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The Identification of Local Parameters for Steam Condensation with the Presence of Air in Plate Heat Exchanger Based on Process Mathematical Model

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The condensation of vapour in the presence of noncondensing gas inside the channels of Plate Heat Exchanger (PHE) is studied on the example of steam condensation from its mixture with air. The mathematical one dimensional model of the process is developed. It is represented by the system of ordinary differential equations accounting for the change of process local parameters along the channel length. The identification of correlations for heat and mass transfer coefficients, as also pressure losses in condensing two-phase flow, is performed based on a comparison of modelling results with experimental data. The experimental model of PHE channel corrugated field consists of four corrugated plates forming three channels. The plate length is 1 m, its width is 0.225 m and a height of corrugations 0.005 m. The corrugations inclination angle to the main flow direction is 60 °. The correlations for two-phase flow are based on single-phase correlations obtained for a considered channel. For mass transfer coefficient, the most accurate results are obtained using approach accounting for transverse mass flux influence based on a stagnant film model with correction for density variation across the turbulent boundary layer. For the prediction of pressure drop in two-phase condensing flow the separate phases, model is used at low Reynolds numbers of the liquid film and dispersed annular flow model further on channel length, with growing mass flow rate of condensed liquid. The resulting mathematical model with correlations identified on experimental data can be used for the design of PHE for condensation of steam from its mixture with air.

1. Introduction

In many industrial processes, a considerable amount of heat is wasted with off and flue gases of different compositions, carrying not only sensible heat but also latent heat of condensable gas contained in the outgoing gaseous mixture. The condensation of vapours on enhanced surfaces is a subject of a number of researches, e.g. by Kukulka et al. (2019). The presence of non-condensable gases during condensation is much-influencing process intensity and very important in desalination plants, condensers, refrigeration systems, utilisation of heat from flue gases and the nuclear industry (Arsenyeva et al., 2016). The efficient utilisation of heat in these processes is possible in compact heat exchangers with intensified heat and mass transfer, among which plate heat exchanger (PHE) is one of the most successful types for many industrial applications. The construction and operation principles of PHE are well described in the literature (e.g. Klemeš et al., 2015).
The PHE consists of corrugated plates that are assembled in a pack are forming channels of intricate geometry that promote high levels of turbulence inflows of heat exchanging streams stipulating heat and mass transfer enhancement. The intensity of these processes during condensation in the presence of noncondensing gas is determined by the nature of species, their concentrations, surface geometry and character of process

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development along the channel. The correct design of PHE requires reliable correlations for calculation of heat and mass transfer coefficients and pressure drop in such channels. Correct design of PHE requires the information that allows calculating the local characteristics of process on small increments of channel length, as it is shown by Tovazhnyanskyy et al. (2018) with the use of empirical approach based on a theory of turbulent boundary layer with suction. In the present study, the most suitable correlations are identified based on comparison of results obtained with further developed mathematical model and tests data for condensation of the steam-air mixture in the sample of PHE channel obtained on a specially developed test rig.

2. Mathematical model

The mathematical model of steam condensation from its mixture with air in an experimental sample of PHE channel is described in a paper by Tovazhnyanskyy et al. (2018). It is consist of the system of ordinary differential equations with nonlinear right parts. This system does not permit an analytical solution and is solved numerically by the finite differences method implemented as software for PC.

Correlations for prediction of heat transfer coefficients and friction factor in single-phase flow inside PHE channels formed by plates with different geometries of corrugations are proposed in the paper of Arsenyeva et al. (2014). These correlations are used as a basis for prediction of heat and mass transfer in the gaseous phase of condensing stream, as also for a pressure drop of two-phase flow in PHE channels. The mathematical modelling of the process for conditions of experimental study is performed to check the validity of these correlations for two-phase condensing flow in the PHE channel with the use of different correction factors. The obtained results are compared with experimental data obtained for condensation in the PHE channel of steam from its mixture with air. The accuracy of prediction by mathematical modelling of process parameter *P* is estimated by calculation of root-mean-square error *(RMSE),* and relative root-mean-square error (*RRMSE*) determined as follows:

$$RSME_P = \sqrt{\frac{\sum_{i=1}^{n}\left(P_{i.\exp} - P_{i.clc}\right)^2}{n}}; \quad RRSME_P = \sqrt{\frac{\sum_{i=1}^{n}\left(\frac{P_{i.\exp} - P_{i.clc}}{P_{i.\exp}}\right)^2}{n}} \tag{1}$$

where n is the number of experimental runs; $P_{i.exp}$ is the experimental value of a parameter; $P_{i.clc}$ is the calculated value of a parameter.

3. Experimental part

The experimental study on the condensation of steam from its mixture with air performed at a model of PHE channel corrugated field at experimental set up described by Vasilenko et al. (2018). The model is made from four corrugated plates made by stamping from the sheet of AISI 304 stainless steel, thickness 1 mm. The plates are welded together at longer sides to form three channels with width 225 mm and length 1 m. The model was placed vertically along its length, and the steam-air mixture was supplied into the central channel from the top. The cooling water is going in two periphery channels in counter-current flow. The corrugations on the plates are made with angle β = 60 ° to a vertical axis. The height of corrugations is 5 mm and pitch 18 mm.

The temperatures of the air-steam mixture and cooling water are measured by copper-constantan thermocouples with accuracy ± 0.1 ºC. The measuring points are placed at the inlet and outlet of hot and cold streams and at six points along the channel length. The pressure of the condensing stream is measured by pressure gauges at the inlet and exit of the channel with an accuracy ± 0.005 bar. The mass flow rate of cooling water is determined with the use of orifice flow meter, which accuracy is ±1 %. The flow rate of air is measured by the set of flow meters, which minimal accuracy was ± 2 %. The volumetric flow rate of water condensate was measured by a set of measuring vessels with an accuracy of ± 1 %. The steam flow rate is determined by summing the water condensate flow rate with the flow rate of not condensed steam exiting channel with the outgoing steam-air mixture at saturation conditions. The experiments included 58 tests with different conditions of gaseous mixture condensation. The experiments were performed with air content in the mixture varied from 3 % to 80 % on volume and absolute pressure from 1.05 bar to 3.0 bar. The local velocity of the gaseous stream was in the range from 46 m/s to 4.1 m/s. The temperature of the gaseous stream changed in the range from 88.2 °C to 115.1 ºC, and the temperature of cooling media varied from 23.8 °C to 71.5 ºC.

4. Results and discussion

4.1 Heat and mass transfer

For every of 58 test runs, the calculations were performed with experimentally measured flow rates of cooling water, incoming steam and air. For the given experimental values of all parameters at the steam-air inlet, the values of temperatures and pressure at the outlet and their distribution along the channel are calculated. Estimation of accuracy for experimental data prediction is made by the root-mean-square error (RMSE) of calculated results compare to the experimental value of the corresponding parameter. By the first approach, the heat- and mass transfer coefficients are calculated by the analogy of heat and mass transfer with the use of correlations for heat transfer in single-phase flow inside the investigated channel. It allows calculating thermal Nusselt number Nu_0 and diffusional Nusselt number Nu_{D0} without transverse mass flux influence.
The results of modelling without accounting for transverse mass flux gave discrepancies in calculations of steam-air mixture temperature at channel outlet up to 12 ºC with its $RMSE_t$ = 4.8 ºC. It is not acceptable for the design of PHE for condensing duties and requires the introduction of correction for the influence of transverse mass flux.
For estimation of transverse mass flux influence, two approaches were examined. The first one is proposed by Tovazhnyansky et al. (2004) for PHE channel using a semi-empirical approach of Kutateladze and Leont'ev (1985) for turbulent boundary layer with suction. The coefficients of heat transfer h_{cv} and mass transfer β_D are calculated according to heat and mass transfer analogy with correction for the influence of transverse mass flux as follows:

$$h_{cv} = \frac{\lambda}{d_e} \cdot \Psi_H \cdot Nu_0 \tag{2}$$

$$\beta_D = \frac{D_D}{d_e} \cdot \Psi_D \cdot Nu_{D0} \tag{3}$$

Here λ is the thermal conductivity of gaseous mixture, W/(m K); D_D is the diffusion coefficient, m^2/s; d_e is equivalent diameter of the channel.
The relative factors of heat and mass transfer for PHE channel:

$$\Psi_{H(D)} = 4 \cdot \left(1 + 0.85 \cdot b_{H(D)}\right) \cdot \left(1 + \sqrt{\frac{\rho}{\rho_f}}\right)^{-2} \tag{4}$$

Here b_H and b_D are heat and diffusivity parameters:

$$b_H = \frac{c_{Pv}}{c_P} \cdot \frac{j_v \cdot \mathrm{Re} \cdot \mathrm{Pr}}{\rho \cdot W \cdot Nu_0} \quad b_D = \frac{j_v \cdot \mathrm{Re} \cdot \mathrm{Pr}_D}{\rho \cdot W \cdot Nu_{D0}} \tag{5}$$

where W is the velocity of the air-steam mixture in the channel, m/s; ρ and ρ_f are the density of gaseous mixture in flow bulk and at gas-liquid interphase, kg/m^3; c_P and c_{Pv} are heat capacities of gaseous mixture and vapour, J/(kg·K); j_v is the transverse flux of condensing vapour, kg/(m^{2}s).
The results of modelling with this first approach are compared with experimental data. The comparison gave an error for outlet temperature $RMSE_{tKL}$ = 1.72 ºC and for the average temperature of the gas-steam mixture, $RMSE_{atKL}$ = 2.39 ºC.
For the prediction of transpiration effects on mass transfer during air-vapour condensation on a flat plate in a paper by Bucci et al. (2008) was examined theoretical approach based o stagnant film model, which in more details discussed in the book of Lienhards (2018) with the introduction of mass transfer driving force for condensing vapour:

$$B_m = \frac{m_v - m_{vf}}{m_{vf} - 1} \tag{6}$$

where m_v and m_{vf} are mass fractions of vapour in flow bulk and on condensate surface.
By this second approach, the relative factor of mass transfer is:

$$\Psi_{D-2} = \frac{\ln\left(1 + B_m\right)}{B_m} \tag{7}$$

The comparison with experimental data of calculations with the model using Eq(7) for mass transfer is shown in Figure 1 for outlet temperature. The error is $RMSE_{t2}$ = 1.95 ºC. For calculation of the average temperature for gas-steam mixture $RMSE_{at2}$ = 2.56 ºC. The accuracy somewhat lower than with a first approach but can be improved by the introduction in Eq(7) the term from Eq(4) accounting for the change of gaseous mixture density across the flow from fluid bulk to condensate surface as follows:

$$\Psi_{D-3} = \frac{\ln\left(1+B_m\right)}{B_m}\cdot 4\cdot\left(1+\sqrt{\frac{\rho}{\rho_f}}\right)^{-2} \tag{8}$$

The modeling with Eq.(8) gave $RMSE_{t3}$ = 1.68 ºC and $RMSE_{at3}$ = 2.19 ºC. This corrected equation is giving the smallest error and is not containing any empirical coefficients that allow assuming it as valid for different channels geometries. The relative error of heat load calculation based on a comparison of its measurement with cooling water heat balance is $RRMSE_Q$ = 0.0215. In the majority of considered experimental runs, the total resistance to heat transfer was controlled by heat and mass transfer at the side of the steam-air mixture. On this estimation of integrally determined heat load it can be concluded that the error in the calculation of local coefficients of heat transfer h_{cv} and mass transfer β_D with probability 95 % is not exceeding ±5 %, that is acceptable for practical engineering applications.

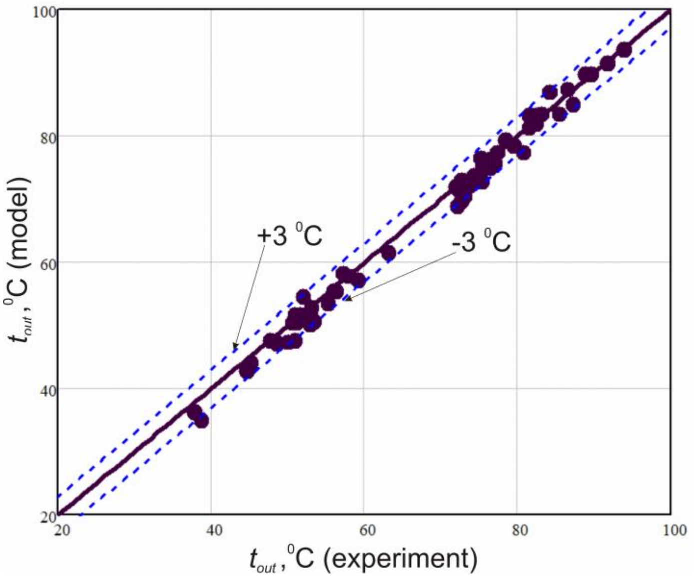

Figure 1: Prediction of air-steam outlet temperature vs experimental data

4.2 Pressure drop

The pressure drop in the condensing two-phase flow was calculated based on a correlation for the friction factor for single-phase fluid using two approaches. The first one is based on parameters proposed by Lockhart and Martinelli (LM) with the flow model of separated phases in the following form:

$$X_{LM} = \sqrt{\frac{dP_L}{dP_G}};\quad \Phi_G = \sqrt{\frac{dP_{TP}}{dP_G}};\quad \Phi_L = \sqrt{\frac{dP_{TP}}{dP_L}}; \tag{9}$$

Here dP_{TP} is a pressure drop in a two-phase flow; dP_L and dP_G are pressure drops of liquid and gaseous phases flowing alone in the channel, Pa, calculated by correlation for single-phase flow (Arsenyeva et al., 2014).
The attempt of modelling all data on pressure drop based on parameter Φ_G with the optimised value of the coefficient in LM correlation gave the following equation with relative error $RRSME_{LMG}$ = 0.211:

$$\Phi_G^2 = 1 + 57.6\cdot X_{LM} + X_{LM}^2 \tag{10}$$

The second considered approach (BK) was proposed by Boyko and Kruzhilin (1967) for dispersed annular flow model:

$$dP_{TP} = dP_{0L} \cdot \left[1 + x_G \cdot \left(\frac{\rho_L}{\rho} - 1\right)\right] \tag{11}$$

Here dP_{0L} is the pressure drop calculated for a total flow rate of two-phase flow and properties of the liquid, Pa; x_G is the mass fraction of gaseous phase; ρ_L is the density of the liquid.
The modelling with this Eq(11), which have no empirical parameters, gave $RRSME_{BK}$ = 0.193. Both KL and LM models are giving similar results on accuracy being applied for all channel length. However the use of LM model for condensate film Reynolds numbers $Re_{film} \le 125$ and BK model for all other channel gave better accuracy $RRSME_{BKLM}$ =0.144. It can be explained by changing of flow structure. At the channel, entrance gas is going in the centre of the channel and condensate starting to form a liquid film on the wall. This corresponds to a model of the separate flow of phases and pressure drop can be calculated as follows:

$$for\, Re_{film} \le 125 \quad \Phi_G^2 = 1 + 355 \cdot X_{LM} + X_{LM}^2 \tag{12}$$

When more condensate is formed, part of the condensed liquid becomes entrained into gas flow core and dispersed annular flow is starting and at $Re_{film} > 125$ pressure drop is corresponding to Eq(11).
It is proposed to use instead of Eq(11) the combination of two approaches to improve the accuracy of correlating the experimental data in the following form:

$$dP_{TP} = dP_L \cdot \sqrt{1 + x_G \cdot \left(\frac{\rho_L}{\rho} - 1\right)} \cdot \left(1 + \frac{3.02}{X_{LM}} + \frac{0.02}{X_{LM}^2}\right) \tag{13}$$

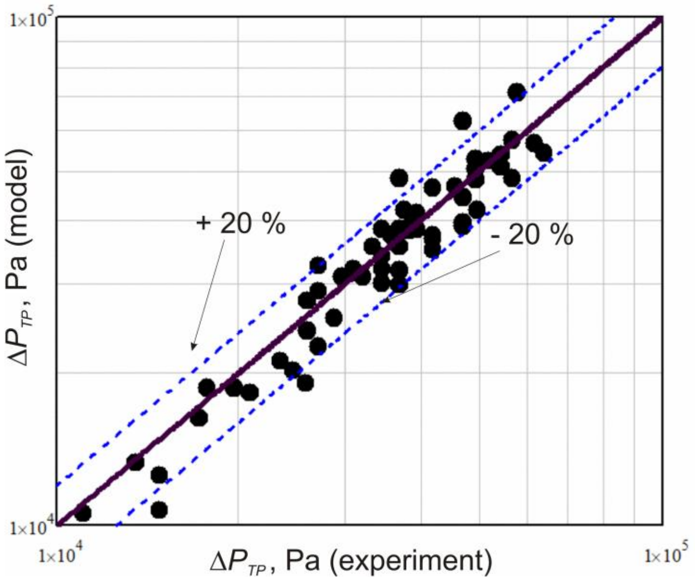

Figure 2: Prediction of pressure drop in two-phase flow with Eq(12) and Eq(13) vs experimental data

The use of Eq(12) for condensate film Reynolds numbers $Re_{film} \le 125$ and Eq(13) for all other channel gave better accuracy $RRSME_{LMBKLM}$ = 0.127, that can be estimated graphically in Figure 2. These two Eq(12) and Eq(13) can be used for calculation of local parameters during condensation of the air-steam mixture in PHE channels of investigated geometry. However, there is a difference in application range with the mass transfer correlation (8), which is having the same form for pipes and channels of different geometries. For calculation of pressure drop during condensation were proposed quite a number of various correlations for different channels and condensing substances which analysis is presented in a paper by Kim and Mudawar (2014). For application correlations of proposed by Eq(11) and Eq(12) form in a wider range of channel geometries and process parameters more experimental data are needed with identification of correlations according to the method proposed in this paper.

5. Conclusions

The method of mathematical modelling with empirical identification of correlations for local process parameters is successfully applied to the process of steam condensation in the presence of air inside the PHE channel of specific geometry. It is demonstrated that the heat and mass transfer coefficients in condensing vapour-gas mixture can be calculated based on single-phase heat transfer correlations for the considered channel with a proposed form of correction factor accounting for transverse mass flux, not depending on the geometry of channel. For condensing two-phase flow of steam-air gaseous mixture and condensed water, two possible flow structures are identified: separate phases flow structure close to the entrance of the channel and dispersed annular flow further along the channel length. The corresponding to these flow models pressure drop correlations are proposed. For the application of pressure drop correlations in a wider range of channel geometries and process parameters, it is required further experimental research of gas-vapour condensation in PHE channels of different geometries with identification of correlations according to presented in this paper methodology.

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