Analysis of heat transfer for gas flow in the multichannel plate heat exchanger depending on the gas Prandtl number

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Abstract. The analysis of an experimental investigation of heat transfer for a gas flow into the multichannel plate heat exchanger depending on the gas Prandtl number is presented. The gas mixture with Prandtl number of 0.23 and 0.7 were used. It is shown that the design procedure by the method a local thermal balance possesses an adequate accuracy for verification design equation for Nusselt number, including definition of the conditions validity of the model of flow regime heat exchanger.

1. Introduction
Modern high-power reactors have ports with the complicated geometry of a heat exchange surface. It impedes the study of performances of flow and heat exchange parameters in such channels required for the designing the heat exchangers based on use of various computing methods. Therefore additional verification of a procedure performance of such computation in some cases is required. In particular, at use heat carrier from a mixture of gases with strongly various molecular weights, there is a problem of computation of heat exchange because Prandtl number Pr for such mixtures can have the meanings of the order of 0.2, while the majority of trusty rated dependences is applicable for gases with Pr > 0.5 [1, 2]. In particular, identification of the influence degree of the duct entrance region for a turbulent flow in the channels of the intricate cross-section form is required [1, 3]. In our work, a series of experiments for the analysis of adequacy of computational methods of the multichannel plate heat exchanger was executed for the mixed gas with Prandtl number Pr = 0.23.

2. Experimental part
An experimental investigation of heat transfer for a gas flow into the multichannel plate heat exchanger on the closed circuit hydraulic was executed. The heat exchanger has two equality sections A, B isolated from each other. Each section has 20 plate channels. Schematic of the plate heat transfer exchanger, arrangement of channels and streamlines of the heat-transfer medium (for a case of counterflow) are shown in figure 1. The common open area of each section is $S_{\text{cross}} = 57.7 \text{ mm}^2$, total area of heat exchange surface of each section is $S_{\text{heat}} = 0.83 \text{ m}^2$. Heat-transfer medium was feed into and discharge out from heat exchanger through pipe nipples in diameter of 16 mm (figure 1). The dimensions of the plate channel of the heat exchanger were as follows: length $L = 283.8 \text{ mm}$, width $H = 73.7 \text{ mm}$, height $h = 0.78 \text{ mm}$, wall thickness from stainless steel $\delta = 0.32 \text{ mm}$. Estimated value of the hydraulic diameter of the channel was $D_h = \frac{4 \cdot S}{\Pi} = 1.549 \text{ mm}$. Water was pumped into the heat exchanger for a cooling from the regulated thermostat.
The experiments were carried out for the mode of cooling the gas mixture with distilled water. The thermal-hydraulic measuring circuit is shown in Figure 2. The main measured parameters are: volume flow rate $Q$, temperature $T$ and pressure $P$ of heat transfer media at the inlet and outlet of the heat exchanger, heat load $W$ in the device of pre-heating the gas mixture. The temperature and flow rate of water inflowing the heat exchanger $T_{w1}$, $Q_w$ were set so that the gas temperature at the exit from the heat exchanger $T_{g2}$ was equal to the gas temperature at the entrance to the preheating section $T_{g0}$. Such a method made it possible to control the amount of heat transferred from gas to liquid in a heat exchanger. The complete heat capacity of the water flow per unit time $C = G \cdot C_p$ was calculated and compared with the warming of water $(T_{w1} - T_{w0})$ by heat load $W$.

![Figure 1. Schematic of the heat transfer exchanger.](image1)

![Figure 2. Thermal-hydraulic measuring circuit.](image2)

A series of preliminary tests of a heat exchanger using distilled water Pr $\sim$ 7 as a coolant and dry air Pr $\sim$ 0.7 showed that the laminar flow regime of cooling water can be used to calculate the friction and heat transfer by the method proposed in [4]. To reduce the error of heat transfer calculations in the heat exchanger, the temperature and flow rate of the gas mixture were selected so that the flow of water in the cooling channels was laminar. In this case, for calculations with the flow of water in the cooling section, equations (1), (2) from [4] were used:
Nu = \left[ 1 - \frac{0.8}{\frac{\nu}{\text{Pe} \cdot D_h \cdot \Phi^3}} \right]^{\frac{1}{2}} + \frac{4.218}{\frac{x}{\text{Pe} \cdot D_h \cdot \Phi^3}} \right]^{\frac{1}{2}} \right]

(1)

\Delta P = \Psi \cdot \frac{64 \cdot L}{\text{Re} \cdot D_h} \cdot \frac{\rho \cdot u^2}{2}

(2)

where \( \Phi = 1 + \frac{(c - 1) \cdot 3 \cdot (0.5 \cdot d_s)^{\frac{1}{6}} - (1 + d_s)}{c + 0.25} \); \( \Psi = 1 + \frac{(c - 1) \cdot 3/8 \cdot d_s^2 \cdot (3 - d_s) - 1}{c + 0.33 \cdot d_s^{2.25}} \); \( c = \frac{S}{S_h} \);

d_s = \frac{D_s}{D_{max}} \), and accordingly for size of the plate channels in the heat exchanger \( \Phi = 0.998, \Psi = 1.475 \).

For the laminar mode of water flow, the pressure drop in the plate straight channel \( \Delta P_p \) is proportional to the flow velocity in the first degree. But due to the pressure drop at the inlet and outlet nozzles, the total pressure drop on the heat exchanger non-linearly depends on the water flow rate due to the pressure loss on local hydraulic resistances, which can be calculated using the Weisbach equation, knowing the value of the local drag coefficient \( \xi_{loc} \). Figure 3 shows the pressure drop \( \Delta P_e = P_{w1} - P_{w2} \) depending on the mass flow rate of water \( G \). The graph also shows the values of the pressure drop in the plate straight channel \( \Delta P_p = \Delta P_e - 2 \cdot \xi_{loc} \cdot (\rho_w \cdot U_w / 2) \) for different values of the local coefficient of friction resistance \( \xi_{loc} (0.25, 0.35, 0.45) \). It can be seen that for \( \xi_{loc} \) values equal to 0.25 and 0.45, the dependence has a nonlinear appearance and the curves are “bent” in different directions with respect to the straight line obtained with the value \( \xi_{loc} = 0.34 \). The obtained value of the local friction coefficient \( \xi_{loc} = 0.34 \) was also used in the calculations for the flow of the gas mixture.

![Figure 3 Pressure drop for water depending on mass flow rate.](image)

3. Result and Discussion

To calculate heat transfer during gas movement in the heat exchanger channels, various dependences were used for the Nusselt number \( \text{Nu} \) and friction coefficients \( \zeta \) depending on the flow regime of the coolant ([1], [5], [6]). The calculations were carried out using the heat balance equations along the flow, using local values of the friction coefficients \( \zeta \) and heat transfer \( \text{Nu} \) on the walls of the heat exchanger, taking into account the temperature difference on the wall separating gas and water. For a flat wall, the heat fluxes into liquid \( q_w, \) through the wall \( q_{wall}, \) from gas \( q_g \) are equal \( q_g = q_{wall} = q_w. \) For gas in section \( x \) from the channel entrance, the heat flux is

\[
q_g = \frac{\text{Nu}_g \cdot \lambda_g}{D_h} \left[ T_g(x) - T_{g, wall}(x) \right],
\]

the heat
transfer coefficient is \( \alpha_g = \frac{\nu_g \cdot \lambda_g}{D_h} \), for water in section \( x \) from the channel entrance, the heat flux is
\[
q_w = \frac{\nu_w \cdot \lambda_w}{D_h} \left[ T_{w,wall}(x) - T_w(x) \right],
\]
the heat transfer coefficient is \( \alpha_w = \frac{\nu_w \cdot \lambda_w}{D_h} \), for wall in section \( x \) from the channel entrance, the heat flux is
\[
q_{wall} = \frac{\lambda_{wall}}{\delta_{wall}} \left[ T_{g,wall}(x) - T_{w,wall}(x) \right].
\]
Accordingly, the temperature difference for the flow of gas in section \( x \) is
\[
\left[ T_g(x) - T_{g,wall}(x) \right] = \frac{\alpha_g \cdot \lambda_{wall} \cdot \left( \alpha_g + \alpha_w \right) + \alpha_w \cdot \alpha_g \cdot h_{wall}}{\lambda_{wall}} \]
and in the local heat exchange area of \( \delta S \) for gas flow cooling it will be
\[
\delta T_g(x) = \frac{\nu_g \cdot \lambda_g}{D_h \cdot C_g} \left[ T_g(x) - T_{wall} \right] \delta S
\]
and for water flow warmup it will be
\[
\delta T_w(x) = \frac{\nu_w \cdot \lambda_w}{D_h \cdot C_w} \left[ T_w(x) - T_{wall} \right] \delta S.
\]
Experimental values of temperature and pressure at the inlet to the heat exchanger \( T_{g1}, P_{g1} \) were taken as the initial values in the counterflow mode for the gas mixture, and the values at the outlet of the heat exchanger \( T_{w2} \) and \( P_{w2} \) were taken as the initial values for the water. The mass flow was calculated from the results of measuring the volume flow and reference data for density depending on the temperature and pressure at the outlet of the flow meter \( G_g = Q_g \cdot \rho_g(T_{g0}, P_{g0}) \) and \( G_w = Q_w \cdot \rho_w(T_{w0}, P_{w0}) \). At performing thermal-hydraulic calculations, the number of discretization cells along the length of the channel was chosen so that the accuracy of calculations based on temperature exceeded the accuracy of measurements by 3–4 times. For example, Fig. 4 shows the calculated curves for the mixture gas with \( \Pr = 0.23 \) at gas volume flow rate \( Q_g = 16 \text{ l/s} \), water flow rate \( Q_w = 0.03 \text{ l/s} \), using various dependences for the Nusselt number \( \nu \) from Reynolds number \( Re \), the Prandt number for \( Pr \) and the dimensionless distance from the channel entrance \( x/D_h \).

The computation of \( E_1 \) was carried out according to the Petukhov-Popov-Gnielinski equation (3) [4, 5], which takes into account the change in the Nusselt number \( \nu \) along the channel and along the flow regime.

\[
\nu = (1 - \gamma) \cdot \nu_{lam} + \gamma \cdot \nu_{nr}, \quad \text{(3)}
\]

where
\[
\nu_{lam} = \frac{(\zeta/8) \cdot (Re-1000) \cdot Pr}{1 + 12.7 \sqrt{(\zeta/8) \cdot (Pr^{2/3} - 1)}} \left[ 1 + (D/x)^{2/3} \right] \left( \frac{Pr_{wall}}{Pr} \right)^{0.11}, \quad \gamma = \frac{Re-2300}{4000-2300}, \quad 0 \leq \gamma \leq 1,
\]

\[\nu_{nr} \xrightarrow{\nu_g \geq 8.235} 8.235 \quad \text{on conditions UWF}, \]

\[\nu_{nr} \xrightarrow{\nu_g < 8.235} 7.541 \quad \text{on conditions UWT}, \]

condition UWF is a constant heat flux on the wall, UWT is a constant wall temperature. The computation of \( E_2 \) was carried out according to the Dittus-Boelter equation (4):

\[
\nu = 0.023 \cdot Re^{0.8} \cdot Pr^n, \quad \text{(4)}
\]

where \( n = 0.4 \) when heating the coolant flow, \( n = 0.3 \) when cooling the coolant flow. The computation of \( E_3 \) was carried out using the Shah-London dependence [6] with a constant value of the Nusselt number \( \nu = 8.235 \) (steady laminar fluid flow in the slotted channel under condition UWF).

The graph shows that the computations of \( E_1, E_2, E_3 \) give distinct values along the length of the channel for each temperature of gas, water and wall.
Comparing the results of computations for the flow of water and gas mixture in the heat exchanger for heat balance; differential pressure $\Delta P$; cooling of gas coolant $\Delta T = (T_{g1} - T_{g2})$ with experimental data, it is possible to determine which dependence describes adequately the hydrodynamics and heat transfer in the gap channels of the heat exchanger. We can use parameter $\theta = \Delta T_{e}/\Delta T_{c}$ as one of the criteria of adequacy. If parameter $\theta$ is equal to one, then there is a complete agreement between the calculations and the experimental data.

Figure 5 shows how parameter $\theta$ changes for different options for computing the cooling value of the gas coolant $\Delta T$ depending on the mass flow rate of gas $G_{g}$. The upper axis represents the values of Reynolds number $Re_{Dh}$ corresponding to the gas flow rate $G_{g}$. Options $A1$, $A2$, $A3$, $A4$ use formulas for laminar flow; $A1$, $A2$ are constant values of the Nusselt number $Nu_{g} = 8.235$ UWF for $A1$, $Nu_{g} = 7.541$ UWT for $A2$. For cases $A3$, $A4$, the values of Nusselt number were calculated using the equation (1) $Nu_{e} = 8.235$ for $A3$, $Nu_{e} = 7.541$ for $A4$. For case $A5$, the values of Nusselt number were calculated by equations (3). For case $A6$, the values of Nusselt number were calculated by the formula (4). It can be seen that depending on the magnitude of the gas flow the dependences for which the parameter $\theta$ is close to unity change. The Shah-London dependence (UWT) gives the best agreement between the results of calculations and experimental data under the laminar flow regime ($G < 50$ g/s); the Petukhov-Popov-Gnielinski equation gives the best agreement for the transition and turbulent flow regimes.

The pressure drop $\Delta P$ in the flow of dry air ($Pr \sim 0.7$) through the heat exchanger corresponds to the calculated values for the flow of gas in a smooth slotted channel [5, 10] with the value of the local drag coefficient at the inlet and outlet nozzle $\xi_{loc} = 0.34$. Figure 6 shows the experimental $\Delta P_{e}$ and calculated data $\Delta P_{c}$ for a complete pressure drop during the flow of a gas mixture with Prandtl number $Pr = 0.23$ depending on the mass flow rate $G_{g}$. A good agreement between the calculated and
experimental data is seen. The graph also shows the values \( \Delta P_p = \Delta P_p - \frac{2}{3} \cdot \xi \cdot \rho \cdot U_e / 2 \) of the pressure drop in the plate straight channel. For flow rates less than 45 g/s (Re_Dh < 1800), the gas flow regime is laminar and, accordingly, \( \Delta P_p \) depends linearly on the flow rate \( G_e \). For flow rates greater than 40 g/s, \( \Delta P_p \approx G_e^{2 - 1/4} \), which corresponds to the transitional and turbulent flow regime. That is, for a gas mixture with Prandtl number \( Pr = 0.23 \), the pressure drop in the slotted channel is calculated in the same way as for air (Pr = 0.7).

![Figure 6. Pressure drop for the mixture gas with Pr = 0.23 flow depending on mass flow rate.](image)

**Conclusion**

Study has shown that convective heat exchange design procedure in the plate heat exchanger by means of heat-balance equations along a stream, with the use of local values of coefficient of frictions and a heat transfer on walls make it possible with a sufficient accuracy verification of the equations for computing of Nusselt number and to spot a range applicability the used equations depending from a heat-transfer agent flow regime in heat exchanger channels. For experimental verification in a regime of heat exchange with heat-transfer agents "gas-gas", the procedure of definition of local coefficient resistance on the heat exchanger inlet and outlet is of great importance. For improvement of the method, it is necessary to use models, allowing an estimate of the influence of stream redistribution of heat-transfer agent among a channels on the flow pattern. To improve the accuracy of verification, reliable data on the full heat capacity of the gas flow depending on the static pressure and average temperature are required.

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