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AN INVESTIGATION OF HEAT TRANSFER COEFFICIENT DURING REFRIGERANTS CONDENSATION IN VERTICAL PIPE MICROCHANNEL

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Abstract: This article presents the results of experimental research of R404A, R407C and R410A high-pressure refrigerants condensation in vertical pipe microchannels with an internal diameter d_w below 1 mm. The study determined the local and average heat transfer coefficient in the full range of vapor quality, x = 1-0. On the basis of experimental investigations, the dependence of heat transfer coefficient on the vapor quality x, the mass flux density G and the channel internal diameter d_w was obtained.

Keywords: Heat Transfer Coefficient, Pressure drop, heat flux density, microchannel, vapor quality

1. INTRODUCTION

The dynamic expansion in global development forces the unceasing need for innovative solutions in design and construction of equipment. These solutions aim at intensifying utility parameters such as power, efficiency and reducing their size while lowering production costs. Researchers are therefore looking for the best solutions from a technical and economic point of view. Increase of efficiency factor results in considerable usage of equipment and that generates increase in production costs [3]. One of the main causes of wear is the inefficient cooling. Due to the limited size of conventional heat exchangers, micro-condensers fit perfectly into the needs of the market. But we still have to be aware of ecological considerations. The smaller the size of the system, the less the risk to the environment – in case of leak of the refrigerant to the environment.

Compact refrigeration systems are commonly used, for example, in personal computers or smartphones. However, miniature heat exchangers will be gradually displaced on the market by "micro" heat exchangers [1]. Growing demands and requirements for heat exchangers constructors forces new solutions. The authors assume that the efficiency of the micro-heat exchangers is mostly influenced by: the nature of the refrigerant flow, the type of refrigerant used, and the influence of the surface tension forces. Not meaningless stays the type of heat reception and channels orientation.

There is still a small number of heat exchanger designed with an air-cooled microchannels. A similar situation occurs with vertical channels.

Paper by Chen et. al. [2] FC-72 describes refrigerants condensation model in rectangular channel with hydraulic diameter d_h = 1mm. The model was confronted with research results available in the world literature. It has been noted that the vapor velocity of the medium is significantly higher than that of its condensate. There was also found that an increase in the size of the bubbles goes along with the increase in the mass flux density.

A new, universal approach to predict the heat transfer coefficient in minichannels and microchannels were presented by Kim and Mudawar [10]. The concept applies to a number of fluids with significantly different thermo-physical properties, as well as for variable channel geometry and flow parameters. The authors collected a database of 28 different sources, including over 4000 measurement points. The data concern the condensation of 17 types of working fluids in single channels and multiports, and their hydraulic diameters are in the range: d_h = 0.424-6.22 mm. Mass flow factor G = 53-1403 kg·m^{-2}·s^{-1}. Two new correlations were proposed: the first for the annular flow, the second for plug flow and bubbly flow.

Mikielewicz et al. [13] investigated the flow resistance and pressure drop during condensation in the two-phase flow in cylindrical minichannel of 2.3 mm inner diameter. They analyzed the effect of heat flux, mass flux, vapor quality and saturation temperature on two-phase pressure drop of HFE 7100 and HFE 7000.

Fronk and Garimella [4] investigated the heat transfer coefficient and flow resistance during carbon dioxide (CO_2) condensation in rectangular ducts. The channels hydraulic diameters were d_h = 0.1 mm and 0.16 mm. Channels were made of copper by electroforming. Thermo-flow parameters were measured at mass flux density G = 400, 600 and 800 kg/m^2·s and the full range of refrigerant dryness x. The authors compared the results with the investigations data from several existing correlations. The best agreement for pressure drop was obtained with Garimella correlation [5]. For the heat transfer coefficient, the best compatibility was obtained for authors [6, 7, 8, 9] data.

Due to significant differences between the values of the heat transfer coefficient in the case of different methods of receiving condensation heat and some deficiencies in the world literature, the authors decided to carry out their own experimental studies on refrigerant R404A, R407C and R410A condensation in air-cooled vertical tube microchannels with inner diameters d_w = 0.5 and 0.7 mm to create an experimental database.

2.EXPERIMENTAL STAND

The heat and flow studies of the condensation process were carried out at a test stand that is shown in Figure 1. The superheated steam of the refrigerant was forced through the measuring section by the compressor after pre-cooling. A heat exchanger was installed in front of the measuring section, which was used to determine the vapor quality of the medium by the balance method. For this purpose, the volume flow of the cooling water and the refrigerant was measured, as well as the water and refrigerant temperature at the inlet and outlet of the exchanger. Adjustment of the water flow rate allowed to control the parameters of the medium (eg. vapor quality) at the inlet to of the measuring section. Then the refrigerant condensed inside a vertical 950 mm stainless steel vertical tube microchannel (flow vertically down). microchannel was placed in a rectangular duct measuring 60×120×1100 mm. A counterflow currant of air at a speed of 17 m/s was forced into the duct. Refrigerant pressure was measured at the inlet and outlet of the measuring section with piezoresistive sensors fitted with the Endress + Hauser PMP 131-A1401A1W Transmitter, made in measuring class 0.5. In addition, a local pressure drop of 100 mm was measured with the Deltabar SPMP in class 0.075. Second heat exchanger was installed to obtain a homogenous liquid at the outlet of the measuring section. After cooling with water in the exchanger, the liquid medium was sent to the Coriolis 34XIP67 flowmeter in measuring class 0.52, where its mass flow was measured. The flow rate of the water through the exchanger was controlled by a RTU-06-160 type rotameter with accuracy class 2.5.

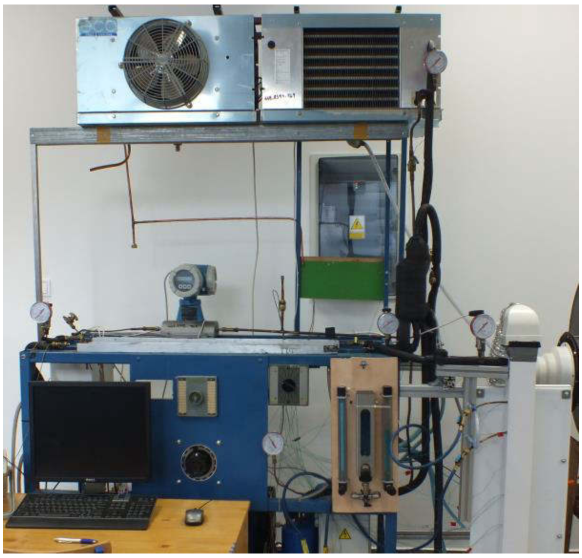


Fig. 1. The overall view of the experimental stand

The measuring equipment installed on the test bench allowed direct measurement of following values:

- the temperature of the refrigerant T_{F1} at the inlet and outlet T_{F2} from the measuring section,
- surface temperature of the outer wall of the tube microchannel using thermocouples in nine sections T_{W1} to T_{W9},
- cooling air temperature in three measuring sections in the air duct over the length of the measuring section T_{P1}-T_{P3},
- refrigerant mass flow rate through the measuring section $\dot{m}_r$,
- air mass flow rate through the measuring section $\dot{m}_p$,
- refrigerant pressure at the inlet and outlet of the measuring section p_{n1}, p_{n2},
- refrigerant pressure drop during the flow through the measuring section ($\Delta p/L$),
- refrigerant temperature T_{F01}, T_{F02} at the inlet and outlet of the pre-cooling heat exchanger,

- T_{H01} temperature of cooling water at the inlet and T_{H02} at outlet of the exchanger,
- the mass flow rate of the water through the exchanger $\dot{m}_{H2O}$.
- Indirectly the following values were calculated:
- heat flux density q,
- density of mass G,
- vapor quality x.

Measuring section is presented in Figure 2.

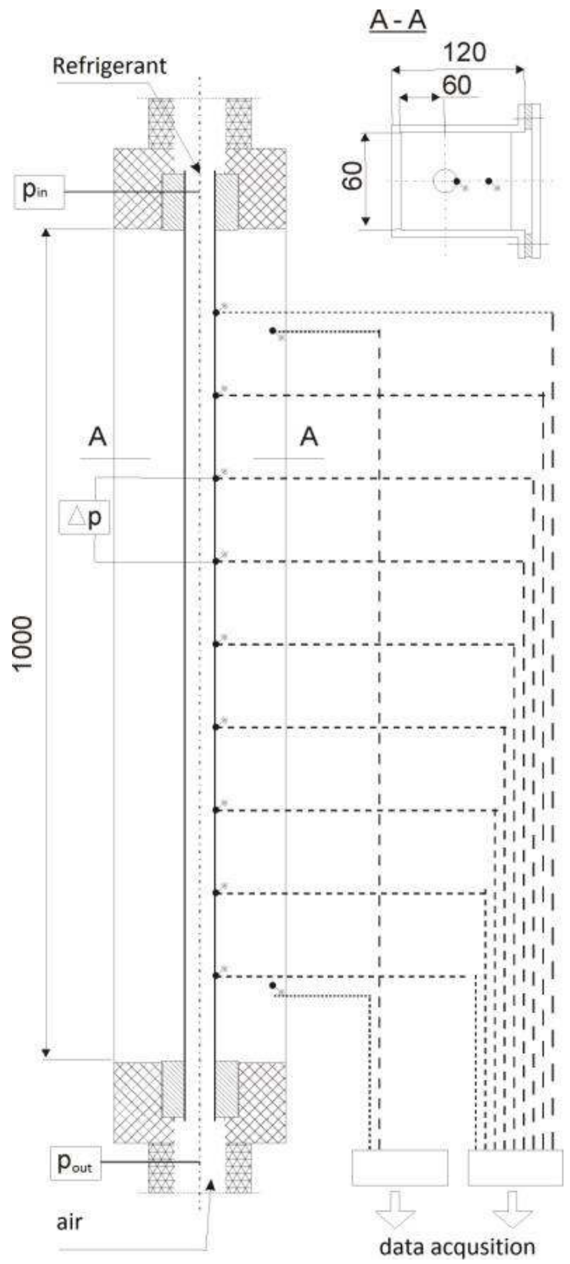


Fig. 2. Framework of the measuring section

The schematic diagram of experimental facility is shown in Figure 3.

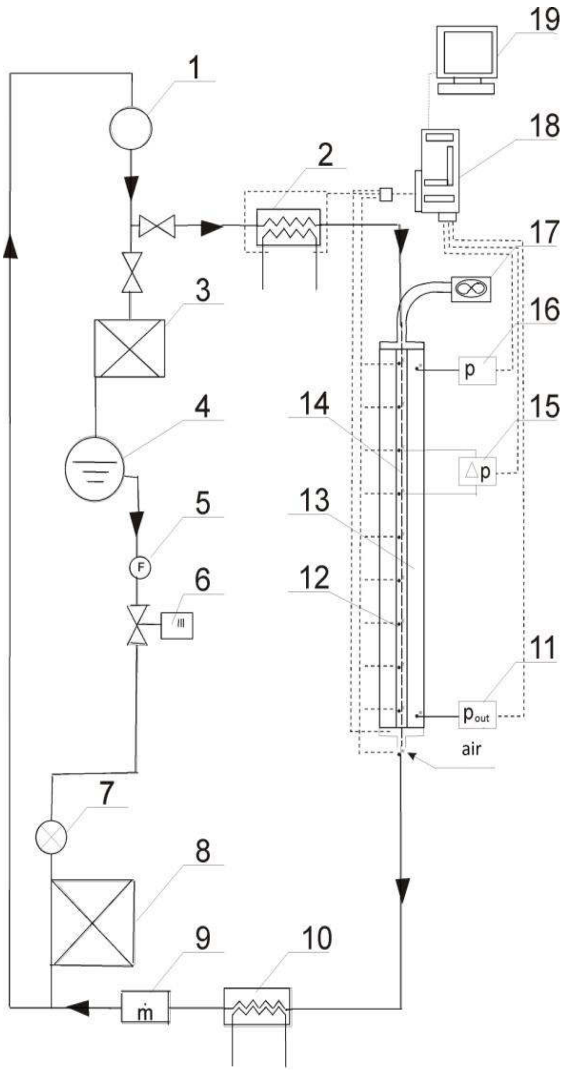


Fig. 3. Schematic diagram of the test stand: 1) compressor, 2) heat exchanger, 3) condenser, 4) refrigerant liquid tank, 5) filter, 6) electromagnetic valve, 7) expansion valve, 8) fan cooler, 9) refrigerant mass flow meter, 10) heat exchanger, 11) inlet pressure sensor, 12) K-type thermocouples, 13) air channel, 14) pipe microchannel, 15) differential pressure measurement, 16) outlet pressure sensor, 17) fan, 18) data acquisition system, 19) computer of the measuring section

3. EXPERIMENTAL RESULTS

One of the key elements of the analysis of the refrigerant condensation process in the tube microchannels is to determine the distribution of the value of the heat transfer coefficient α_x

Fig. 4 shows the dependence of the mass flux density G on local heat transfer coefficient in a microchannel with a diameter of d_w = 0.5 mm while refrigerants R410A, R407C and R404A condenses.

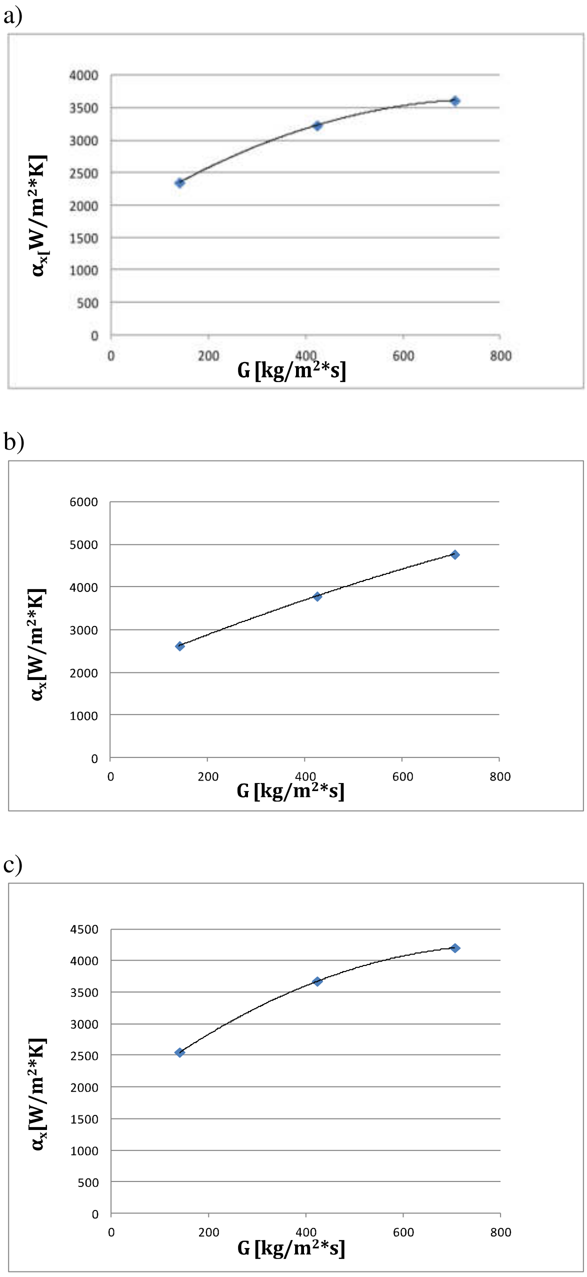


Fig. 4. Experimental influence of mass flux density on local heat transfer coefficient during condensation of refrigerants a) R410A, b) R407C, c) R404A in vertical pipe minichannel with internal diameter $d_w = 0.5$ mm

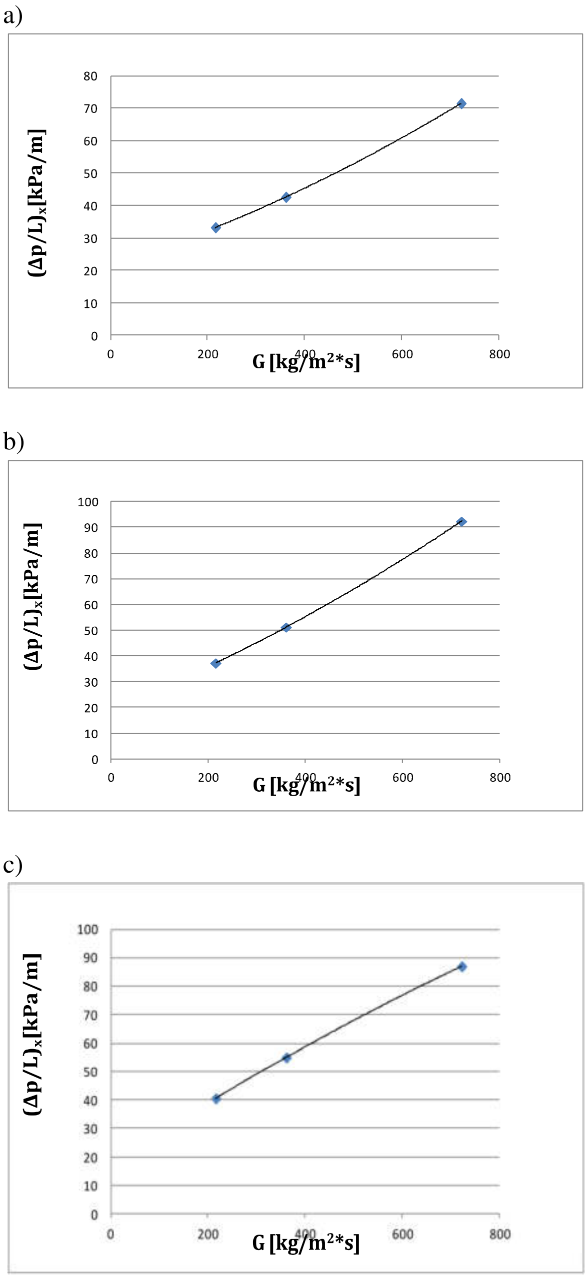


Fig. 5. The influence of mass flux density G on local pressure drop *$(\Delta p/l)_x$* during refrigerants a) R410A, b) R407C and R404A condensation in microchannel with inner diameter $d_w = 0.7$ mm

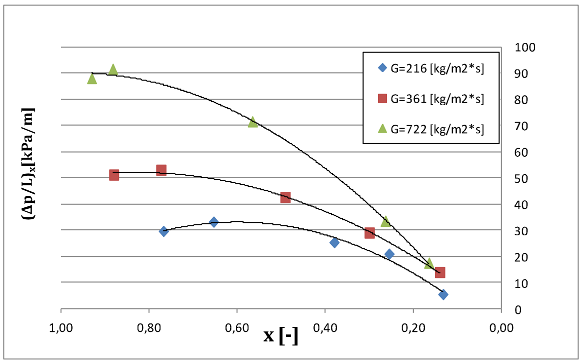
The local heat transfer coefficient increases along with the increase of mass flux density – regardless to the type of refrigerant. The heat transfer coefficient takes highest values during R407C refrigerant's condensation.

Figure 5 shows the dependence of values of local pressure drop on the mass flux density *G* during refrigerant condensation (a) R410A, (b) R407C and (c) R404A in tube microchannel with diameter $d_w = 0.7$ mm. As it can be seen along with the increase in density of the mass flux, the local pressure drop increases.

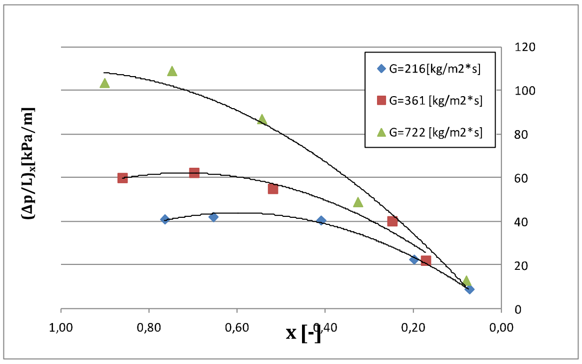
In all three cases the tendencies are similar. Value of local pressure drop *$(\Delta p/l)_x$* during refrigerant R410A condensation oscillates in the range of 35-72 kPa/m, for R404A it is 40-87 kPa/m and for R407C the range is between 37 and 92 kPa/m. As it can be seen in the microchannel with an internal diameter $d_w = 0.7$ mm refrigerant R407C shows slightly larger flow resistance then R404A does. The smallest pressure drops occurs during R410A condensation.

Figure 6 presents the dependence of dryness *x* on local pressure drop *$(\Delta p/l)_x$* of all three examined refrigerants for three chosen mass flux density rates. It was found that the dryness *x* determines pressure drop during condensation. The biggest pressure drop values were observed in the main condensing area, which is $x = 0.6$-0.8.

a)



b)



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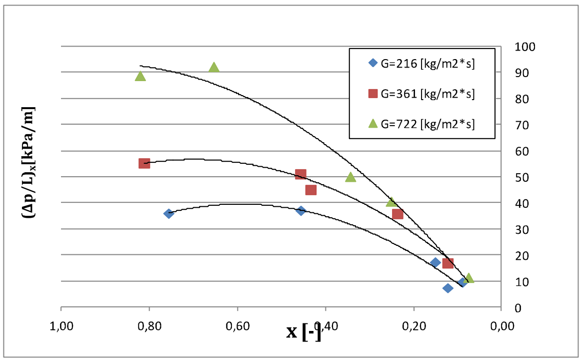


Fig. 6. The influence of dryness x on local pressure drop during refrigerant condensation a) R404A, b) R407C and c) R410A in microchannel with inner diameter $d_w = 0.5$ mm

The local pressure drop increases with the increase of mass flux density.

It was also found that along with the decrease of internal diameter d_w local heat transfer coefficient increases. The dependence of microchannel's internal diameter is characteristic for all three types of tested refrigerants.

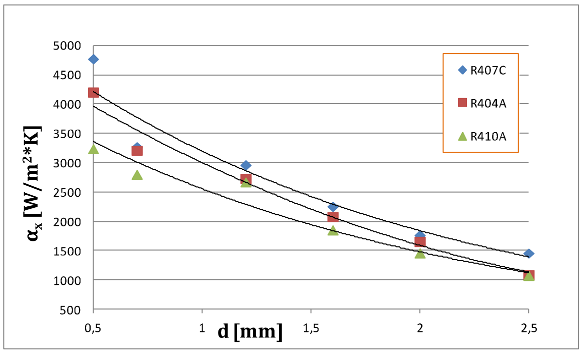


Fig. 7. Comparison of experimental local HTC in microchannel local HTC in minichannel for all three examined refrigerants

The influence of internal diameter on heat transfer coefficient becomes more pronounced with the crossing of 1 mm internal diameter boundary (were the micro scale begins).

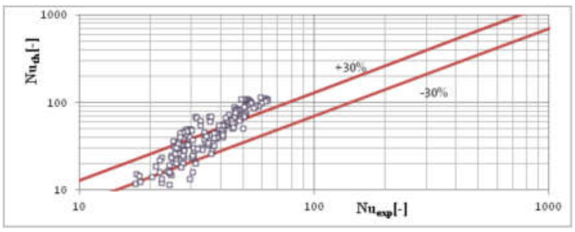
Based on correlation of various authors and dimensional analysis a new correlation for local heat transfer coefficient and local pressure drop was obtained. Correlation parameters were determined using nonlinear regression model estimated by *Levenberg – Marquardt* method (in Statistica's standard pacage). Local heat transfer coefficient α_x. was calculated by:

$$Nu_x = 0.63 \cdot Re_l^{0.35} \cdot p_r^{-0.49} \cdot Pr_l^{0.79} \cdot \left(\frac{x}{1-x}\right)^{0.22}, \quad (1)$$

$$\alpha_x = \frac{Nu \cdot \lambda_l}{d_w}. \quad (2)$$

Experiment results were compared with calculations of own and other authors correlation (Fig. 8).

a)



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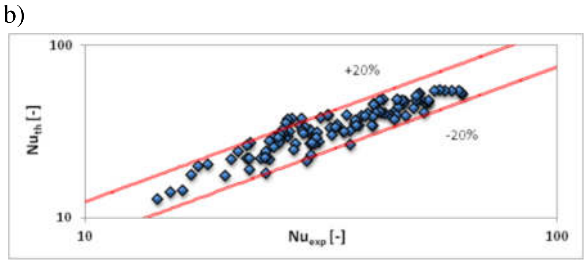


Fig. 8. Comparison of experimental and theoretical local Nusselt number results from the correlation by: a) Thome [11], b) author's own correlation for condensation of R407C in microchannel with internal diameter $d_w = 0.7$ mm

The results of the comparison with the other authors correlations were subject to a significant error (MAE) of over 50%. Such a large discrepancy may result with different ranges of applicability of particular correlations.

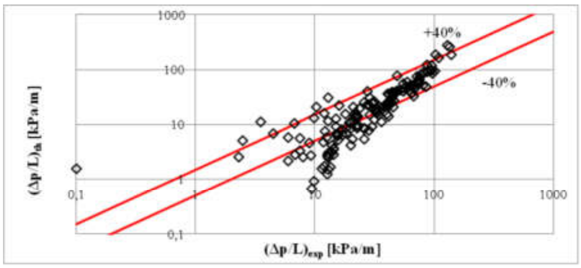
Local pressure drop was calculated by:

$$\left(\frac{\Delta p}{L}\right)_{Fr} = \left(\frac{\Delta p}{L}\right)_{lo} \cdot \Phi_{lo}^2 \,, \quad (3)$$

$$\Phi_{lo}^2 = \left[2.62 \cdot {p_r}^{0.28} \cdot E^{0.71} + 245.77 \cdot \left(\frac{F^{1.34}+H^{2.11}}{We^{1.15}}\right)\right]. \quad (4)$$

Experiment results were compared with calculations of own and other authors correlation (Fig. 9).

a)



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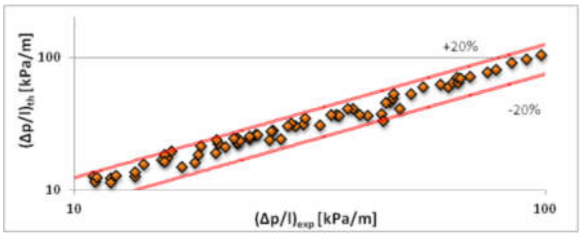

Fig. 9. Comparison of experimental and theoretical local pressure drop results from the correlation by: a) Friedel [12], b) author's own correlation for condensation of R407C in microchannel with internal diameter d_w = 0.5 mm

After comparing calculations of Friedel's corre-lation with the results of the experimental studies, it was found that for the investigated case the total error of the MAE was 45% R407C refrigerant.

4. CONCLUSIONS

1. The heat and flow measurements of R407C, R404A and R410A refrigerants condensation was conducted in pipe microchannels with an internal diameter d_w = 0.7 and 0.5 mm. The study included the determination of average and local pressure drop and heat transfer coefficient in the model conditions.
2. It was found that the heat transfer coefficient depends not only on the size of the inner diameter d of the pipe microchannel but also on the mass flux density G and local vapor quality.
3. Based on the above it is recognized that there is a need for further research and elaboration of own empirical correlation describing heat exchange during condensation of refrigerants in vertical pipe microchannels.
4. Based on the results experiment, author's own thermal and flow correlations were obtained. Discrepancies between calculated results and experimental results did not exceed the range of 20%.

Nomenclature

Symbols

$(\Delta p/l)_x$ – local pressure drop, kPa/m
α_x – local heat transfer coefficient, W/m^2·K
d_w – internal diameter, mm
G – mass flux density, kg/(m^2·s)
q – heat flux, kW/m^2

Acronyms

HE – Heat Exchanger
HTC – Heat Transfer Coefficient
MC – Microchannel

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Biographical notes

Tadeusz Bohdal is the author of more than 300 scientific and technical publications (domestic and foreign), 10 books and over 100 documented studies for business entities. He directed 6 MNiSWW and NCN grants, promoted 8 doctors, is the author of numerous qualification testimonials, research grants and academic papers. He is a member of the Committee of Thermodynamics and Combustion of the Polish Academy of Sciences, Scientific Committee of the monthly "Chłodnictwo" and a scientific consultant of the monthly "Refrigeration and Air Conditioning Technique". He is also an expert of SIMP and an expert at Koszalin District Court in the field of heat and heat measurement and refrigeration. He has promoted more than 200 engineers and engineering masters in the field of Machine Building and Engineering and Agricultural and Forestry Engineering (specializations: Thermal Power Engineering and Refrigeration, Food Processing and Chemical Engineering, Computer Engineering, Engineering and Management, Food Engineering, Food Processing Technology).

Marcin Kruzel received his M.Sc. degree in Economics at the Institute of Economics and Management of Koszalin University of Technology (2008). Until 2010 academic and didactic staff of the Institute of Economics and Management. Currently a PhD student at the Faculty of Mechanics of Koszalin University of Technology. In his work he deals with refrigeration and the economic and technical aspects of using renewable sources of energy. He is an author of 18 papers printed in national and international magazines. Since 2016 works as a scientific specialist at Laboratory of Energetics in Koszalin University of Technology

Małgorzata Sikora received her M.Sc. degree in Environmental Engineering (specialization: Heating and air conditioning) and next Ph.D (with honors) degree in Machinery Construction and Operation from Koszalin University of Technology, in 2008 and 2011 respectively. Since 2011 she has been an assistant in the Department of Heating and Refrigeration Engineering at the Koszalin University of Technology. Currently she works as an assistant professor in Department of Power Engineering. Her scientific interests concern a heat and flow phenomenon during refrigerants condensation, refrigeration, heat pumps, etc. She has participated a in 4 national research projects, 1 international education project (Tempus Energy). She presenting results of her work at 4 international and numerous national conferences, she published 4 articles in journals from the Philadelphia list and 15 articles in national magazines and 46 papers printed in national and international conferences materials. Dr. Eng. Małgorzata Sikora is also co-author of 1 monograph published in English, and 1 didactic textbook.