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Condensation of refrigerant R407C in multiport minichannel section

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Abstract Analysis of the state of-the-art in research of refrigerant condensation in miniature heat exchangers, so-called multiports, was made. Results of refrigerant R407C condensation in a mini condenser made in the form of two bundles of tubular minichannels from stainless steel with an inside diameter 0.64 mm and length 100 mm have been presented. Two exchangers consisted of four minichannels and 8 minichannels have been investigated. The values of average heat transfer coefficient and frictional pressure drops throughout the condensation process were designated. The impact of the vapor quality of refrigerant and the mass flux density on the intensity of heat transfer and flow resistance were illustrated. A comparative analysis of test results for various refrigerants in both mini heat exchangers were made.

Keywords: Condensation; Minichannels; R407C refrigerant; Heat exchanger

Nomenclature

A	–	inner surface of the heat transfer, m^2
d	–	diameter, m
f	–	functional symbol
G	–	mass flux density, $kg/(m^2s$
L	–	length, m
$\dot{m}$	–	mass flux, kg/s

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n	–	number of minichannels
p	–	pressure, Pa
Δp	–	pressure drop, Pa
q	–	heat flux density, W/m^2
$\dot{Q}$	–	heat flux from electric heating, W
T	–	temperature, °C
x	–	vapour quality,

Greek symbols

α	–	heat transfer coefficient, W/(m^{2}K)
ρ	–	mass density, kg/m^3

Subscripts

a	–	average
f	–	fluid
s	–	saturation
w	–	wall

1 Introduction

At the turn of the XX and XXI century was observed a rapid development of technological progress. It is manifested by qualitative and quantitative increase in production of equipment, especially in the two fields: in the aerospace and electronics. These fields are inextricably linked and determine the trend of these devices miniaturization. It should be noted that the absolute value of thermal power in computer systems is not very large, but the heat flux density, with is the amount of heat transferred by the heat exchange surface area reaches a significant value, even more than 1000 W/cm^2, as indicated by Baummer *et al.* [1]. The traditional ways of transmitting or receiving such heat flux density are not very useful. Discussion of current methods and recommendations for future includes work of the Obhan and Garimella [15]. The use of two-phase flow, mediating in the heat exchange is a priority in these situations. The practical utilization is reduced to implementation of the convective heat exchange intensification methods with the phase changes. One of the passive methods of the convective heat transfer intensification process is to reduce the channels internal diameter for the agents with implementing the process of heat transfer [12,14]. The measure of the effectiveness of this intensification is increase of the heat transfer coefficient. It can be said that the construction of modern refrigeration and air conditioning heat exchangers should meet the technical and environmental criteria, like: compact dimensions, high heat transfer efficiency, and minimal impact on the environment [3,4,13].

Due to the withdrawal of refrigerant R134a from use, more often are used new, high pressure refrigerants like R404A, R410A, and R407C. From the world literature review results that R407C refrigerant is recommended to use in near future. This zeotropic refrigerant is problematic because of a high temperature glide. Review of the modern calculation methods for heat transfer coefficient and pressure drop in the condensation in conventional and minichannels was reviewed and discussed in [9]. Authors show the usefulness of Silver-Bell-Ghaly [17,3], Thome [18], and Cavallini *et al.* [8] calculation methods. These methods are developed based on a mechanism of zeotropic mixtures flow condensation. Honda *et al.* [10] performed a comparative study of heat transfer during condensation of R407C and R22 refrigerants in a horizontal pipe with an internal diameter approximately 5.38 mm with microfins. It was shown that the heat transfer coefficient of R407C was lower than that of R22, and significant differences existed for smaller values of the vapor quality. In the paper by Lie *et al.* [11] were presented the experimental results of R407C and R134a refrigerants boiling in horizontal smooth tubes with internal diameters of 0.83 mm and 2 mm for mass flux density G = 200–400 kg/ m^{2}s, heat flux density q = 5–15 kW/m^2, vapour quality x = 0.2–0.8, and saturation temperature T_s = 5–15 °C. In the paper was shown that the friction pressure drop for R407C boiling in minichannels is lower than for R134a. The authors developed their own empirical correlations. The paper of Zhang *et al.* [19] presented the results of R22, R410A, and R407C refrigerant condensation in single circular minichannels with internal diameters of 1.088 and 1.289 mm, with parameters in the ranges of: T_s = 30–40 °C, G = 300–600 kg/m^{2}s, x = 0.1–0.9. As expected, it was found that pressure drops increase when the mass flux density, G, and vapor quality, x, increase, but in the range of $x > 0.8$, this influence is much smaller. This underlined the dependence of the two-phase flow regime type on the flow resistance. For higher values of x, there was observed a transition from the annular flow to the mist flow. Additionally, the influence of the channel diameter and refrigerant type on pressure drops was shown [5].

2 Experimental investigations

2.1 Object of the experiments

The object of the experimental studies were two bundles of tubular minichannels (multiports) of the design shown in Fig. 1. Bundle of the tubular minichannels called, MULTI-4 consisted, of 4 minichannels made from stainless steel with an internal diameter $d = 0.64$ mm and length $L = 100$ mm. In the case of tube bundle called MULTI-8 uses an 8 tubular minichannels with the same internal diameter and length.

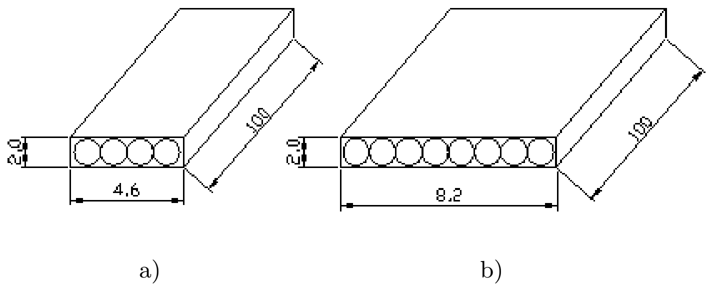


Figure 1: Dimentional diagram of the testing minichannels bundles: a) MULTI-4, b) MULTI-8.

2.2 Testing facility

The experimental investigations were made on the test stand, which diagram is showed in Fig. 2, and external appearance in Fig.3 [6,7]. The basic element of the test stand was a operating distance (1), along with the testing mini heat exchanger. It was placed in the horizontal axis of the rectangular water channel 2 made of aluminum with internal dimensions of 28×24 mm. To measure section (1) refrigerant was brought from the compressor (3) discharge side. Prior to flow to the minichannel inlet section the superheated steam of refrigerant flows through a tube in tube heat exchanger (10) cooled with water. The application of this heat exchanger is not only allowed to the removal of overheating heat, but also for the preparation of refrigerant in the form of dry saturated steam with a vapor quality $x = 1$ (or close to this state). After condensation of the refrigerant vapor flow through miniature heat exchanger, the refrigerant liquid flow to sub cooler (11), from which the flow rate of refrigerant through was measured by Coriolis flowmeter (15). Control posts were also measured the refrigerant mass flow rate through the vascular system hallmarked. Then the refrigerant returns to the refrigeration system supplied with the unit (3), with air-cooled condenser (4) and the lamelled air cooler (8). Tubular minichannels included in the multiport were supplied parallel of refrigerant according to the diagram in Fig. 3.

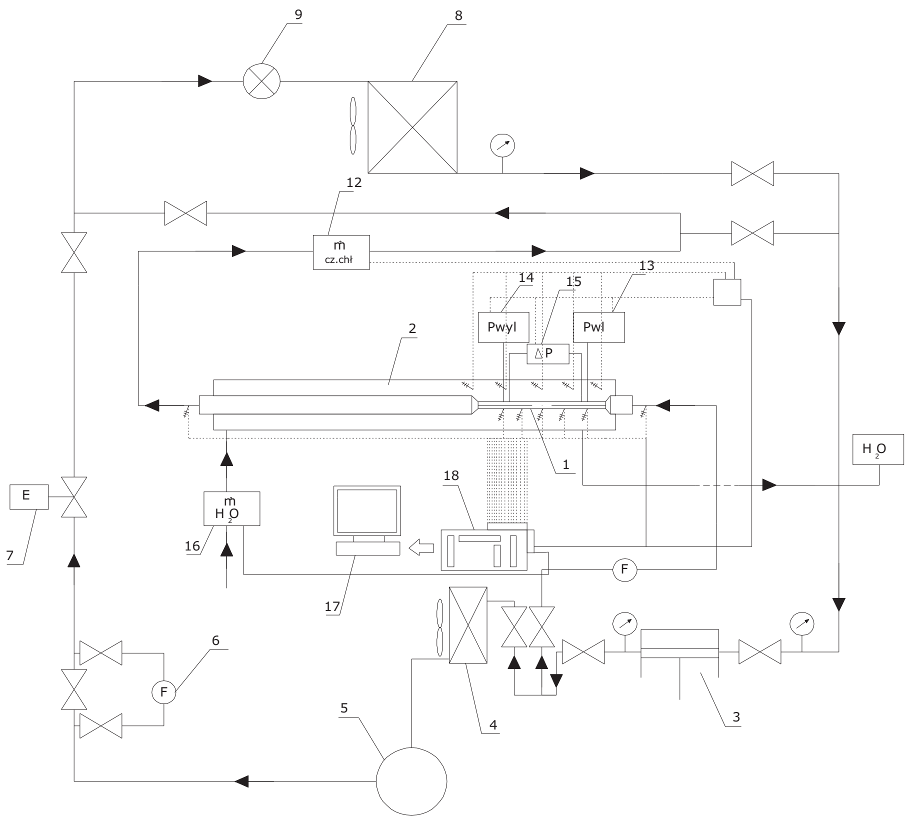


Figure 2: Schematic diagram of the test stand: 1 – test section, 2 – water channel, 3 – refrigeration compressor installation, 4 – air cooled condenser, 5 – liquid vessel, 6 – filter–dryer of refrigerant, 7 – electromagnetic valve, 8 – lamelled air cooler, 9 – expansion valve, 12 – electronic flowmeter of refrigerant, 13 – refrigerant's pressure pickup on the inlet to the measuring section, 14 – refrigerant's pressure pickup on the outlet to the measuring section, 15 – refrigerant's differential pressure transducer, 16 – water electronic flowmeter, 17 – computer, 18 – data acquisition system.

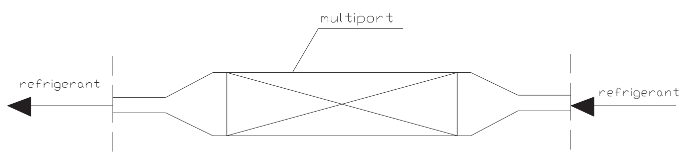


Figure 3: Supply diagram of the multiports minichannels by the refrigerant.

Multiports were mounted interchangeably as a part of measuring section (1) of the test stand shown in Fig. 2. On the length L of the multiport mounted nine K-type thermocouples to measure temperature distribution on multiports outer wall. After taking into account the thermal resistance and bringing the related corrections, specified average temperature of the pipe minichannels inner wall surface in that section. The cooling water temperature distribution in nine sections was also measured. Directly measured mass flux density of refrigerant condensing in the flow through the multiport and the heat flux density by the methodology proposed by the Shin and Kim [16].

This method consists of heat transfer coefficient determination from the water side at a specific temperature difference of the minichannel wall, T_w, and water, T_f. Test section was heated by electric heater. The construction of the test section for this aim is shown in Fig. 4.

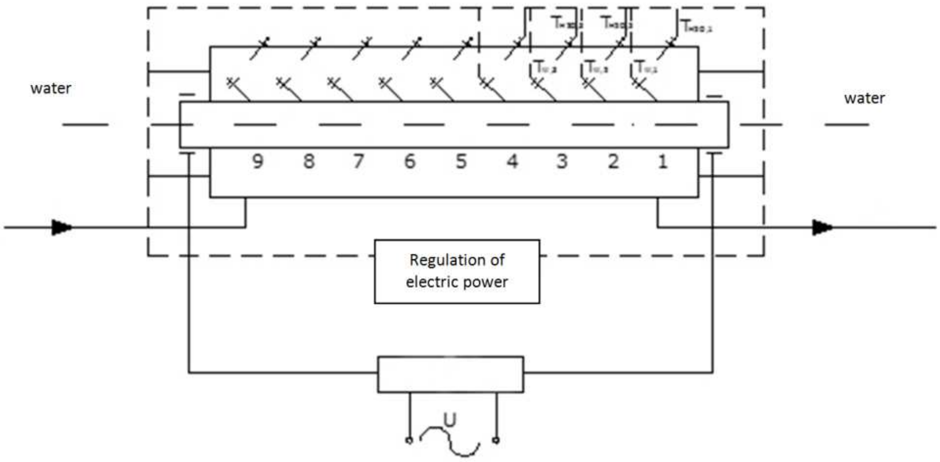


Figure 4: Scheme of the test section used for the indirect determination of heat amount received by cooling water in minichannel elecrically heated.

In line with the law of Joule the heat flux density on a minichannel was specified :

$$q_i = \frac{\dot{Q}_i}{\pi d L_i} \, , \tag{1}$$

where L_i is the length of minichannel, $\dot{Q}$ is the heat flux from electric heating, and subscript i denotes the number of cross-section.

Using the temperatures of channel wall T_w and cooling water T_f in chosen cross-section i, characteristics of $q_i = f(T_{w,i} - T_{f,i})$ was made, which allow to calculate the heat flux density during condensation of refrigerant after measuring of wall and cooling water temperature (here f is the functional symbol).

Directly was measured also the refrigerant pressure at the inlet and outlet of multiport as well as the pressure drop, Δp, in the flow. These parameters allowed for determination of an average value of the vapor quality, x_a, and medium mass flux density G, pressure drop $(\Delta p/L)$, which enabled subsequently elaboration of the experimental thermal-hydraulic characteristics of the condensation process.

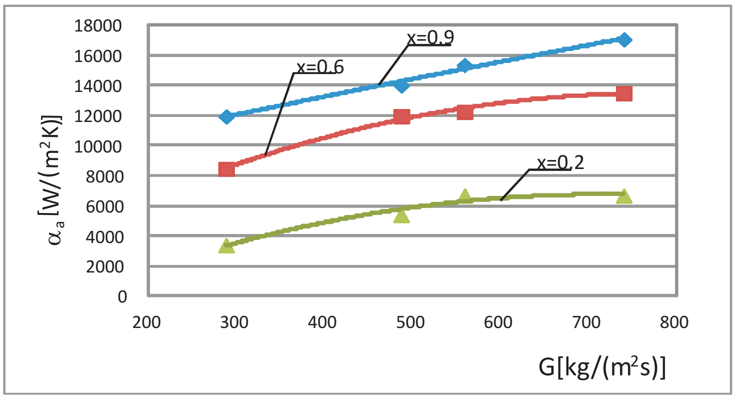
3 Results of the investigations

An experimental heat-flow researches were performed for condensation of the R407C refrigerant in multiports MULTI-4 and MULTI-8 (Fig. 1). The average values of heat transfer coefficient and pressure drop were defined. Mass flux density of the refrigerant was determined from direct measurement of the mass flow rate of refrigerant at the inlet to multiport by using the relationship

$$G = \frac{\dot{m}}{n\frac{\pi d^2}{4}} , \tag{2}$$

where $\dot{m}$ is the mass flow, n is the number of minichannels parallel supplied ($n = 4$ for MULTI-4 and $n = 8$ for MULTI-8), and d is the internal diameter of the minichannel ($d = 0.64$ mm). The heat transfer density, q, is related to 1 m^2 multiport inner surface of the heat transfer: $A = n\,\pi\,d\,L$, where L is the length of minichannels in multiport.

Figures 5–7 shows the dependence of the average (marked with subscript a) heat transfer coefficient, α_a, and flow resistance, $(\Delta p/L)_a$, in two models of interpretation, first when the mass flux density, G, with $x_a =$ const, and second when it dependence on the x_a, the $G =$ const.



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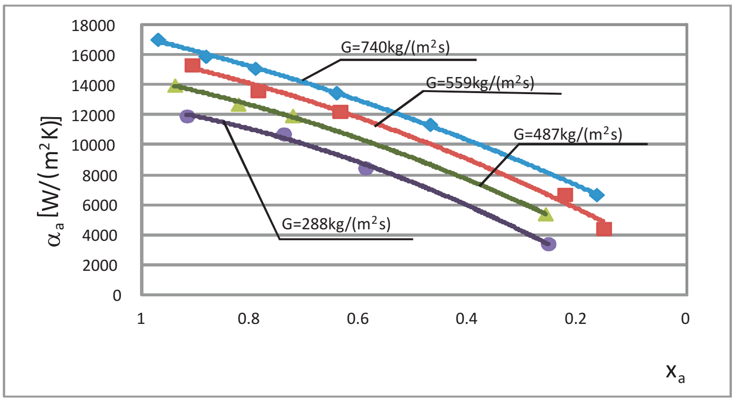
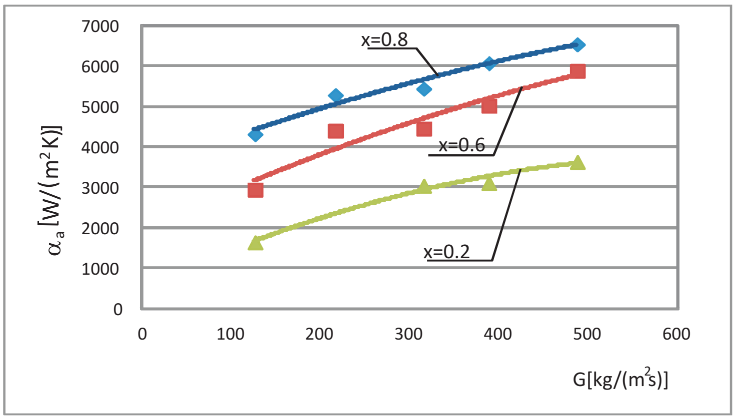


b)

Figure 5: Experimental investigation results of the dependence of the average heat transfer coefficient, α_a, on: a) mass flux density, G, for x_a = const; b) average vapor quality, x_a, for G = const.; multiport MULTI-4, refrigerant – R407C.



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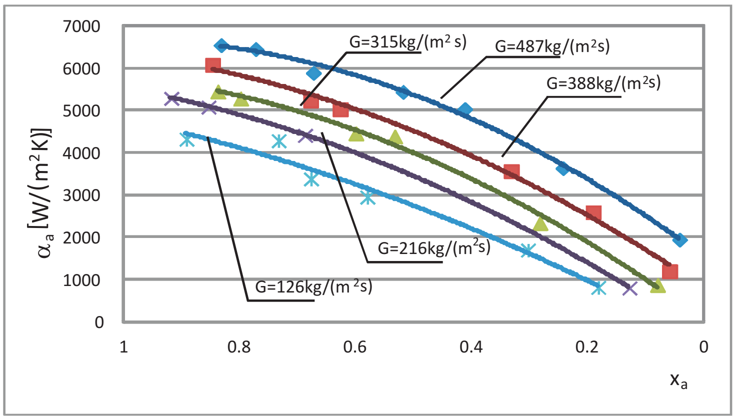
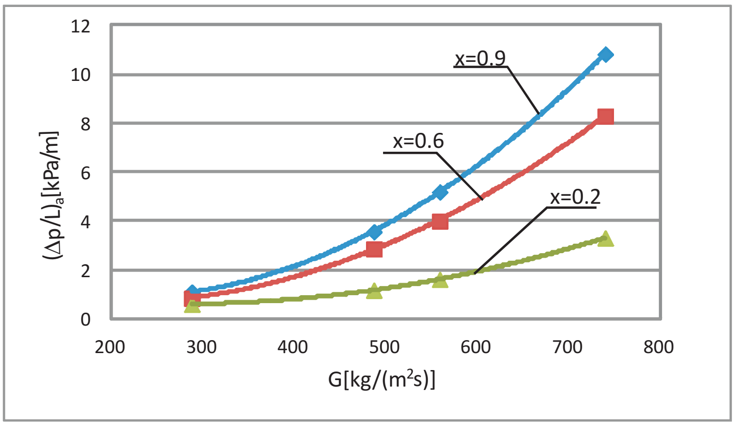


b)

Figure 6: Experimental investigation results of the dependence of the average heat transfer coefficient, α_a, on: a) mass flux density, G, for x_a =const; b) average vapor quality, x_a, for G = const.; multiport MULTI-8, refrigerant – R407C.



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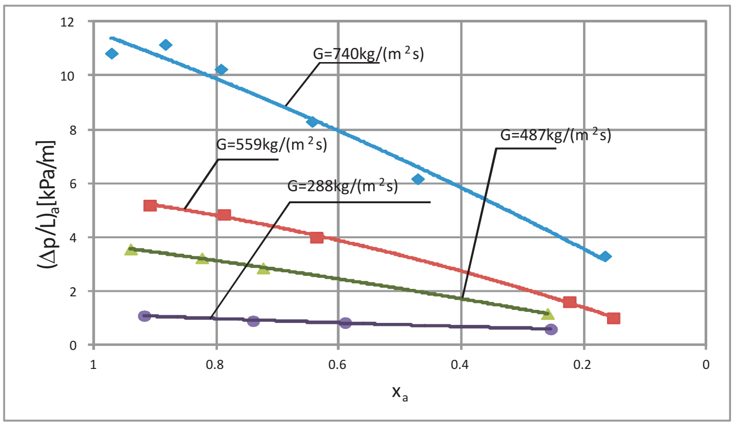
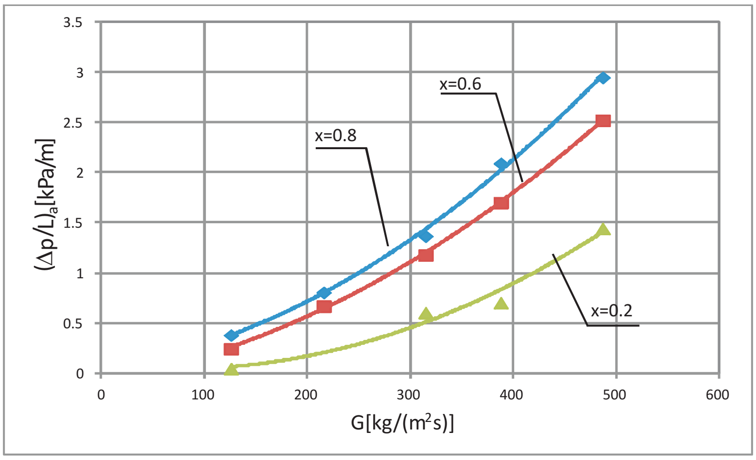


b)

Figure 7: Experimental investigation results of the dependence of the average pressure drop, $(\Delta p/L)_a$, on: a) mass flux density, G, for x_a = const; b) average vapor quality, x_a, for G = const.; multiport MULTI-4, refrigerant – R407C.



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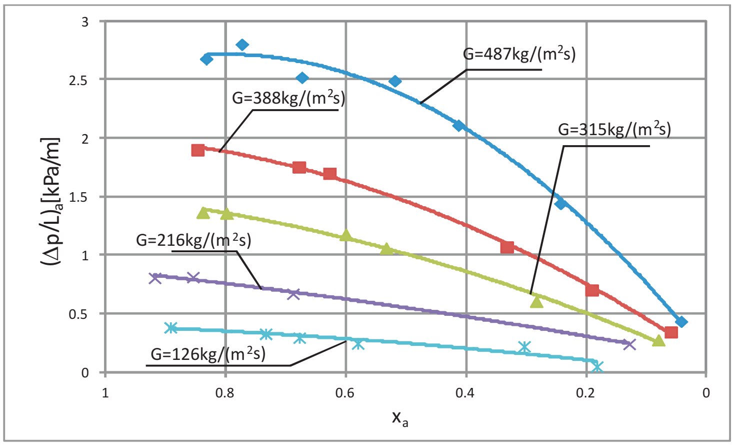


b)

Figure 8: Experimental investigation results of the dependence of the average pressure drop, $(\Delta p/L)_a$, on: a) mass flux density, G, for x_a = const; b) average vapor quality, x_a, for G = const.; multiport MULTI-8, refrigerant – R407C.

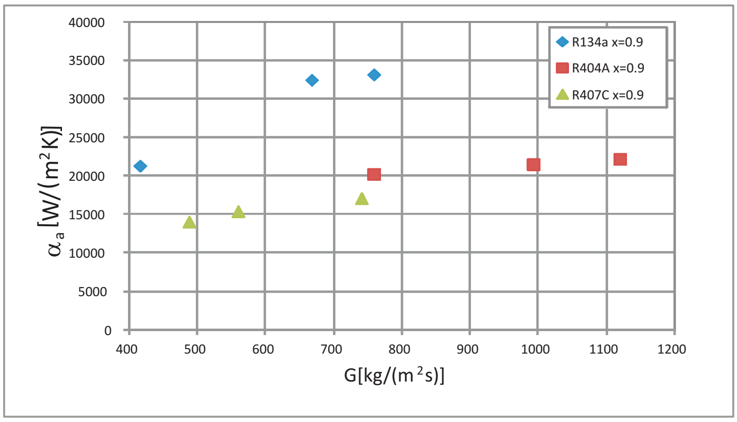
4 Analysis of experimental results

The heat-flow characteristics of R407C refrigerant condensation allow for a comparative analysis of the test parameters. They concerned the condensation in tubular multiports MULTI-4 and MULTI-8 (Fig. 1), which differed in the number of tabular minichnnels. Based on the analysis, the following observations can be made.

Thermal characteristics of condensation in type $\alpha_a = f(G)$, for a constant level of average vapor quality x_a = const showed an increase in the average heat transfer coefficient, α_a, with increase in mass flux density, G of refrigerant. On the basis of diagrams the characteristics of $\alpha_a = f(x_a)$, with G = const shows that the α_a coefficient decreases when average vapor quality, x_a, decrease too. The nature of these average thermal characteristics direction and the average flow resistance characteristics is similar to the characteristics obtained for the condensation process in single minichannels [12].

In the case of multiport, which is fed by n number parallel minichannels, mass flux density was calculated from Eq. (1). This means that for the same values of mass flow rate, $\dot{m}$, the average mass flux density, G, in multiports made from n minichannals is n times smaller (and decreases with increasing on number of parallel minichannels). This results in effects on the value of the heat transfer coefficient, α_a. For the same value $\dot{m}$ = const flowing to the multiport, value of the average heat transfer coefficient, α_a, is growing when the process takes place in a single minichannel, and for multiport MULTI-4 is also higher than for the MULTI-8. It is interesting to compare the characteristics of condensation process obtained according to the type of a refrigerant [7]. Figures 9 and 10 show a example characteristics for MULTI-4 and MULTI-8.

Figure 9 presents a comparative approach to the effects of three refrigerants (R134a, R407C, and R404A) on the thermal and flow characteristics of condensation in multiport from 4 minichannels MULTI-4. The results show that for R134a refrigerant obtained the highest values of the average heat transfer coefficient α_a (and the pressure drop $(\Delta p/L)_a$). Refrigerant R134a is an intermediate pressure refrigerant (in terms of pressure and saturation temperature during the condensation process). Refrigerants R407C and R404A belong to a group of high-pressure refrigerants. Value of the α_a coefficient and pressure drop $(\Delta p/L)_a$ are higher for the R404A refrigerant then for R407C, but this increase is in the range from 10 to 15%. Figure 10 shows a comparison of thermal and flow characteristics for R134a



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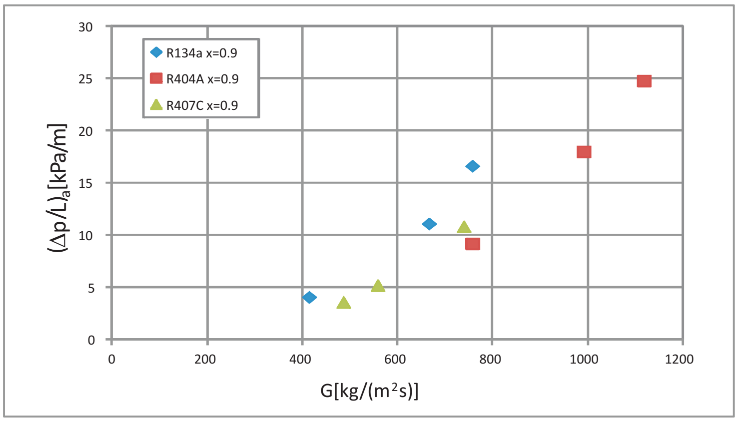
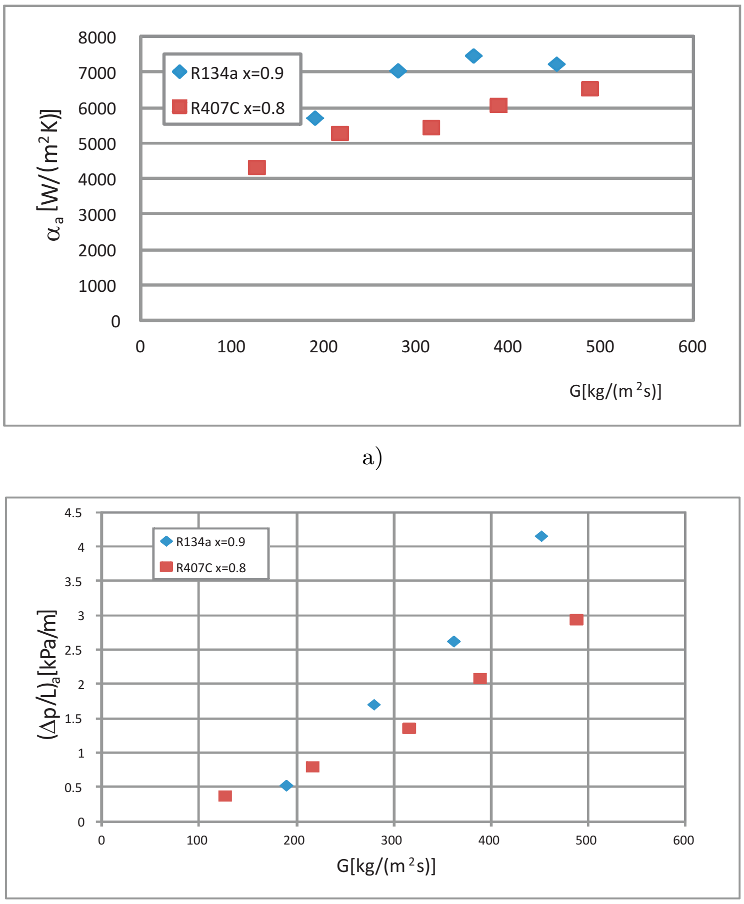


b)

Figure 9: Sample comparative characteristics of the condensation of R134a, R407C and R404A refrigerants in multiport MULTI-4, for $x_a = 0.9$: a) $\alpha_a = f(G)$, b) $(\Delta p/L)_a = f(G)$.

and R407C refrigerants for multiport MULTI-8. The increase in flow resistance for R407C refrigerant is a lack of acceptance for the analysis of the test results at a lower average vapor quality.



a)

b)

Figure 10: Sample comparative characteristics of the condensation of R134a, R407C and R404A refrigerants in multiport MULTI-8, for $x_a = 0.9$: a) $\alpha_a = f(G)$, b) $(\Delta p/L)_a = f(G)$.

5 Conclusions

Analysis of the diagrams Leeds to the following conclusions:

1. Construction of the compact condensers built on the minichannels contain elements of the multiports forming the heat exchange area. Current state of knowledge in the field of the heat transfer and pressure drop during refrigerants condensation in multiports is definitely unsatisfactory.

2. The results of thermal flow experimental tests of the R134a, R404A, and R407C condensation in multiports from 4 – and 8 minichannels parallel fed, showed that the highest values of average heat transfer coefficient and pressure drop were obtained for R134a.

3. Knowledge of the average values of the heat transfer coefficient and pressure drop during condensation of the refrigerants in multipors is very important for the designers of this miniature heat exchangers type. Raising awareness of energy transfer mechanisms during phase changing in multiports should include their local assessment, depending on vapor quality x changes. This is also the need to test for other sections of the multiport.

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