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The Influence of Plate Corrugations Geometry on Performance of PHE as Condenser of Steam from its Mixture with Air

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The condensation of steam from its mixture with air in channels of plate heat exchanger (PHE) is studied experimentally and with calculations based on a mathematical model developed earlier and based on correlations for local process parameters. The experiments are done for five experimental samples modelling the corrugated field of PHE channel with corrugation angle to the main flow direction 30 degrees, 45 degrees, 60 degrees and channel spacing 5 mm, 7.5 mm and 10 mm. The mathematical model validity is confirmed by comparison of calculated temperature program for cooling of air-stream mixture with experimental data for all five samples. The influence of plates corrugations geometry on overall PHE performance in air-stream condensation and local process parameters distribution is discussed. The analysis reveals that to satisfy the required temperature program, PHE with smaller corrugations angle should have plates of bigger length, as well as PHE with bigger channel spacing. Both these parameters can be used as independent variables in the optimisation of plates corrugations geometry for specified process conditions using examined mathematical model.

1. Introduction

The intensification of heat recuperation is an important factor in increasing the efficiency of energy use at industrial enterprises, enabling the reduction of fossil fuel consumption for energy generation and corresponding carbon dioxide and other harmful emissions to the environment (Klemeš et al., 2018). In a number of industrial applications, the recuperation of heat from condensing gaseous streams is involved. The use of efficient heat transfer equipment can increase this process intensity and amount of recuperated heat. The compact heat exchangers can be the best choice for this purpose, among which Plate Heat Exchanger (PHE) is one of the most promising types (Klemeš et al., 2015). It has been confirmed that in a number of thermal processes, PHEs have advantages compare to traditional shell-and-tube heat exchangers in obtaining efficient and economically viable solutions, as, e.g. in liquid-liquid applications (Hajabdollahi et al., 2016), in the chemical industry (Kapustenko et al., 2009), carbon dioxide capture (Perevertaylenko et al., 2015) and other cases. PHE of special construction with increased channel spacing also can be used for condensation processes of pure steam (Arsenyeva et al., 2011), multicomponent vapours mixtures (Tovazhnyansky et al., 2004) and air-stream mixtures (Kapustenko et al., 2021). The process of vapour condensation in the presence of noncondensable gases (NCG) takes place in many industrial applications, such as waste heat recovery from the flue gas after fuel combustion (Famileh and Esfahani, 2017), refrigeration plants (Charef et al., 2018), desalination systems (Caruso and Di Maio, 2014), petrochemical industry (Gu et al., 2015), nuclear powers (Bian et al., 2017) and others. The survey of this process studies in tubular heat exchangers (Huang et al., 2015) reveals its complexity compared to single-phase heat transfer and the necessity to consider heat and mass transfer in gas and liquid

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phases with the change of process parameters along heat transfer surface. The process in the channels of PHE is even more complicated by the complexity of the geometrical channel form (Kapustenko et al., 2021). However, the strong influence of plate corrugations geometrical parameters on transport and hydraulic characteristics of the channel enables the optimisation of PHE geometrical parameters for specific process conditions. It can be used for further improvement of PHE performance in the processes of vapour condensation from its mixture with NCG. This paper has presented the results of an experimental study supplemented by mathematical modelling of local process parameters for condensation of steam from the steam-air mixture in PHE channels formed by plates with different geometrical forms of corrugations. The influence of the main geometrical parameters on PHE performance in different process conditions is discussed.

2. Experimental part

The experimental study of steam condensation from its mixture with air was performed at the test rig described in the paper by Arsenyeva et al. (2020). Five experimental samples of PHE channels formed by plates with different corrugations geometry were tested. Each sample consists of four rectangular corrugated plates stamped from stainless steel AISI 304. The corrugations are situated at straight lines inclined with the angle β to the plate longer side. The corrugations are of triangular cross-section and are stamped with a pitch S on a plate surface, as shown in Figure1. On assembly, two adjacent plates are overturned at 180° in such a way that their corrugations are crossing each other and have multiple contact points at the edges. Four plates are welded on longer sides to form three similar channels with width 225 mm and length L 1 m. The sample is placed vertically, and in the central channel from the top is supplied saturated air-steam mixture through the welded connection; in two other channels from the bottom is directed cooling water. The geometrical parameters of the channels and corrugations on plates for all five samples are presented in Table 1.

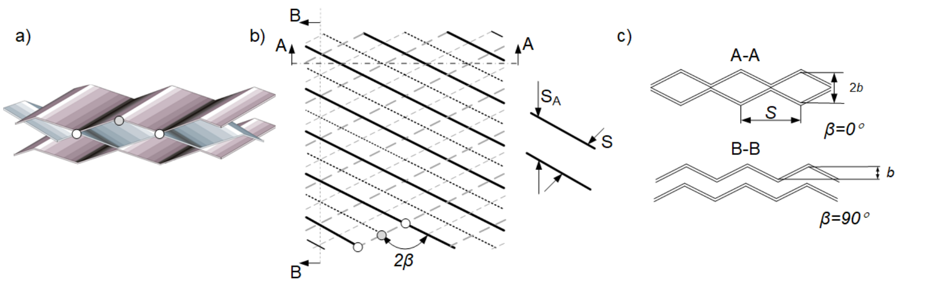

Figure 1: Schematic drawing of tested channels geometries: a) 3D drawing of plates corrugations intersection; b) 2D drawing of plates corrugations intersections; c) channel cross-sections at β = 0° and β = 90°

Table 1: The parameters of tested samples

Designations of PHE samples	S1	S2	S3	S4	S5
Corrugations pitch S, mm	18.0	27.0	36.0	18.0	18.0
Depth of corrugations b, mm	5.0	7.5	10.0	5.0	5.0
Corrugations angle β, degrees	60	60	60	45	30
Corrugated field length L, m	1.0	1.0	1.0	1.0	1.0
Corrugated field width, m	0.225	0.225	0.225	0.225	0.225
Cross section area of channels f_{ch}, cm^2	11.25	16.90	22.50	11.25	11.25
Sample heat transfer area $2*F_{pl}$, m^2	0.518	0.518	0.518	0.518	0.518

The temperatures of the air-steam mixture and cooling water were measured at the inlet and exit of heat-exchanging streams and at six points along the channel. The pressure of the air-steam mixture was measured at the channel inlet and exit. The process parameters during tests varied in the following ranges: the volume concentration of air in a mixture from 3.0 % to 80.0 %; the absolute pressure of gaseous stream from 103 kPa to 310 kPa; the temperatures of the gaseous stream from 88.0 ºC to 115.1 ºC; the temperatures of cooling water from 23.6 ºC to 71.7 ºC.

3. The mathematical modelling

To study the influence of plate corrugations geometry on PHE performance in condensation of steam from the steam-air mixture, the one-dimensional model reported in a paper by Kapustenko et al. (2021) is used. To confirm the model validity for all experimental samples with different corrugation angle β and corrugation pitch S, the calculations of exhaust mixture temperature are compared with its value obtained in experiments, as it is presented on a graph in Figure 2. The root-mean-square error for prediction exhaust temperature of steam-air mixture for all three samples is 1.71 °C with maximal deviation not exceeding ±3 °C for 95 % of experiments. The sufficient for engineering calculations accuracy of pressure drop prediction was shown in a paper by Kapustenko et al. (2021). It allows to use of the mathematical model for calculation of local process parameters in all samples of PHE channel and to predict PHE performance with different channel length L and different geometrical parameters of corrugations. It is illustrated in the next subsection.

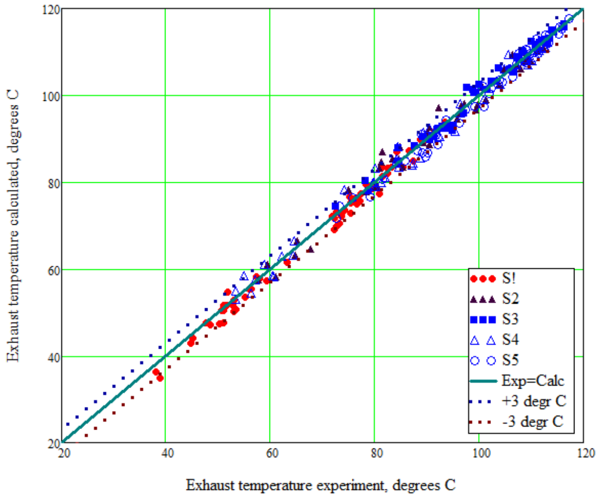

Figure 2: Comparison of calculated by model exhaust temperature with experiment

4. Results and discussion

The influence of the corrugations inclination angle β on process intensity can be illustrated by a comparison of air-steam mixture condensation in experimental samples S1, S4 and S5, which are having the same cross-section area, and the same form of corrugation, but different β. In Figure 3 are presented the distributions of the air-steam mixture and cooling water temperatures along the channels at similar mixture flowrates and gas volumetric concentrations ε_g about 5 %. There are also presented the calculated distributions of pressure drop from the channel entrance. The intensity of condensation in a channel of S1 with β = 60° is highest, and the mixture is cooled to the lowest temperature, 75 °C. In the channel of S5 with angle β = 30°, the mixture is cooled down only to 110 °C and in S4 with β = 45° to intermediate 85 °C. However, this intensification is at the expense of increased pressure drop from 6 kPa in S5 to 21 kPa in S1, when in S4, it has a value of 10 kPa.

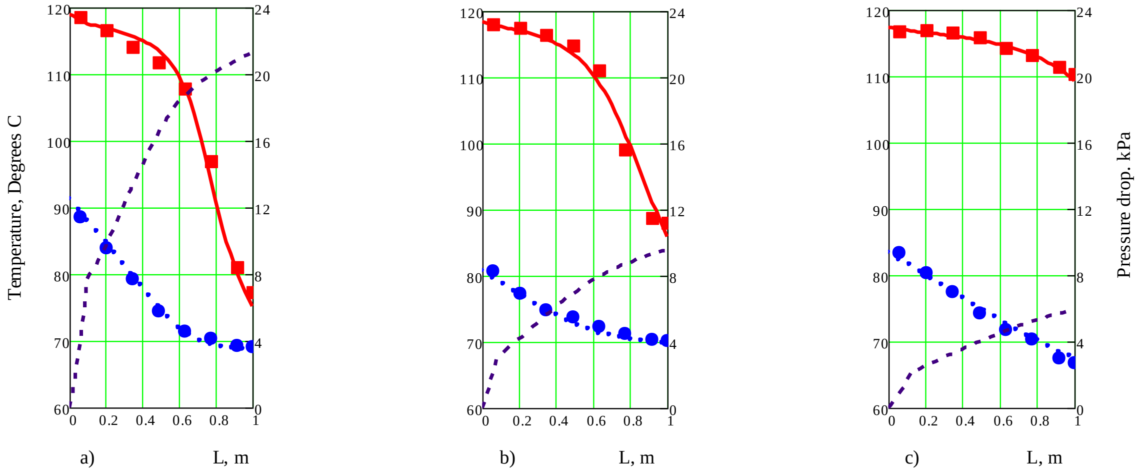

Figure 3: The distribution of mixture temperature (squares – experiment, line – calculated), cooling water temperature (dots – experiment, dotted line – calculated), and pressure drop (dashed line – calculated) along the channel length at S = 18 mm, mixture flowrate ~ 0.031 kg/s, volume gas concentration ε_g ~ 0.05: a) – sample S1 – β = 60°; b) – sample S4 - β = 45°; c) – sample S5 - β = 30°.

In real-life applications, it is required that the heat exchanger has to satisfy specified conditions on temperature program and pressure drop. Figure 3 follows that the cooling mixture to 110 °C in sample S4 require much less channel length than in the channel of sample S5. When pressure drop is not limited, the channel S1 with the highest β is the best solution. It is also better for cooling down the mixture to 85 °C, as in sample S4 requiring less channel length with not limited pressure drop. However, if the pressure drop is limited to 10 kPa (equal to full pressure drop in S4), the length of channel S1 has to be limited to about 0.2 m, but the required temperature is not be achieved, so the flowrate must be reduced in some series of calculations with a mathematical model. The temperature of 85 °C can also be achieved with the channel of β = 30° (Figure 3c), but by increasing channel length, that also requires calculation with the mathematical model. The data on graphs in Figure 3 are selected just for illustration of mathematical model validity and general comparisons. There are some differences in total pressure and cooling water flowrates, as data were selected from already made sets of experiments. More accurate comparisons can be made with mathematical modelling of the process.

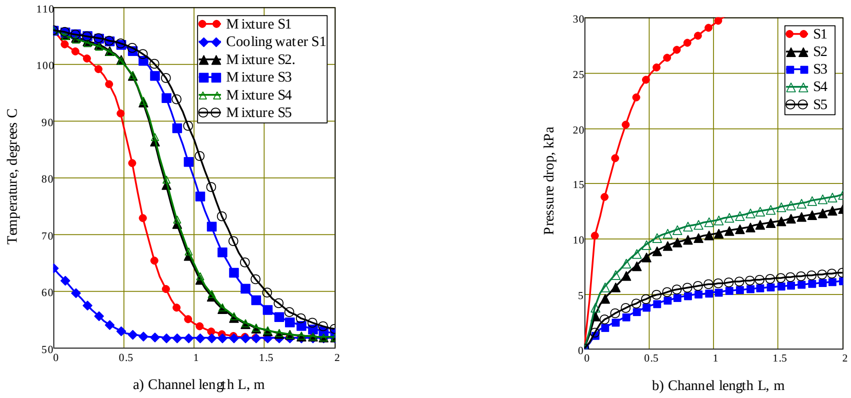

Figure 4: The calculated distribution along the length of the channel for mixture flowrate 0.03 kg/s and volumetric air content ε_g = 3 %; a) mixture and water temperature; b) pressure drop from the channel entrance.

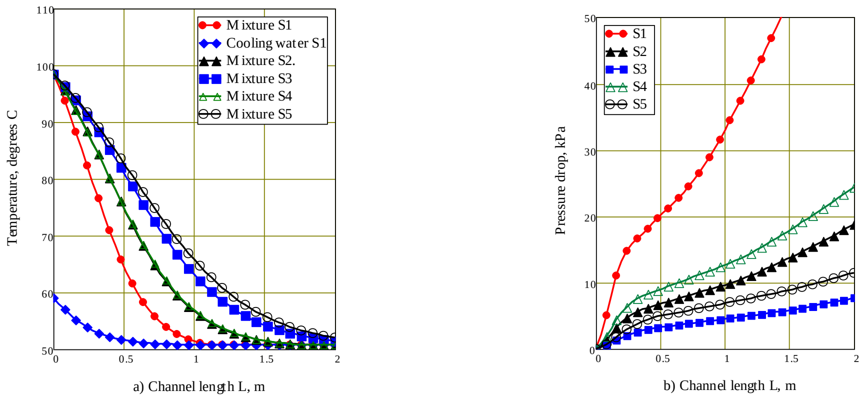

Figure 5: The calculated distribution along the length of the channels for mixture flow rate 0.025 kg/s and volumetric air content ε_g = 46 %; a) air-steam mixture and water temperature; b) pressure drop from the channel entrance.

The calculated temperature and pressure drop profiles during air-steam mixture condensation in all five samples, but with length L extended to 2 m, are shown in Figure 4. The parameters of incoming mixture with a small air content of 3 % on volume are taken the same for all samples, as also water temperature and flowrate on its channels entrance. The forms of temperature and pressure drop curves are similar for all channels geometries. But the intensity of the processes is different. The highest rate of temperature drop along the channel length, which corresponds to the highest intensity of heat and mass transfer, is observed in a channel of sample S1 with β = 60° and channel spacing b = 5 mm. However, the pressure drop in this channel is also highest. The decrease of β and increase of channel spacing b are leading to smaller pressure drop but lower heat and mass transfer intensities. The temperature curves for the channel with β = 60° and b = 7.5 mm (S2) and for the channel with β = 45° and b = 5 mm (S4) are very close (see Figure 4a), while the pressure drop in S2 is lower on about 10 % (see Figure 4b). The temperature change in the channel with β = 60° and b = 10 mm

(S3) is somewhat bigger than in channel with β = 30° and b = 5 mm (S5) (see Figure 4a), but the pressure drop in S3 is even 5 % lower than in S5. It can be concluded that in these conditions (small volumetric air content 3 %), the increase of channel spacing is leading to a somewhat smaller loss in heat and mass transfer intensity at the same pressure drop compared to a decrease of corrugations angle β. The trend becomes even more emphasised with an increase of gas volumetric concentration to ε_g = 46 %, as it is shown in Figure 5.

The analysis of graphs in Figure 4 and Figure 5 reveals that to satisfy the required temperature program, PHE with smaller corrugations angle β should have plates of bigger length, as well as PHE with bigger channel spacing b. For example, to cool down the air-steam mixture to 60 °C in conditions of Figure 4a, the length of the plate with β = 60° and b = 5 mm (S1) can be 0.8 m. With β = 45° and the same b = 5 mm (S4) it is 1.1 m and β = 30° and b = 5 mm (S5) it is 1.5 m. With geometry of sample S5 the heat transfer area is 1.5 * 0.225 = 0.3375 m^2. At these plates lengths, the pressure drops according to Figure 4b are approximately 28 kPa, 12 kPa and 6 kPa, respectively. The similar picture is with increase of channel spacing b for β = 60° and b = 7.5 mm (S2) and β = 60°, b = 10 mm (S3). It can be concluded that for limited pressure drop 6 kPa and the same numbers of plates in PHE (same flow rate in one channel) the use of plates with β = 30° and b = 5 mm (S5) or increased spacing β = 60° and b = 10 mm (S3) is required but increased length of plates would lead to proportionally bigger heat transfer area of PHE.

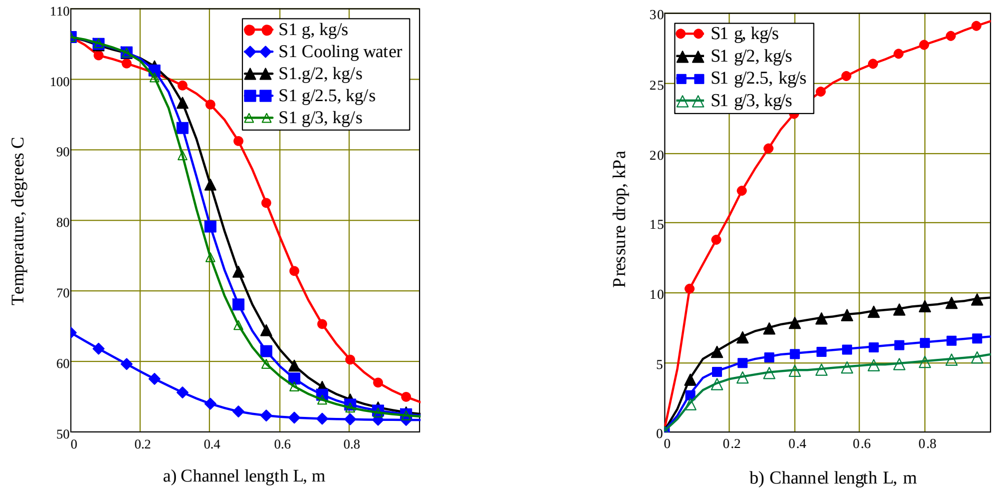

Figure 6: The calculated distribution along the length of the channels for mixture flowrate 0.03 kg/s and volumetric air content ε_g = 3 % in sample S1: a) mixture and water temperature; b) pressure drop from the channel entrance.

There is another way to satisfy the required pressure drop by decreasing the mixture flow rate in one channel. In PHE, it can be by increasing the plate width or adding the plates and a corresponding number of channels for the same total flow rate in PHE. Figure 6 are presented the results of calculation by mathematical model for the channel with β = 60° and b = 5 mm (S1) with flow rate shorter in 2, 2.5 and 3 times. The decrease of flow rate 2.5 times (Figure 6b) leads to a decrease of pressure drop to 6 kPa at the channel length 0.58 m. At this channel length, the temperature drops to 60 °C.

To achieve this flow rate, the width of the channel must be 2.5 times bigger than in experimental samples and the heat transfer area 0.58 * 0.225 * 2.5 = 0.3263 m^2. It is only about 3.3 % smaller than channel S5, with a length of 1.5 m, which is having these values of temperature and pressure drops. It shows that it was possible to strictly satisfy required process conditions on temperature and pressure drop with any β and b inside investigated range by adjusting the length of the PHE plate. With the heat transfer area of PHE as an objective function, the optimal values of β and b and corresponding plate length can be found.

The considered examples are concerned with the models of the main heat transfer area of PHE channels. In industrial PHE applications, the channels entrance and exit zones must be accounted for, as well as possible constraints on PHE sizes and the number of plates in one heat exchanger.

5. Conclusions

The geometrical form of corrugations is significantly affecting the thermal and hydraulic performance of PHE in the process of steam condensation from the air-steam mixture. The decrease of the corrugations inclination angle is reducing the intensity of heat and mass transfer processes, as also the pressure drop in the channel.

This effect is similar to an increase of channel spacing at the same corrugations angle. In both these cases, for cooling the air-steam mixture to the same temperature, a bigger length of plates is required. The character of local process parameters distribution is also influenced by the air content in the incoming air steam mixture, while the trend of corrugation angle influence remains the same. By using the developed mathematical model of air-steam condensation in PHE channels of different geometries, it is possible to calculate the geometrical parameters that allow satisfying the required thermal program and pressure drop in PHE. It enabled to use of the mathematical model for optimisation of PHE geometrical parameters for specified conditions of air-steam mixture condensation. The development of algorithm and software for such optimisation of PHE in industrial applications is a problem for further research.

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