Experimental study of pressure drops in multichannel microchannel heat exchanger at adiabatic two-phase flow of R134a with condensation

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Abstract. The pressure drop in the heat exchanger with parallel microchannels with throttling of the flow at the inlet and outlet of the heat exchanger was investigated. The experiments were carried out during condensation of the two-phase flow of R134a and under adiabatic conditions in the mass flow range from 200 to 900 kg/m2s. The pressure gradient in microchannels was determined from a change in the saturation temperature under adiabatic conditions. The obtained data were compared with the known relations for calculating the pressure drop taking into account the sudden expansion and constriction of the flow.

1. Introduction
Currently, multi-port pipes with small channels have been widely used as condensers in various cooling and air conditioning systems. The microchannel technology allows reducing the size and increasing the efficiency of heat exchangers. One of the important parameters for the design of the heat exchanger is the pressure drop. Despite many correlations that predict the pressure drop of two-phase flows, the existing formulas are well applicable to the specific conditions under which they were developed and require verification under various conditions [1,2]. It is worth noting that in a larger number of works, only friction losses of the two-phase flow inside the microchannels are studied, and the correlations used to calculate pressure losses associated with a sudden constriction and expansion of the flow are not verified. When designing two-phase heat exchangers, it is important to calculate not only the pressure loss inside the microchannels of the heat exchanger, but also the pressure loss, taking into account rotations, sudden constriction and expansion of the flow, and friction pressure loss in the input-output chambers. For effective operation of the microchannel heat exchanger, a uniform distribution between the microchannels of the two-phase flow supplied to the input is necessary. Flow throttling is one way to evenly distribute a gas-liquid stream. In this work, we conduct an experimental study of the pressure drop in a microchannel heat exchanger with throttling of a two-phase flow at the inlet to the microchannels and compare the data obtained with various models.

2. Experimental equipment and treatment
The experiments on the pressure drop in a multi-port microchannel heat exchanger were carried out on an experimental stand in the form of a closed loop. The working fluid was fed to the vapor generator, in which the flow reached the required vapor quality, through the flow controller. After the vapor generator, a two-phase flow was directed to the experimental section, and then to the condenser, in which complete vapor condensation occurred.
As an experimental section, a multi-port microchannel heat exchanger (Fig. 1) containing 21 microchannels with a width of 300 μm, a depth of 930 μm, and a channel spacing of 685 μm was used. The length of the microchannels was 40 mm. At the inlet and outlet of the microchannels there were slotted chambers with a gap of 300 μm, a width of 20 mm, and a length of 10 mm. There was transition section between the slotted chamber and microchannels with the length of 2.5 mm (Fig. 1). At a distance of 3.5 mm from the outer edge of the slotted chambers there were pressure drops. The heat exchanger was located horizontally. A copper block, cooled by Peltier elements, was mounted below the microchannels. Along the microchannels, at a distance of 5, 15, 25, 35 mm, the wall temperature was measured. The flow temperature at the inlet and at the outlet of the heat exchanger was measured as well. Inlet pressure was measured by SMH5051 detector with 1500 Pa accuracy and differential pressure was measured by SMH5440 detector with 150 Pa accuracy. The experimental setup and the data processing to heat flux calculation were described in more detail in [3]. Refrigerant R134a was used as the working fluid.

Before starting the experiments with the two-phase flow, tests with single-phase fluid flow were carried out. A comparison of the measured pressure drop with the calculation was made. The total pressure drop $\Delta P_{\text{Total}}$ was calculated as

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\Delta P_{\text{Total}} = \Delta P_{\text{Ch1}} + \Delta P_{\text{Friction}} + \Delta P_{\text{Ch2}}
$$

Figure 1. Scheme of test section.

where $\Delta P_{\text{Ch1}}$, $\Delta P_{\text{Friction}}$, $\Delta P_{\text{Ch2}}$ is the pressure drop due to friction in the inlet chamber, inside the microchannels and in the outlet chamber, $\Delta P_{\text{Ext1}}$, $\Delta P_{\text{Ext2}}$ is the pressure drop associated with the expansion of the flow in the transition chamber at the outlet of the inlet chamber and at the outlet of the microchannels, $\Delta P_{\text{Constr1}}$, $\Delta P_{\text{Constr2}}$ are associated with the flow constriction at the entrance to the microchannels and upon transition to the exit slotted chamber. The experiments were carried out in the range of Reynolds numbers up to 1400 inside microchannels and 2100 inside slotted chambers, and in the calculations the flow was assumed to be laminar. The friction pressure loss was calculated as for a developed laminar flow in a rectangular channel according to [4], and the influence of the inlet section was taken into account according to [5]. The pressure drop associated with the constriction and expansion of the flow was calculated according to [6]. Figure 2a shows the calculated pressure drop along the length of the heat exchanger between the pressure taps at a mass flow rate of liquid equal to 550 kg/m$^2$s. Comparison of the measured differential with the calculation by eq. (1) is shown in Fig. 2b; the experimental data are in good agreement with the calculation.
Figure 2. The pressure drop along the length of the heat exchanger at a liquid mass flow rate in the microchannels of 550 kg/m$^2$s (a). Dependence of the total pressure drop on the mass flow rate, points — experimental data, line — calculation (b).

3. Results

After the single-phase testing, experiments with an adiabatic two-phase flow were carried out. Under adiabatic equilibrium two-phase flow, the wall temperature equals the temperature of saturated flow. Thus, the pressure gradient, along the length of the microchannels can be determined by the change in the saturation temperature along the microchannels. In the experiments, the temperatures were considered steady when the wall temperature drift did not exceed 0.05°C for 15 minutes. Figure 3 shows an example of determining the pressure gradient from the data of measuring wall temperatures at a mass flow rate of 339 kg/m$^2$s with a vapor quality of 0.544.

Figure 3. Changes in the saturation pressure of a two-phase flow along the length of microchannels under adiabatic conditions.
Figure 4. Comparison of experimental data with calculation by models [7] (a) [8] (b)

Experiments under adiabatic conditions are carried out in the range of mass flow rates from 200 to 600 kg / m² s and vapor qualities from 0.1 to 0.8. The experimental data are compared with calculations for separate flow models from the works of Chisholm [7], Fig. 4a, and Kim Mudawar [8], Fig. 4b. The Chisholm model [7] shows the best fit to the experimental data.

A series of experiments to measure the total pressure drop under conditions of flow condensation was also performed. The experiments were carried out in the range of mass flow rates from 400 to 900 kg / m² s and vapor qualities from 0.05 to 0.85 at a system pressure in the range from 5.5 to 10 bar. The heat flux density on the inner walls of the microchannels did not exceed 14 kW / m² and the decrease in vapor quality on the microchannel heat exchanger did not exceed 5%. In the conditions of condensing flow, in order to correctly calculate the pressure drop, it was necessary to additionally take into account the deceleration of the flow. The pressure drop associated with deceleration was calculated using the methods presented in [8]. Under the conditions of the experiments, the contribution of deceleration to the total pressure drop was small. Therefore, it was impossible to correctly choose a model for the void fraction when calculating the pressure drop associated with deceleration according to the data of the experiments. Below in the given data, when calculating the contribution of deceleration, the Zivi model [9] was used to calculate the void fraction [9].

To calculate the pressure drop associated with an abrupt flow area changes, it is recommended to use the relations from [10]. It is worth noting that when calculating the total pressure drop across the heat exchanger, it is necessary to take into account the change in gas density along the length of the heat exchanger. Comparison of the total measured pressure drop during condensation and under adiabatic conditions with that calculated at friction computed on the model from [7], and pressure loss at flow expansion and constriction computed according to [10] is shown in Fig. 5 by triangular symbols. The calculation is in good agreement with the data at small pressure drops. With increasing dynamic pressure, the calculated values tend to underestimate the measured pressure drop. In [11], relations were obtained for the pressure drop caused by abrupt flow area changes in capillary tubes. When using the relations from [11] with the calculation of friction according to [7], the calculation corresponds to the experimental values of the pressure drop over the entire range of data obtained, see round symbols in Fig. 5.
Figure 5. Comparison of experimental data with calculation. Friction pressures drop by [7]. Pressure drop caused by abrupt changes in flow area: triangle symbol [10], round symbol [11].

Conclusions
An experimental study of the pressure drop in a heat exchanger with parallel microchannels at two phase flow of R134a refrigerant under condensation and adiabatic conditions with throttling of the flow at the inlet to the microchannels has been performed.

It has been determined that under such conditions the relation recommended by Kim and Mudawar [8] for calculating the effect of friction on the pressure drop gives an underestimated value of 40%. The best fit to the experimental data is shown by the Chisholm ratio [7].

When calculating the pressure drop caused by abrupt changes in the flow area, the ratios from Collier Thome [10] are in good agreement with experimental data only at low dynamic pressures and show an underestimated pressure drop in comparison with experimental data at high dynamic pressures. The best agreement with the experimental data is found for calculations according to the ratios from Abdellal et al. [11].

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References
[1] Adams D C et al. 2006 The Int. Refrigeration and Air Conditioning Conf. R 174
[2] Sakamatapan K, Wongwises S 2014 Int J of Heat and Mass Transfer 75 31–9
[3] Shamirzaev A S, Kuznetsov V V 2019 Journal of Physics: Conf Series 1382(1) 012118
[4] Shah R K, London A L 1978 Laminar flow forced convection in ducts
[5] Steinke M E, Kandlikar S G 2005 ASME 3rd International Conference on Microchannels and Minichannels 291
[6] Kays W M, London A L 1984 Compact heat exchangers
[7] Chisholm D 1967 Int J of Heat and Mass Transfer 10(12) 1767
[8] Kim S, Mudawar I 2014 Int J of Heat and Mass Transfer 77 74
[9] Zivi S M 1964 J. Heat Transfer – Trans. ASME 86 247
[10] Collier J G, Thome J R 1994 Convective Boiling and Condensation
[11] Abdellal F F et al. 2005 Experimental Thermal and Fluid Science 29(4) 425