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Investigations on mixture preparation for two phase adiabatic pressure drop of R134a flowing in 5 mm diameter channel

TOMASZ MUSZYŃSKI[a∗]
RAFAŁ ANDRZEJCZYK[a]
CARLOS A. DORAO[b]

[a] Gdańsk University of Technology, Narutowicza 11/12, 80-233 Gdańsk, Poland

[b] Norwegian University of Science and Technology, Department of Energy and Proces Engineering, N-7491 Trondheim, Norway

Abstract The article presents detailed two-phase adiabatic pressure drops data for refrigerant R134a. Study cases have been set for a mass flux varying from 200 to 400 kg/m^{2}s, at the saturation temperature of 19.4 °C. Obtained experimental data was compared with the available correlations from the literature for the frictional pressure drop during adiabatic flow. Influence of mixture preparation on pressure drop was investigated, for varying inlet subcooling temperature in the heated section. The flow patterns have also been obtained by means of a high-speed camera placed in the visualization section and compared with literature observations.

Keywords: Adiabatic flow; Frictional pressure drop; Two-phase flow; Flow pattern map; R134a

Nomenclature

A	–	surface area, m^2
a	–	heat transfer coefficient, W/m^{2}K
c_p	–	specific heat, J/kgK
D	–	diameter, m

∗Corresponding Author. Email tommuszy@pg.gda.pl

f	–	friction factor
G	–	mass flux, kg/m^{2}s
g	–	gravitational acceleration, m/s^2
h	–	specific enthalpy, kJ/kg
L	–	channel length, m
I	–	current, A
MAD	–	mean absolute deviation, %
$\dot{m}$	–	mass flow of refrigerant, kg/s
ΔP	–	pressure drop, Pa
$\dot{q}$	–	heat flux, kW/m^2
$\dot{Q}$	–	rate of heat, kW
T	–	temperature, °C
x	–	vapour quality

Greek symbols

α	–	heat transfer coefficient
ρ	–	density, kg/m^3

Superscripts

el	–	electric
i	–	inner
in	–	inlet
l	–	liquid
out	–	outlet
sat	–	saturation
sub	–	subcooling
v	–	vapour
W	–	wall

1 Introduction

Developers of many modern devices are faced with two conflicting trends: the need to dissipate increasing amounts of heat, and the quest for more compact and lightweight designs. These trends have spurred unprecedented increases in heat dissipation per volume and per surface area, forcing the research in heat removal enhancement in air cooling [1] and single-phase liquid cooling solutions [2]. Cooling demands in these and many other applications have resulted in a paradigm shift from single-phase to two-phase cooling strategies to capitalize upon the coolant's sensible and latent heat rather than sensible heat alone. Phase change cooling solutions come in a variety of configurations that could meet the system requirements of the application in question. These include pool boiling [3], channel flow boiling [4], mini/microchannels [5], jet [6,7], boiling on enhanced surfaces [8–10], and hybrid cooling techniques [11].

The flow of vapors and liquids in pipes, channels, equipment, etc. is frequently encountered in industry and has been studied intensively for many years. The reliable prediction of flow parameters in two-phase flows is thereby an important aim; yet, heat transfer coefficients and pressure gradients predicted using leading methods often differ by more than 50% according to various reports [12–14].

Instability of two-phase flow in a channel with evaporating fluid is closely related to the existence of pulsations of pressure and flow rate. Significant pressure pulsations may be dangerous. They can lead to channel wall deformations and, in consequence, emergency shutdown. Changes of mass flow rate accompanying pressure fluctuations may, on the other hand, lead to boiling crisis, which in effect lead to the reduction of heat transfer effectiveness in heat exchangers [15]. Therefore the issue of local stability (flow in the channel with evaporating fluids), as well as general evaporator stability, is important to the thermal design engineers [16].

Dorao *et. al* [17] studied the effect of the heating profile of the characteristics of pressure drop oscillations (PDO). The experiments were performed in a 2 m long horizontal test section with 5 mm internal diameter, with R134a as working fluid. The PDOs were characterized by superimposing high- and low-frequency oscillations for varying range of heat flux. It was observed that at low and high heat fluxes with a uniform heating profile the high-frequency oscillations vanish. In addition, a decreasing power distribution can increase their occurrence.

Two-phase flow maldistribution in systems of heated parallel channels with a subcooled inlet state was investigated by Oevelen *et.al* [18]. Such maldistribution can result from the nonmonotonic behavior of channel pressure drop as a function of flow rate. A pressure drop model applied to every individual channel was integrated together with a pump curve into a system model. Multiple different flow distributions can occur for a given operating condition; the stability of each flow distribution is assessed by solving a generalized eigenvalue problem. Parametric effects of inlet subcooling, heat flux, and flow rate on the stability of the uniform distribution and on the severity of maldistribution were also investigated. Authors observed that there is a minimum inlet subcooling below which the uniform distribution is always stable and maldistribution cannot occur, regardless of the boiling number.

Lee *et al.* [19] investigated the minimum mass flux conditions in which a stable flow is sustainable. Flow parameters were identified *via* 47 flow instability data points and were compared with relevant correlations. The results implied flow excursion points that were close to the onset of a significant void. The visualization of the flow excursion using a high-speed camera was achieved, clearly demonstrating that the flow excursion is triggered by the coalescence of facing bubbles or wavy vapors on opposing heated surfaces. Presented data showed deviation from predictions, therefore new empirical correlation that reflect the gap size effect was suggested.

As indicated in the literature experimental data tend to deviate greatly from predictions. Therefore, the main objective of this paper was to investigate the frictional pressure drop during adiabatic flow in the 5 mm diameter channel with varying heat flux at a two-phase mixture preparation section. The objective of the present study is to run the two-phase flow in-tube cases under different experimental conditions in order to characterize the frictional pressure drop for the refrigerant R134a, for vapor qualities ranging from 0 to 1.

2 Experimental setup

2.1 Experimental test facility

The experimental facility is the R134a loop consisting of a main reservoir or refrigerant, pump, preheater (conditioner), heated test section, sight glass, and an adiabatic section and condenser. The loop is schematically represented in Fig. 1.

The fluid pressure is set by controlling the temperature in the main tank where the refrigerant is at saturation conditions. The fluid is driven by a magnetically coupled gear pump which prevents any leakage of working fluid. The conditioner is a shell and tube heat exchanger with glycol in the shell side which is used for adjusting the R134a inlet temperature. Before entering the heated section the refrigerant flows through a Coriolis type mass flow meter. A mass flow rate accuracy of 0.2% of the reading was given by the supplier. The heated section is made from two meters long stainless steel tube. The section is electrically heated by Joule effect with the use of a low voltage AC power supply. The section is thermally insulated with a thick layer of mineral wool, thus thermal losses are neglected. The tube dimensions are 5 mm and 8 mm internal and external diameter respectively. Nine galvanically separated thermocouples are distributed along the wall surface, additional two are inside the tube in order to measure the local fluid temperature. The thermocouple accuracy after in house calibration

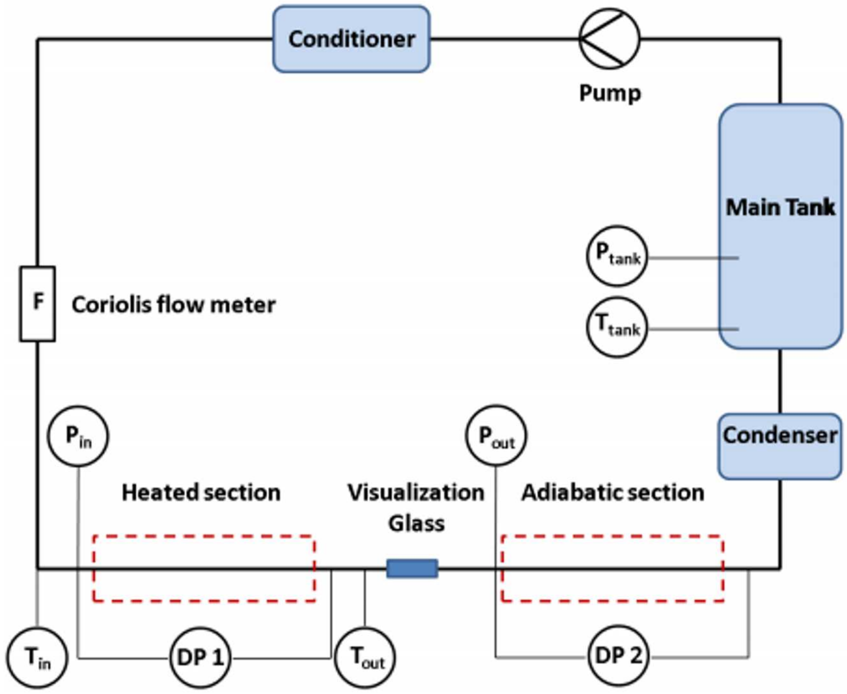

Figure 1: Schematic diagram of test facility: P_{tank} – pressure in the tank, P_{out} – pressure at the outlet of heated section, T_{out} – temperature at the outlet of the heated section, T_{in} – temperature at the inlet of the heated section, T_{tank} – temperature in the tank, DP1 – pressure drop at the heated section, DP2 – pressure drop at the adiabatic section.

was found to be 0.05 K.

The adiabatic test section is a 1 m long stainless steel pipe with an inner diameter of 5 mm and 8 mm outer diameter. It is thermally insulated with polyurethane foam insulation. The adiabatic and heated section are arranged horizontally in line, with a 300 mm distance between them.

2.2 Experimental procedure and data acquisition

The measurements have been performed with the aid of computer connected to a National Instruments Compact RIO data acquisition system. The signal from measuring devices was processed with the aid of the LabVIEW application. The temperatures, absolute pressures, pressure differences and mass flow rates were acquired at a frequency of 2 Hz.

For every experimental point $\sim$100 data points were acquired, thus every point was obtained by averaged values from ˜50 s measurements. Additionally, every experimental point was recorded twice, with 5 min interval, in order to exclude heat capacity effect of tube and insulation.

Two-phase total pressure drop was directly measured with a differential pressure transducer, same type differential pressure transducer was used to measure the adiabatic pressure drop. A differential pressure accuracy of 0.075% full-scale was given by the supplier. An absolute pressure accuracy of 0.04% full-scale was given by the supplier. This accuracy was checked by in house calibration. The heat flux and two-phase heat transfer coefficient were determined according to relations

$$\dot{q} = \frac{\dot{Q}_{el}}{A_W} , \tag{1}$$

$$\alpha = \frac{\dot{q}}{(T_W - T_{sat})} . \tag{2}$$

The absolute pressure at the inlet and outlet of the heated section was also recorded and was used for checking the saturation temperature T_{sat} of the fluid based on the equilibrium thermodynamic properties. The refrigerant quality at the inlet of the adiabatic section was determined from mass and energy conservation equations using NIST REFPROP fluid database [20]:

$$x_{out} = \frac{\dot{q}\pi D_i L - \dot{m} c_p \Delta T_{sub}}{\dot{m} h_{lv}} . \tag{3}$$

The general expression for describing the total two-phase pressure drop Δp_{total} is

$$\Delta p_{total} = \Delta p_{mom} + \Delta p_{frict} + \Delta p_{static} , \tag{4}$$

where Δp_{static} is the elevation head pressure drop and is neglected in a horizontal tube, Δp_{mom} is the momentum pressure drop created by the acceleration of the flow in a heating/cooling process, and Δp_{frict} is the two-phase frictional pressure drop.

The adiabatic two-phase pressure drops were obtained at the vapor quality leaving the horizontal boiling test section. No additional heat flux allows steady two-phase flow inside tube without bubble growth, thus the momentum pressure drop in the adiabatic section is zero. Therefore, we can determine the two-phase frictional pressure drops in the adiabatic test section directly from the measured values:

$$\Delta p_{total} = \Delta p_{frict} . \tag{5}$$

Using electric current to supply heat in the diabatic test section to evaporate the refrigerant, the wall temperature undergoes a slight temperature change while the phase changing refrigerant stays at nearly the same saturation temperature. The local heat flux is calculated as a function of generated Joule heat and is assumed to be constant during the evaporation process along the length of the tube.

In order to determine the reliability of the experimental results, an uncertainty analysis was conducted on all measured quantities as well as the quantities calculated from the measurement results. For the heat flux, the error coming from the propagation is the error associated with the voltage and current measurements. Nevertheless, the thermal heat flowing to the fluid under stationary conditions was calibrated against the electrical value for different temperatures and conditions for single phase liquid considering the heat exchange with the surroundings arriving to a final accuracy of 3%. Uncertainties were estimated according to the standard procedures described by National Institute of Standards and Technology (NIST) [20]. Overall, the uncertainty in the calculated vapor quality is lower than 10%.

3 Experimental data analysis

A summary of the experimental conditions of adiabatic two-phase pressure drop of R134a are presented in Tab. 1.

Table 1: Experimental conditions of pressure drop experiments.

Parameter	Unit	Operating range
D	m	0.005
L_h	m	2
L_{ad}	m	1
T_{sat}	°C	19.4
G	kg/m^{2}s	200–400
q''	W/m^2	100–69000

In order to verify the assumed accuracy of the measurement and the correctness of the experimental procedure the preliminary studies were accomplished involving the determination of the pressure drop for the single-phase adiabatic flow of the test fluid R134a. The obtained experimental data, in the form of a pressure drop at the length of the 2 m long channel were compared with the Darcy-Weisbach correlation, assuming the friction coefficient according to the Haaland [21] equation, with pipe roughness height given by supplier to be below 0.03 mm. The pressure drop can be obtained by the following expression:

$$\Delta p = f \frac{G^2}{2\rho} \frac{L}{D} , \tag{6}$$

where f is the frction factor It turns out that the majority of the experimental data fits in the range of ±10% of the consistency with predictions. The maximal absolute deviation of 30% was present in the lower range of Reynolds numbers, as can be seen in Fig. 2. This can be attributed to larger error of pressure transducers. Most of the experimental points presented in this article are higher than 2 kPa.

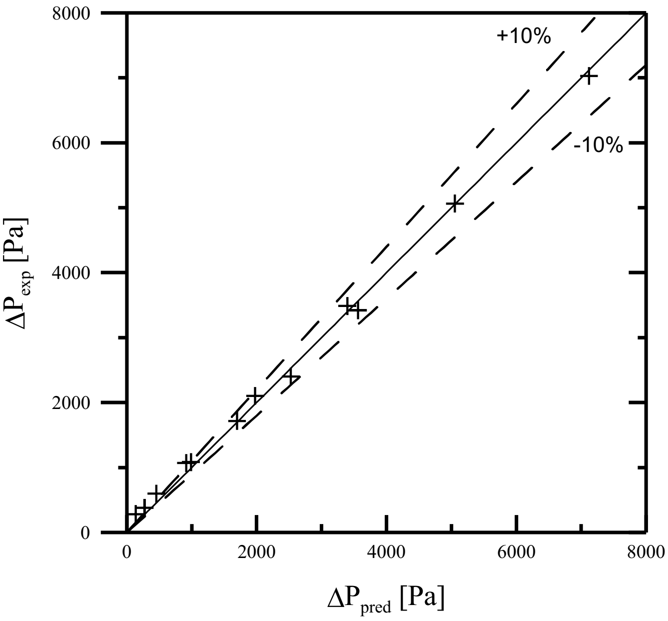

Figure 2: Comparison of single-phase pressure drop experimental data for the R134a with the predicted Darcy-Weisbach correlation.

In authors opinion that result can be deemed as satisfactory, particularly in the light of the fact that the recorded pressure drop rarely drops below a value of 1 kPa, with a maximum of 50 kPa in the accomplished experiment. The deviation between predicted and obtained results are close to the measurement accuracy achieved by the applied pressure transducers.

Adiabatic pressure drop

Adiabatic pressure drops as indicated in the introduction can be measured after preparation of vapour-liquid mixture in heated section. Varying the inlet subcooling temperature of working fluid will influence the amount of heat necessary to obtain same vapor quality. Figure 3 shows the experimental two-phase pressure drop of the 5 mm tube as a function of mass flux and heat flux of R134a at different subcooling temperatures, at a saturation temperature of 19.4 °C.

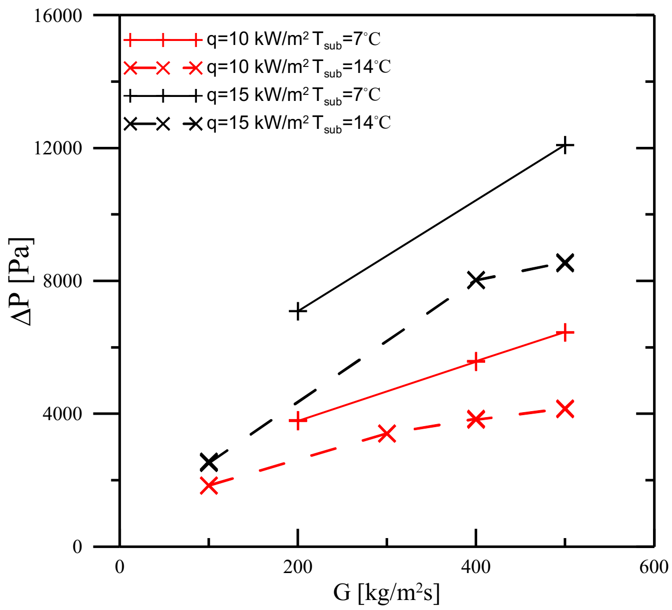

Figure 3: Experimental two-phase pressure drop in the 5 mm tube as a function of mass flux and heat flux of R134a at different subcooling temperatures.

The experimentally obtained adiabatic pressure drop values are compared with well-known correlations from the literature. Because inlet vapor quality is obtained from energy balance equation on heated section all data regarding adiabatic flow with the same mass flux and various subcooling are plotted on a single figure. The pressure drop gradients are calculated by dividing measured pressure difference by the test section length, thus each experiment test condition allows to gather a single point. Since the flow is adiabatic the experimental data presents a frictional pressure drop. Figures 4 to 6 show the experimental adiabatic frictional pressure drops, plotted versus values predicted with selected correlations. Pressure drop models used in this comparison are presented in detail in the previous section.

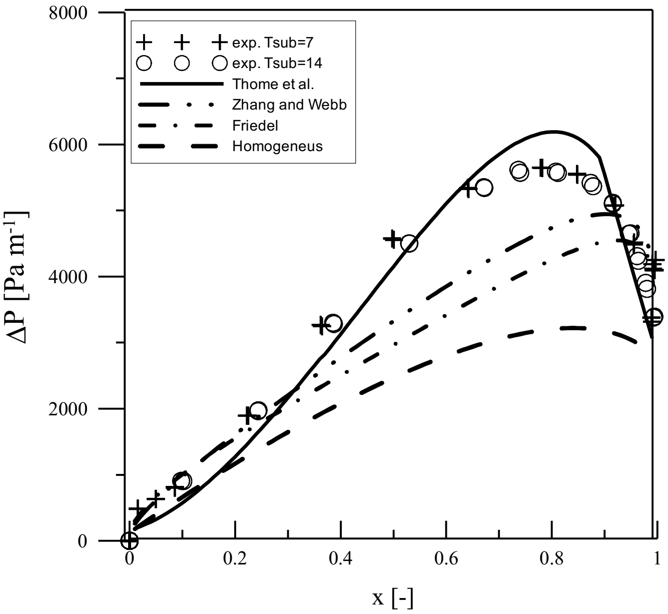

Figure 4: Experimental adiabatic pressure drop of R134a as a function of vapor quality in comparison with literature models for $G = 200$ kg/m^{2}s.

The adiabatic pressure drop along channel increases exponentially with mass flux. It can also be seen that the two-phase pressure drop is increasing for higher exit vapor quality with a maximum about vapor qualities around x = 0.7-0.9, as reported in the literature. The local maxima of pressure drop are observed to be more shifted towards higher vapor qualities at a higher mass fluxes. It can be seen that pressure drop values of each mass flux corresponds to single phase pressure drop values. It can also be seen from that the difference between the two phase pressure drops is higher for higher mass flux. That corresponds to the trends reported in literature, e.g., by Ould Didi *et al.* [22] for refrigerants flow in macrotubes of 10.92–12 mm and by Tran *et al.* [23], for small channels.

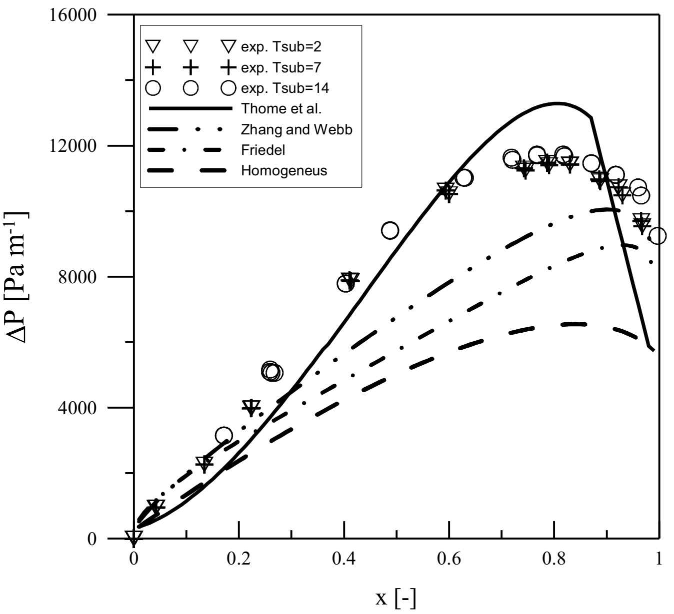

Figure 5: Experimental adiabatic pressure drop of R134a as a function of vapor quality in comparison with literature models for $G = 300$ kg/m^{2}s.

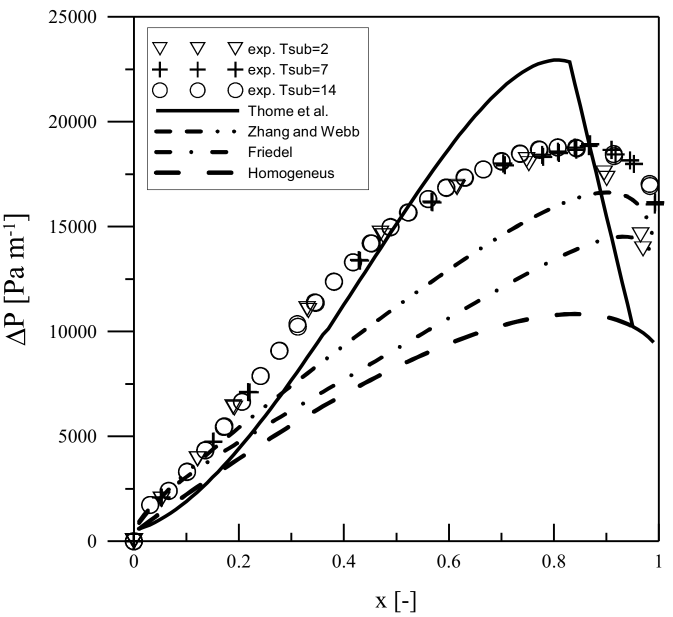

Figure 6: Experimental adiabatic pressure drop of R134a as a function of vapor quality in comparison with literature models for $G = 400$ kg/m^{2}s.

From an overall comparison of figures, 4 to 6 good agreement with experimental data for correlations of Zhang and Webb [24] and Thome *et al.* [25] clearly stands out as best in describing pressure drop variations with varying vapor quality. Most importantly one can observe the difference between the experimental data series for varying inlet subcooling. For higher values of inlet subcooling series experimental pressure drop is shifted towards higher values, what is visible for 0.8–1 vapor qualities.

Table 2: Literature correlations predictions confidence levels.

	Zhang and Webb	Thome *et al.*	Friedel	Homogeneous
Mean abolute deviation	0.17	0.18	0.24	0.36
Confidence level $>30\%$	0.93	0.80	0.58	0.27
$>20\%$	0.60	0.63	0.36	0.09
$>10\%$	0.30	0.32	0.22	0.04

The Zhang and Webb [24] correlation is a modified form of the Friedel [26] correlation. This correlation predicts most of the data within $\pm 30\%$. However, only 30% of the data is predicted within 10% error band.

The correlation of Thome et al. [25] for predicting the frictional pressure drop is flow pattern based. Thus this method includes the effect of interfacial flow structure via the flow pattern map, and better follows the variation in pressure gradient with vapor quality. Also, it captures the peak in the pressure gradient at high vapor qualities. It is based on the actual mean velocities of the phases via the void fraction equation rather than superficial velocities. It has to be pointed that pressure drop experimental values during the experiments were also visually investigated in order to validate flow pattern map developed by Wojtan *et al.* [27].

Predictions given by the homogeneous model assume equal velocities of vapor and liquid, also the fluid is considered as one single phase with averaged properties. This model under predicts the data with a mean aqbsolute deviation (MAD) of 36%. It should be noted that the prediction trend in the results is good and that the deviation is decreasing with increased mass flux, indicating that with a change in the leading constant this correlation would give better predictions. In low flow ranges, prediction error seems to be more pronounced in vapor qualities in a range of $x = 0.4$–0.8.

In order to find an answer for a shift in pressure drop values, flow visualization was performed. Figure 7 shows the flow pattern map for a representative set of experimental conditions for the mass flux $G = 400$ kg/m^{2}s. Selected points confirming structure shown on the flow pattern map are depicted in Figs. 8 to 12. This map is based on a recent version of the Kattan *et al.* [28] flow map, proposed by Wojtan *et al.* [27], and also includes an improved method for the effect of heat flux on the transition to mist flow. D represents the transition zone between annular and mist flow.

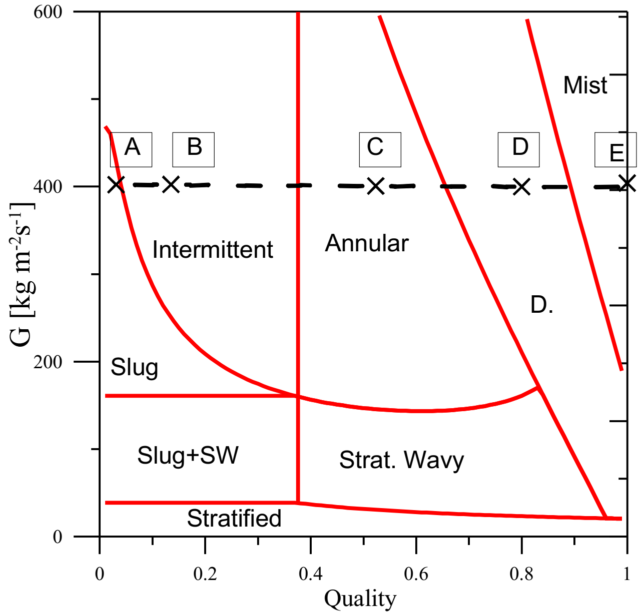

Figure 7: Flow pattern map evaluated for R134a at $T_{sat} = 19.4$ °C in a 5 mm internal diameter tube for $\dot{q} = 35$ kW/m^2 using $G = 400$ kg/m^{2}s to calculate the void fractions.

All of the obtained visualizations were similar in terms of flow structure without a visible difference in the flow pattern. As mentioned in the previous section, the refrigerant was heated in horizontal channel of approx. 2 m length. Vapor quality at the inlet of the adiabatic test section was calculated from energy balance equation. Entrained droplets can be observed in Fig. 12, while the calculated vapor quality is equal to 1 (Fig. 7). In authors' opinion the difference between presented test runs may be explained by the

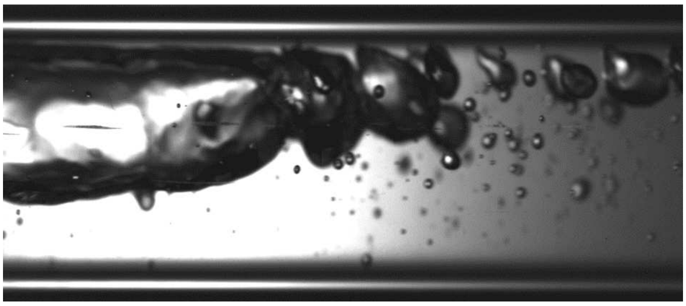

Figure 8: Experimentally obtained slug adiabatic flow of R134a for $G = 400$ kg/m^{2}s with $\dot{q} = 4.7$ kW/m^2, point A on flow pattern map.

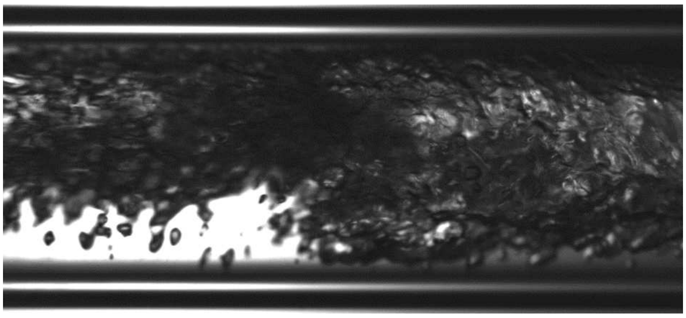

Figure 9: Experimentally obtained intermittent adiabatic flow of R134a for $G = 400$ kg/m^{2}s with $\dot{q} = 9.5$ kW/m^2, point B on flow pattern map.

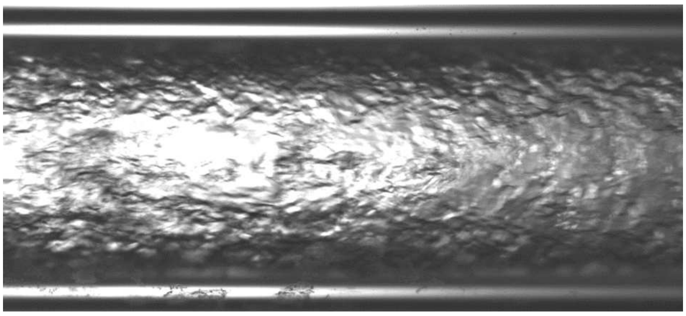

Figure 10: Experimentally obtained annular adiabatic flow of R134a for $G = 400$ kg/m^{2}s with $\dot{q} = 28.6$ kW/m^2, point C on flow pattern map.

entrainment effect, which is influenced by the supplied heat flux. Unfortunately, in most prediction tools the parameters at which the two-phase mixture is prepared are not taken into account. It is also possible that the

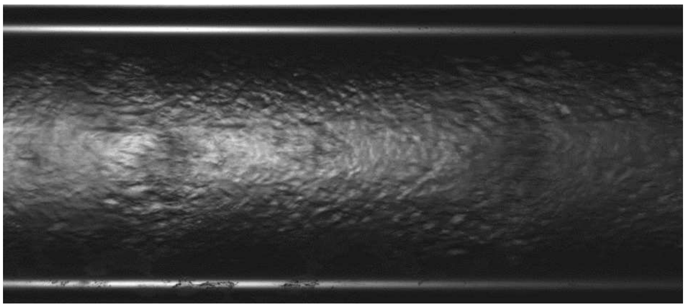

Figure 11: Experimentally obtained dryout adiabatic flow of R134a for $G = 400$ kg/m^{2}s with $\dot{q} = 39$ kW/m^2, point D on flow pattern map.

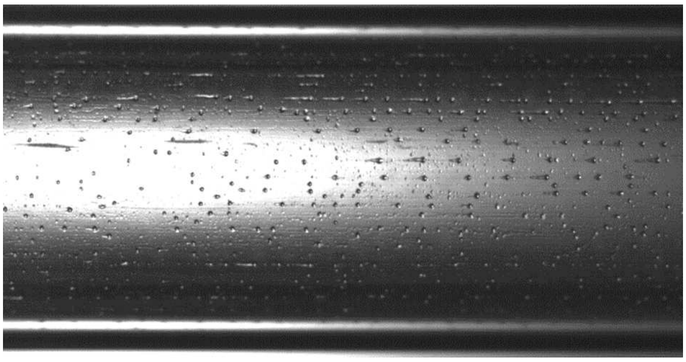

Figure 12: Experimentally obtained mist adiabatic flow of R134a for $G = 400$ kg/m^{2}s with $\dot{q} = 47$ kW/m^2, point E on flow pattern map.

pressure drop is affected by an average void fraction which may be different than in a single snapshot.

4 Conclusions

As the first step of this work, a comprehensive experimental study was undertaken in order to obtain accurate two-phase pressure drop values during the adiabatic flow of refrigerants R134a in a horizontal tube. The flow conditions were chosen to obtain experimental values over a wide range of test parameters so that the effect of each parameter could be easily identified. The range of experimental conditions covered were three mass velocities and vapor quality covering the entire range from 1 to 0, and three inlet subcooling temperatures. The existing experimental facility allowed to run tests under adiabatic conditions to obtain two-phase pressure drop values for nearly every flow regime, and to validate the data reduction procedure used to obtain the pressure drop, and to validate existing flow pattern maps. The experimental campaign acquired over 500 experimental two-phase pressure drop values covering all flow regimes except bubbly because of operating limitations.

Obtained data is used as a validation of the literature models, a set of graphs showed comparisons, for a representative set of experimental conditions, of the two-phase frictional pressure gradients for the adiabatic test section. Agreement including reliability of the measurements as well as the correctness of the data reduction protocol and choice of void fraction model was shown to be quite good. A slight shift from predictions given by Thome *et al.* [25] model was attributed to entrained droplets in vapor flow with fluid enthalpy calculated from energy balance higher than saturated vapor.

Verification of the pressure drop for single-phase adiabatic flow showed that for Zhang and Webb correlation 93% of experimental data fits in the range of +/-30%. The model proposed by Thome *et al.* in other hand predicts almost 33% of data within 10% error, but only 80% of the data is predicted within 30% error.

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References

[1] Muszynski T., Koziel S.M.: *Parametric study of fluid flow and heat transfer over louvered fins of air heat pump evaporator.* Arch. Thermodyn. **37**(2016), 3, 45–62. DOI:10.1515/aoter-2016-0019.

[2] Bohdal T., Charun H., Sikora M.: *Empirical study of heterogeneous refrigerant condensation in pipe minichannels.* Int. J. Refrig. **59**(2015), 210–223. DOI:10.1016/j.ijrefrig.2015.07.002.

[3] Mudawar I., Howard A.H., Gersey C.O.: *An analytical model for near-saturated pool boiling critical heat flux on vertical surfaces.* Int. J. Heat Mass Transf. **40**(1997), 2327–2339.

[4] Mikielewicz D., Andrzejczyk R., Jakubowska B., Mikielewicz J.: *Analytical model with nonadiabatic effects for pressure drop and heat transfer during boiling and*

condensation flows in conventional channels and minichannels. Heat Transf. Eng. **37**(2016), 1158–1171.

[5] Tran T.N., Wambsganss M.W., France D.M.: *Small circular-and rectangular-channel boiling with two refrigerants*. Int. J. Multiph. Flow. **22**(1996), 485–498.

[6] Muszynski T., Mikielewicz D.: *Comparison of heat transfer characteristics in surface cooling with boiling microjets of water, ethanol and HFE7100*. Appl. Therm. Eng. **93**(2016), 1403–1409. DOI:10.1016/j.applthermaleng.2015.08.107.

[7] Muszynski T., Andrzejczyk R.: *Applicability of arrays of microjet heat transfer correlations to design compact heat exchangers*. Appl. Therm. Eng. **100**(2016), 105–113, DOI:10.1016/j.applthermaleng.2016.01.120.

[8] Nakayama W., Nakajima T., Hirasawa S.: *Heat sink studs having enhanced boiling surfaces for cooling of microelectronic components*. ASME Pap. 84-WA/HT-89(1984).

[9] Marto P.J., Lepere V.J.: *Pool boiling heat transfer from enhanced surfaces to dielectric fluids*. J. Heat Transfer. **104**(1982), 292–299.

[10] Andrzejczyk R., Muszynski T.: *Performance analyses of helical coil heat exchangers. The effect of external coil surface modification on heat exchanger effectiveness*. Arch. Thermodyn. **37**(2016), 4, 137–159. DOI:AOT-00010-2016-04.

[11] Muszynski T., Andrzejczyk R.: *Heat transfer characteristics of hybrid microjet – microchannel cooling module*. Appl. Therm. Eng. **93**(2016), 1360–1366. DOI:10.1016/j.applthermaleng.2015.08.085.

[12] Andrzejczyk R., Muszynski T., Alberto Dorao C.: *Experimental investigations on adiabatic frictional pressure drops of R134a during flow in 5mm diameter channel*. Exp. Therm. Fluid Sci. **83**(2017), 78–87. DOI:10.1016/j.expthermflusci.2016.12.016.

[13] Mikielewicz D., Jakubowska B.: *Prediction of flow boiling heat transfer coefficient for carbon dioxide in minichannels and conventional channels*. Arch. Thermodyn. **37**(2016), 2, 89–106. DOI:10.1515/aoter-2016-0014.

[14] Mikielewicz D., Andrzejczyk R.: *Comparative study of flow condensation in conventional and small diameter tubes*. Arch. Thermodyn. bf 33(2012), 2, 67–83. DOI:10.2478/v10173-012-0011-2.

[15] Mikielewicz J., Mikielewicz D.: *Thermal-hydraulic issues of flow boiling and condensation in organic Rankine cycle heat exchangers*. Arch. Thermodyn. **33**(2012), 1, 41–66. DOI:10.2478/v10173-012-0002-3.

[16] Muszynski T.: *Design and experimental investigations of a cylindrical microjet heat exchanger for waste heat recovery systems*. Appl. Therm. Eng. **115**(2017),782–792. DOI:10.1016/j.applthermaleng.2017.01.021.

[17] Dorao C.A., Langeland T., Fernandino M.: *Effect of heating profile on the characteristics of pressure drop oscillations*. Chem. Eng. Sci. **158**(2017), 453–461. DOI:10.1016/j.ces.2016.10.009.

[18] Van Oevelen T., Weibel J.A., Garimella S.V.: *Predicting two-phase flow distribution and stability in systems with many parallel heated channels*. Int. J. Heat Mass Tran. **107**(2017), 557–571. DOI:10.1016/j.ijheatmasstransfer.2016.11.050.

[19] Lee J., Chae H., Chang S.H.: *Flow instability during subcooled boiling for a downward flow at low pressure in a vertical narrow rectangular channel.* Int. J. Heat Mass Tran. **67**(2013), 1170–1180. DOI:10.1016/j.ijheatmasstransfer.2013.08.049.

[20] *N. REFPROP, standard reference database 23. NIST Thermodyn. Prop. Refrig. Mix. Database (REFPROP).* Version. 9 (2002).

[21] Haaland S.E.: *Simple and explicit formulas for the friction factor in turbulent pipe flow.* J. Fluids Eng. **105**(1983), 1, 89–90.

[22] Ould Didi M.B., Kattan N., Thome J.R.: *Prediction of two-phase pressure gradients of refrigerants in horizontal tubes.* Int. J. Refrig. **25**(2002), 935–947. DOI:10.1016/S0140-7007(01)00099-8.

[23] Tran T.N., Chyu M.-C., Wambsganss M.W., France D.M.: *Two-phase pressure drop of refrigerants during flow boiling in small channels: An experimental investigation and correlation development.* Int. J. Multiph. Flow. **26**(2000), 11, 1739–1754.

[24] Zhang M., Webb R.L.: *Correlation of two-phase friction for refrigerants in small-diameter tubes.* Exp. Therm. Fluid Sci. **25**(2001), 3–4, 131–139. DOI:10.1016/S0894-1777(01)00066-8.

[25] Thome J.R.: *Engineering Databook III.* Wolverine Tube Inc. 2010 (2004).

[26] Friedel L.: *Improved friction pressure drop correlations for horizontal and vertical two-phase pipe flow.* In: Eur. Two-Phase Flow Gr. Meet. Pap. E, 1979.

[27] Wojtan L., Ursenbacher T., Thome J.R.: *Investigation of flow boiling in horizontal tubes: Part II—Development of a new heat transfer model for stratified-wavy, dryout and mist flow regimes.* Int. J. Heat Mass Tran. **48**(2005), 2970–2985.

[28] Kattan N., Thome J.R., Favrat D.: *Flow boiling in horizontal tubes: Part 1 – Development of a diabatic two-phase flow pattern map.* J. Heat Transfer. **120**(1998), 1, 140–147.