Rationale for New Formulas for Determining the Speed of Heat Carriers, Recovery Coefficients and Efficiency of Heat Exchangers of the Heat Supply System

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Abstract: The article presents new formulas for calculating heat carriers' velocities, recovery factors and efficiency of heat exchangers in the heat supply system. The formulas have been obtained after a profound analysis of literature in the field of design calculations of heat exchangers in the heat supply system and mathematical transformations. Approximate calculations based on reliable data of famous scientists were made in order to determine the validity of new formulas. The results of the calculation of the error of the average velocity of heating fluid and heated coolants were processed. The formulas can be used to design plate devices, as well as for operational monitoring of parameters in order to determine the operating modes and efficiency of heat exchangers. The formula for determining the coefficient of recovery of tubular and plate heat exchangers has been obtained as a result of an analysis of literature in the field of calculating the regenerators of gas turbine plants and mathematical transformations. By monitoring the change in performance characteristics over time, it is possible to predict deterioration in the condition of heat exchangers, provided that the tests are performed under identical conditions.

1. Introduction

In the classical version [2], the average velocity of the heating fluid (HF) in the heat exchanger channels was determined from the equation:

$$w_1 = 2 \cdot \sqrt[3]{\frac{K \cdot \Theta_m \cdot \Delta P_1}{\xi_1 \cdot C_{p1} \cdot (T_1 - T_2) \cdot \rho_1}}, \qquad (1)$$

where K - heat transfer coefficient, W/(m^2·K); Θ_m - average log temperature, K: $\Theta_m = (\Theta_1 - \Theta_2)/\ln(\Theta_1/\Theta_2)$, where $\Theta_1 = (T_1 - \tau_2)$, $\Theta_2 = (T_2 - \tau_1)$ - the largest and smallest temperature difference between HF and the heated coolant respectively, τ_1 and τ_2, K. If $0{,}5 \le \frac{\Theta_1}{\Theta_2} \le 2$, the average temperature head can be defined as the arithmetic mean $\Theta_m = (\Theta_1 + \Theta_2)/2$ (the error is up to 4 %) [1, 2]; ΔP_1 - pressure loss according to HF, Pa; ξ_1 - coefficient of total hydraulic resistance of the reduced length of one channel of HF; C_{p1} - average heat capacity of HC, J/(kg·K); $(T_1 - T_2) = \Delta T$ - average temperature difference in HF, K; ρ_1 - average density of HF, kg/m^3.

After the analysis of thermal calculations of different types of heat exchangers (HE) in the heat supply system and hot water supply, it is established that there is no equation for the direct calculation of the average velocity of the coolant. It is adopted, as in CS-41-101-95, or calculated by the method of successive approximations [3]. An exception is the equation for the direct calculation of the average velocity of combustion products in the regenerators of gas turbine plants [1].

2. Materials and methods

It is known from the source [2] that the coefficient of the total hydraulic resistance of the reduced length of one HF is:

$$\xi_1 = \frac{B_1}{\mathrm{Re}_1^{0,25}} = \frac{\nu_1^{0,25} \cdot B_1}{w_1^{0,25} \cdot d_{1e}^{0,25}}, \tag{2}$$

where $\mathrm{Re}_1 = \frac{w_1 \cdot d_{1e}}{\nu_1}$ - Reynolds number, w_1 - velocity of the heating fluid, m/s; $d_{1э}$ - equivalent diameter of one channel of HF, m; ν_1 - average kinematic viscosity of the coolant, m^2/s; B_1 - empirical coefficient, depending on the type of a plate.

Substituting the value of ξ1 into formula (1), we obtained:

- for HF $w_1^3 = 8 \cdot \frac{w_1^{0,25} \cdot d_{1e}^{0,25} \cdot \Theta_m \cdot \Delta P_1 \cdot K}{\nu_1^{0,25} \cdot B_1 \cdot C_{p1} \cdot \Delta T \cdot \rho_1^2}$ or $w_1^{2,75} = 8 \cdot \frac{d_{1e}^{0,25} \cdot \Theta_m \cdot \Delta P_1 \cdot K}{\nu_1^{0,25} \cdot B_1 \cdot C_{p1} \cdot \Delta T \cdot \rho_1^2}$, (3)

- for heated coolant $w_2^{2,75} = 8 \cdot \frac{d_{2e}^{0,25} \cdot \Theta_m \cdot \Delta P_2 \cdot K}{\nu_2^{0,25} \cdot B_2 \cdot C_{p2} \cdot \Delta\tau \cdot \rho_2^2}$, (4)

where $\Delta\tau = \tau_2 - \tau_1, K$.

The ratio of the average speeds of HF and heated coolant:

$$\frac{w_1}{w_2} = \left(\left(\frac{d_{1e}^{0,25} \cdot \Delta P_1}{\nu_1^{0,25} \cdot B_1 \cdot C_{p1} \cdot \Delta T \cdot \rho_1^2}\right) / \left(\frac{d_{2e}^{0,25} \cdot \Delta P_2}{\nu_2^{0,25} \cdot B_2 \cdot C_{p2} \cdot \Delta\tau \cdot \rho_2^2}\right)\right)^{0,36364}. \tag{5}$$

An equation is known from the source [4, 5] (with allowance for heat losses, η):

$$\frac{\Delta T}{\Delta\tau} = \frac{T_1 - T_2}{\tau_2 - \tau_1} = \frac{W_2}{\eta \cdot W_1} = \frac{M_2 \cdot C_{p2}}{\eta \cdot M_1 \cdot C_{p1}}. \tag{6}$$

For engineering calculations of heat exchangers, η = 1 [4]. From equation (6) we have obtained the formula $\frac{C_{p2} \cdot \Delta\tau}{C_{p1} \cdot \Delta T} = \frac{M_1}{M_2}$ (without allowance for thermal losses, η = 1), which is substituted into formula (5). Then:

$$\frac{w_1}{w_2} = \left(\frac{\Delta P_1 \cdot B_2 \cdot d_{1e}^{0,25} \cdot \nu_2^{0,25} \cdot M_1 \cdot \rho_2^2}{\Delta P_2 \cdot B_1 \cdot d_{2e}^{0,25} \cdot \nu_1^{0,25} \cdot M_2 \cdot \rho_1^2}\right)^{0,36364}. \tag{7}$$

It is known that the heat transfer coefficient of HE [3], W / m^2 · K:

$$K = \frac{\beta}{\frac{1}{\alpha_1} + \frac{\delta}{\lambda_{wall}} + \frac{1}{\alpha_2}} \approx \frac{\varphi \cdot \beta}{\frac{1}{\alpha_1} + \frac{1}{\alpha_2}} \approx \frac{\varphi \cdot \beta \cdot \alpha_1}{1 + \frac{\alpha_1}{\alpha_2}} \tag{8}$$

where $\varphi = K / K_{wall} = 0{,}89 \div 0{,}9$ - the ratio of the value of the heat transfer coefficient (K), taking into account the thermal resistance of the plate wall $(\delta_{wall} / \lambda_{wall})$ to the value of the heat transfer coefficient (K_{wall}) without taking into account the resistance $(\delta_{wall} / \lambda_{wall})$; β - coefficient that takes into account the decrease in the heat transfer coefficient due to the thermal resistance of scale and impurities on the plate, depending on the water quality, β = 0.7 ÷ 0.85; α_1 - heat transfer coefficient of HF, W/(m^2·K); α_2 - heat transfer coefficient of heated coolant, W/(m^2·K); δ_w - plate wall thickness, m; λ_w - coefficient of wall thermal conductivity, W/(m·K).

It is known that the coefficient of heat transfer from HF under turbulent motion of the coolant in lamellar HE (Re1> 50) [2], W/(m^2·K) is:

$$\alpha_1 = \frac{Nu_1 \cdot \lambda_1}{d_{1e}} = C \cdot \mathrm{Re}_1^{0,73} \cdot \mathrm{Pr}_1^{0.43} \cdot \left(\frac{\mathrm{Pr}_1}{\mathrm{Pr}_{wall}}\right)^{0,25} \cdot \frac{\lambda_1}{d_{1e}} = C \cdot \left(\frac{w_1 \cdot d_1}{\nu_1}\right)^{0,73} \cdot \left(\frac{\mathrm{Pr}_1^{0,68}}{\mathrm{Pr}_{wall}^{0,25}}\right) \cdot \frac{\lambda_1}{d_{1e}} \tag{9}$$

where C - empirical coefficient depending on the type of a plate.

Heat transfer coefficient of HE from heated coolant [2], W/(m^2·K):

$$\alpha_2 = \frac{Nu_2 \cdot \lambda_2}{d_{2e}} = C \cdot \mathrm{Re}_2^{0,73} \cdot \left(\frac{\mathrm{Pr}_2^{0,68}}{\mathrm{Pr}_{wall}^{0,25}}\right) \cdot \frac{\lambda_2}{d_{2e}}. \tag{10}$$

Hence, the ratio of heat transfer coefficients is:

$$\frac{\alpha_1}{\alpha_2} = (C \cdot \mathrm{Re}_1^{0,73} \cdot \left(\frac{\mathrm{Pr}_1^{0,68}}{\mathrm{Pr}_{wall}^{0,25}}\right) \cdot \frac{\lambda_1}{d_{1e}})/(C \cdot \mathrm{Re}_2^{0,73} \cdot \left(\frac{\mathrm{Pr}_2^{0,68}}{\mathrm{Pr}_{wall}^{0,25}}\right) \cdot \frac{\lambda_2}{d_{2e}}) =$$

$$= \left(\frac{\mathrm{Re}_1}{\mathrm{Re}_2}\right)^{0,73} \cdot \left(\frac{\mathrm{Pr}_1}{\mathrm{Pr}_2}\right)^{0,68} \cdot \frac{\lambda_1}{\lambda_2} \cdot \frac{d_{2e}}{d_{1e}} = \left(\frac{w_1 \cdot d_{1e} \cdot \nu_2}{w_2 \cdot d_{2e} \cdot \nu_1} \cdot\right)^{0,73} \cdot \left(\frac{\mathrm{Pr}_1}{\mathrm{Pr}_2}\right)^{0,68} \cdot \frac{\lambda_1}{\lambda_2} \cdot \frac{d_{2e}}{d_{1e}} =$$

$$= \left(\frac{\Delta P_1 \cdot B_2 \cdot d_{1e}^{0,25} \cdot \nu_2^{0,25} \cdot M_1 \cdot \rho_2^2}{\Delta P_2 \cdot B_1 \cdot d_{2e}^{0,25} \cdot \nu_1^{0,25} \cdot M_2 \cdot \rho_1^2}\right)^{0,2655} \cdot \left(\frac{d_{1e} \cdot \nu_2}{d_{2e} \cdot \nu_1} \cdot\right)^{0,73} \cdot \left(\frac{\mathrm{Pr}_1}{\mathrm{Pr}_2}\right)^{0,68} \cdot \frac{\lambda_1}{\lambda_2} \cdot \frac{d_{2e}}{d_{1e}} =$$

$$= \left(\frac{\Delta P_1 \cdot B_2 \cdot M_1 \cdot \rho_2^2}{\Delta P_2 \cdot B_1 \cdot M_2 \cdot \rho_1^2}\right)^{0,2655} \cdot \left(\frac{\mathrm{Pr}_1}{\mathrm{Pr}_2}\right)^{0,68} \cdot \left(\frac{d_{1e}}{d_{2e}}\right)^{0,2036} \cdot \left(\frac{\nu_2}{\nu_1}\right)^{0,7964} \cdot \left(\frac{\lambda_1}{\lambda_2}\right). \tag{11}$$

Substituting the values of α1 from formula (11) and the relation (α_1 / α_2) from formula (12) into formula (8), we obtained:

$$K \approx \frac{\varphi \cdot \beta \cdot \alpha_1}{1 + \frac{\alpha_1}{\alpha_2}} \approx \frac{\varphi \cdot \beta \cdot \left(C \cdot \left(\frac{w_1 \cdot d_{1e}}{\nu_1}\right)^{0,73} \cdot \left(\frac{\mathrm{Pr}_1^{0,68}}{\mathrm{Pr}_{wall}^{0,25}}\right) \cdot \frac{\lambda_1}{d_{1e}}\right)}{\left(1 + \left(\frac{\Delta P_1 \cdot B_2 \cdot M_1 \cdot \rho_2^2}{\Delta P_2 \cdot B_2 \cdot M_2 \cdot \rho_1^2}\right)^{0,2655} \cdot \left(\frac{\mathrm{Pr}_1}{\mathrm{Pr}_2}\right)^{0,68} \cdot \left(\frac{d_{1e}}{d_{2e}}\right)^{0,2036} \cdot \left(\frac{\nu_2}{\nu_1}\right)^{0,7964} \cdot \left(\frac{\lambda_1}{\lambda_2}\right)\right)}. \tag{12}$$

Substituting the values of K from formula (12) into formula (4), we obtain:

$$w_1^{2,75} = 8 \cdot \frac{d_{1e}^{0,25} \cdot \Theta_m \cdot \Delta P_1 \cdot K}{\nu_1^{0,25} \cdot B_1 \cdot C_{p1} \cdot \Delta T \cdot \rho_1^2} = 8 \cdot \frac{d_{1e}^{0,25} \cdot \Theta_m \cdot \Delta P_1}{\nu_1^{0,25} \cdot B_1 \cdot C_{p1} \cdot \Delta t \cdot \rho_1^2}.$$

$$\cdot\frac{\varphi\cdot\beta\cdot\left(C\cdot\left(\frac{w_1\cdot d_{1e}}{\nu_1}\right)^{0,73}\cdot\left(\frac{\Pr_1^{0,68}}{\Pr_{wall}^{0,25}}\right)\cdot\frac{\lambda_1}{d_{1e}}\right)}{\left(1+\left(\frac{\Delta P_1\cdot B_2\cdot M_1\cdot\rho_2^2}{\Delta P_2\cdot B_2\cdot M_2\cdot\rho_1^2}\right)^{0,2655}\cdot\left(\frac{\Pr_1}{\Pr_2}\right)^{0,68}\cdot\left(\frac{d_{1e}}{d_{2t}}\right)^{0,2036}\cdot\left(\frac{\nu_2}{\nu_1}\right)^{0,7964}\cdot\left(\frac{\lambda_1}{\lambda_2}\right)\right)}. \quad (13)$$

After the reduction, we got:

$$w_1^{2,02}=\left(\frac{8\cdot\varphi\cdot\beta\cdot C\cdot\Pr_1^{0,68}\cdot\lambda_1\cdot\Theta_m\cdot\Delta P_1}{d_{1e}^{0,02}\cdot\Pr_{wall}^{0,25}\cdot B_1\cdot\nu_1^{0,98}\cdot C_{p1}\cdot\Delta T\cdot\rho_1^2}\right)/(1+\left(\frac{\Delta P_1\cdot B_2\cdot M_1\cdot\rho_2^2}{\Delta P_2\cdot B_1\cdot M_2\cdot\rho_1^2}\right)^{0,2655}\cdot$$

$$\cdot\left(\frac{\Pr_1}{\Pr_2}\right)^{0,68}\cdot\left(\frac{d_{1e}}{d_{2e}}\right)^{0,2036}\cdot\left(\frac{\nu_2}{\nu_1}\right)^{0,7964}\cdot\left(\frac{\lambda_1}{\lambda_2}\right)). \quad (14)$$

As a result of mathematical transformations, new formulas were obtained for calculating the average velocity of heat carriers [8]:

- heating coolant in the channels of lamellar HE, m/s:

$$w_1=\left(\left(\frac{8}{d_{1e}^{0,02}}\cdot\frac{\varphi\cdot\beta\cdot C\cdot\Pr_1^{0,68}\cdot\lambda_1\cdot\Theta_m\cdot\Delta P_1}{\Pr_{wall}^{0,25}\cdot B_1\cdot\nu_1^{0,98}\cdot C_{p1}\cdot\Delta T\cdot\rho_1^2}\right)\Big/1+\left(\frac{\Delta P_1\cdot B_2\cdot M_1\cdot\rho_2^2}{\Delta P_2\cdot B_1\cdot M_2\cdot\rho_1^2}\right)^{0,2655}\cdot$$

$$\cdot\left(\frac{\Pr_1}{\Pr_2}\right)^{0,68}\cdot\left(\frac{d_{1e}}{d_{2e}}\right)^{0,2036}\cdot\left(\frac{\nu_2}{\nu_1}\right)^{0,7964}\cdot\left(\frac{\lambda_1}{\lambda_2}\right)\Bigg)\Bigg)^{0,49505} \quad (15)$$

- heated coolant in the channels of HE, m/s:

$$w_2=w_1\Big/\left(\frac{\Delta P_1\cdot B_2\cdot d_{1e}^{0,25}\cdot\nu_2^{0,25}\cdot M_1\cdot\rho_2^2}{\Delta P_2\cdot B_1\cdot d_{2e}^{0,25}\cdot\nu_1^{0,25}\cdot M_2\cdot\rho_1^2}\right)^{0,36364} \quad (16)$$

To verify the validity of the obtained formulas, the initial data for a plate heat exchanger with plates 0.3 with the technical characteristics presented in the literature [2, p. 60] were used.

Initial data: area HE - F = 20 m^2; the number of moves - $X_1 = X_2 = 1$; the reduced length of the channel - $L_{pr1} = L_{pr2} = 1.12$ m; consumption of HF - $M_1 = 30$ t/h; consumption of heated coolant - $M_2 = 35$ t/h; $\varphi = 0.89$; $\beta = 0.85$; $B_1 = B_2 = 19.3$; C = 0.1; $d_{1e}=d_{2e}=0{,}008$ m; $T_1 = 353$ K; $T_2 = 333$ K; $\tau_1 = 319$ K; $\tau_2 = 336.2$ K; $\Delta P_1 = 8338$ Pa; $\Delta P_2 = 10889$ Pa.

Calculation*:* $C_{p1} = 4199.8\ \frac{J}{kg\cdot K}$; $C_{p2} = 4194.1\ \frac{J}{kg\cdot K}$; $\rho_1 = 976.4\ \frac{kg}{m^3}$; $\rho_2 = 983.5\ \frac{kg}{m^3}$; $\Theta_m = 15.36$ K; $\nu_1 = 0.41\cdot10^{-6}\ \frac{m^2}{s}$; $\nu_2 = 0.51\cdot10^{-6}\ \frac{m^2}{s}$; $\lambda_1 = 0.658\frac{W}{m\cdot K}$; $\lambda_2 = 0.645\frac{W}{m\cdot K}$; $\Pr_1 = 2.5$; $\Pr_2 = 3.1$; $\Pr_{wall} = 2.77$; $w_1 = 0.228\ \frac{m}{s}$; $w_2 = 0.257$ m/s; Q = 699996 W; K = $2389\frac{W}{m^2\cdot K}$.

Average speed of the heating fluid (HF), m/s:

$$w_1=\left(\left(\frac{8}{0{,}008^{0,02}}\cdot\frac{0{,}89\cdot0{,}75\cdot0{,}1\cdot2{,}5^{0,68}\cdot0{,}658\cdot15{,}36\cdot8338}{2{,}77^{0,25}\cdot19{,}3\cdot(0{,}41\cdot10^{-6})^{0,98}\cdot4199{,}7\cdot20\cdot976{,}4^2}\right)\Big/(1+\right.$$

$$+\left(\frac{8388\cdot 19{,}3\cdot 30\cdot 983{,}5^2}{10889\cdot 19{,}3\cdot 35\cdot 976{,}4^2}\right)^{0{,}2655}\cdot\left(\frac{0{,}008}{0{,}008}\right)^{0{,}2036}\cdot\left(\frac{0{,}51\cdot 10^{-6}}{0{,}41\cdot 10^{-6}}\right)^{0{,}7964}\cdot\left(\frac{0{,}658}{0{,}645}\right)\Bigg)\Bigg)^{0{,}49505}=0{,}238.$$

The error of the calculation, %: $\delta_1=\frac{w_{10}-w_1}{w_{10}}\cdot 100=\left|\frac{0{,}238-0{,}224}{0{,}238}\right|\cdot 100=5{,}9$.

Average speed of the heated coolant, m/s:

$$w_2 = 0{,}238\Big/\left(\frac{8338\cdot 19{,}3\cdot 0{,}008^{0{,}25}\cdot(0.51\cdot 10^{-6})^{0{,}25}\cdot 30\cdot 983{,}5^2}{10889\cdot 19{,}3\cdot 0{,}008^{0{,}25}\cdot(0{,}41\cdot 10^{-6})^{0{,}25}\cdot 35\cdot 976{,}4^2}\right)^{0{,}36364} = 0{,}255\,.$$

The error of the calculation, %:

$$\delta_2=\frac{w_{20}-w_2}{w_{20}}\cdot 100=\left|\frac{0{,}257-0{,}255}{0{,}257}\right|\cdot 100=0{,}78.$$

Heat transfer coefficient, W/ (m^2·K):

$$K=\frac{w_1\cdot C_{p1}(t_1-t_2)\cdot\rho_1\cdot d_{1э}}{4\cdot\Theta_m\cdot L_{pr}\cdot X_1}=\frac{0{,}238\cdot 4199{,}8(80-60)\cdot 976{,}4\cdot 0{,}008}{4\cdot 15{,}35\cdot 1{,}12\cdot 1}=2270{,}6\,.$$

The error of the calculation, %: $\delta_3=\frac{K_o-K}{K_o}\cdot 100=\left|\frac{2389-2270{,}6}{2389}\right|100=4{,}96$.

Area of the heat exchange surface, m^2: $F=\frac{Q}{K\cdot\Theta_m}=\frac{699996}{2270{,}6\cdot 15{,}36}=20{,}07$.

The error of the calculation, %: $\delta_4=\frac{F_o-F}{F_o}\cdot 100=\left|\frac{20{,}07-20}{20{,}07}\right|\cdot 100=0{,}35$.

For the operational control of the parameters of the operation modes of the apparatus in urban heat supply systems, the recovery factor has been received and is recommended to use additionally [4, 5, 6].

The algorithm of mathematical transformations is the following [4, 6]:

1. According to the equation of heat balance and heat transfer:

$$Q=M_1C_{p_1}\eta(T_1-T_2)=M_2C_{p_2}(\tau_2-\tau_1)=KF\Theta_m\,, \tag{17}$$

whence it follows that: $$KF=\frac{M_2\cdot C_{p2}(\tau_2-\tau_1)}{\Theta_m}. \tag{18}$$

2. The equation, according to the sources [6, 7]: $$KF=\frac{2\cdot W_1\cdot W_2\cdot(T_1-\tau_1-\Theta_m)}{(W_1+W_2)\cdot\Theta_m}. \tag{19}$$

3. From equations (18 and 19) we obtain:

$$\frac{M_2\cdot C_{p2}(\tau_2-\tau_1)}{\Theta_m}=\frac{2\cdot W_1\cdot W_2\cdot(T_1-\tau_1-\Theta_m)}{(W_1+W_2)\cdot\Theta_m}, \tag{20}$$

whence it follows that: $(T_1-\tau_1)-\Theta_m=\frac{(W_1+W_2)\cdot(\tau_2-\tau_1)}{2\cdot W_1}$,

or: $$\Theta_m=(T_1-\tau_1)-\frac{(W_1+W_2)\cdot(\tau_2-\tau_1)}{2\cdot W_1}=\left[1-\left(\frac{W_1+W_2}{2\cdot W_1}\right)\cdot\left(\frac{\tau_2-\tau_1}{T_1-\tau_1}\right)\right]\cdot(T_1-\tau_1). \tag{21}$$

Paragraph 3 contains a complex which characterizes the degree of recovery:

$$\left(\frac{W_1 + W_2}{2 \cdot W_1}\right) \cdot \left(\frac{\tau_2 - \tau_1}{t_1 - \tau_1}\right) = R \,. \tag{22}$$

4. Equation 3, with equation (21) taken into account, is transformed into the following:

$$\Theta_m = (1 - R) \cdot (t_1 - \tau_1)\,. \tag{23}$$

5. Hence, according to the literature [6, 8]: $R = 1 - \dfrac{\Theta_m}{t_1 - \tau_1}$. (24)

We modernize the known characteristics, such as ε - efficiency HE (dimensionless specific heat load of the apparatus), NTU - the number of units of heat transfer, presented in the works of scientists Zinger N.M., Migai V.G., Sokolov E.Y., taking into account equation (23):

- counterflow: $$\varepsilon = \frac{q}{W_{\min}} = \frac{Q}{(t_1 - \tau_1) \cdot W_{\min}} = \frac{Q(1 - R)}{\Theta_m \cdot W_{\min}} =\leq 1; \tag{25}$$

- straight flow: $$\varepsilon = \frac{1 - \exp\left[-NTU\left(1 + \dfrac{W_{\min}}{W_{\max}}\right)\right]}{1 + \dfrac{W_{\min}}{W_{\max}}} \leq 1\,. \tag{26}$$

The number of units of heat transfer: $$NTU = \frac{KF}{W_{\min}} = \frac{Q}{\Theta_m \cdot W_{\min}} = \frac{\varepsilon}{1 - R} \tag{27}$$

where $W_{\min}$, $W_{\max}$ - smaller and larger values of the water equivalent of heat carriers, $W = M \cdot C_p$.

The average mass heat capacities of water are calculated by equations [1]:

$$C_{p_1} = 4{,}187 + 1{,}05 \cdot 10^{-10} \cdot (t_{mean} + 35)^4 \tag{28}$$;

$$C_{p_2} = 4{,}187 + 1{,}05 \cdot 10^{-10} \cdot (\tau_{mean} + 35)^4 \tag{28a}$$

The average logarithmic temperature of the heat carriers (θ_m) was calculated by equations [1, 2], K:

- for plate-type devices:

- counterflow: $$\Theta_m = \frac{(t_1 - \tau_2) - (t_2 - \tau_1)}{\ln \dfrac{t_1 - \tau_2}{t_2 - \tau_1}}\,; \tag{29}$$

- straight flow: $$\Theta_m = \frac{(t_1 - \tau_1) - (t_2 - \tau_2)}{\ln \dfrac{t_1 - \tau_1}{t_2 - \tau_2}}\,;$$

(30)

- for shell-and-tube steam-water devices:

$$\Theta_m = \frac{\tau_2 - \tau_1}{\ln \dfrac{t_н - \tau_1}{t_н - \tau_2}}\,, \tag{31}$$

where $t_н$ - coexistence temperature of water vapor, $°C$, which is determined by the vapor pressure in the apparatus according to the tables of the thermophysical properties of water and steam.

In plate heat exchangers, in general, a counterflow motion of coolants is organized. The mode of the coolants motion is determined by the value of the Reynolds number (Re). If Re > 50, then the regime is turbulent, if Re < 50, then the mode of the coolants motion is laminar [2]. Therefore, the equation for calculating the Nusselt number is selected depending on the mode of motion of the coolant in the heat exchanger channels.

To test the validity of the formulas (24; 25; 27), we compile Table 1 with the initial data and the results of the calculation. Table 1 shows the initial data for plate HEs with the area of heat exchange of one plate F1 = 0.3 m^2 from the literature source [2], according to the specification for FP-14-73-1 ("Funke-Tyumen" firm), and for shell-and-tube steam-water HE (HWH-5000-3,5-8) are the initial data from the source of literature [1].

In Table 1 $T_1 = T_н = 403{,}83\,K$ according to literature [1] for HWH.

Table 1. Initial data and calculation of efficiency coefficients of HE

№	Parameter	Size	Value			Note
			Plate		Tubular HWH-5000--3,5-8	
			0,3	FP-14		
1.	Q	kW	700.0	589.2	164808.0	Manufacturer's data
2.	T_1	K	353	393	403	
3.	T_2	K	333	336	-	
4.	T_{av}	K	343.0	364.5	-	$T_{av} = \frac{T_1 + T_2}{2}$
5.	$C_{p_г}$	$\frac{kJ}{\kappa g \cdot K}$	4.200	4.214	-	Equation (28)
6.	M_1	$\frac{\kappa g}{s}$	8.333	2.453	-	Manufacturer's data
7.	$W_1 = W_{\min}$	$\frac{\kappa W}{K}$	34.996	10.337	-	Equation (6)
8.	τ_1	K	319.0	333.0	378.7	Manufacturer's data
9.	τ_2	K	336.2	353.0	402.0	
10.	τ_{av}	K	327.6	343.0	390.4	$\tau_{av} = \frac{\tau_1 + \tau_2}{2}$
11.	C_{rv}	$\frac{kJ}{kg \cdot K}$	4.1941	4.1997	4.2440	Equation (28a)
12.	M_2	$\frac{kg}{s}$	9.72	7.01	1666.67	Manufacturer's data
13.	$W_2 = W_{\max}$	$\frac{kW}{K}$	40.78	29.44	7073.30	Equation (6)
14.	Θ_m	K	15.35	14.28	8.89	Equations (29, 31)
15.	R	-	0.549	0.762	0.647	Equation (24)
16.	$\varepsilon = \frac{Q(1-R)}{\Theta_m \cdot W_{\min}} = \leq 1$	-	0.588	0.950	0.930	Equation (25)
17.	$NTU = \frac{\varepsilon}{1 - R}$	-	1.30	3.99	2.64	Equation (27)

3. Conclusion

1. The results of calculation of errors in the average HF speed (up to 6%) and heated coolant speed (up to 1%) show that the reliability of formulas (15 and 17) is satisfactory. The formulas can be used for the design of plate devices, as well as for operational monitoring of parameters in order to determine the operating modes and efficiency of heat exchangers (HEs).

2. The analysis of Table 1 shows that when determining the efficiency of HF, the recovery factor for tubular and plate heat exchangers of the heat supply system is within the limits of $R = 0{,}5 \div 0{,}7$. Additional efficiency factors of HE (ε - dimensionless specific heat load of apparatus, NTU - number of heat transfer units) are determined considering the value of R. Calculation of such characteristics as R, ε, NTU shows the effectiveness of HE operation provided that the tests are performed under identical conditions.

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