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Investigations of Environmentally Friendly Refrigerants' Phase Changes in Minichannels

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1. Introduction

The continuous development of the world economy increases demand for energy in various forms. This applies however mainly to electricity, which is the most valuable and wanted. The increase of its production entails some certain environmental changes. The classic way of its production is associated with an increase of greenhouse gases production and harmful effects on the human environment. New methods of the green energy production are still expensive and inefficient. Construction of nuclear power plants in many cases encounters social resistance. It is therefore the rational use of energy in production whereas the costs, effectiveness of this process and aspects of environmental protections. This has a direct relationship related to the implementation of the sustainable development concept. This concept has to rely on prosperity improvement of society in long-term, by the desire to maintain a balance between energy security, to meet the needs and environmental protection.

An important role is also rational management of the energy. This means reducing of energy consumption in the implementation of various technological processes and supplies. In every area of life, both in manufacturing and in the case of an individual using of the final product should be used with efficient forms of energy. The practical implementation of this goal demands change the current way of thinking, about the production of various products and their use. Very important is implementation of product eco-design, this is the account of its harmful impact on the environment both during the design and later work.

The most important role in the eco-design of energy devices and equipment have two aspects, i.e. rational use of energy and environmental protection. Modern trends in the development of such systems are moving towards their miniaturization. By miniaturization need to understand the trend in the decreasing of system size, while maintaining the functionality of constant use. In this issue important part is designing of miniature heat exchangers, with can exchange big amount of heat by small heat transfer area. It is important because of systems miniaturization in areas such as electronics, medicine, aerospace, transport, etc. The intensification of heat transfer by small area is an absolute necessity. This can be seen very clearly in the increasing production of computer units with have more processing power, which need intensive cooling.

The use of heat exchangers with using single phase flow (such as water or air) isn't longer sufficient. Therefore, it is necessary to use two-phase flow of refrigerants (boiling and condensation) implemented in the channels of small diameter. In this has created an concept of compact heat exchanger (evaporators and condensers) included in the mini cooling systems or micro power station, the concept described in [6]. It will be used to produce electricity and heat for domestic use. According to the authors in the future, it has to replace conventional boilers for heating. Its advantage is the compactness and small size. Because of the small internal size of the micro- and minichannels meet is also an ecological criterion, because in case of leakage in the installation, amount of refrigerant with like out into the environment is negligible.

However, the designers of compact heat exchangers today have a difficult problem of appropriate selection of the calculation in case of heat transfer and flow resistance in channels with hydraulic diameter less than 3 mm [7]. Known in the literature correlations, proven theoretically and experimentally for conventional channels ($d > 3$ mm), usually don't work for micro-and minichannels [4, 15]. It follows the need to identify best calculation formulas for use in compact heat exchangers [1, 5].

The current state of knowledge including conventional channels proves that you can't treat condensation and boiling as symmetrical phenomenon. Besides, there is another mechanism of energy and momentum transfer in the process of boiling and condensation in the flow, in both the conventional and miniature channels.

The paper presents some results of experimental investigations conducted in the Laboratory of the Thermal and Refrigeration Technology Department in Koszalin University of Technology. They concern the processes of environmentally friendly refrigerants boiling and condensation in minichannels of compact heat exchangers.

2. The subject of experimental investigations

The subject of experimental studies were pipe minichannels with an internal diameter d ≤ 3.3 mm. Available for research team were minichannels with internal diameter: 0.31, 0.45, 0.64, 0.98, 1.40, 1.60, 1.94, 2.30 and 3.30 mm, made of stainless steel . Figure 1 shows views of pipe minichannels used in experimental studies.

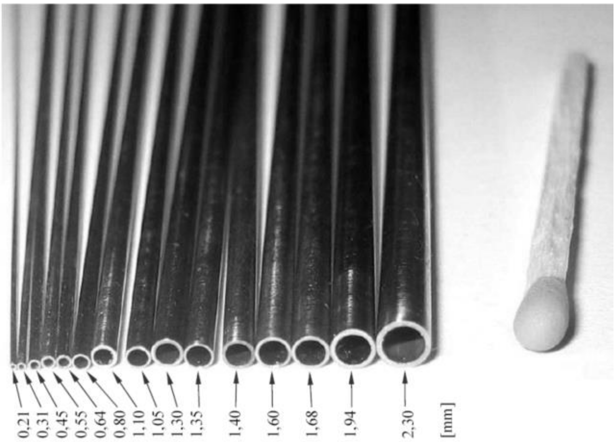


Fig. 1. View of the tubular mini-channels used for the purpose of the experimental research
Rys. 1. Widok minikanałów rurowych zastosowanych w badaniach eksperymentalnych

3. Boiling of environmentally friendly refrigerants

Experimental investigations of the environmentally friendly refrigerants R134a and R404A was carried out in the test stand in laboratory, shown in fig. 2 [9].

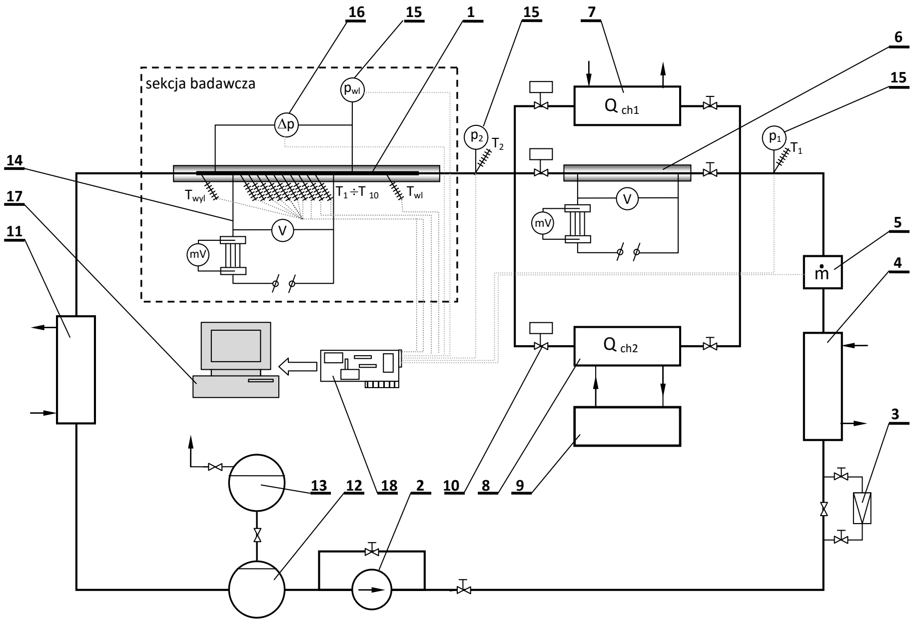


Fig. 2. Schematic diagram of testing facility; 1 – minichannel, 2 – refrigerant's pump, 3 – filter, 4 – refrigerant's precooler, 5 – flowmeter, 6 – electric heater, 7 – cooler no. 1 (cooling with water), 8 – cooler no. 2 (cooling with additional refrigerant), 9 – auxiliary refrigerating system, 10 – electromagnetic valves, 11 – condenser, 12 – tank for refrigerant liquid, 13 – supplementary tank for refrigerant, 14 – electric heating system, 15 – pressure sensor, 16 – pressure difference sensor, 17 – computer, 18 – data acquisition system

Rys. 2. Schemat ideowy stanowiska do badań eksperymentalnych wrzenia w minikanałach rurowych: 1 – minikanał rurowy, 2 – pompa czynnika chłodniczego, 3 – filtr czynnika, 4 – wstępna chłodnica czynnika, 5 – przepływomierz czynnika chłodniczego, 6 – elektryczny podgrzewacz czynnika, 7 – chłodnica nr 1 (chłodzenie wodą), 8 – chłodnica nr 2 (chłodzenie dodatkowym czynnikiem chłodniczym), 9 – pomocniczy układ chłodniczy, 10 – zawory elektromagnetyczne, 11 – skraplacz czynnika chłodniczego, 12 – zbiornik cieczy czynnika, 13 – uzupełniający zbiornik czynnika, 14 – układ ogrzewania elektrycznego odcinka pomiarowego, 15 – czujnik ciśnienia czynnika, 16 – czujnik pomiaru różnicy ciśnienia, 17 – komputer, 18 – system akwizycji danych

The study of heat transfer and flow resistance was carried out in measurement series for each minichannel diameter. The refrigerant flow rate in each series was maintained at a level such the mass flux density was comparable size. In each series of measurements increased and then reduced heat flux. The maximum value of the heat flux was q = 90 kW/m^2. As a result of calculations obtained were local values of the heat transfer coefficient and flow resistance.

Figure 3 shows, example characteristics of the boiling process according to the local heat transfer coefficient α_{exp} from local equilibrium vapor quality x. The characteristics illustrate the rapid growth of heat transfer coefficiant in the supercooled boiling ($x < 0$) and an increase in the local values of heat transfer coefficient in the only small area of the developed boiling ($x > 0$). In a further part is to achieve the maximum, after which the value of the local heat transfer coefficient is set at a certain level or decreases. The increase in heat flux dencity causes increase of the local heat transfer coefficient for the corresponding vapor quality. The resulting characteristics of the course is consistent with the characteristics offered in the literature.

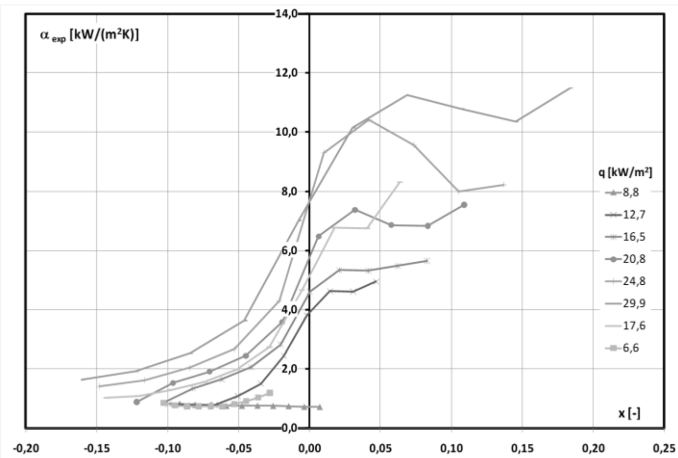


Fig. 3. Experimental dependence of local heat transfer coefficient α_{exp} on equilibrium vapour quality x; R134a, $(w\rho)$ = 436.4 kg/(m^{2}s), d = 0.80 mm
Rys. 3. Eksperymentalna zależność lokalnego współczynnika przejmowania ciepła α_{exp} od równowagowego stopnia suchości x; R134a, $(w\rho)$ = 436,4 kg/(m^{2}s), d = 0,80 mm

Figure 4 shows the characteristics on the form of pressure drop $(\Delta p/\Delta L)_{exp}$ dependence on equilibrium vapour quality x for $(\Delta p/\Delta L)_{exp} = f(x)$ for d = 0.45 mm and $(w\rho)$ = const. This figure demonstrates that when vapour quality increase the dry two-phase flow resistance increases too. For two-phase flow resistance significantly affects the value of the mass flux density. The higher flow rate of two-phase refrigerant flow causes the increase of the flow resistance value. Figure 4 shows example characteristics of the three mass flux density levels. A similar trend occurred in the other conditions. The conducted study also showed a clear effect of minichannal internal diameter on value of the two-phase flow resistance. With the increase of internal diameter flow resistance is smaller in the same conditions of experiment.

The study of the single-phase flow resistance in minichannels confirmed that the calculation of frictional resistance coefficient λ can be used correlations for conventional channels, i.e. the *Hagen-Poiseuille* formula for laminar flow and *Blasius* formula - for the turbulent flow [10, 11]. Was confirmed that the threshold value of the *Reynolds* number at which is the transition from laminar flow to turbulent flow is Re ≈ 2000. On the basis of single-phase heat transfer experiments, it was found that the value of the local *Nusselt* number for laminar flow can be correctly calculated from the dependence of *Shah* and *London*, and for turbulent flow from the correlation of *Gnielinski* for conventional channels [8].

The experimental studies results of the two-phase flow resistance in a pipe minichannels indicate that there is the possibility of using a homogenous model and classical “separated” models (*Lockhart-Martinelli* and *Friedel*) also their modifications with are recommended for use in minichannels, to calculate the two-phase flow resistance of refrigerants in minichannels. The results of heat transfer allowed for the conclusion that it is possible to calculate the heat transfer coefficient in supercooling boiling zone by using dependences used for conventional channels (eg *Shah*, *Kandlikar* and *Thome*). Ther is no way to use classic correlations and theirs modifications for minichannels to calculate the heat transfer coefficient in the expanded bubble boiling [9].

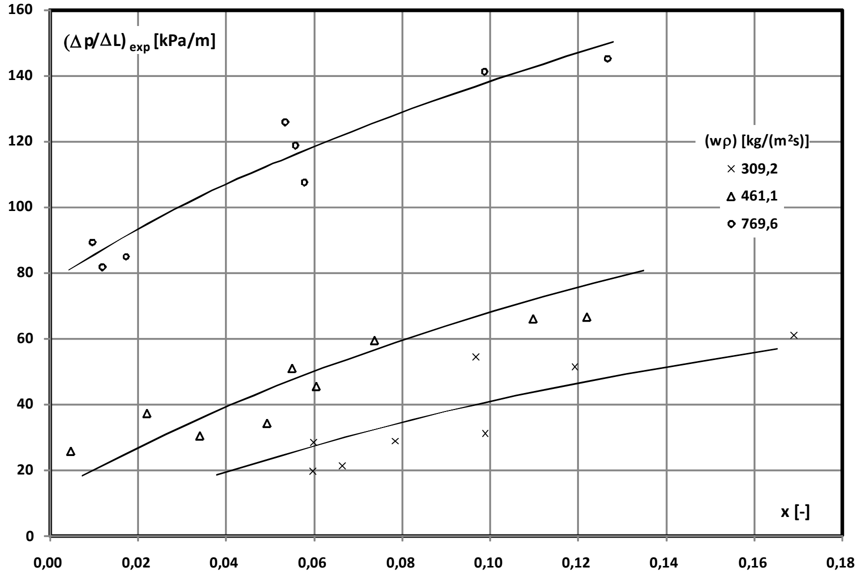


Fig. 4. Dependence of two-phase flow resistance on equilibrium vapour quality *x*; R134a, *d* = 0.45 mm, ($w\rho$) = const
Rys. 4. Zależność wartości oporu przepływu dwufazowego od równowagowego stopnia suchości *x*; R134a, *d* = 0,45 mm, ($w\rho$) = const

Own correlations to calculate the heat transfer coefficient were shown during the developed boiling and the two-phase flow resistance.

Based on the dimensional analysis the was obtained dimensional correlation to calculate the *Nusselt* number during developed bubble as:

$$Nu_{TP} = 0.41 \cdot \mathrm{Re}_l^{0.848} \cdot Bo^{0.66} \cdot Co^{-0.62} \cdot \left(\frac{\rho_l}{\rho_v} \right)^{1.28} . \tag{1}$$

Correlation (1) takes into account the range of parameters:

- heat flux density q = 0–90 000 W/m^2;
- mass flux density ($w\rho$) = 300–1400 kg/(m^{2}s);
- equilibrium vapor quality x = 0–0.2 (R404A) i x = 0–0.4 (R134a);
- the criteria number: Re_l = 550–20 000, Bo = 5·10^{-5}–7·10^{-4}, Co = 0.26–1.76;
- ratio of the refrigerant densities (ρ_l/ρ_v) = 5–30.

The proposed method for calculating the frictional component of the two-phase flow resistance based on a modification of the *Lockhart-Martinelli* method, where *Chisholm* and *Laid* proposed analytical character relations for calculation correction factors to the two-phase flow resistance – *C* [10]. Based on made dimensional analysis, in relation to their own experimental research, obtained criteria dependence as:

$$C = 150 \cdot Co^{-1.4} \cdot \left(\frac{\mathrm{Re}_l}{\mathrm{Re}_v} \right)^{1.25} \cdot \left(\frac{x}{1-x} \right)^{0.44} . \tag{2}$$

Correlation (2) was tested in the range of parameters:

– mass flux density ($w\rho$) = 300–1400 kg/(m^{2}s),
– equilibrium vapour quality x =0–0.2 (R404A) and x = 0–0,4 (R134a),
– the criteria number: Re_l= 550–20 000, Re_v= 30–13 000, Co = 0.26–1.76.

4. Condensation of environmentally friendly refrigerants

The experimental investigations of the R134a, R404A, R407C and R410A refrigerants condensation In pipe minichannels was conducted on test stand, with was describe in articles [1–3]. This papers included description of research methodology and development of research results.

The purpose of the experimental tests on the refrigerants condensation in pipe minichannels determined the local heat transfer coefficient α_x. Figure 5 presents, an example investigations results as a dependence $\alpha_x = f(x)$ during condensation in minichannels.

A comparative analysis of the results of the experimental investigations with the results of calculations according to the correlations by various authors demonstrated, both in the local heat transfer coefficient and the flow resistances, that the application of the abovementioned correlations is limited. For this reason, the authors have developed their own experimental correlation with the use of mathematical statistics principles and with a selection of the model's parameters with *quasi-Newton* and *Symplex* methods. An experimental correlation was obtained that describes the local heat transfer coefficient in the following form:

$$Nu_x = 25.084 \cdot \mathrm{Re}_l^{0.258} \cdot \mathrm{Pr}_l^{-0.495} \cdot p_r^{-0.288} \cdot \left(\frac{x}{1-x}\right)^{0.266} \tag{3}$$

hence:

$$\alpha_x = \frac{Nu_x \cdot \lambda_l}{d}, \tag{4}$$

where:

$$p_r = \frac{p_s}{p_{cr}} \tag{5}$$

is a reduced pressure, p_s is a saturation pressure, and p_{cr} is a critical pressure.

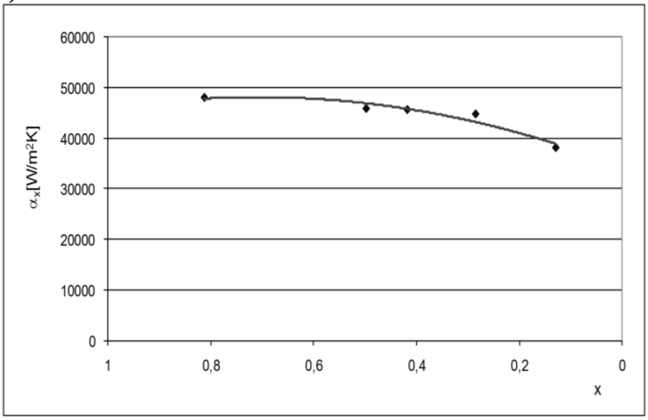
It was found that the results of the experimental tests and calculations from correlations (3) fall within the compatibility range of ± 25%. The experimental correlation developed was verified in the following range for refrigerants R134a, R404A, R407C and R410A: d = 0.31–3.30 mm; x = 1–0; T_s = 20–50°C; and $(w\rho)$ = 100–1300 kg/(m^2·s).

Figure 6 shows an example of experimental results as a dependence of the local pressure drop $(\Delta p/L)_x$ on the vapor quality x in the condensation of the R407C refrigerant for fixed values of the mass flux density $(w\rho)$ = const, in the interval $(w\rho)$ = 469–1336 kg/(m^2·s) in pipe minichannel with internal diameter d = 1.40 mm and d = 1.92 mm in the interval $(w\rho)$ = 317–1336 kg/(m^2·s).

Similar results were obtained for other internal diameters of minichannels. The study shows that the diagram of $(\Delta p/L)_x = f(x)$ for $(w\rho)$ = const has a characteristic change in the course of the vapour quality x = 1–0. With a decrease of the vapour quality x is initially a slight increase in the value of local resistance, and then the decrease. The increase in mass flux density $(w\rho)$ causes an increase in local flow resistance of the condensing refrigerant.

Using the results of experimental investigations own correlation was checked allows to specify value of the frictional pressure drop for refrigerant R134a, which is shown in paper [2]. It was found that it can be used with good results also for condensation R404A, R407C and R410A refrigerants in minichannels.

a)



b)

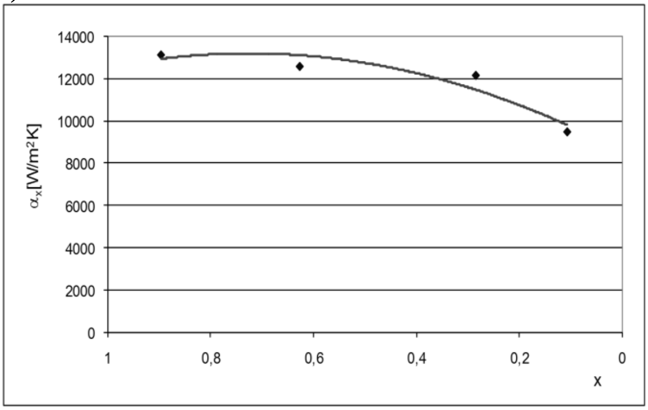


Fig. 5. Experimental results as dependence of α_x = f(x) during condensation of R407C refrigerant in pipe minichannels with internal diameter: a) d = 0.31 mm, b) d = 0.98 mm
Rys. 5. Wyniki badań eksperymentalnych zależności α_x = f(x) podczas skraplania czynnika chłodniczego R407C w minikanale rurowym o średnicy wewnętrznej: a) d = 0,31 mm, b) d = 0,98 mm

The verifying calculations of the results of the experimental investigations demonstrated that in the range covered by an analysis of this correlations there occurred the following two-phase flow structures for both refrigerants: annular and annular-stratified structures. It was found that the results of the experimental tests and calculations from correlations (6) fall within the compatibility range of ± 20%.

a) b)

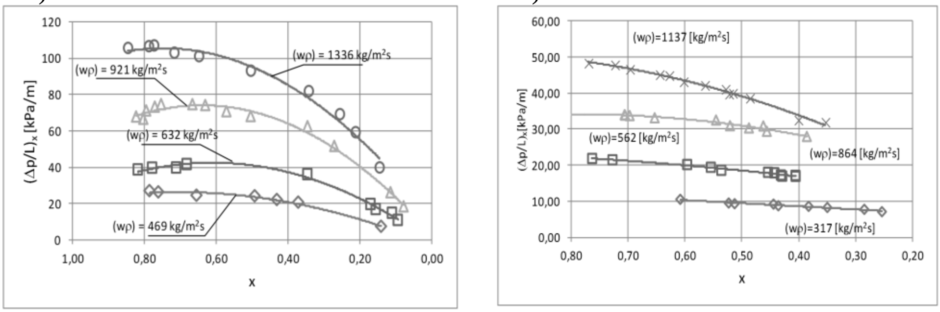


Fig. 6. The results of the experimental characteristics $(\Delta p/L)_x = f(x)$ during the refrigerant R407C condensation in minichannels with internal diameter:
a) d = 1.40 mm, b) d = 1.92 mm

Rys. 6. Wyniki badań eksperymentalnych charakterystyki $(\Delta p/L)_x = f(x)$ podczas skraplania czynnika R407C w minikanałach o średnicy;
a) d = 1,40 mm; b) d =1,92 mm

5. Experimental investigations of compact heat exchangers

The subject of experimental research were two bundles of pipe minichannels (multiports) of the design shown in fig. 7. Bundles of pipe minichannels called MULTI-4 consisted of four minichanels made from stainless steel with an internal diameter d = 0.64 mm and length L = 100 mm. In the case of tubular bundle MULTI-8 was consisted of eight pipe minichannels with the same internal diameter and length.

a) b)

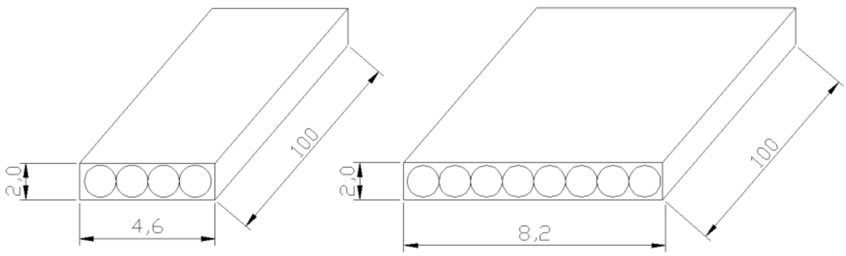


Fig 7. Dimensional diagram of tested minichannels bundles: a) MULTI-4, b) MULTI-8

Rys. 7. Schemat wymiarowy badanych pęczków minikanałów: a) MULTI-4, b) MULTI-8

Fig. 8 to 10 shows the dependence of the average heat transfer coefficient α_A and pressure drop $(\Delta p/L)_A$ characteristics in the two versions of interpretation, namely the mass flux density $(w\rho)$, with x_A = const and the x_A, at $(w\rho)$ = const.

a) b)

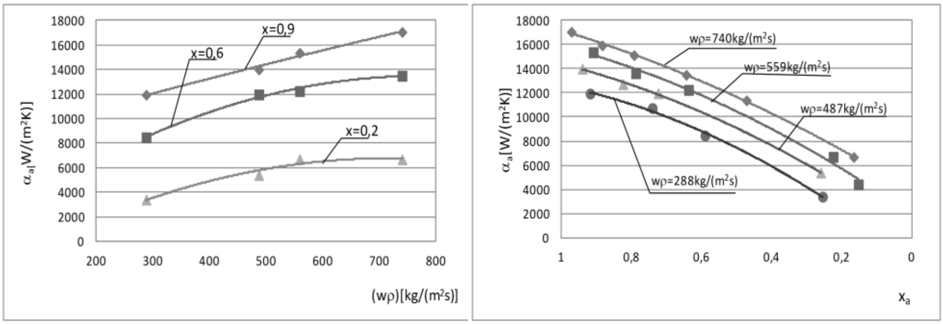


Fig. 8. Experimental results according to the average heat transfer coefficient αA from: a) mass flux density (wρ) for xA = const, b) the average vapour quality xA, for (wρ) = const; MULTI-4, the refrigerant R407C
Rys. 8. Wyniki badań eksperymentalnych zależności średniego współczynnika przejmowania ciepła α_A od: a) gęstości strumienia masy czynnika $(w\rho)$, dla x_A = const; b) średniego stopnia suchości x_A, dla $(w\rho)$ = const.; MULTI-4, czynnik chłodniczy R407C

The thermal characteristics of a condensation $\alpha_A = f(w\rho)$, for a constant level of the average vapour quality x_A = const showed a clear increase in the average heat transfer coefficient α_A, with an increase in the mass flux density $(w\rho)$ of refrigerants. Based on diagrams showed characteristics of $\alpha_A = f(x_A)$, with $(w\rho)$ = const it appears that when decrease the value of α_A coefficient, the average vapour quality x_A decrease too.

The course nature of these thermal characteristics and the characteristics of flow resistance under flow averaged conditions is similar to the characteristics obtained for the condensation process in a single minichannel [5]. The results show that for R134a refrigerant achieved the highest values of average heat transfer coefficient α_A (and pressure drop $(\Delta p/L)_A$)). Values of the heat transfer coefficient α and the flow resistance $(\Delta p/L)$ are higher for the R407C refrigerant then for R404A, and the increase is in the range of 10 to 15%.

a) b)

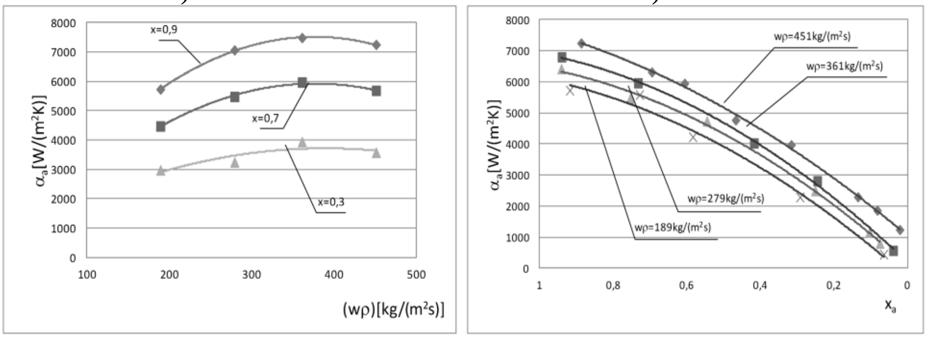


Fig. 9. Experimental results dependence of the average heat transfer coefficient αA on: a) mass flux density (wρ) for xA = const, b) the average vapour quality xA, for (wρ) = const, MULTI-8, R134a

Rys. 9. Wyniki badań eksperymentalnych zależności średniego współczynnika przejmowania ciepła α_A od: a) gęstości strumienia masy czynnika (*wρ),* dla x_A = const; b) średniego stopnia suchości x_A, dla (*wρ*) = const, MULTI-8, czynnik R134a

a) b)

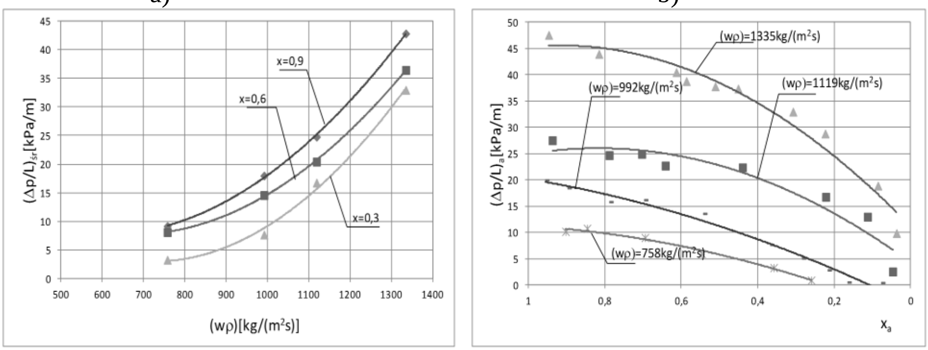


Fig. 10. Experimental results dependence of the flow resistance (Δp/L)A of: a) mass flux density (wρ) for xA = const, b) the average vapour quality xA, for (wρ) = const.; MULTI-4, R404A

Rys. 10. Wyniki badań eksperymentalnych zależności oporu przepływu $(\Delta p/L)_A$ od: a) gęstości strumienia masy czynnika (*wρ*), dla x_A = const; b) średniego stopnia suchości x_A, dla (*wρ*) = const; MULTI-4, czynnik chłodniczy R404A

6. Instability of phase changes

Work of devices, conventional refrigeration equipment and compact heat exchangers (with use of mini-or microchannels) should be stable and secure. In the experimental conditions for devices working can affect disorders that cause various types of instability, which may occasionally result in a transition to the area of non-equilibrium states. This is a very important issue in the two-phase flow, because of disorder and instability propagated in them have wave properties. This also applies to phase transitions of refrigerants.

In the Department of Heat and Refrigeration Koszalin University of Technology conducted are research of the instability in two phase flow, especially during boiling and condensation of refrigerants in conventional channels and minichannels. Number of published papers including the problem of instability in phase transitions in minichannels is according to the current state of knowledge is relatively small. Therefore, the aim of research is to identify the problem and determine whether the phenomenon of the spread of instability in minichannels during condensation can be comparable to those occur in the convencional channels. Below are showed some results instability during condensation of environmental friendly refrigerants in pipe minichannels. Specified speed of movement disorders in the condensing refrigerant, and an assessment of the causes and consequences of the instability caused by the internal and external interaction [12]. Experimental research was carried out in the laboratory, which is described in papers [1, 2, 3]. External interference of a periodic character transferred to the process of refrigerant condensation process in pipe minichannel. This disturbances formed by interaction of valve on refrigerant flow. The consequence of these disturbances was the mass flow pulsation and resulting saturation pressure p_s pulsations and heated wall temperature T_w of the tested channel. The experiments result analysis showed that the generated disturbances with were periodicals have a significant effect on the change of the refrigerant condensing pressure during the flow in the pipe minichannel. There was a delay in registering the refrigerant pressure p_s by presser sensors along the road at the measuring section. This proves the finite speed v_p with which transit the signal of the pressure changes after open or close the valve [13].

The dependence of the propagation disturbance speed on the frequency and minichannel internal diameter has been shown (Fig. 11).

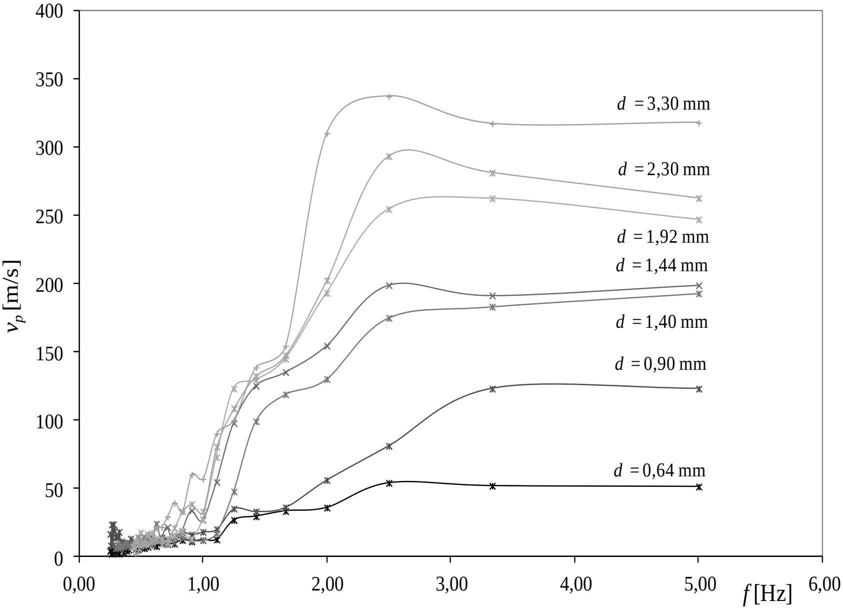


Fig. 11. Experimental dependant on the speed v_p of movement pressure instability of the periodic distribution generation during refrigerant R134a condensation in pipe minichannels of internal diameter d = 0.64–3.30 mm
Rys. 11. Eksperymentalna zależności prędkość v_p przemieszczania się niestabilności ciśnieniowych od częstotliwości generowania zakłóceń periodycznych podczas skraplania czynnika chłodniczego R134a w minikanałach rurowych o średnicy wewnętrznej d = 0,64–3,30 mm

Based on experimental studies, it was found that in contrast to the conventional channels, the velocity spread of the limit pressure disturbances movement v_p in minichannels is obtained at much lower frequencies f of hey generation [14]. With a decrease in internal diameter d of minichannel occurs a reduction in disturbance propagation of pressure velocity v_p. The real speed of the pressure change signal also depends on the void fraction ϕ, what is shown in Figure 12.

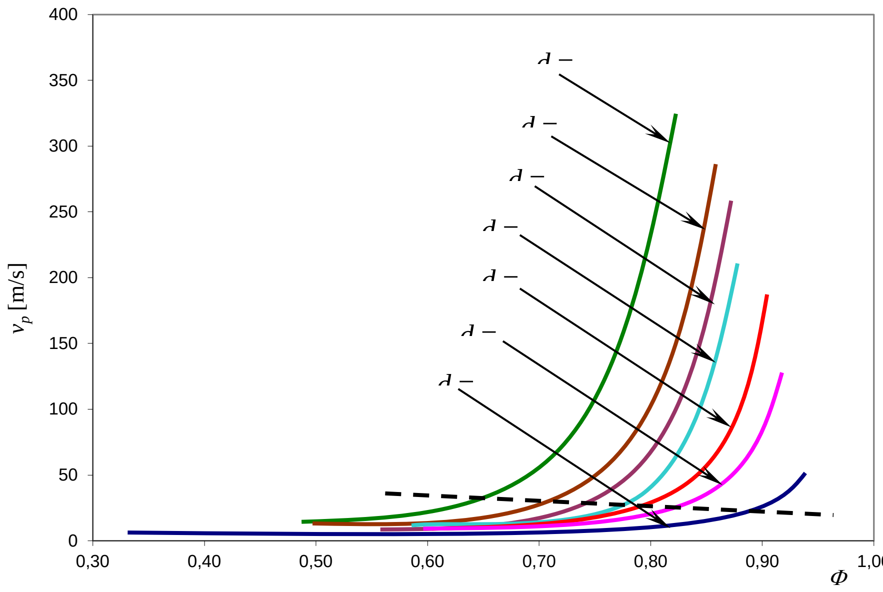


Fig. 12. Summary of experimental results characteristics $v_p = f\,(\Phi)$ during condensation of refrigerant R134a in pipe minichannels with internal diameter d = 0.64, 0.90, 1.40, 1.44, 1.92 and 3.30 mm in a periodic instability
Rys. 12. Zestawienie wyników badań eksperymentalnych charakterystyki $v_p = f(\Phi)$ podczas skraplania czynnika chłodniczego R134a w minikanałach rurowych o średnicy wewnętrznej d = 0,64; 0,90; 1,40; 1,44; 1,92 i 3,30 mm w warunkach niestabilności periodycznych

7. Summary

The investigations results of environmental friendly refrigerants (R134a, R404A, R410A and R407C) phase changes in pipe minichannels with an internal diameter d = 0.31–3.3 mm. For boiling and condensation process determined were values of heat transfer coefficient and flow resistance. The effect of the vapour quality, mass flux density, internal diameter on heat transfer intensity was illustrated. Developed own dependence allowing to calculate value of heat transfer coefficient and flow resistances during phase changing of environmental friendly refrigerants in pipe minichannels.

This paper presents the thermal investigations results of the average heat transfer coefficient in multiports from: four and eight channels with internal diameter d = 0.64 mm parallel powered by refrigerants R134a, R404A and R407C. Developed characteristics provide thermal dependence of the average heat transfer coefficient α_A on the mass flux density ($w\rho$) and the vapour quality x_A for these refrigerants. It has been shown that the highest value of α_A were obtained for R134a refrigerant.

The research of refrigerants condensation in external disturbances conditions occurring in cooling devices were conducted. Development or disappearance of condensation induced by changing the parameters of a two-phase flow, such as pressure and mass flux density in the tubular channel. It was confirmed that the two-phase flows exhibit wave and induced disturbance moving with finite speed.

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Nomenclature:

Bo	–	Boiling number
C	–	coefficient, formula (2)
Co	–	coefficient, formula (1 and 2)
d	–	inner diameter, mm
Nu	–	*Nusselt* number
p	–	pressure, Pa
$\Delta p/\Delta l$	–	pressure drop, kPa/m
Pr	–	*Prandtl* number
$\dot{q}$	–	heat flux density, W/m^2
Re	–	*Reynolds* number
T	–	temperature, ^{o}C
$(w\rho)$	–	mass flux density, kg/(m^2·s)

v	–	wave velocity, m/s
x	–	quality,

Greek letters

α	–	heat transfer coefficient, W/(m^2·K)
ϕ	–	void fraction
λ	–	conductivity, W/(m·K)
ρ	–	density
ΔT	–	differential temperature, K

Subscripts relate to the following:

A	-	average value
cr	–	critical value,
l	–	liquid,
p	–	pressure,
r	–	Reduced value,
s	-	saturation,
TP		two phase flow
w	-	wall,
x	-	local value,
v	-	vapour,

Badanie przemian fazowych proekologicznych czynników chłodniczych w minikanałach

Streszczenie

Przedstawiono wyniki badań przemian fazowych proekologicznych czynników chłodniczych R134a, R404A, R410A i R407C w minikanałach rurowych o średnicy wewnętrznej d = 0,31–3,3 mm. Dla procesu wrzenia i skraplania wyznaczono wartości współczynnika przejmowania ciepła i oporów przepływu. Zilustrowano wpływ stopnia suchości pary czynnika, gęstości strumienia masy oraz średnicy wewnętrznej minikanału na intensywność wymiany ciepła. Opracowano nowe zależności pozwalające wyznaczać wartości współczynnika przejmowania ciepła i opory przepływu podczas przemian fazowych proekologicznych czynników chłodniczych w minikanałach rurowych.

W opracowaniu przedstawiono wyniki badań cieplnych średniego współczynnika przejmowania ciepła w multiportach cztero- i ośmiokanałowym

o średnicy wewnętrznej d = 0,64 mm zasilanych równolegle czynnikami chłodniczymi R134a, R404A i R407C. Opracowane charakterystyki cieplne podają zależność średniego współczynnika przejmowania ciepła α_A od gęstości strumienia masy ($w\rho$) i stopnia suchości x_A dla wymienionych czynników. Wykazano, że najwyższe wartości α_A uzyskano dla czynnika R134a.

Przeprowadzono badania skraplania czynników chłodniczych w warunkach zaburzeń zewnętrznych występujących w urządzeniach chłodniczych. Rozwój lub zanik skraplania wywoływano poprzez zmianę parametrów układu dwufazowego, np. ciśnienia i gęstości strumienia masy w kanale rurowym. Potwierdzono, że ośrodki dwufazowe wykazują własności falowe a wywoływane zaburzenia przemieszczają się ze skończoną prędkością.