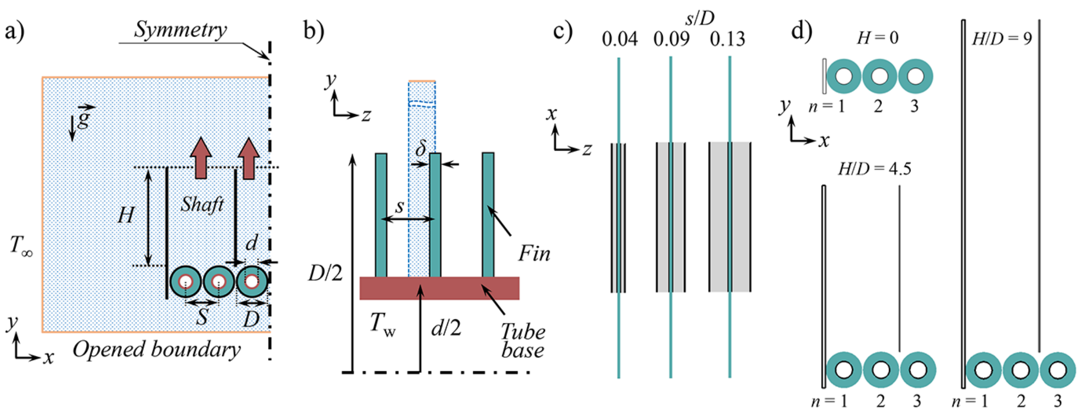

Fig. 1. The computational domain shown in two planes: a) z = Const and b) x = Const; the cases with different geometrical configurations of the tube bundle, variable parameter: c) the fin pitch (s/D) and d) the shaft height (H/D)

The following geometrical parameters of the tube bundle were set. The external diameter of the carrying tube was equal to $d = 26.4$ mm, the diameter of the solid aluminum fins was $D = 2.15d$, the transversal distance between tubes S corresponded to the dimensionless parameter $\sigma = S/D = 1.02$, and the fin thickness $\delta = 0.01D$. The fin pitch along the tube s was varied from $s_0 = 0.04D$ (the most compact configuration of the bundle) to $0.13D$ (Fig. 1c). Calculations with different tube bundle fashions were carried out both for the cases with a rectangular exhaust shaft installed above the bundle (the height H was equal to $4.5D$ and $9D$) and for the cases without a shaft ($H = 0$). Schemes of the computational domain for the cases with different values of H are shown in Fig. 1d. The shaft contains a plane wall (barrier) located above the tubes with the numbers $n = 2$ and $n = 3$ (n is counted from the bundle side) to prevent the large-scale upward airflow.

The computational setup includes a side plate that is located near the bundle and prevents global airflow movement in the x-direction (Fig. 1d, case $H = 0$). The width and the height of the plate were about $D \times 0.09D$. The boundaries far from the tube bundle are located at least at a distance of $6D$ from the bundle and the shaft.

Note that the geometrical configuration with the most dense arrangement of the fins ($s = s_0$) without a shaft is close to that considered in the experimental research [8].

1.2. Boundary conditions

The following conditions were set for the problem at the boundaries of the computational domain. The boundaries far from the tube bundle were open; the ambient air temperature was imposed to $T_0 = 22.1°C$ on them. The temperature of the tube base, T_w, was assumed to be constant and varied from 37 to 244°C. The corresponding values of the relative temperature difference (the buoyancy parameter) $\varepsilon_T = (T_w - T_0)/T_0$ varied from 0.05 to 0.7. The values of buoyancy velocity estimated as $V_b = (g\varepsilon_T d)^{0.5}$ were in the range 12 ... 44 cm/s. The values of the Grashof number $Gr = (\rho V_b d/\mu)^2$ varied from 3.7×10^4 to 5.5×10^5, and the dynamic viscosity μ and density ρ of air were taken at the ambient temperature T_0. The values of the Rayleigh number $Ra = Gr{\cdot}Pr$ were in the range of 2.6×10^4 to 3.9×10^5, and the Prandtl number was $Pr = 0.7$. The given values indicate that the airflow near the tube bundle was laminar.

The side plate and the shaft with the barrier (Fig. 1d) were considered to be thermally insulated; at these boundaries the no-slip conditions were set.

1.3. Mathematical model and computational aspects

The numerical simulation is carried out on the basis of the Navier–Stokes system of governing equations, written for a perfect viscous gas with variable physical properties. The problem was solved in a conjugate formulation: heat transfer along the fins of the tubes was calculated together with the flow.

The unstructured grids consist of hexahedral elements that clustered to the wall surfaces were used. The size of the grids was varied from 140 ($H = 0$) to 300 ($H = 9D$) thousand cells. Automatic generation of computational grids for the cases with various geometrical parameters was organized using ANSYS Meshing. Computations have been performed using the CFD package ANSYS Fluent. The spatial discretization was done with the second-order upwind scheme for convective terms, and also the second-order pressure interpolation method was assigned.

The second-order implicit time integration scheme was applied. The time step was equal to 0.02 s, and it was chosen to provide the values of the Courant number less than 1. The duration of the processed sample, related to the statistically steady flow regime, was about 120 s, which for all the cases provided at least 540 characteristic times, estimated as $t_s = d/V_b$.

2. Results and discussion

2.1. Airflow structure inside the tube bundle: influence of the fin spacing

Typical airflow structure that forms near the tube bundle is shown in Fig. 2a using visualizations of the dimensionless velocity V/V_b and temperature $T^* = (T - T_0)/(T_w - T_0)$ fields in the middle section of the inter-fin space for the most compact geometrical configuration ($s = s_0$) and at $\varepsilon_T = 0.7$ ($Gr = 5.5{\times}10^5$). Note that for all the cases considered, the flow near the tube bundle is significantly unsteady. The thermal plume formed above the bundle periodically (approximately once every half of the time period, t_s) detached from the surface of the tubes. The fields and integral characteristics of heat transfer presented and discussed in the paper are averaged in time.

Under the action of buoyancy forces, the air below the bundle moves towards the heated tubes with area-averaged velocity values of about $V_{in} = 0.09V_b$ (the corresponding value of the Reynolds number $Re = \rho V_{in} d/\mu = 68$), passes through the inter-fin space of the tubes, where the values of velocity locally reach $0.4V_b$, and the inter-tube gap with the higher values of about $1.2V_b$. Above the tube bundle, the area of the increased velocity and temperature values take place, reflecting the presence of the thermal plume. In the inter-fin space, the temperature field is almost uniform.

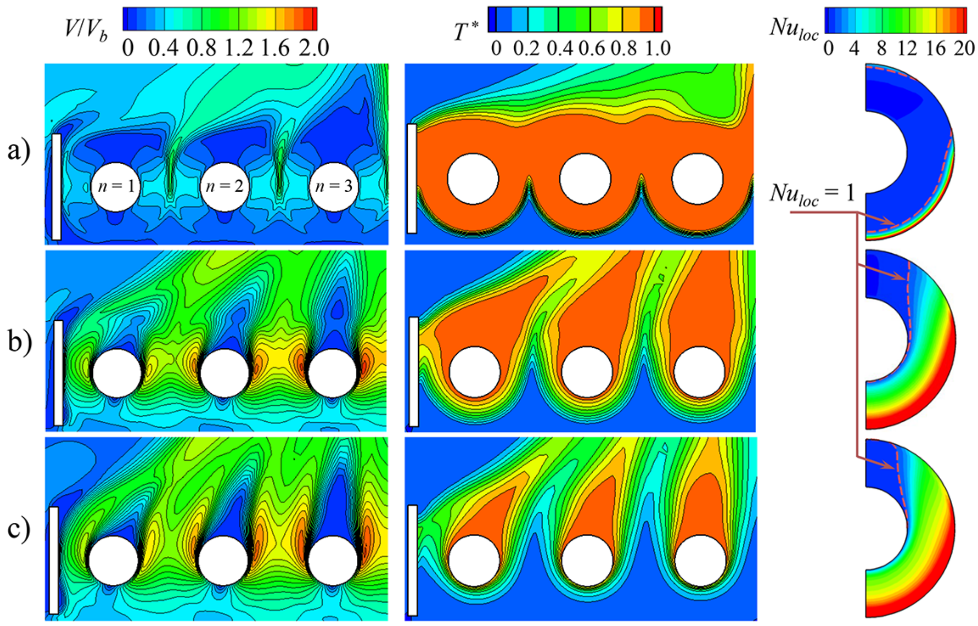

Fig. 2. Distributions of velocity magnitude (left column) and temperature (middle column) in the plane passing through the middle section of the inter-fin space and the local Nusselt number at the wall surface for $n = 3$ (right column) for the cases with different fin spacing s/D: a) 0.04, b) 0.09 and c) 0.13, for all the cases — $\varepsilon_T = 0.7$, $H = 0$

In the absence of a shaft, increasing the fin pitch by two times or more noticeably intensifies the airflow in the tube bundle and above it (Fig. 2). For the case with increased fin pitch, the values of velocity in the bundle are comparable with buoyancy velocity. The Reynolds number for the case with $s/D = 0.09$ is equal to $Re = 240\,(V_{in} = 0.33V_b)$, and for the case with $s/D = 0.13 - Re = 305\,(V_{in} = 0.41V_b)$. The temperature field in the tube bundle has become spatially non-uniform.

2.2. Heat transfer parameters of the tubes: influence of the fin spacing

The distributions of a local Nusselt number $Nu_{loc} = q_{w,loc}\, d/\lambda\,(T_s - T_0)$ at the surface of the fins are presented in Fig. 2 (right column) for the cases with different fin spacing and at $\varepsilon_T = 0.7$. In this formula, $q_{w,loc}$ is the local heat flux, T_s is the local fin surface temperature, and the thermal conductivity coefficient was calculated based on the ambient temperature (the value of $\lambda = 0.0261$ W/(mK) was taken). In the absence of a shaft, very low efficiency of the fin surface takes place for the case with the most compact configuration (Fig. 2a): heat removal is realized mainly from the periphery of the fins (indicated by dashed lines), and the contribution of their inner part is relatively small. The low efficiency occurs due to uniform temperature distribution in the inter-fin space.

With an increase in the inter-fin pitch, the values of the local Nusselt number are also increased noticeably (Fig. 2). The integral (averaged over the tube bundle surfaces) values of the Nusselt number $<Nu>$ related to the value of s/s_0 for the cases with different fin pitch and values of the buoyancy parameter are shown in Fig. 3a. It has been established that by varying the fin pitch, it is possible to escalate the effective heat transfer from the tube bundle surface by more than two times compared to the configuration with the largest number of the fins. The optimal (among those considered) fin pitch with the maximum value of the effective Nusselt number depends on the buoyancy parameter, and it is observed for the cases with $s/D = 0.09$ at $0.2 \leqslant \varepsilon_T \leqslant 0.7$ and for the case with $s/D = 0.11$ at a fixed $\varepsilon_T = 0.05$.

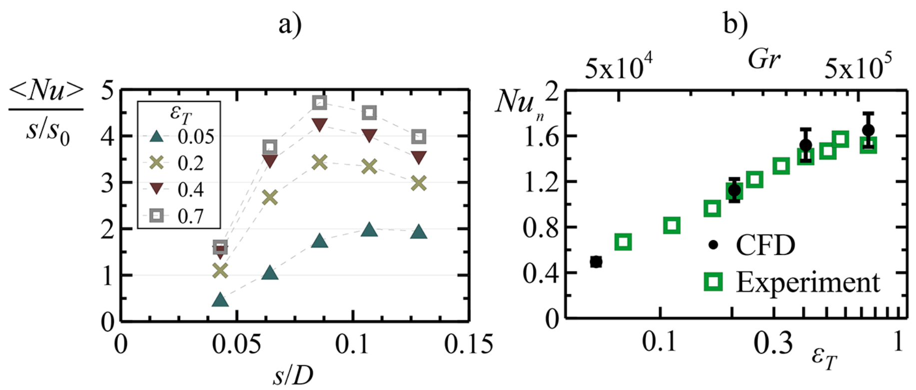

Fig. 3. a) The effective Nusselt number for the cases with different fin spacing at $H = 0$, b) comparison of computational and experimental [8] data on Nu_n values dependent on the Grashof number, the case with $s/D = 0.04$, $H = 0$

For the case without a shaft and with the most compact configuration of the tube bundle, the comparison with experimental data available from the literature [8] has been performed. Fig. 3b presents the experimental (for the tube with $n = 3$) and computational (indicated as "CFD") data of the Nusselt number $Nu_n = <q_{w,n}>\, d/\lambda\,(T_w - T_0)$, where $<q_{w,n}>$ is the mean heat flux over the surface of tube n. CFD data plotted in the range of Nu_n values for the tubes with the numbers $n = 1$, 2 and 3.

The values of dimensionless heat transfer obtained in calculations, averaged over the surface of the middle tube of the bundle, are in good agreement with the experimental data in the examined range of the Grashof number. Computations show that for the side tube (with $n = 1$), the value of Nu_n is noticeably lower than for the central tube (with $n = 3$); maximum differences in Nu_n values are detected for the case with $\varepsilon_T = 0.7$ and reach 24%.

2.3. Effects of the shaft height

In the presence of the shaft, the airflow through the bundle is significantly more intense in the region above the bundle, as expected, and less concentrated near the symmetry boundary. The flow distribution for the case with $s = s_0$ is presented in Fig. 4a,b for $H/D = 4.5$ and 9, correspondingly. The volume flow rate under the bundle increases with the growth of the H/D ratio, and it corresponds to the values of the Reynolds number $Re = 260$, $V_{in} = 0.35\,V_b$ ($H/D = 4.5$), and $Re = 440$, $V_{in} = 0.6V_b$ ($H/D = 9$). The maximum values of the velocity in the tube bundle significantly exceed V_b.

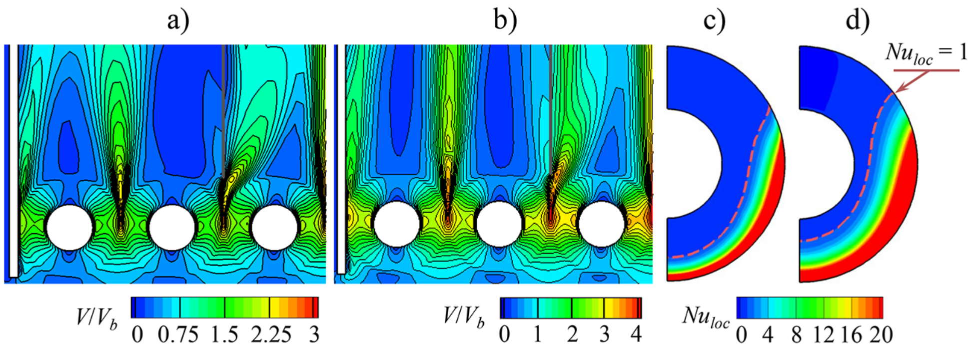

Fig. 4. a,b) Distributions of the velocity magnitude in the plane passing through the middle section and c,d) the local Nusselt number at the wall surface for $n = 3$ for the cases with different shaft height H/D: a,c) 4.5, b,d) 9, for all the cases – $s = s_0$, $\varepsilon_T = 0.7$

Figs. 4c,d show the distributions of the local Nusselt number (Nu_{loc}) over the surface of the fin (for the tube with $n = 3$), calculated for the cases with different H/D ratios. The area of the fin with $Nu_{loc} > 1$ (indicated by the dashed line) significantly increases with increasing the height of the shaft. The values of the mean effective Nusselt number for the cases with different heights of the shaft and by varying the buoyancy parameter are presented in Fig. 5.

The effective Nusselt number for the most compact bundle increases by about three times for the case with the shaft height of $H/D = 4.5$ and by six times for the case with $H/D = 9$. For the optimal fin pitch ($s/D = 0.09$), the shaft encourages the Nu number to increase by two times for the case with $H/D = 9$ and by one and a half for the case with $H/D = 4.5$. A further increase in fin pitch leads to a relatively small effect of the shaft: shaft installation whose height is 10 times greater than the diameter of the fins leads to an increase in the Nusselt number by only 1.5 times. Data on the effective Nusselt number for the cases with the shaft demonstrates that the most compact configuration with the densest arrangement of the fins becomes well suited for

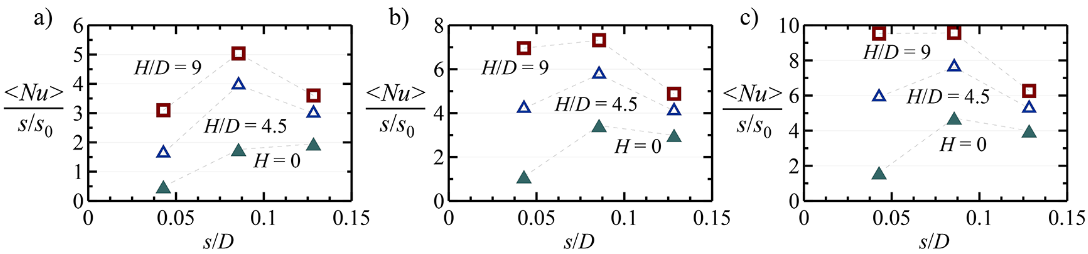

Fig. 5. The mean effective Nusselt number for the cases with different shaft height and fin spacing; the buoyancy parameter ε_T: a) 0.05, b) 0.2 and c) 0.7

the shaft with the height $H/D \geqslant 9$. Among the considered cases, the optimal value of the fin pitch is $s/D = 0.09$, since it provides the highest level of heat removal in a wide range of values of the buoyancy parameter ε_T and shaft height H.

Conclusion

Numerical modeling of the airflow and heat transfer in the single-row bundle of heated finned tubes under the action of free convective forces was carried out. The cases without and with the shaft installation above the tube bundle were considered.

The validation of the methods and tools used was performed for the case with the most compact tube bundle configuration: good agreement was obtained between the calculated average heat transfer from the surface of the tube and the experimental data available from the literature in the considered range of the buoyancy parameter (or the Grashof number).

In the absence of a shaft, the flow movement through the bundle with dense tube location is characterized by a relatively low level of velocity fields comparable with the values of buoyancy velocity. The compact tube bundle configuration provided a low level of the effective Nusselt number. The configuration with optimal fin pitch provides the highest values of airflow velocity and the effective Nusselt number.

In order to intensify significantly free convective airflow in the tube bundle, installation of the exhaust shaft could be used. The mean effective Nusselt number for the most compact bundle was greatly increased by increasing the shaft height. The customized combination of the fin pitch (spatial occupancy of the bundle) and the shaft height led to the optimal tube bundle configuration with the highest level of heat removal.

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Численное исследование течения и теплообмена при термогравитационной конвекции воздуха в однорядном пучке горизонтальных оребренных труб

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Аннотация. Представлены результаты численного моделирования термогравитационного течения воздуха и теплообмена в горизонтальном однорядном пучке из шести оребренных нагретых труб. Расчеты выполнены для пучка с тесным расположением труб при варьировании шага оребрения. Рассмотрены постановки задач с установленной над пучком вытяжной шахтой прямоугольного сечения разной высоты и без шахты. Для наиболее сжатого пучка без шахты получено хорошее согласие рассчитанного интегрального числа Нуссельта на поверхности несущей трубы с доступными из литературы данными экспериментов в рассмотренном диапазоне умеренных значений числа Грасгофа (до $5.5{\times}10^5$). Сочетание шага ребер (пространственного расположения на трубах) и высоты шахты позволяет получить оптимальную конфигурацию пучка с наибольшей интенсивностью отвода тепла.

Ключевые слова: воздушное охлаждение, теплообменные аппараты, трубные пучки, оребренные трубы, межреберное расстояние, вытяжная шахта, термогравитационная конвекция, численное моделирование.