Journal of Engineering Physics and Thermophysics, Vol. 80, No. 3, 2007

HEAT CONDUCTION AND HEAT TRANSFER IN TECHNOLOGICAL PROCESSES

RENORMALIZATION TECHNIQUE IN DESIGNING A HEAT EXCHANGE APPARATUS

G. V. Gembarzhevskii

UDC 536.246:621.373.8

A renormalization technique of thermal design of the heat exchanger is proposed to solve the problems of designing and optimizing the gas–liquid coolant heat-exchange apparatus. In this design, the leading process is the heat transfer in the gas, and renormalization of the heat exchanger efficiency is carried out with the aim of taking into account the influence of the thermal processes in the liquid and the heat conduction of the coolant boundary. For the Nusselt criterion, the one-parameter approximation (not counting the dependences on the Prandtl number) was used. The proposed technique is economical in design, admits application of different goal functions in the optimization, and permits setting limits on the domain of variability of parameters. The efficiency of the proposed technique has been demonstrated for the problem of optimization of parameters of a ribbed-pipe heat exchanger for the facility used in investigating the interaction of a high-speed gas flow with the discharge.

Introduction. In developing a heat engineering device, especially at the initial stage of designing, it may turn out that it is necessary to consider a wide spectrum of designs and parameters of its heat exchange apparatus with the aim of optimization at a given angle of view of the operation of the device on the whole taking into account the fact that a noticeable contribution to the volume and mass of the entire facility may come from the gas heat exchanger. The heat exchange apparatus is characterized by external parameters (heat power, efficiency, hydraulic resistance, dimensions) and internal parameters, i.e., design variables. By the design of the heat exchanger we will imply the expression of its external parameters in terms of the internal ones. Accordingly, the problem of optimizing the heat exchanger design at given external parameters should be considered. In a more general formulation, it is necessary to consider the problem of optimizing the operation of the device on the whole, i.e., at varied external parameters of the heat exchanger. To this end, it is desirable to use a general unified method for designing the heat exchange apparatus, no matter whether it provides the best attainable design accuracy but permits simplifying it to a formalism convenient for optimization, covering thereby a wide range of designs. Upon solving the optimization problem in the first approximation (in the widest possible range of parameters), if necessary, calculations can be made more precise in the vicinity of the optimum localized in the first approximation.

A popular variant of the heat exchange apparatus is a device of heat transfer from the gas to the liquid coolant by induced convection of both coolants with the formation of boundary layers [1]. In this case, in the process of heat transfer, there exists an obvious bottleneck — heat transfer in the gas determining the design of the device. Therefore, it makes sense to make design calculations of such a heat exchange apparatus by the renormalization technique, i.e., take into account in the initial formulation only the leading process — convective heat transfer in the gas, and in the following steps — the influence of the less significant effects. Among these is the heat conduction in a solid surface dividing coolant flows and thermal processes in the liquid coolant.

Institute of Problems of Mechanics, Russian Academy of Sciences, 101-1 Vernadskii Ave., Moscow 119526; email: gvgemb@ipmnet.ru. Translated from Inzhenerno-Fizicheskii Zhurnal, Vol. 80, No. 3, pp. 3–15, May–June, 2007. Original article submitted July 19, 2005; revision submitted February 27, 2006.

1062-0125/07/8003-0425

Method of Design Calculation of Heat Transfer by Laminar Induced Convection. The design calculation of convective heat transfer in the gas is based on the relations generalizing the experimental data for the mean Nusselt number of one row of heat-transfer surfaces [2]. (On this basis, to obtain analytical results of calculating typical multirow apparatuses, one usually uses ordinary differential equations for the averaged description of the heat transfer process). In so doing, for heat-transfer surfaces of different geometries and for particular ranges of parameters, different determining dependences of the Nusselt number are used, which is not always convenient, in particular, in optimization problems, when, in essence, correct approximation of not so much the Nusselt criterion as its derivatives is required. Nonetheless, from general theoretical considerations for laminar flows along channels formed between arbitrary heat-transfer surfaces, it may be expected that the universal relation for the Nusselt number will be fulfilled. For instance, in the regimes of the initial thermal and hydrodynamic parts of the flow this role is played by the Polhausen relation for the heat transfer from the plate [1]. The above relation has the form (with allowance for the transition in the Nusselt and Peclet numbers from the determining dimension — plate, channel length L to the channel diameter D)

$$\mathrm{Nu}_D = 0.664\ \mathrm{Pr}^{-1/6}\left(\mathrm{Pe}_D\, D/L\right)^{1/2} .$$

For another limiting case of the thermal initial part of the hydrodynamically stabilized flow in the channel, the theory uses the Levek formula [3, 4]

$$\mathrm{Nu}_D = 1.62\left(\mathrm{Pe}_D\, D/L\right)^{1/3} .$$

Passing to the practical solution of the problem of constructing a universal relation for the Nusselt number under laminary induced convection, we use the data generalized in [3]. For the laminary flow conditions ($\mathrm{Re}_{D^*} < 2\cdot10^3$) of the coolant (both gas and liquid) through a channel with an arbitrary geometry of the heat-transfer surface in the region of parameters, where thermal stabilization of the flow has not yet occurred and the relation

$$\mathrm{Pe}_{D^*}\, D^*/L' > 50 \tag{1}$$

holds, we have a universal dependence for the mean Nusselt number as a result of the data interpolation [3] in the parameter region $0.72 \le \mathrm{Pr} \le 500$; $50 \le \mathrm{Pe}_{D^*} D^*/L' \le 2\cdot10^3\ (D^*/L')^2\mathrm{Pr}$:

$$\mathrm{Nu}_{D^*} = \left(0.93 + 0.42/\mathrm{Pr}^{0.17}\right)^{-1}\left(\mathrm{Pe}_{D^*}\, D^*/L'\right)^{\alpha},\quad \alpha = 0.37 + 0.12/\mathrm{Pr}^{0.2} . \tag{2}$$

The error of the Nusselt number values used is estimated [3] to be ~15–20% on the basis of comparison with the experimental data, which can be thought to be sufficient for design calculations. It should be noted that the smooth approximation (2) is in good agreement with the Polhausen formula in the region of the initial hydrodynamic part of the flow, i.e., at a Prandtl number Pr = 1, and approaches the Levek dependence in the process of hydrodynamic stabilization of the flow at $\mathrm{Pr} \to 500$. The simple exponential form of the universal dependence (2) of the Nusselt number on the only (except for the Prandtl number) determining parameter $\mathrm{Pe}_{D^*} D^*/L'$ is obtained due to the use of the "physical" variables L' and D^* [3], and the Prandtl number therewith can be considered as a constant for gas flows. The flow length L' is defined as a quotient obtained when the surface area of the body A participating in the heat transfer is divided by the body perimeter projection P_{pr} on the surface perpendicular to the flow direction. This is the mean length of the boundary layer formation in the channels between the heat-transfer surfaces

$$L' = A/P_{\mathrm{pr}} . \tag{3}$$

According to [3], the mean hydraulic diameter of the channels D^* between the packing elements is

$$D^* = 4\Psi L' S_0^A / A \,. \tag{4}$$

The quantity S_0^A is the unencumbered cross-section of the channel per one transverse spacing in the arrangement of the heat-transfer surfaces (pipes). The mean encumbering of the channel $1 - \Psi$ is determined by the relative fraction of solid bodies (pipes) V_s in the unencumbered volume of the gas channel V_Σ

$$\Psi = 1 - (V_s / V_\Sigma) \,. \tag{5}$$

The Nusselt number in relation (2) has been determined by the maximum temperature difference (the inlet gas temperature and the heat-transfer surface temperature) and the hydraulic diameter D^*. The Peclet number $\mathrm{Pe}_{D^*} = \langle w \rangle D^* / a_g$ has been calculated by the hydraulic diameter D^* and the mean flow velocity $\langle w \rangle$:

$$\langle w \rangle = w_0 / \Psi \,. \tag{6}$$

The parameter $w_0 = \overset{*}{V} T_m / S_0 T''$ is the gas velocity in the unencumbered cross-section of the channel S_0, and the gas temperature thereby is equal to the determining temperature T_m. The gas flow rate V^* is given at the outlet from the heat exchanger (since we assume that gas-drawing fan is located there).

As a performance index of the heat exchange apparatus designed for cooling the gas, we take its efficiency $\hat{\Theta}$ [3]:

$$\hat{\Theta} = \frac{T' - T''}{T' - T_2'} \,, \tag{7}$$

where T' is the liquid gas temperature at the heat exchanger inlet and T_2' is the inlet temperature of the liquid coolant. The apparatus has a counterflow scheme. The efficiency calculated with account for only the convective heat transfer in the gas, when the heat-transfer temperature is assumed to be equal everywhere ($T_f = T_2'$), is expressed in terms of the introduced parameters [3]

$$\theta_i = \frac{4\mathrm{Nu}_{D^*}}{\mathrm{Pe}_{D^*} D^* / L'} \,. \tag{8}$$

The subscript i in the efficiency means that the quantity pertains to the ith, in the gas flow direction, streamlined body (to the pipe with a row number i). If the heat exchanger consists of z rows having the efficiency θ_i respectively, then the efficiency of the heat exchanger on the whole is

$$\theta = 1 - \prod_{i=1}^{z} (1 - \theta_i) \,. \tag{9}$$

The union of relations (2)–(9) gives a compact analytical representation of the efficiency of the heat-exchange apparatus θ with an arbitrary geometry of the heat-transfer surfaces with account for only the heat transfer in the gas (for the laminary flow conditions of the gas and when condition (1) is fulfilled).

For the popular type of recuperative heat exchangers assembled from staggered (rhombic cell) ribbed pipes (of diameter d at the rib basis; with ribs in the form of disks of thickness δ and height h arranged with a pitch s along the pipe axis), we have the volume concentration of voids in the channel (5)

$$\Psi = 1 - \frac{\pi}{4ab}\left[1 + 4\left(\frac{\delta h}{sd}\right)\left(1 + \frac{h}{d}\right)\right], \tag{10}$$

the flow length (3), and the hydraulic diameter (4)

$$L' = \frac{\pi d}{2}\left[1 + \left(\frac{2h}{s}\right)\left(1 + \frac{\delta + h}{d}\right)\right]\left[1 + \frac{2h}{s}\right]^{-1}, \quad D^* = \frac{2\Psi ad}{1 + (2h/s)}. \tag{11}$$

Here the parameter $a = s_{\mathrm{tr}}/d$ is the relative pitch of the pipe bundle in the direction across the gas flow; $b = s_{\mathrm{lon}}/d$ is the relative longitudinal pitch (along the flow). The total area of the pipe surface $A = \pi d L_{\mathrm{p}}\varphi$, where $\varphi = 1 + (2h/s)(1 + (h + \delta)/d)$ is the ribbing coefficient of the pipe [2].

Account of the Influence of the Thermal Resistance of the Heat-Transfer Surface Ribbing on the Apparatus Efficiency. Let us denote the sought renormalized efficiency of a row of heat-exchanger pipes determined with respect to the liquid coolant temperature T_2 as Θ_i (the liquid temperature T_2 therewith is assumed to be uniformly distributed in the heat exchanger):

$$\Theta_i = \frac{T_i' - T_i''}{T_i' - T_2}. \tag{12}$$

Above, in estimating the efficiency of the row θ_i in the chief approximation of (8), it was assumed that the ribbing temperature is everywhere the same and equal to the coolant temperature, and now, in calculating the renormalized efficiency (12), the distribution of the heat-transfer surface temperature should be found. To this end, we solve the equation of heat conduction in the ribbing plates in the known approximation of uniform distribution of the heat flux density at the plate–gas boundary. Having calculated the ribbing–liquid temperature drop averaged over the plate surfaces, we find the efficiency of one row of the heat exchanger Θ_i determined with allowance for the thermal resistance of the ribs (12), expressing it in terms of the previously determined startup efficiency of the row θ_i (8):

$$\Theta_i = \frac{\theta_i}{1 + \varepsilon\theta_i}. \tag{13}$$

The parameter ε determining the effect of thermal resistance of the ribbing is a sum of two terms

$$\varepsilon = \varepsilon_{\mathrm{f}} + \varepsilon_{\mathrm{b}}, \tag{14}$$

$$\varepsilon_{\mathrm{f}} = \left[\frac{adsw_0\rho C_p}{2\pi\delta\lambda_{\mathrm{f}}}\right]\frac{(1 + \delta/R)^2 \ln(R/r) - [1 - (r/R)^2]\,[0.75 + (\delta/R) - 0.25\,(r/R)^2]}{[1 + (\delta/R) - (r/R)^2 + r\,(s - \delta)/R^2]^2}, \tag{15}$$

$$\varepsilon_{\mathrm{b}} = \left[\frac{adw_0\rho C_p}{2\pi r_{\mathrm{b}}}\right]R_{\mathrm{b}}. \tag{16}$$

With the help of ε_{f} the effect of thermal resistance of the pipe ribbing material is taken into account, and by means of ε_{b} the expression for the determining parameter ε is generalized in order to take into account the thermal contact resistance of the technological gap between the surfaces of a bimetal pipe R_{b} [2]. Here we used the following designations: λ_{f} — heat conductivity coefficient of the ribbing material; ρ and C_p – density and specific heat capacity of the gas; $r = d/2$, $R = r + h$, and r_b — pipe radii at the base and top of the ribs and at the contact boundary of the materials.

To calculate the determining parameter ε_f more precisely, let us express it in terms of the known efficiency coefficient of a rib E_i [2] on a pipe of the row i;

$$\varepsilon_f = \left(\frac{1}{\langle E_i \rangle} - 1\right)\left(\frac{1}{\theta_i} - 0.5\right), \quad \langle E_i \rangle = E_i \mu_E \psi_E \left(1 - (1 - \delta/s)/\varphi\right) + (1 - \delta/s)/\varphi \,. \tag{17}$$

The quantity $\langle E_i \rangle$ is the mean value of the efficiency of the pipe surface on the whole. According to the adopted formalism [2] in (17) coefficients ψ_E and μ_E were introduced with the aim of taking into account the influence of the inhomogeneity of the heat flux density on the rib surface and the change in the rib thickness over the radius on the rib efficiency and, accordingly, on the value of ε_f.

The multiplicative relation (9) holds also for the renormalized, with allowance for the ribbing resistance, efficiencies of the rows and the heat exchanger on the whole Θ:

$$\Theta = 1 - \prod_{i=1}^{z} (1 - \Theta_i) \,. \tag{18}$$

Account of the Influence of the Liquid-Coolant Flow Rate Finiteness on the Heat Exchanger Efficiency. Below we correct the calculation of the efficiency of the heat exchange apparatus with the aim of taking into account the effect of downstream heating of the liquid coolant because of the finiteness of its heat capacity and flow rate. Accordingly, we introduce into the consideration the heat exchanger efficiency renormalized with regard for the coolant heating

$$\overline{\Theta} = \frac{T' - T''}{T' - T_2'} \tag{19}$$

((19) coincides in form with (7), but here the temperature gradient of the liquid across the pipes is ignored). Coolant heating leads to a decrease in the "driving" temperature difference at each point of the heat-exchange surface with a single proportionality coefficient $(1 - \Delta)$. The dependence of both the gas and the coolant temperature on the pipe row number is actually the same (approximately exponential), differing only in the scale. For exact fit of the temperature profiles of the gas and the liquid coolant in the heat exchanger (averaged within the pipe row), the following relation between the gas and coolant heat capacities C_2 is formally necessary and sufficient:

$$C_p(T) = \text{const}\; C_2\left(T_\infty + \Delta\,(T - T_\infty)\right), \quad T_\infty = (T_2' - \Delta T'')/(1 - \Delta) \,. \tag{20}$$

The parameter T_∞ is the temperature of complete relaxation of the heat transfer in a hypothetic heat exchanger with a packing infinitely extended in the direction of the gas flow. In practice, the relation between the heat capacities (20) is provided due to their weak temperature dependence. Hence it is possible to express the efficiency of the counter flow heat exchanger with allowance for the coolant heating $\overline{\Theta}$ in terms of the previously introduced (see (18)) efficiency Θ renormalized only to the account of the ribbing thermal resistance:

$$\overline{\Theta} = \frac{1 - (1 - \Theta)^{1-\Delta}}{1 - \Delta\,(1 - \Theta)^{1-\Delta}} \,. \tag{21}$$

We determine the attenuation coefficient of the heat flux $1 - \Delta$ from the energy conservation law in the heat transfer, expressing the transferred quantity of heat in terms of the temperature differences between the liquid coolant and the gas on the heat exchanger

$$\Delta = \frac{\overset{*}{V}\rho C_p}{\overset{*}{m}_2 C_2} = \frac{T_2'' - T_2'}{T' - T''} . \tag{22}$$

Account of the Influence of the Heat Transfer Efficiency on the Side of the Liquid Coolant on the Efficiency of the Heat Exchanger. For completeness of the physical picture, it is also necessary to take into account the influence of the liquid heating in the direction of the normal to the heat-exchange surface (along the pipe radius). In other words, it is necessary to take into account the difference between the mean-mass temperature of the liquid flow (it is this quantity that appeared in the previous section, in particular, in (19) and (21)) and its temperature at the boundary with the heat-exchange surface. Taking into account the invariance of the temperature distribution profile of the gas and the coolant on the heat exchanger, we obtain the attenuation coefficient of the driving temperature difference in the form $(1 - \Delta - \hat{\Delta})$, where the additional parameter $\hat{\Delta}$ appears:

$$\hat{\Delta} = \frac{\overset{*}{Q}\left(\dfrac{d_2}{\mathrm{Nu}_2^{\ln}\lambda_2}\right)\ln\left(\dfrac{T' - T_\infty}{T'' - T_\infty}\right)}{A_2\,(T' - T'')} . \tag{23}$$

Noteworthily, the law of renormalization of efficiency (21) actually holds, but is now determined by the overall heat flux attenuation coefficient $(1 - \Delta - \hat{\Delta})$ because of the effects of "longitudinal and transverse (to the heat-exchange surface) heating of the coolant:"

$$\hat{\Theta} = \frac{1 - (1 - \Theta)^{1-\Delta-\hat{\Delta}}}{1 - \Delta\,(1 - \Theta)^{1-\Delta-\hat{\Delta}}} . \tag{24}$$

Choice of the Determining Gas Temperature. The sequential renormalization computational procedure (2), (8), (13), (18), (24) permits economic calculation of the efficiency of a given heat exchanger design. It is only required to choose the determining temperatures of the rows $T_{\mathrm{m}i}$ in order to calculate the determining criteria of the rows $(\mathrm{Pe}_{D^*}D^*/L')$. To this end, in [3] it is recommended to use the relation

$$T_\mathrm{m} = 0.5\,(T' + T'') - \left[0.5\,(T' + T'') - T_\mathrm{f}\right]\left[\frac{40 + 0.1\mathrm{Pr}}{72 + \mathrm{Pr}}\right], \tag{25}$$

where T_f is the temperature of the (solid) heat-exchange surface. The application of (25) to the calculation of the row efficiencies complicates the calculations (since the determining temperatures of the rows $T_{\mathrm{m}i}$ will depend on their number), and its direct use for the whole of the heat exchanger is rough (at large temperature drops on the apparatus). To simplify the calculation, it makes sense to indicate a single determining temperature for the whole of the heat exchanger, and, according to (13), (18), this temperature T_m should satisfy the equation

$$\sum_{i=1}^{z} \ln\left(1 - \theta_i\,(T_{\mathrm{m}i})/(1 + \varepsilon\theta_i\,(T_{\mathrm{m}i}))\right) = z\ln\left(1 - \theta_i\,(T_\mathrm{m})/(1 + \varepsilon\theta_i\,(T_\mathrm{m}))\right).$$

At a large number of pipe rows in the heat exchanger $z >> 1$ and, accordingly, at small values of the parameters $\theta_i << 1$ the products of the row efficiencies θ_i should be averaged in fact. Let us also take into account that the determining temperature appears in the row efficiency $\theta_i \sim (T_{mi})^{\gamma}$ as a cofactor to some power γ (if we make use of the power approximation of the temperature dependences of the gas transport coefficient). According to the foregoing, averaging of products of the determining temperatures of the rows T_{mi} yields the expression for the single determining temperature of the multirow heat exchanger on the whole T_m:

$$\ln (T_m/T_\infty) = \frac{\left[\ln\left[1+\beta\left(\frac{T'}{T_\infty}-1\right)\right]\right]^2 - \left[\ln\left[1+\beta\left(\frac{T'}{T_\infty}-1\right)\left(\frac{T''-T_\infty}{T'-T_\infty}\right)\right]\right]^2}{2\ln\left(\frac{T'-T_\infty}{T''-T_\infty}\right)},$$

$$\beta = 1 - \left(\frac{40+0.1\mathrm{Pr}}{72+\mathrm{Pr}}\right)\left(\frac{1-\Delta-\hat{\Delta}}{1+\varepsilon\theta_i}\right). \tag{26}$$

For the apparatus with a gas circulation loop, the inlet T' and outlet T'' temperatures of the gas under stable operating conditions of the heat exchanger are determined by the heat power $\overset{*}{Q}$ fed into the gas loop (and taken off by the heat exchanger), by the heat exchanger efficiency $\hat{\Theta}$, and by the volumetric rate of flow $\overset{*}{V}$ and pressure p of the gas. It is convenient to give the two quantities at the heat exchanger outlet — where the fan is situated and where the gas temperature changes to the least extent with a change in the input power $\overset{*}{Q}$:

$$T'' = T_2'\left[1-\left(\frac{1}{\hat{\Theta}}-1\right)\left(\frac{\overset{*}{Q}R_g}{\overset{*}{V}C_p p\mu}\right)\right]^{-1}, \tag{27}$$

$$T' = T_2'\left[1+\frac{\overset{*}{Q}R_g}{\overset{*}{V}C_p p\mu}\right]\left[1-\left(\frac{1}{\hat{\Theta}}-1\right)\left(\frac{\overset{*}{Q}R_g}{\overset{*}{V}C_p p\mu}\right)\right]^{-1}. \tag{28}$$

Calculation of the Hydraulic Resistance of the Heat Exchange Apparatus as to the Gas. Let us estimate the resistance of the multirow ($z >> 1$) heat exchanger by [2, 4]

$$\Delta p/p = z\,\mathrm{Eu}\,\rho\,(\max w)^2/2p\,. \tag{29}$$

The maximum gas velocity ($\max w$) is attained in the most contracted section of the gas flow (localized in the minimum clearance of the transverse or the diagonal row of pipes). In particular, for the staggered rhombic bundle of ribbed pipes

$$\max w = \max\left[w_{tr};\, w_d\right],\quad w_{tr} = w_0 a\left(a-1-\frac{2h\delta}{sd}\right)^{-1},$$

$$w_d = w_0\frac{a}{2}\left(a_d-1-\frac{2h\delta}{sd}\right)^{-1},\quad a_d = \sqrt{0.25a^2+b^2}\,. \tag{30}$$

In calculating the Euler number Eu for (29), in [2] it is recommended to use the approximation

$$\mathrm{Eu} = 5.4 \left(l / d_{\max}\right)^{0.3} (\mathrm{Re}_l)^{-0.25}, \quad \mathrm{Re}_l = l\,(\max w)/\nu\,. \tag{31}$$

The hydraulic diameter of the channel $d_{\max}$ in the section where the maximum gas velocity max w is realized is

$$d_{\max} = 2d\left[a_{\max} - 1 - \frac{2h\delta}{sd}\right]\left[1 + \frac{2h}{s}\right]^{-1}. \tag{32}$$

The parameter $a_{\max}$ is the relative pitch of the transverse (a) or diagonal (a_{d}) row of pipes in the bundle, for which the maximum gas velocity (30) is realized. It is recommended to calculate the determining dimension l in (31) by the formula

$$l = d\left[\frac{1 - (\delta/s)}{\varphi}\right] + \sqrt{\pi h\,(d + h)}\left[1 - \frac{1 - (\delta/s)}{\varphi}\right]. \tag{33}$$

However, it seems reasonable not to complicate the calculation and use the already introduced mean length of the flow past the ribbed pipe:

$$l \equiv L'\,. \tag{34}$$

Problem of Optimization of the Heat Exchange Apparatus. As an example of practical application of the proposed computational procedure, consider the problem of optimizing the gas–liquid heat exchange apparatus employed in a closed-gas-cycle facility. The aim of the optimization is to attain the minimum temperature of the gas at the outlet from the apparatus T'' at given flow rate $\overset{*}{m}_2$, heat capacity C_2, and inlet liquid coolant temperature T_2' and at a fixed heat power $\overset{*}{Q}$ of the heat exchanger. The dimensions of the unencumbered channel section S_0, as well as the pressure p and the volumetric rate of flow of the gas $\overset{*}{V}$ at the outlet from the heat exchanger, are also given. A restriction on the hydraulic resistance of the apparatus as to the gas $\Delta p/p \le K$ is taken. The solution is sought in the class of designs from ribbed pipes of cylindrical symmetry assembled into a staggered bundle. The contact thermal resistance of the bimetal pipes is ignored ($R_{\mathrm{b}} = 0$).

Obviously, at a fixed heat power $\overset{*}{Q}$ and under the above additional conditions the gas temperature minimum at the outlet from heat exchanger is realized for an apparatus having the maximum efficiency (formally, this conclusion follows from (27)). Accordingly, we obtain a natural problem of optimizing the heat exchanger efficiency under the additional conditions listed above. Suppose that in the optimum region the row efficiencies are low ($\theta_i << 1$), and, to start with, we shall neglect the thermal processes in the liquid ($\Delta = \hat{\Delta} = 0$). Then the expression for the heat exchanger efficiency will take on the form

$$\hat{\Theta} = \Theta = 1 - \exp\left(-\frac{z}{(\theta_i)^{-1} + \varepsilon}\right). \tag{35}$$

In accordance with (35) we have the goal function of optimization to the maximum $z(\theta_i^{-1} + \varepsilon)^{-1}$, where the parameters θ_i and ε are determined by (2), (8), (15), and (16), and the Pr number is assumed to be 0.72. By means of (29) we formulate the restriction on the hydraulic resistance of the apparatus:

$$z\left(\frac{a}{a - 1 - 2\dfrac{h\delta}{ds}}\right)^2 \le k \quad \text{for} \quad a < 0.5\,(4b^2 - 1)\,; \quad z\left(\frac{0.5a}{\sqrt{0.25a^2 + b^2} - 1 - 2\dfrac{h\delta}{ds}}\right)^2 \le k \quad \text{for} \quad a > 0.5\,(4b^2 - 1)\,;$$

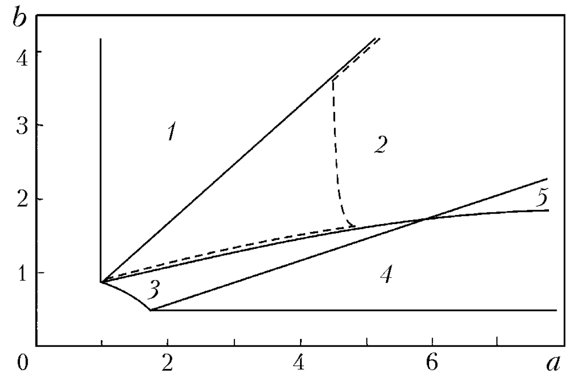

Fig. 1. Map of domains of analyticity of functions in the process of optimization of the ribbed-pipe heat exchanger (solid lines); optimal geometry of the pipe bundle (dashed line).

$$k = \frac{2pK}{\rho\,(w_0)^2\,\mathrm{Eu}}\,. \tag{36}$$

Then, in optimizing the heat exchanger efficiency, we shall assume k to be a constant, neglecting the weak variability of the Euler number in (36) determined by (31)–(34). (Incidentally, note that in using the approximation of the Nusselt number of the row according to (5.1) [2], the mathematical form of the goal function will be more complicated for the analytical optimization, namely it will contain an exponential function with variable arguments both at the base and in the exponent.)

For the optimization to be carried out, it is essential that relations (30), (36), and others, which will be given below, have a different form in different ranges of values of dimensionless pitches of the pipe bundle a and b. Figure 1 gives the map of the parameters a and b, in which the solid lines show the boundaries of transitions between the regimes of dependences of different functional forms. Altogether, there are five ranges ($1 \le n \le 5$) in the case of fairly thin ribs $\chi \equiv 2h\delta/ds(a-1) < (2/\sqrt{3}) - 1$; the boundaries between the ranges are determined by relations (37), (38), and (39) and are marked on the map by solid lines. The entire range of admissible values of pitches lies in the first square and by virtue of the impermeability of the packing pipes is bounded below by a ray $b = 0.5$, on the left — by a ray $a = 1$, and in the region of the origin of coordinates — by an ellipse arc $0.25a^2 + b^2 = 1$. Demarcation of the form of the dependence (30) and (36) in the case of fairly thin ribs $\delta/s = 0$ occurs on the curve

$$b = 0.5\,\sqrt{2a+1}\,. \tag{37}$$

The maximum height of a rib on the pipe depending on the pitches between pipes a, b is limited by the presence of the closets neighboring pipe either in the transverse or longitudinal or diagonal row of the bundle. According to these three cases, the range of admissible values of the parameters a and b is split by the rays

$$b = 0.5\,\sqrt{3}\,a\,, \tag{38}$$

$$b = (0.5/\sqrt{3})\,a \tag{39}$$

into three sectors (Fig. 1). Above ray (38), the limiting height of the rib is determined by the presence of a neighboring pipe in the transverse row, and below ray (39) it is limited by the presence of a neighboring pipe in the longitudinal row (the number of this neighboring pipe differs by two units from the row number of the pipe under consideration). In the sector between rays (38), (39), the maximum height of the pipe rib is limited by the neighborhood in the diagonal row:

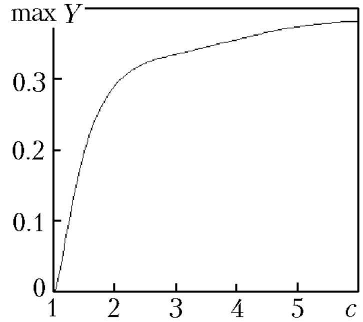

Fig. 2. Dependence of the optimization goal function cofactor max Y (45) on the reduced rib height $c = 1 + 2h/d$ (42).

$$\frac{2 \max h}{d} = \sqrt{0.25a^2 + b^2} - 1 \quad \text{for} \quad (0.5/\sqrt{3})\, a \le b \le 0.5\,\sqrt{3}\, a\,. \tag{40}$$

We carry out optimization of the goal function $z(\theta_i^{-1} + \varepsilon)^{-1}$ by steps. In the first step, we vary the ribbing pitch s. We obtain that the optimum is localized on the surface $\chi = 1/3$. It should be remembered that the sufficiently small values of the parameter s/δ satisfying this relation are not always plausible. Therefore, we further assume that the parameter $\chi = \text{const} \le 1/3$ is fixed.

In the second step of optimization, we vary the pipe diameters d and the rib thickness $\delta \sim d$ so that the parameters a, b, h, s remained fixed. Note that the goal function monotonically increases with decreasing diameter of the pipes d up to the moment of contacting the ribs of the neighboring pipes. Therefore, we further consider (and illustrate in Fig. 1) only these optimal configurations of bundles, for which the pipe ribs come in contact either in the transverse or diagonal or longitudinal row. Now the three parameters a, b, and h/d are definitevely related by the relation conveniently given in the form

$$c = c\,(a, b)\,, \tag{41}$$

where the quantity of the relative height of the rib on the pipe

$$c = 1 + 2h/d \tag{42}$$

was introduced.

In the third step of optimization, we seek the maximum of the goal function $z(\theta_i^{-1} + \varepsilon)^{-1}$, varying the parameters d and $h \sim d$ so that the parameters a, b, and s remain fixed. We obtain the relation that can be considered as the equation for determining the optimal pipe diameter:

$$\varepsilon\theta_i = \left(\frac{\partial \ln \theta_i}{\partial \ln d}\right) \Big/ \left(\frac{\partial \ln \varepsilon}{\partial \ln d}\right), \tag{43}$$

where the logarithmic derivatives were taken at fixed values of the above parameters. Taking into account the inequalities $2h/s >> 1$ and $\delta/d << 1$, we reduce relation (43) to the form

$$\varepsilon\theta_i \approx 0.5\,. \tag{44}$$

In the fourth step of optimization, we vary the ribbing pitch of the pipes s at fixed values of the parameters a and b. In so doing, we reveal that the goal function $z(\theta_i^{-1} + \varepsilon)^{-1}$ infinitely increases with decreasing pitch s (and with decreasing optimal geometric parameters of the ribbing δ, h, and d). However, the transition of the flow condi-

TABLE 1. Comparison of the Efficiencies of the Heat Exchanger Pipe Row θ_i Calculated by the Proposed and Standard [2] Methods for the Case of $\mathrm{Re_s} = 200$, $d/s = 3$, $a = 3$

Pipe row efficiency	Longitudinal pitch $b = 0.5\sqrt{7}$	Longitudinal pitch $b = 1.5\sqrt{3}$	Gain in efficiency $\theta_{i(1.5\sqrt{3})}/\theta_{i(0.5\sqrt{7})}$
Value calculated by the proposed technique	0.1857	0.3776	2.033
Value calculated by (5.1) [2]	0.1965	0.3457	1.759
Ratio of values calculated by the two methods	0.945	1.092	1.156

tions into the region of thermal stabilization sets a limit to the process of infinite increase in the goal function. To this change of the heat transfer regime there corresponds formally the going of the parameters to the boundary of inequality (1). Let us denote the point of their appearance at the boundary of (1) as s_0. A further increase in the goal function with decreasing ribbing pitch s is now possible only along the boundary of inequality (1).

As a result of the four optimization steps, the goal function takes on the form

$$z\,(\theta_i^{-1} + \varepsilon)^{-1} = \frac{\left[\dfrac{0.73\sqrt{\pi}\chi}{\sqrt{\mathrm{Pe_s^0} s_0/x}}\right]^{2/3} k\,(1-\chi)^2}{[1 + 0.5\,(s/s_0)^2]}\, Y\,, \tag{45}$$

$$\mathrm{Pe_s^0} = w_0 s_0/a_\mathrm{g}\,,\quad x = \frac{3\pi\lambda_\mathrm{f}}{w_0 \rho C_p}\,,\quad Y = \left[\frac{c+1}{\psi\,(a, b, \chi)}\right]^{1/3} \frac{(a-1)^{7/3}}{a^3}\,.$$

In the fifth step of the optimization procedure, we vary the pipe bundle pitches a and b at a fixed value of the relative height of the rib c. We seek the cofactor Y extremum under a simplified assumption $\chi \to 0$. Accordingly, we obtain that the optimum of the function $Y(a, b)$ is reached for the configuration of the pipe bundle shown by a dashed broken line in Fig. 1. It lies within the limits of region 2, and at small or large values of the parameter c it coincides with the boundaries of this region, and its middle link represents a hyperbolic segment:

$$b = 1.451 + \frac{0.0542}{a - 4.422}\,,\quad 4.4448 \le a \le 4.7443\,. \tag{46}$$

Next, for the sake of simplification, we approximate the hyperbolic arc of the straight line $a = 4.5$ [Note that by means of the standard approximation of the experimental data for the Nusselt number (formula (5.1) in [2]) it is impossible to obtain this optimum for values of the parameter $c \le 1.7$, since it is situated practically at the boundary of the range of applicability of approximation (5.1) at $c \le 1.7$.]

In the sixth step, we optimize the relative rib height c. For the optimum configuration of the pipe bundle $a = a(c)$, $b = b(c)$ obtained in the fifth step of the process, the function max $Y(c)$ is given by the relations

$$\max Y = \frac{(c-1)^{7/3}}{c^3}\left[\frac{c+1}{1 - \pi\sqrt{3}/6c^2}\right]^{1/3} \quad \text{for} \quad c \ge 4.5\,, \tag{47}$$

$$\max Y \approx \frac{(3.5)^{7/3}}{(4.5)^3}\left[\frac{c+1}{1 - \pi/18\sqrt{c^2 - (2.25)^2}}\right]^{1/3} \quad \text{for} \quad 2.75 < c < 4.5\,, \tag{48}$$

TABLE 2. Basic Parameters of the Optimized Variant of the Heat Exchanger on the Facility for Investigating the Interaction of the Glow Discharge with a High-Speed Gas Flow

Basic characteristics of the heat exchanger	Number of pipe rows	
	8	12
Heat exchanger efficiency calculated by the proposed technique	0.960	0.992
Heat exchanger efficiency calculated by (5.1) [2]	0.938	0.984
Hydraulic resistance of the heat exchanger for gas $\Delta p/p$	0.012	0.018

$$\max Y = \frac{(2c-2)^{7/3}}{(2c-1)^3}\left[\frac{c+1}{1-\pi/4(2c-1)\sqrt{c-0.25}}\right]^{1/3} \quad \text{for} \quad c \le 2.75\,. \tag{49}$$

This dependence is given in Fig. 2. The cofactor of the goal function max $Y(c)$ first rapidly increases with increasing rib height c, and then becomes constant, and the maximum thereby is attained at $c > 7$. A much smaller height of ribs on the pipes is practically possible; to actually high values of the parameter c there corresponds the widely used design of a sheet-stack heat exchanger.

The solution of the optimization problem with a fixed flow rate of the liquid coolant will not change appreciably as a result of taking into account the thermal processes in the liquid ($\Delta + \hat{\Delta} \neq 0$), since in this case the parameter $\Delta + \hat{\Delta} \approx \Delta$ (22), by means of which the efficiency of the heat exchanger is renormalized according to (24), is practically fixed.

In domestic practice, in the thermal design of recuperative heat exchangers from bimetal ribbed pipes with helical aluminum ribs, the criteria relations developed at the Central Boiler-Turbine Institute, Kiev Polytechnical Institute or at the Arkhangel'sk State Technical University) [2] are used. According to the foregoing, to estimate the reliability of the results obtained in the optimization problem, let us compare the values of the heat exchanger efficiency calculated by the proposed method to the values obtained by the standard method (in (5.1) [2]) for the parameter domain where both computational procedures are totally applicable. Let us take two characteristic points on the map of parameters (Fig. 1) — at the domain 2 boundary, at a value of a = 3. The values of the other parameters are as follows: the Reynolds number of the flow Re_s = 200 (calculated by the characteristic size — the pipe ribbing pitch, as is customary in [2]); the parameter d/s = 3; the rib height is maximally admissible. As a result of the calculations of the pipe row efficiency θ_i, we obtained the values presented in Table 1. As may be concluded from these data, the difference in the values of the pipe row efficiency calculated by the two methods is within the limits of 10%, which is quite admissible. In such an event, let us try to widen the range of applicability of the results obtained for the Reynolds number. As would be expected, an attempt to calculate the heat exchanger by the proposed method in the case of turbulent flow conditions leads to an underestimated efficiency of the pipe row (almost twice lower for the value of Re_s = 2000). A more interesting (for the discussion) result is that the dependence of the row efficiency on the pipe bundle pitch is well taken into account by the proposed method also in this case. This means that the results obtained by the optimal geometry of the pipe bundle (Fig. 1) are evidently valid, in general, under turbulent gas flow conditions as well.

It is proper to note that according to the accepted scheme of optimization of water-cooling heat exchange apparatuses [2, 5], the heat-transfer surfaces and the corresponding apparatuses are directly compared (and optimized) under the condition of equality of energy expenditures for the gas pumping referred to a unit of surface area. Unlike this scheme, in the present paper the efficiency of the heat exchanger is directly optimized at a varied heat-transfer area but with fixed total (for the heat exchanger) expenditures for the gas pumping and at a fixed unencumbered cross-section of the channel as to the gas. The available data on optimal arrangements of pipe bundles ([6], Par. 1.9) are indefinitive, which is explained, in particular, by the diversity of additional conditions required for the optimization; however, there is no indication of a change in the optimal arrangement of bundles of ribbed pipes with a variation of the ribbing height (in our consideration, of the parameter $c = 1 + 2h/d$).

Choice of Heat-Exchanger Parameters on the Facility for Investigating the Effect of Glow Discharge on the Pulsation Characteristics of a High-Speed Gas Flow. At the Institute of Problems of Mechanics, Russian Academy of Sciences, investigations of vortex plasma flows after bluff bodies [7, 8] are conducted on the Lantan-2 facility having a vacuum-sealed loop through which the working mixture of gases circulates at a pressure equal to ~0.05 of atmospheric pressure. In the operating chamber cross-section, a discharge is glowing, and downstream of the gas a heat-exchanger, where practically the whole of the power input into the discharge is taken off, and a one-stage axial-flow gas-circulating fan are located. For the investigation being conducted, it is important to obtain high values of the electric discharge power $\overset{*}{Q}$. Therefore, a high efficiency of the heat exchanger is required to prevent an increase in the gas temperature at the inlet into the discharge chamber, which inevitably causes a decrease in the highest possible discharge power. The heat exchanger located on the facility does not meet this requirement and is not subject to modernization. According to the gas temperature measurements at the inlet into the discharge chamber, its efficiency is of the order of 0.8. The question of replacing the heat exchanger arose. There is a possibility to make a heat exchanger from bimetal (stainless steel–aluminum) rolled pipes with parameters d = 12 mm, h = 6 mm, δ = 0.5 mm with a minimum ribbing pitch s = 2.5 mm. According to these parameters of the pipes (and taking into account the technological spacing between pipes), an optimum value of the transverse pitch $a \sim 3$ at a longitudinal pitch $b \sim 1.5$ is obtained. [This point falls outside the validity range of the standard approximation ([2], (5.1)) as to the Reynolds flow number.] The efficiency was calculated with allowance for the contact thermal resistance of the bimetal pipe and the resistance of the ribbing material. To take into account the possible consequences of gas superheating, in the calculations the value of the contact resistance $R_b = 1.1 \cdot 10^{-3}$ m^2·K/W was taken, which turned out to be admissible at a low density of the cooled gas. The basic parameters of the chosen variant of the heat exchanger are given in Table 2. The calculated value of the energy input $\overset{*}{Q}$ = 25 kW, and the small corrections for the finiteness of the parameters Δ and $\hat{\Delta}$ were neglected.

Conclusions. The application of the renormalization method to the calculation of a recuperative gas–liquid heat exchange apparatus has been considered. The basic characteristic of such an apparatus — efficiency — is determined primarily by the process of heat transfer in the gas. Accordingly, an actually one-parameter form of approximation of the Nusselt number for the regime of induced convection on an arbitrary heat-transfer surface, which is convenient for analytical calculations of laminar flows, has been found. This has made it possible to reduce the compact general analytical relation for calculating the heat exchanger efficiency taking into account only the convection in the gas, which was then renormalized by two transforms in order to take into account the influence of the secondary physical processes on the apparatus efficiency. To demonstrate the efficiency of the method for designing the heat exchange apparatus, the problem of minimizing the cooled gas temperature at its outlet has been considered. The problem takes into account the restrictions on the cross-section of the gas channel, the rate of flow, the pressure, and the hydraulic resistance of the heat exchanger for gas. The solution was sought in the class of designs of the apparatus assembled from staggered ribbed pipes. It has been shown that to solve this optimization problem, account of only one renormalization transform is enough. We have found a sequence of steps in which the process of optimization problem solution can be carried out practically to the end in analytic form. For the first time an optimum configuration of a staggered pipe bundle as a function of the available ribbing height has been obtained.

The author wishes to thank V. I. Polezhaev, G. S. Glushko, E. V. Teodorovich, V. S. Moskalev, and V. G. Baidulov for useful discussions of the problem. The work was supported by the Russian Foundation for Basic Research, grant 05-01-00901.

NOTATION

a, transverse relative pitch of the pipe bundle; a_d, diagonal relative pitch of the bundle; a_{max}, pitch of a row of pipes, in whose narrowness the maximum flow velocity is attained; a_g, thermal diffusivity of the gas, m^2/sec; A, area of the heat-transfer surface of the body (pipe) for gas, m^2; A_2, area of the heat exchanger surface for liquid, m^2; b, longitudinal relative pitch of the bundle; c, reduced height of the rib; C_p and C_2, specific heat capacities of the gas and liquid, respectively, J/(kg·K); d, pipe diameter at the level of the ribbing base, m; d_2, inner diameter of pipes, m; d_{max}, hydraulic diameter of the channel in the cross-section where the maximum flow velocity is attained, m; D and D^*, hydraulic diameter and mean hydraulic diameter of the channel, m; E_i, efficiency coefficient of the rib; $\langle E_i \rangle$, efficiency coefficient of the pipe surface; Eu, Euler number of the pipe row; h, height of ribs on pipes, m; K and k, hydraulic resistance and reduced hydraulic resistance; l, determining dimension in the Re_l criterion, m; L, length of the channel, plate, m; L', mean length of flow, m; L_p, pipe length, m; $\overset{*}{m}_2$, mass flow of liquid, kg/sec; Nu_D and Nu_{D^*}, Nusselt numbers determined from the values of D and D^* and the difference between the inlet temperature of the heat-transfer agent and the heat-transfer surface temperature [3]; $\mathrm{Nu}_2^{\ln}$, Nusselt number of the liquid flow calculated from the d_2 value and the logarithmic mean difference between the mass mean temperature of the liquid and the heat-transfer surface temperature; p, gas pressure, Pa; Pe_{D^*} and Pe_D, Peclet numbers calculated from the mean flow velocity $\langle w \rangle$ and the values of D^* and D; Pe_s^0, Peclet number calculated from the flow velocity ω_0 and size of s; P_{pr}, projection perimeter of the heat-transfer surface, m; Pr, Prandtl number; $\overset{*}{Q}$, thermal capacity of the heat exchanger, W; r and R, pipe radius at the base and top of ribs, m; R_b, thermal contact resistance, $m^2 \cdot K/W$; r_b, radius of the contact surface of metals in the pipe, m; Re_{D^*}, Reynolds number of the gas flow determined by the size of D^* and the mean flow velocity $\langle w \rangle$; Re_s, Reynolds number calculated by (5.1) [2]; R_g, universal gas constant, J/(mole·K); s, rib pitch along the pipe, m; s_{tr} and s_{lon}, transverse and longitudinal absolute pitches of bundle division, m; S_0 and S_0^A, total area of the unencumbered cross-section of the channel and area per one pitch of transverse division of the bundle, m^2; T, gas temperature; T' and T'', gas temperatures at the heat exchanger inlet and outlet, K; T_{mi} and T_m, determining gas temperatures for row i and the whole of the heat exchanger, K; T_∞, temperature of the full thermal relaxation of heat-transfer agents, K; T_f, heat-transfer surface temperature at the boundary with gas, K; T_2, (mass-average) temperature of the coolant, K; T_2' and T_2'', coolant temperature at the heat-exchanger input and output, K; V_s and V_Σ, volume of heat-transfer bodies and channel volume, m^3; $\overset{*}{V}$, volumetric rate of gas flow, m^3/sec; $\langle w \rangle$, mean (determining) gas velocity, m/sec; w_0, gas velocity in the unencumbered cross-section of the channel, m/sec; w_{tr} and w_d, gas velocities in the narrowness of the transverse and diagonal rows of pipes, m/sec; x, length scale in (45), m; Y, cofactor in the goal function (45); z, number of pipe rows in the heat exchanger; α, exponent in (2); β, parameter in (26); γ, exponent in the approximation of the temperature dependence of the efficiency θ_i; δ, thickness of ribs on pipes, m; Δ, determining parameter for taking into account the effect of limited liquid flow rate; $\hat{\Delta}$, determining parameter for taking into account the finite heat-transfer efficiency in the liquid; ε, determining parameter for taking into account the effect of thermal resistance of the ribbing; ε_f and ε_b, terms of the parameter ε; θ_i and θ, calculated efficiencies of the ith pipe row and the heat exchanger on the whole, with account for only the heat transfer in the gas; Θ_i and Θ, efficiencies of the ith row of pipes and the heat exchanger on the whole with account for also the thermal resistance of the ribbing; $\underline{\Theta}$, heat exchanger efficiency also taking into account the effect of downstream heating of the coolant; $\hat{\Theta}$, heat exchanger efficiency taking into account all of the effects considered; λ_1 and λ_2, heat conductivity coefficients of the rib material and liquid, W/(m·K); μ, molar mass of the gas, kg/mole; μ_E, cofactor with additional account for the efficiency of the rib; ρ, gas density, kg/m^3; ν, kinematic viscosity of the gas, m^2/sec; φ, ribbing coefficient of the pipe; χ, characteristic of the channel encumbering; Ψ, average volume fraction of voids in the channel; ψ_E, cofactor in the rib efficiency parameter. Subscripts: 0, unencumbered cross-section of the channel; 1, parameter refers to the gas (omitted everywhere); 2, liquid coolant; b, boundary of ribbing materials; d, diagonal row of the bundle; i, row (of pipes) number; f, ribbing material; g, gas; l, characteristic size in the Re_l criterion; lon, longitudinal row of the bundle; m, mean (determining) quantity; n, function analyticity domain number (see Fig. 1); max, at the maximum flow rate; p, pipe; pr, projection; s, solid; tr, transverse row of the bundle; Σ, total; $'$ and $''$, conditions at the input and output of the heat-transfer agent from the heat-exchange apparatus.

REFERENCES

1. H. Schlichting, *Boundary Layer Theory* [Russian translation], Nauka, Moscow (1969).
2. V. B. Kuntysh and A. N. Bessonnyi (Eds.), *Principles of Calculation and Design of Air-Cooled Heat Exchangers: Handbook* [in Russian], Nedra, St. Petersburg (1996).
3. W. Kast, O. Krischer, G. Reinicke, and K. Wintermantel, *Konvektive Wärme- und Stoffübertragung* [Russian translation], Energiya, Moscow (1980).
4. B. S. Petukhov, V. K. Shikova, and O. G. Martynenko (Eds.), *Handbook of Heat Exchangers* [Russian translation], Energoatomizdat, Vols. 1, 2, Moscow (1987).
5. V. F. Yudin, *Heat Transfer of Cross-Finned Tubes* [in Russian], Mashinostroenie, Leningrad (1982).
6. V. K. Migai and E. V. Firsova, *Heat Transfer and Pressure loss of Tube Bundles* [in Russian], Nauka, Leningrad (1986).

7. G. V. Gembarzhevskii, N. A. Generalov, and N. G. Solov'ev, Investigation of the spectrum of velocity fluctuations in a vortex flow of a vibrationally excited molecular gas in a glow discharge, *Izv. Ross. Akad. Nauk, Mekh. Zhidk. Gaza*, No. 2, 81–91 (2000).
8. G. V. Gembarzhevskii and N. A. Generalov, On the model of a turbulent near-wake flow in a glow discharge, *Teplofiz. Vys. Temp.*, **42**, No. 4, 501–505 (2004).