

Heat transfer during condensation of refrigerants in tubular minichannels

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Abstract The present paper describes the results of experimental investigations of heat transfer during condensation of R134a, R404A and R407C in pipe minichannels with internal diameters 0.31–3.30 mm. The results concern investigations of the local heat transfer coefficient. The results were compared with the correlations proposed by other authors. Within the range of examined parameters of the condensation process in minichannels made of stainless steel, it was established that the values of the heat transfer coefficient may be described with Akers *et al.*, Mikielewicz and Shah correlations within a limited range of the mass flux density of the refrigerant and the minichannel diameter. On the basis of experimental investigations, the authors proposed their own correlation for the calculation of local heat transfer coefficient.

Keywords: Condensation in minichannels; Heat transfer; Compact condenser; Refrigeration

Nomenclature

- A – heat transfer area, m²
- d – minichannel inner diameter, m
- d_h – hydraulic diameter, m
- G – mass flux density, kg/m²s
- j_v – dimensionless vapour velocity

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L	–	length, m
\dot{m}	–	mass flow rate, kg/s
p	–	pressure, kPa
Δp	–	pressure drop, kPa
Q	–	heat flux, W
q	–	heat flux density, W/m ²
S	–	slip
T	–	temperature, °C
x	–	vapor quality
Nu	–	Nusselt number
Pr	–	Prandtl number
Re	–	Reynolds number

Greek symbols

α	–	heat transfer coefficient, W/m ² K
λ	–	thermal conductivity coefficient of refrigerants, W/mK
ρ	–	density, kg/m ³
μ	–	dynamic viscosity, kg/ms
χ_{tt}	–	dimensionless two-phase flow multiplier Martinelli

Subscripts

a	–	averaging conditions
cr	–	critical value
exp	–	experimental value
H_2O	–	water
i	–	consecutive section ($i = 1, 2, \dots, 9$)
l	–	liquid
r	–	reduced pressure
s	–	saturation value
th	–	theoretical value
w	–	wall
v	–	vapor
x	–	local value

1 Introduction

The turn of the twenty-first century saw an abrupt technological advance that was evident in the qualitative and quantitative growth in the production of space technology and electronics devices. These two technologies are inseparable, and they mark a trend toward the miniaturisation of such devices. It is worth noting that the absolute value of thermal power in computer systems is not significantly large, while the density of heat fluxes, i.e., the amount of heat transferred by a field unit of the heat transfer area, reaches substantial values, even as much as 1000 W/cm², as indicated by

Baumner *et al.* [5]. The traditional methods for transferring such heat flux densities are either minimally useful or not useful at all. A paper by Obhan and Garimella [24] discusses methods presently used and those recommended for the future. In view of the above, the use of two-phase media becomes a priority. The practical realisation of these media involves implementation of the intensification methods of convective heat transfer with phase change. One of the passive methods used to intensify the convective heat transfer process includes, among others, a reduction of the diameter of those channels through which the agent that realises the heat transfer process flows. Increase in the value of heat transfer coefficients serves as a measure of the effectiveness of this intensification. The following criteria need to be fulfilled for the construction of modern refrigeration and air conditioning heat exchangers: small dimensions, high efficiency of heat transfer and the environmentally friendly technology.

For this purpose, the concept of so-called compact heat exchangers (including refrigeration evaporators and condensers) was proposed. In these compact heat exchangers, the ratio of the heat transfer area to the overall volume is greater than $700 \text{ m}^2/\text{m}^3$, Shah [26]. The fulfilment of this criterion requires the use of channels with hydraulic diameters $d_h = 1\text{--}6 \text{ mm}$. Such a criterion was proposed by Mehendale *et al.* [20]. At present, the classification proposed by Kandlikar [17] is commonly used. This classification specifies in detail the measurements of channels in the construction of heat exchangers. According to this classification, the following can be used in compact exchangers: for microchannels, $d_h = 10\text{--}200 \mu\text{m}$; for minichannels, $d_h = 200 \mu\text{m} - 3 \text{ mm}$. In comparison, conventional channels have hydraulic diameters $d_h \geq 3 \text{ mm}$.

Recent years have seen an increased interest and a growing number of papers about heat and flow processes in compact exchangers where channels are used with dimensions smaller than conventional ones, Thome [32], Kandlikar *et al.* [18], Giaasian [15], Mikielewicz [22]. Also, the number of studies concerning heat transfer and flow resistance in channels with small diameters is growing as is the number of studies that deal with the condensation process. One of the reasons for this increased interest might be that, initially, the boiling and condensation phenomena in these types of channels were treated as being symmetrical to each other, which, even concerning the flow in conventional channels, proved to be too great of a simplification, Mikielewicz [21]. This problem was confirmed in papers by Thome *et al.* [33,34], which presented a new heat transfer model in horizontal pipes appli-

cable to hydraulic diameters $d_h = 3.1\text{--}21.4$ mm, and in Cavallini *et al.* [9], who dealt with heat transfer in minichannels. Many authors warn against transferring to micro- and minichannels those correlations that have been specified and verified for the flow with the phase change in conventional channels. However, it is also to be noted that the correlations developed for low- or medium-pressure refrigerants (e.g. R134a) do not have to be appropriate for high-pressure refrigerants such as R404A and R410A.

Designing compact heat exchangers is also made difficult because, according to the present state of knowledge, there is no unequivocal indication of a unified model of the condensation of refrigerants in micro- and minichannels. Investigations concerning the realisation of this postulate are being conducted in many countries, while an introductory attempt to summarise these was provided, among others, in papers by Sun and Mishima [27] and Garcia-Cascales *et al.* [14].

This study presents research problems that occur in compact condensers with refrigerant R134a. Homogenous refrigerant R134a belongs to the hydrofluorocarbon (HFC) group and is one of the substitutes of R12, while the near-azeotropic mixture (composed of three refrigerants from the HFC group: R134a, R125 and R134a) is a substitute for R22. On 1 January 2010, R22 freon was withdrawn from active application in refrigeration technology (it is still possible to service devices that were designed prior to that date).

2 Selection of the proper correlation for the calculation of heat transfer coefficients

When designing a compact condenser composed of minichannels, the same engineering problems need to be solved as in a conventional condenser. This task involves the calculation of the required available heat exchange area and the driving power of the movement generators of those agents that realise the heat exchange process. To this end, it is necessary to have access to information concerning the establishment of heat transfer coefficients from the side of the refrigerants, i.e. the condensing and refrigerating agents, as well as the flow resistances of both these agents. The determination of these values for the refrigerant is relatively simple, but for the phase change during condensation, especially in the flow in a minichannel, it is difficult [7].

In papers concerning condensation in minichannels, the following issues of heat exchange and flow resistance are addressed:

1. The flow takes place in a single smooth pipe minichannel with a circular or noncircular section.
2. The flow occurs in a multichannel system, which is composed of several to several dozen parallel channels that are fed at the same time.
3. The flow takes place in channels with internal fins and with a small diameter (e.g., with microfins on the internal surface) in systems with single channels and with many channels.
4. The structure of the two-phase flow dominating the condensation process in the minichannel must be determined.

The issues presented above do not exhaust all the problems, including the problems of great importance, such as the impact of the shape of the section of the hydraulic diameter of the channel or the refrigerant type. Computational correlations are generally proposed in one of two forms:

1. An adaptation of those correlations that were verified for conventional channels in relation to minichannels.
2. Proposals of new correlations that are based on investigations (theoretical or experimental) conducted with the use of minichannels.

The following correlations are useful in the calculations of heat exchange during condensation in conventional channels and that can be adopted for minichannels: Akers *et al.* [2,3], Dobson and Chato [13], Shah [26], Cavallini and Zecchin [10], Tang [31], Thome *et al.* [33,34].

At the same time, a growing number of papers deal with the results of investigations into heat exchange in mini- and microchannels. The research conducted by Yang and Webb [35] and Yan and Lin [35] confirmed that in comparable ranges of condensation parameters, the values of heat transfer coefficient in minichannels are higher, by at least 10%, than the values obtained in conventional channels. In addition, the decrease of minichannel diameter leads to increase of the heat transfer coefficient, which is usually accompanied by the increase of flow resistance.

According to the present trends, Garcia-Cascales *et al.* [14], the results of these investigations are particularly important for the construction of computational correlations for the determination of the values of heat transfer coefficient, α , where the influence of surface tension in minichannels and the capillarity effect in micro- and nanochannels is taken into account. In this area, recent studies by Tabatabai and Faghri [29] as well as Akbar *et al.* [1] are noteworthy. Papers by Yang and Webb [36], Bandhauer *et al.* [4]

and Cavallini *et al.* [9] are also of particular significance. On the basis of an extended database of various researchers, Garcia-Cascales *et al.* [14], that are presently interesting and worth recommending for the calculation of the heat transfer coefficient, were developed by Webb *et al.* [38], Wang *et al.* [37], and Koyama *et al.* [19]. Previously developed correlations proposed by Akers *et al.* [2,3] and Shah [26] can also be used in suitably defined ranges. The abovementioned correlations were experimentally tested as follows:

- correlation due to Webb *et al.* [38]: investigations in channels with hydraulic diameter $d_h = 1.56$ mm and 1.41 mm (R12, R134a), saturation temperature $T_s = 65$ °C, and mass flux density $400 < G < 1.400$ kg/(m²s);
- correlation by Wang *et al.* [37]: investigations in a rectangular channel $d_h = 1.46$ mm (R134a), $T_s = 61.5$ – 66 °C, and $75 < G < 750$ kg/(m²s);
- correlation by Koyama *et al.* [19]: investigations in a multi-port type channel (R134a), $d_h = 0.81$ and 0.89 mm, $T_s = 60$ °C, and $100 < G < 1300$ kg/(m²s);
- correlation by Cavallini *et al.* [9]: investigations in a multiport type channel (R134a, R410A), $T_s = 40$ °C, and $200 < G < 1000$ kg/(m²s); correlation by Bandhauer *et al.* [4]: $d_h = 0.4$ – 4.9 mm (R134a) and $150 < G < 750$ kg/(m²s).

Apart from the correlations by Akers *et al.* [2, 3] and Shah [26], it also possible to use ‘older’ correlations, including those due to Dobson and Chato [13], Tandon *et al.* [30], Moser *et al.* [23], Tang [31] and Wilson *et al.* [39]. When making use of any correlation concerning condensation in minichannels, one needs always to refer to the source material of its origin and to compare the process parameters and the range of its verification.

Sometimes researchers have contradictory views concerning the same issue, such as the application of a given correlation in the calculation of the heat transfer coefficient. This problem will be presented in an example of the use of the correlation by Shah [26] and Akers [2,3]. In the paper by Webb *et al.* [38], the results of the authors’ own research were compared with the calculations according to the abovementioned correlations. When taking into account the dependence of coefficient α from the mass flux density, G , it was established that the values of this coefficient calculated from Shah’s correlation were too high, while those from the correlation by Akers *et al.* [2,3] were too low in relation to the experimental data. A study by Chang and Wang [12] confirmed the usefulness of Shah’s [26] correlation

in the calculation of coefficient α in minichannels, with low values of quality x (however, this is contradictory to the conclusions provided in the paper by Webb *et al.* [38]). At the same time, the research by Yan and Lin [35] confirms the observations made by Webb *et al.* [38]. Numerous authors have remarked that the results of condensation in minichannels are to be treated with a special care for small values of mass flux densities $G < 70 \text{ kg}/(\text{m}^2\text{s})$ and quality $x < 0.5$.

3 Experimental investigations

3.1 Subject of the investigations

Pipe minichannels with internal diameters $d = 0.31\text{--}3.30 \text{ mm}$ constituted the subject of investigations. Stainless steel minichannels with internal diameters $d = 0.31, 0.45, 0.64, 0.98, 1.40, 1.60, 1.94, 2.30$ and 3.30 mm were used. The sections of these minichannels, with a total length of 1000 mm (and measuring length $L = 950 \text{ mm}$), were alternately assembled at the experimental facility.

3.2 Experimental facility and methodology of investigations

Figure 1 presents a diagram of the experimental facility, while Fig. 2 provides its overall view. Section 1 of the pipe minichannel was the basic element of the experimental facility. This section was placed in the horizontal axis of water channel 2, which was made from aluminium with a rectangular cross-section and internal dimensions of $28 \times 24 \text{ mm}$. The refrigerant was supplied to measuring section 1 from the side of pumping compressor 3 of the compressor installation. Before the refrigerant reached the inlet section of the minichannel, its superheated vapour flew through the tubular type exchanger in pipe 10, which was chilled with water. The use of this exchanger allowed not only the collection of superheat but also the preparation of the refrigerant's state in the form of a dry, saturated vapour with a quality $x = 1$ (or near this state). After the condensation of the refrigerant's vapour in a flow through a pipe minichannel, the refrigerant's liquid flew to subcooler 11 on the outlet, from which the refrigerant's flow rate was measured with the aid of a Coriolis type flowmeter 15. The refrigerant's flow rate was measured for control purposes with the aid of a system of calibrated vessels. Then, the refrigerant returned to the refrigeration system that was fed from unit 3 with a condenser with chilled air 4

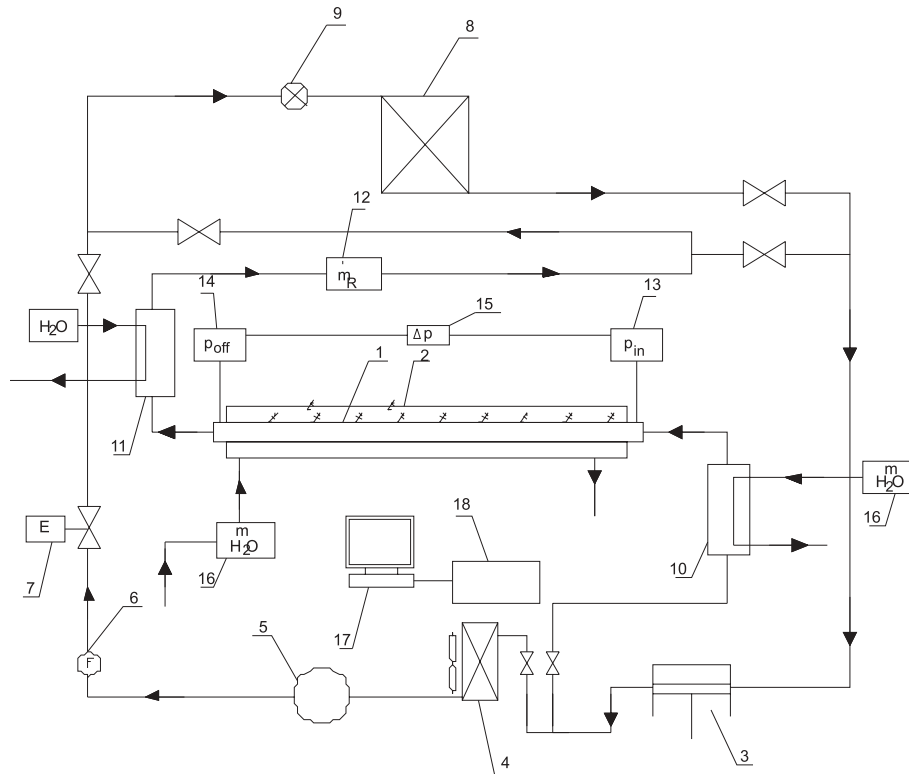


Figure 1. Schematic diagram of the experimental facility: 1 – measuring section of the pipe minichannel, 2 – water channel, 3 – refrigeration compressor installation, 4 – condenser chilled with air, 5 – container of refrigerant liquid, 6 – filter-dryer of refrigerant, 7 – electromagnetic valve, 8 – lamelled air cooler, 9 – expansion valve that feeds the cooler, 10 – heat exchanger to collect refrigerant’s superheat, 11 – subcooler of refrigerant’s liquid, 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.

and a fan cooler 8.

Figure 3 presents a diagram of measuring section 1 from Fig. 1 with its instrumentation, and Fig. 4 gives its view. Along the length $L = 950$ mm of this minichannel section, 9 thermocouples of the K type were installed that were arranged in even distances for the purpose of measuring the external temperature of the minichannel surface. In each of these nine measuring sections, the temperature of the chilling water that flew into water chan-

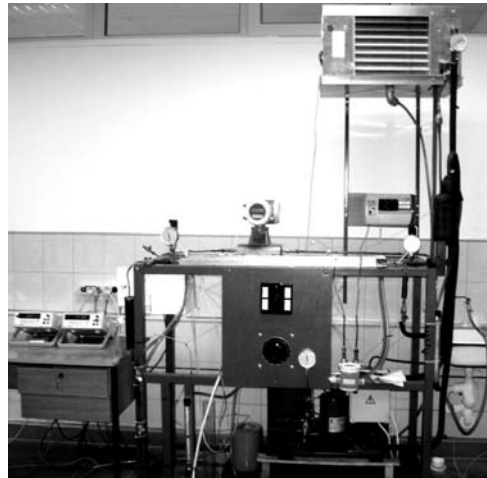


Figure 2. Overall view of the experimental facility.

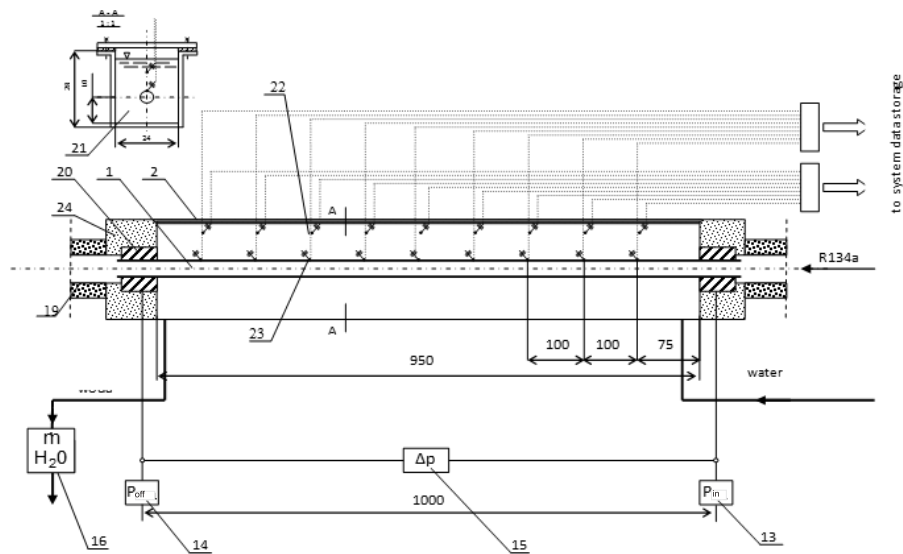


Figure 3. Schematic diagram of the measuring section of the pipe minichannel; 19 – copper pipe $\phi 8/\phi 10$ mm, 20 – pipe connector, 21 – pipe minichannel refrigeration system, 22 – connector of thermocouple wires of water temperature thermocouples, 23 – connector of thermocouple wires of minichannel surface thermocouples, 24 – isolation; (the remaining denotations are as in Fig. 1).

nel 2 from Fig. 1 was measured. The thermocouple welds for the water temperature measurement were placed 19 mm from the bottom of water channel 2 in its vertical axis. The thermocouple wires of the temperature sensors were led to connectors 22 and 23 and to data acquisition system 18. Before installation, all of the thermocouples were recalibrated in relation to the standard glass thermometer with a 0.05 °C scale interval. Individual thermoelectric characteristics were made. The pressure of the refrigerants at the inlet to the measuring section was measured with a piezoresistant sensor with a transducer of the class 0.4, while the pressure drop was measured with the aid of a differential pressure sensor with a transducer of the same class.

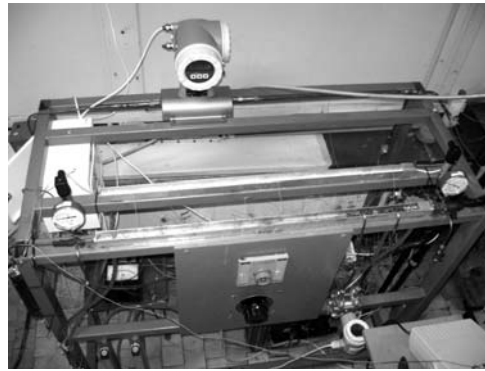


Figure 4. Overall view of measuring section.

Once the tests have been conducted with the pipe minichannel of internal diameter d , measurements were conducted for minichannels with other diameters. Electronic flowmeter 16 was used to measure the chilling water flow rate.

The characteristic experimental values of the condensation heat transfer coefficient of R134a, R404A or R407C in a pipe minichannel were determined using the developed procedures. The following were measured directly at the experimental facility: the temperature of the external surface of the minichannel wall in the i -th section, T_{zi} ($i = 1, 2, \dots, 9$); the flow rate of the refrigerant and its temperature in the inlet and outlet sections of the minichannel; the mass flow rate of chilling water and its temperature in nine sections; the pressure of the refrigerant in these sections; and the pressure drop between the inlet section to the minichannel and a given section.

At a separate measuring facility, tests were conducted of heat flux, Q , which was carried away in the condensation process of the refrigerant on the length $L = 950$ mm of the pipe minichannel and its values in each of the 9 sections. For a comparative method of heat flux measurement, an idea proposed by Shin and Kim [27] was used. For this purpose, nine sections were made of minichannels (with different internal diameters d), each of length L . Thermocouples were installed in the individual sections. At a separate facility, the minichannel sections were switched on interchangeably as resistors in an electric circuit. The electric current power was measured, which, after taking into account the heat losses to the environment, enabled determination of heat flux Q (during the measurement, the refrigerant did not flow inside the minichannel). By referring heat flux, to the internal surface of the minichannel, heat flux density, q , ($q = Q/\pi dL$) was determined. Individual characteristics $q = f(T_{wi} - T_{H_2O,i})$ were made for each measuring section of the minichannel. The temperature of the internal surface of the minichannel in the i -th section was denoted as T_{wi} . The value of this temperature was determined by knowing the measured value T_{zi} of the external surface and the temperature drop as a result of heat conduction on the thickness of the minichannel wall. The local heat transfer coefficient was calculated from the following dependence: where is the saturation temperature in the i -th section. The dryness factor x of the refrigerant's vapour was determined from the energy balance of the heat exchanger denoted with the symbol 10 in Fig. 1. In the calculation of x , the dependence was used that describes the slip between phases S according to Huhn's correlation [16].

4 Experimental results

The purpose of experimental tests on the condensation of refrigerants in pipe minichannels with internal diameters $d = 0.31$ – 3.30 mm was to determine the local heat transfer coefficient α_x . Figure 5 presents, as an example of the results on the investigations of the local heat transfer coefficient the dependence with dryness factor, $\alpha_x = f(x)$, during condensation in minichannels with the diameters of 0.31 mm. Figure 6 presents the test results for minichannels with internal diameters $d = 0.98$ – 3.30 mm.

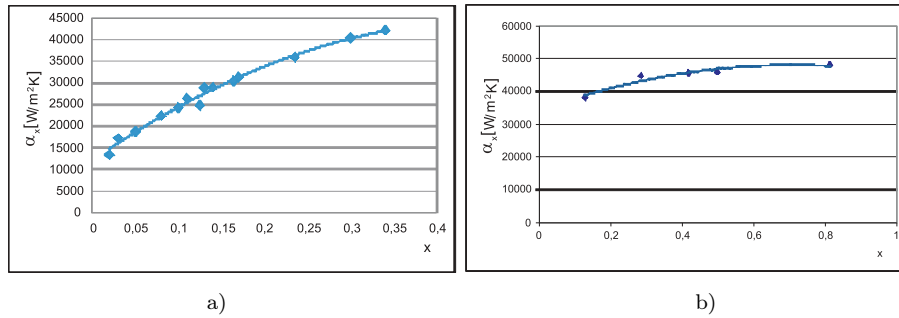


Figure 5. Results of experimental tests of the local heat transfer coefficient during the condensation in pipe minichannels with the diameters of 0.31 mm: a) R134a, b) R404A.

4.1 Results of the analysis

When comparing the results of the authors' own investigations with the results of tests conducted by others, the criteria provided by Cavallini *et al.* [11] were used. They allow an assessment of the limits of the two-phase flow structures of the condensation of refrigerants in channels with the aid of dimensionless vapour velocity, j_v , and χ_{tt} , which is the dimensionless two-phase flow multiplier due to Martinelli, as follows:

$$j_v = \frac{Gx}{\sqrt{gd\rho_v(\rho_l - \rho_v)}}, \quad (1)$$

$$\chi_{tt} = \left(\frac{1-x}{x}\right)^{0.9} \left(\frac{\rho_v}{\rho_l}\right)^{0.5} \left(\frac{\mu_l}{\mu_v}\right)^{0.1}, \quad (2)$$

where G is the mass flux density, g is the gravitational acceleration, ρ is the density, μ is the kinematic viscosity, x is the quality, and the subscripts l and v refer to liquid and vapour phase, respectively.

In most of the cases, the measurements conducted by the authors were realised for the flow parameters $j_v \geq 2.5$ and $\chi_{tt} < 1.6$, which involves the possibility of the occurrence of annular or annular and stratified flow structures. Figure 7 presents a comparison of the results of experimental investigations concerning the heat transfer coefficient with the results of calculations from the classical calculations used for conventional channels. The following correlations were taken into account: Dobson and Chato [13], Cavallini and Zecchin [10], Akers *et al.* [2,3], Shah [26], Mikielewicz [25] and

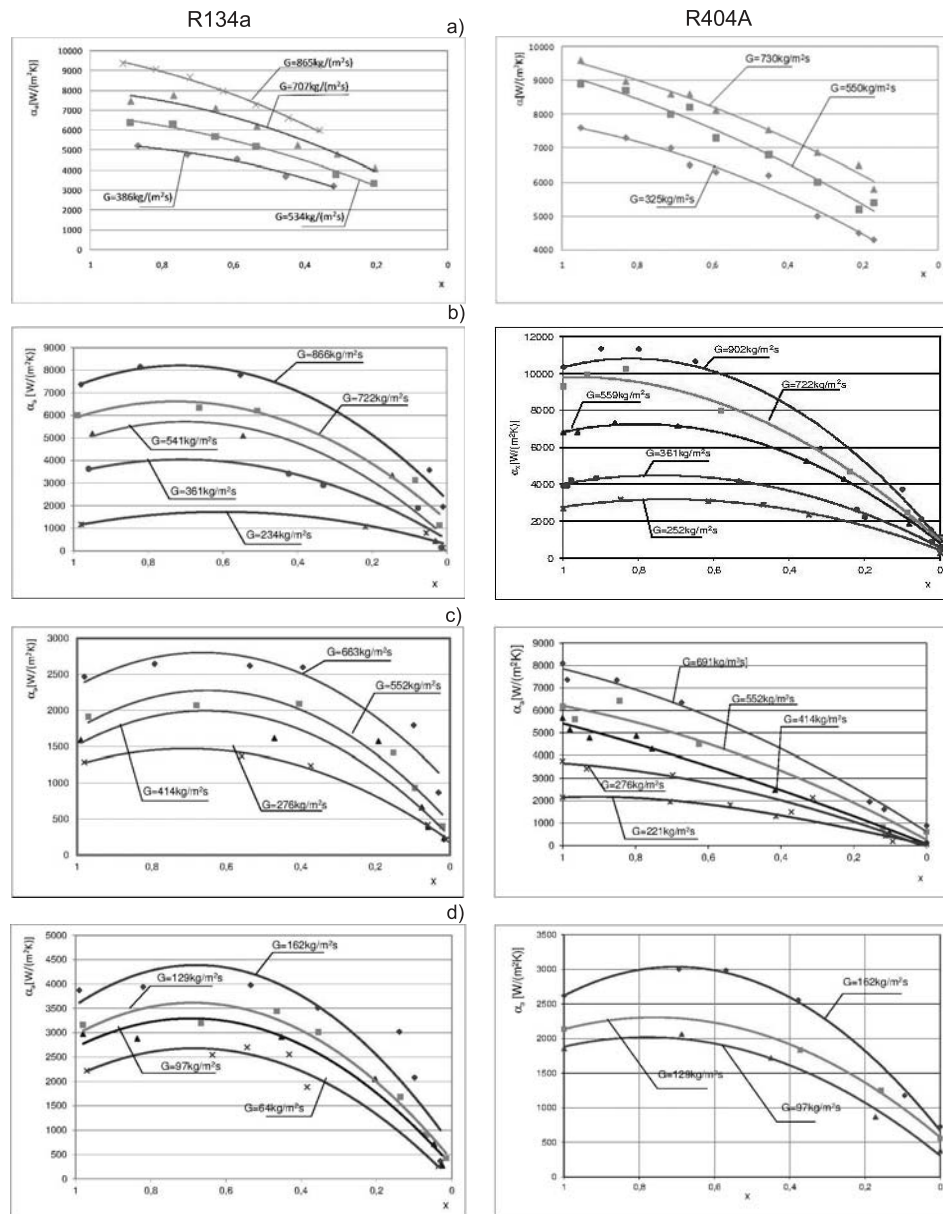


Figure 6. Results of experimental tests concerning the local heat transfer coefficient during the condensation of R134a and R404A in tubular minichannels with internal diameters: a) $d = 0.98$ mm, b) $d = 1.40$ mm, c) $d = 1.60$ mm, and d) $d = 3.30$ mm.

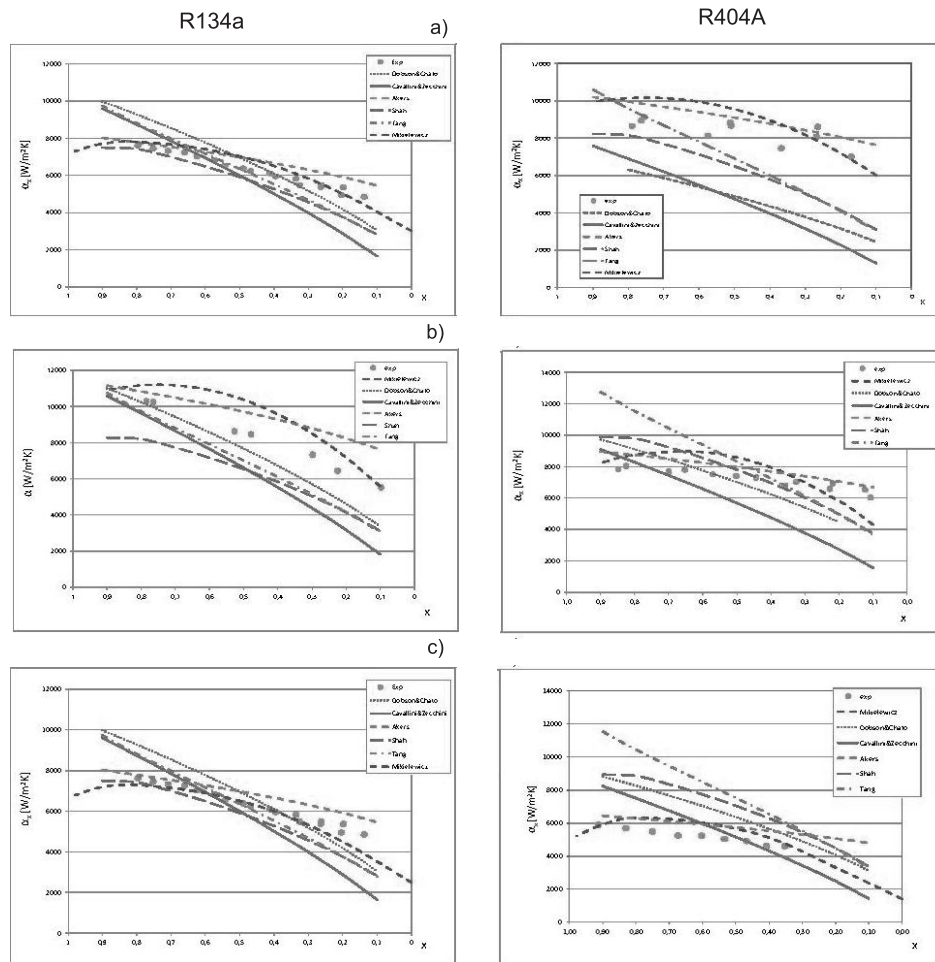


Figure 7. Comparison of the results of the experimental tests with the results of calculations according to the correlations proposed by other authors (Dobson and Chato [13], Cavallini and Zecchin [10], Akers *et al.* [2,3], Shah [26] and Tang [31], Mikielewicz [22], for $G = 550\text{--}600 \text{ kg/m}^2\text{s}$ and the following minichannel internal diameters: a) $d = 0.98 \text{ mm}$, b) $d = 1.40 \text{ mm}$, and c) $d = 2.30 \text{ mm}$.

Tang [31]. It is important that when using the abovementioned selection of correlations in the range of the parameters with R134a, one may take into account the correlations according to Akers *et al.* [2,3] and Shah [26], though with certain reservations. This is particularly evident for R134a. The approximation accuracy not only depends on the internal diameter of

the minichannel, but also on the refrigerant's mass flux density. For mass flux density $G < 400 - 600 \text{ kg}/(\text{m}^2\text{s})$, a good approximation is obtained from the results of the experiment and the calculation from the correlation of Akers *et al.* [2,3]. This is particularly evident concerning the range of minichannel diameters $d < 1.40 \text{ mm}$. For higher values of heat flux density $G > 600 \text{ kg}/(\text{m}^2\text{s})$, better results are obtained when using Shah's correlation [26]. Correlation of Mikielwicz [22] describe experimental investigations results with good accuracy in a part of experimental area. A list of comparisons is presented in Fig. 8, which confirms the abovementioned reservations. For the whole range of diameters $d = 0.31 - 3.30 \text{ mm}$ of the minichannels, with results are clearly visible; thus, the results of this comparisons are unsatisfactory, with the exception of comparison with Mikielwicz correlation.

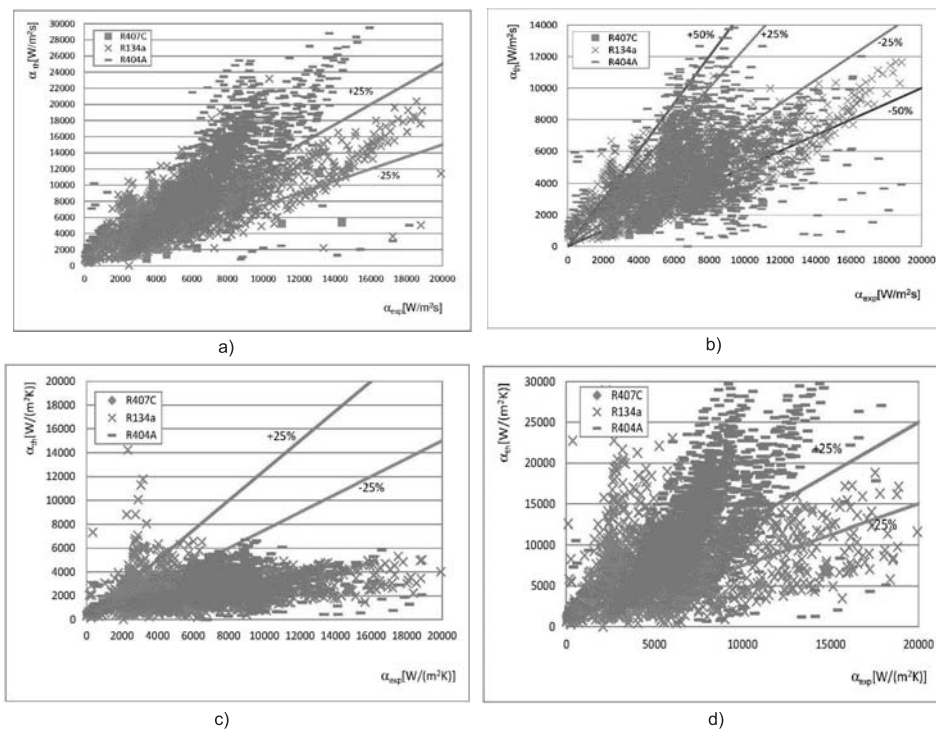


Figure 8. Comparison of the results of the experimental investigations concerning the local heat transfer coefficient of R134a in minichannels with internal diameter $d = 0.31 - 3.30 \text{ mm}$ with the results of calculations according to the correlations of: a) Shah [26], b) Akers *et al.* [2,3], c) Thome [32], d) Tang [31].

A comparative analysis of the results of experimental investigations with the results of calculations according to the correlations by various authors demonstrated, in relation to the local heat transfer coefficient, 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 simplex methods. An experimental correlation was obtained that describes the local heat transfer coefficient in the following form:

$$\text{Nu}_x = 25.084 \text{Re}_l^{0.258} \text{Pr}_l^{-0.495} p_r^{-0.288} \left(\frac{x}{1-x} \right)^{0.266}, \quad (3)$$

hence

$$\alpha_x = \frac{\text{Nu}_x \lambda_l}{d}, \quad (4)$$

where

$$p_r = \frac{p_s}{p_{cr}} \quad (5)$$

is a reduced pressure, p_s is a saturation pressure, and p_{cr} is a critical pressure, λ is the thermal conductivity coefficient, Nu, Pr are the Nusselt and Prandl number, respectively.

Figure 9 presents a comparison of the results of experimental tests with the results of calculations from correlations (3) and (4). The results of the experimental tests fall within the range of $\pm 25\%$. The developed experimental correlation was verified for the following refrigerants R134a, R404A and R407C and a range of parameters $d = 0.31\text{--}3.30$ mm; $x = 1\text{--}0$; $T_s = 20\text{--}50$ °C; and $G = 100\text{--}1300$ kg/(m²s).

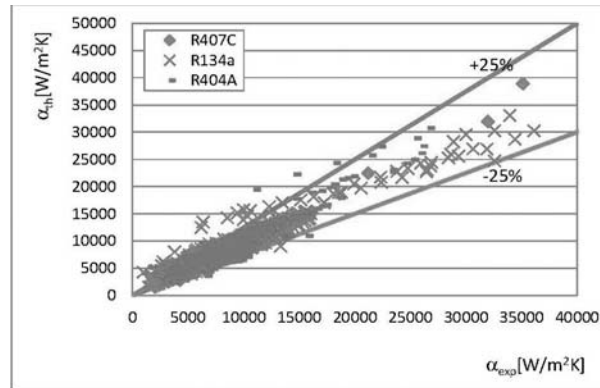


Figure 9. Comparison of the results of experimental tests concerning the local heat transfer coefficient, α_x , during condensation of the refrigerants: R134a, R404A and R407C in pipe minichannels with internal diameter $d = 0.31\text{--}3.30$ mm with the results of calculations from correlation (3).

5 Conclusions

1. Pipe minichannels can be used in the construction of compact refrigeration condensers. These are channels with a circular section, which in the condensation process of refrigerants intensify the heat transfer process while guaranteeing a high degree of structural compactness. The heat transfer coefficient is a measure of this intensification compared with conventional channels. Determination of the value of heat transfer coefficient and the pressure drop during this phase change constitutes the primary problem for designers.
2. The present paper provides comparative results of experimental tests concerning the condensation of refrigerants R134a, R404A and R407C in stainless steel pipe minichannels with internal diameters from 0.31 to 3.30 mm.
3. A comparative analysis was conducted of the results of the experimental tests with the results of calculations from correlations proposed by other authors. It was demonstrated that, of these correlations, which serve to determine the local heat transfer coefficient, those proposed by Akers *et al.* [2,3] and Shah [26] can be used within a limited range depending on the internal diameter of the minichannel and the density of the mass flux of the refrigerant. It was demonstrated, however, that the area of their applications is limited. Mikielewicz correlation [22]

describe experimental results with quite good accuracy and can be used, with limited range, to describe heat transfer coefficient during the condensation in minichannels.

4. An empirical correlation formula (3) was proposed on the basis of the investigations for the calculation of the local heat transfer coefficient within the range of two-phase flow structures, annular and annular-stratified, for the following parameters: saturation temperature 30–40 °C, quality 0–1, and mass flux density 100–1300 kg/(m²s). The results of the experimental tests were within the range of ±25% in relation to the values calculated from correlation expression (3).

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