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Two-phase frictional pressure gradient of R236ea, R134a and R410A inside multi-port mini-channels

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Abstract

In the present paper the pressure drop characteristics of a 1.4 mm hydraulic diameter multiport minichannel tube during adiabatic two-phase flow of HFC refrigerants are discussed. The tube consists of eleven parallel rectangular cross section channels. Although much experimental research has been devoted to investigate the pressure drop characteristics of R134a inside multiport minichannels in the last years, very little information is available on different refrigerants, such as R236ea (low pressure) and R410A (high pressure), whose behaviour is different from R134a. The experimental runs are carried out at mass velocities ranging from 200 to 1400 kg/(m^2 s), depending on the refrigerant under test, at constant value of vapour quality. The frictional pressure gradient is obtained from the saturation temperature drop measurement. The results presented here cover a wide range of the reduced pressure, from 0.1 up to 0.5. The experimental frictional pressure drop data is compared against models available in the literature for prediction of frictional pressure gradient in channels. The comparison of calculated to experimental values shows that the R134a frictional pressure gradient in the multiport minichannel test tube can be fairly well predicted by available correlations, but not satisfactory agreement is found for R236ea data and even worse in the case of R410A data.

1. Introduction

More and more research has been focusing on two-phase heat transfer and fluid-dynamics inside mini and microchannels in the last years. From the knowledge on macroscale convective heat transfer inside channels, it is well known that, at some mass velocity higher heat transfer coefficient can be obtained when reducing the hydraulic diameter at the expense of higher frictional pressure drop. Although this simple mechanism suggests the advantage of lowering the channel diameter, it is not clear to what extent the know-how on macroscale heat transfer and fluid-dynamics remains valid in the microscale phenomena.

Mini and microchannels are used in many different applications, such as heat pipes, electronic equipment and automotive condensers. Automotive condensers use flat, extruded aluminium channels having a minor side length as small as 1.35 mm and hydraulic diameters as small as 0.5 mm. This flat tube reduces the air-side pressure drop and the refrigerant charge in the equipment.

It is clear that, in designing miniaturized heat transfer equipment, characterized by very small cross sections, the prediction of pressure drop is as important as the prediction of heat transfer coefficients.

Recently much experimental research was devoted to measure frictional pressure drop during gas–liquid two-phase flow in small diameter tubes and in microchannels with circular, rectangular and triangular cross section, and several new correlations have been published.

The present work deals with pressure drop during two-phase flow inside multi-port minichannels. A

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picture of the test tube is reported in Fig. 1. In this particular case, the presence of several parallel channels adds a new constraint to the investigator, because of the problems with the liquid–vapour flow distribution in the channels. A single phase flow is maintained at the inlet of the multiport tube to avoid mal-distribution in the minichannels, which would occur if the liquid and vapour phases were both present in the fitting at the entrance of the multiport channels.

In the presence of many parallel mini-channels, the pressure taps for the differential pressure transducer would normally be connected to the fittings before and after the multiport tube. This way, the measured pressure drop would include the local inlet and outlet effects and the frictional pressure drop of the desuperheating length in the channels.

In the present paper a different technique is adopted, which allows to measure the mere local frictional pressure gradient during two-phase flow in the tube, with no need of subtracting other unpredictable components from the measured value. In fact, the frictional pressure drop is obtained from the saturation temperature drop which, in turn, is measured by means of a thermopile in a 1.4 mm hydraulic diameter multiport tube.

The pressure gradient data reported in the paper are referring to the three refrigerants R236ea, R134a and R410A. All three are HFC refrigerants, but they present far different saturation pressures at the same temperature. For instance, at 40 °C saturated R134a has 10.2 bar, while at the same temperature the saturation pressure of R236ea is 3.38 bar and for R410A it is 24.3 bar. Therefore, R236ea is included among the so-called "low-pressure" refrigerants and R410A is considered a "high-pressure" refrigerant. Several studies have been conducted to investigate the R134a heat transfer and pressure drop performance inside multiport minichannels, primarily because of its application in automotive condensers, while very little information is available on the performance of the other refrigerants. The adoption of these three fluids in the present experimental investigation allows to collect data in far different operating conditions, as it results from the values of the reduced pressure at 40 °C: 0.096 for R236ea, 0.49 for R410A. The reduced pressure of R134a remains in the middle ($p_R = 0.25$).

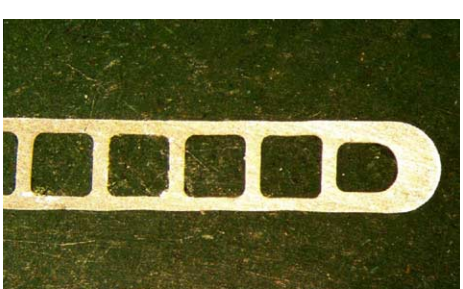

Fig. 1. Enlarged image of the multiport minichannels test tube.

For the future, the multiport minichannels are considered as promising geometries in condensers and evaporators also for some non-mobile air-conditioning and heat pump applications. This perspective can explain the increased interest in the performances of other refrigerants, mostly high pressure ones, inside multiport minichannels.

2. Literature review

The variation of pressure during intube condensation or vapourisation can be obtained from the gas–liquid separated flow momentum equation (the co-ordinate z is positive in the direction of motion), if no component directly caused by surface tension effects is present:

$$-(\mathrm{d}p/\mathrm{d}z) = -(\mathrm{d}p/\mathrm{d}z)_\mathrm{f} - (\mathrm{d}p/\mathrm{d}z)_\mathrm{g} - (\mathrm{d}p/\mathrm{d}z)_\mathrm{a} \quad (1)$$

$-(\mathrm{d}p/\mathrm{d}z)_\mathrm{g}$ is the gravitational pressure gradient and it is important for long vertical tubes while it is nil for horizontal tubes; $-(\mathrm{d}p/\mathrm{d}z)_\mathrm{a}$ is the accelerational pressure gradient. The last term $-(\mathrm{d}p/\mathrm{d}z)_\mathrm{f}$ is the frictional pressure gradient, which can be related to that of the liquid or of the gas phase, flowing by themselves in the tube at their actual flow rates or at the total mass flow rates. For example:

$$-(\mathrm{d}p/\mathrm{d}z)_\mathrm{f} = -\Phi_\mathrm{LO}^2(\mathrm{d}p/\mathrm{d}z)_\mathrm{f,LO} \quad (2)$$

where Φ_LO is the related frictional multiplier and $-(\mathrm{d}p/\mathrm{d}z)_\mathrm{f,LO}$ is the pressure gradient of the liquid phase flowing by itself with the total two-phase flow rate. The single phase pressure gradient in Eq. (2) can be calculated with the classical equations by using the friction factor f. Two of the best known correlations to predict the two-phase multiplier are the Friedel [6,7] correlation and the Lockhart and Martinelli model.

The method by Lockhart and Martinelli uses the two-phase multiplier Φ_L which is referred to the pressure gradient of the liquid phase flowing with its own flow rate $[-(\mathrm{d}p/\mathrm{d}z)_\mathrm{f} = -\Phi_\mathrm{L}^2(\mathrm{d}p/\mathrm{d}z)_\mathrm{f,L}]$. Φ_L is linked to the parameter X through the Chisholm correlation:

$$\Phi_\mathrm{L}^2 = 1 + C/X + 1/X^2 \quad (3)$$

where X is the Lockhart and Martinelli parameter $[X^2 = (\mathrm{d}p/\mathrm{d}z)_\mathrm{f,L}/(\mathrm{d}p/\mathrm{d}z)_\mathrm{f,G}]$ and C is a constant depending on the flow regime of the two phases.

Recently much experimental research was devoted to measure frictional pressure drop during gas–liquid two-phase flow in small diameter tubes ($d < 6$ mm) and in minichannels with circular, rectangular and triangular cross section, and several new correlations have been published.

Cavallini et al. [1] reported a literature review on condensation heat transfer and pressure drop inside channels for AC&R heat transfer applications, including multiport minichannels. For sake of clearness, we adopt here the general classification proposed by Kandlikar and Grande [8], using the dubbing "minichannel" for channels with equivalent hydraulic diameters of 0.5–3 mm.

These small channel dimensions (indicatively $d_h < 3$ mm) and channel geometries different from the circular one influence the gas–liquid two-phase flow characteristics. Flow regimes ranges severely depend on channel hydraulic diameter. Rarely studies developed for larger diameter and different shape tubes can be applied and extended to minichannels.

A state-of-the-art on flow regimes, pressure drop and heat transfer during condensation of refrigerants inside minichannels can also be found in Cavallini [2].

Zhang and Webb [19] and Webb and Ermis [15] presented experimental data and correlations for smooth and microfinned minichannels. From Webb's experimental data it results that, with R134a condensing at 65 °C, at vapour quality $x = 0.5$ and specific mass flow rate $G = 600$ kg/(m^2 s), $-(\mathrm{d}p/\mathrm{d}z)_f$ is 24000 Pa/m for a multichannel with hydraulic diameter equal to 1.33 mm and rectangular channel cross section. With the same flow rate flowing in a single circular tube with $d = 4.52$ mm (the same cross sectional area and then the same mass velocity) the frictional pressure gradient is $-(\mathrm{d}p/\mathrm{d}z)_f = 4500$ Pa/m. In the first case the saturation temperature gradient is 0.6 °C/m, while in the second case is 0.1 °C/m. Parallel flow refrigerant circuiting with decreasing number of tubes in each pass is then used in equipment with minichannels, so as to contain pressure drop, since the pressure gradient diminishes with vapour quality.

Zhang and Webb [19] interpolated their own experimental data and obtained a new correlation:

$$\Phi^2_{LO} = (1-x)^2 + 2.87x^2 p_R^{-1} + 1.68x^{0.8}(1-x)^{0.25} p_R^{-1.64} \qquad (4)$$

to be applied when $1.0 < d_h < 7.0$ mm and $p_R > 0.2$. Webb and Ermis [15] suggested to apply Eq. (4) even to micro-finned channels of their data bank, implying a widening of the hydraulic diameter validity range, that becomes $0.44 < d_h < 7.0$ mm.

Coleman [5] presented many experimental measurements of pressure drop of R134a condensing in different tubes and minichannels, and various correlations were estabilished, one for each geometry and for each flow regime. Niño et al. [13] measured pressure drop during adiabatic two-phase flow in minichannels with R134a, R410A and air–water.

More recently, Yu et al. [17] measured two-phase pressure drop in a 2.98 mm diameter horizontal circular tube with water, finding that the Chisholm correlation Chisholm [4] consistently overpredicted their data.

Similarly, Kawahara et al. [9] found that the Chisholm correlation overpredicted their pressure drop data taken with water and nitrogen gas flowing in a 0.1 mm diameter circular tube.

Koyama et al. [10] investigated the pressure drop characteristics of R134a in four types of multiport minichannel tubes. They found that the Friedel correlation was able to predict well their data, except at low mass velocity. They also developed a new model for frictional pressure drop based on the Mishima and Hibiki [11] correlation.

Yun and Kim [18] investigated two-phase pressure drops of CO_2 inside minitubes with 2.0 mm and 0.98 mm inner diameters. They proposed to modify the Chisholm parameter of the Lockhart and Martinelli model by replacing the constant C in Eq. (3) with a function of the diameter.

In Cavallini [2] pressure drop correlations for minichannels are reviewed, with comparisons between models and experimental data taken by independent researchers. The Friedel [6,7] correlation and the correlation by Zhang and Webb [19] were found to give the best prediction for the available experimental data. Other correlations have been developed by Tran et al. [14], Chen et al. [3], Mishima and Hibiki [11].

A comparison between values predicted by the Friedel equation and experimental data (796 data points) by independent researchers was also reported in Cavallini et al. [1]: the average deviation e_R was +16% and the absolute mean deviation e_A was 38%. The experimental frictional pressure gradient collected by Zhang and Webb and Webb and Ermis in smooth and microfinned multichannels were overestimated by the Friedel equation, with a mean deviation of 52%. Most of Coleman's data points were estimated within ±30%, while Niño et al. data, at lower mass velocities, were estimated with a mean deviation around 64%.

The comparison between experimental data and prediction from the Zhang and Webb [19] equation gave an average deviation e_R of −21% and an absolute mean deviation e_A of 31%. The same correlation tends to underestimate Coleman [5] data, while it does not estimate accurately Niño et al. [13] data. Almost all Coleman's channels, though, present $d_h < 1$ mm.

As already observed, most of the experimental data were taken during two-phase flow of R134a. More experimental work is certainly needed for a further appraisal of the above correlations, above all with the so called high pressure refrigerants, such as R410A and R32.

3. Experimental test rig

The pressure drop data is obtained on the experimental facility depicted in Fig. 2, which consists of the

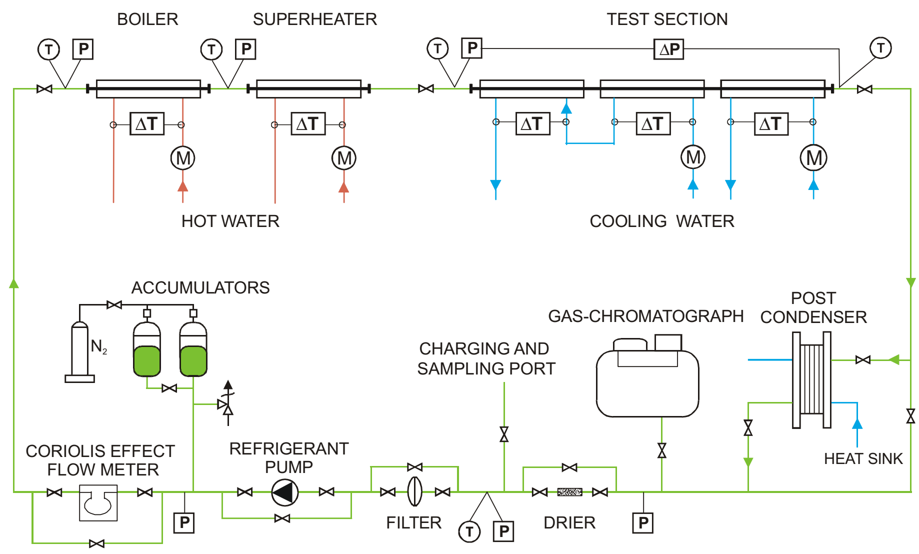

Fig. 2. Experimental test rig.

refrigerant loop, the hot water loop and the cooling water loop.

The refrigerant flowing in the primary loop is vapourised and superheated by hot water flowing in the boiler and superheater. Then it is cooled and partially condensed in the first sector of the test section.

The refrigerant flow can be independently controlled by the magnetic gear pump. The refrigerant mass flow rate is measured by a Coriolis effect mass flow meter inserted downstream of the pump, having an accuracy of 0.4% of the measured value.

All measurements are taken in the test section, which is made of three counter-flow heat exchangers. Each of the three heat transfer sectors can be turned to an adiabatic sector when no secondary fluid flows in it. The test section is realised by means of one 1.8 m long multiport tube and three separated 0.45 m long PVC jackets (Fig. 3). The test tube has 13 parallel channels and a major side length of 25.4 mm, a minor side length of 2 mm, a 0.3 mm wall thickness and a 1.4 mm hydraulic diameter. Two digital strain gauge pressure transducers (absolute and differential transducers) are connected to manometric taps to measure the vapour pressure upstream and downstream of the test tube.

In the test section used for the present experimental investigation, a chamber is installed before the multiport tube, to collect the fluid coming out from a round tube. The need for a good flow distribution requires that a single phase flow is present in the chamber. It means that superheated vapour enters the multiport test tube and, after a partial condensation in the first sector two-phase flow is present inside the channels.

In the first heat exchanger sector, the refrigerant condenses in the minichannels against the cooling water flowing in the jacket. This first sector is used to achieve the desired value of vapour quality: the cooling water flows on both sides of the multiport tube, within the PVC jackets.

The pressure drop is obtained by measuring the saturation temperature drop in the tube, between the inlet of the second sector and the outlet of the third sector (Fig. 3). This measurement is performed by means of one T-type thermopile and two T-type thermocouples fixed to the aluminium tube between adjacent PVC jackets.

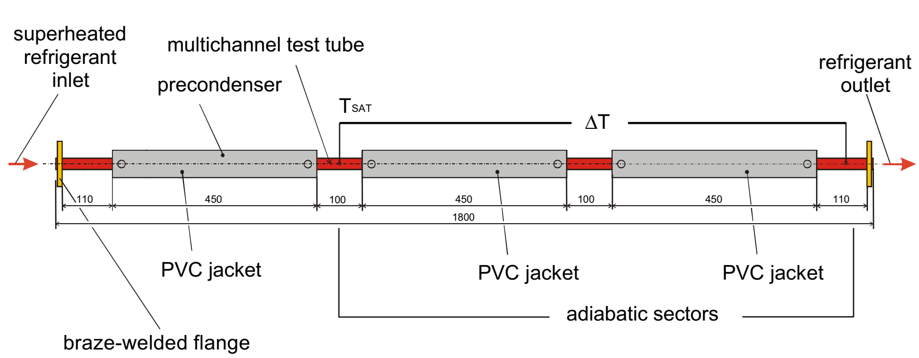

Fig. 3. Experimental test section (measures in millimeters).

The measured pressure drop refers to a 1.13 m long tube.

During two-phase pressure drop tests, no heat transfer is present in the other two sectors, which means that runs are adiabatic. Nevertheless, the test section allows to collect pressure gradient data during diabatic flow too when needed.

The thermopile used in the measurement presents an experimental uncertainty equal to ±0.03 K, which leads to an experimental nominal uncertainty of ±0.71 kPa/m for R134a at 40 °C saturation temperature, ±0.28 kPa/m for R236ea and ±1.6 kPa/m for R410A at the same average saturation temperature.

4. Experimental pressure drop

The two-phase pressure drop tests are carried out inside the 1.4 mm hydraulic diameter multiport minichannel tube with R236ea, R134a and R410A at 40 °C saturation temperature, corresponding to a reduced pressure of 0.096, 0.25 and 0.49, respectively.

The vapour quality entering the test section (x_{in}) is calculated from an energy balance on the first sector. Since the runs are adiabatic, no significant vapour quality change occurs in the tube, except for the vapour quality variation due to the pressure drop. The vapour quality variation due to different pressure between inlet and outlet is around 1–2% for R410A and R134a, while it reaches up to 5% in the case of R236ea.

The first experimental tests were carried out with R134a at mass velocity ranging from 400 to 1000 kg/(m^2 s). The tests were performed at three values of vapour quality: 0.25, 0.5 and 0.75, with variations of 0.01–0.02 from the desired vapour quality.

Experimental values are reported in Fig. 4. The nominal uncertainty of the pressure gradient goes from 1% to 6%, with an average uncertainty of 1.7%. As one could expect, the pressure gradient increases with vapour quality and mass velocity.

The R410A tests were carried out at around 0.25, 0.5 and 0.75 vapour qualities, at mass velocity ranging from 600 to 1400 kg/(m^2 s). The experimental pressure gradients are plotted vs mass velocity at constant vapour quality in Fig. 5. R410A pressure gradients are significantly lower than R134a ones at the same operating conditions. That leads to a higher experimental uncertainty for the R410A data, which ranges from 2% to 18%, with an average uncertainty of 6.4%.

Pressure drop tests of R236ea were conducted at mass velocity ranging from 200 to 600 kg/(m^2 s), at constant vapour quality. Although the target values of vapour quality are the same as for the other two fluids, i.e. 0.25, 0.5 and 0.75, the effective experimental value of the vapour quality can differ up to 5% from those. The experimental conditions and results are presented in Fig. 6, where the pressure gradient is reported vs vapour quality at constant mass velocity. The same R236ea data points are also plotted vs mass velocity in Fig. 7. The nominal uncertainty of the pressure gradient goes from 0.2% to 2.9%.

One could easily compare the pressure drop behaviour of the three fluids in Fig. 8. At 600 kg/(m^2 s) and 0.5 vapour quality, R410A presents a pressure gradient of 19.5 kPa/m vs 53.0 kPa/m of pressure gradient measured during R134a flow at the same operating conditions. At the same mass velocity and slightly higher

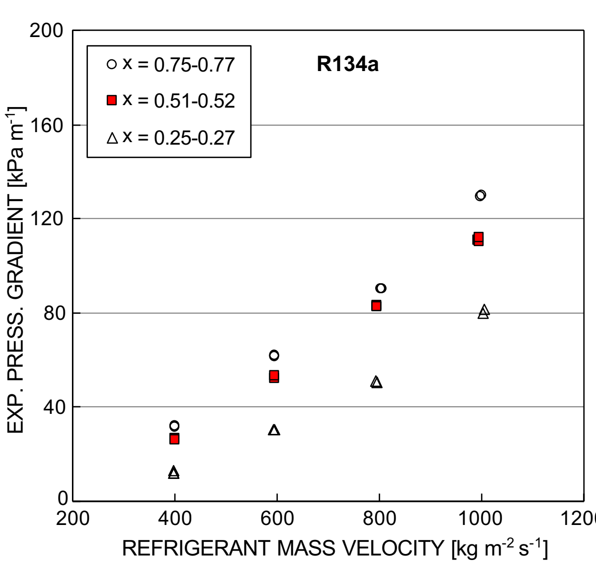

Fig. 4. R134a experimental frictional pressure gradient vs mass velocity.

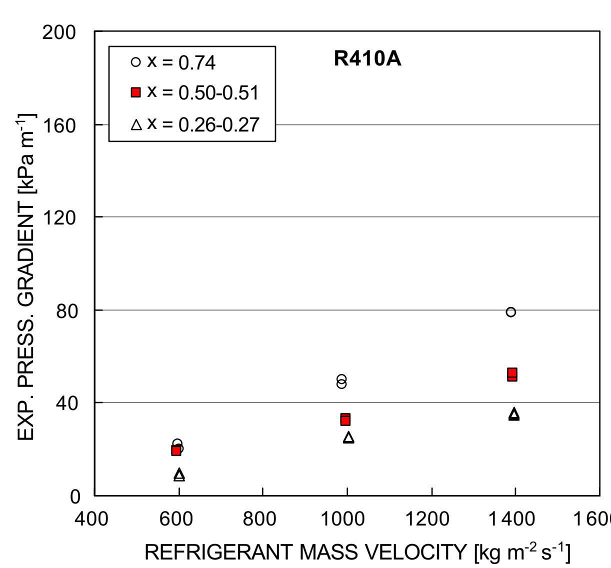

Fig. 5. R410A experimental frictional pressure gradient vs mass velocity.

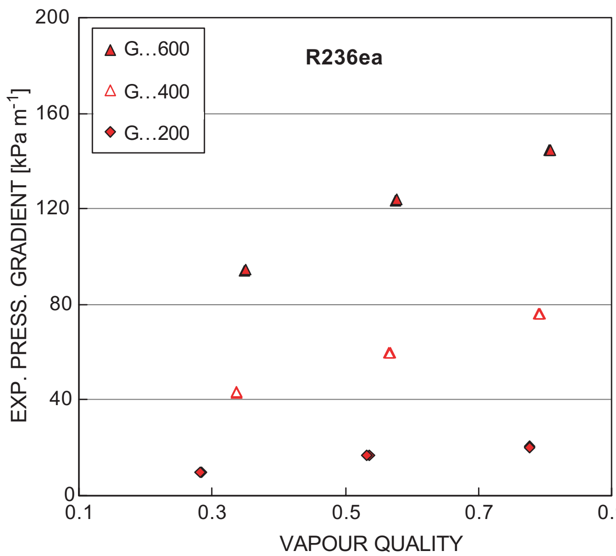

Fig. 6. R236ea experimental frictional pressure gradient vs vapour quality.

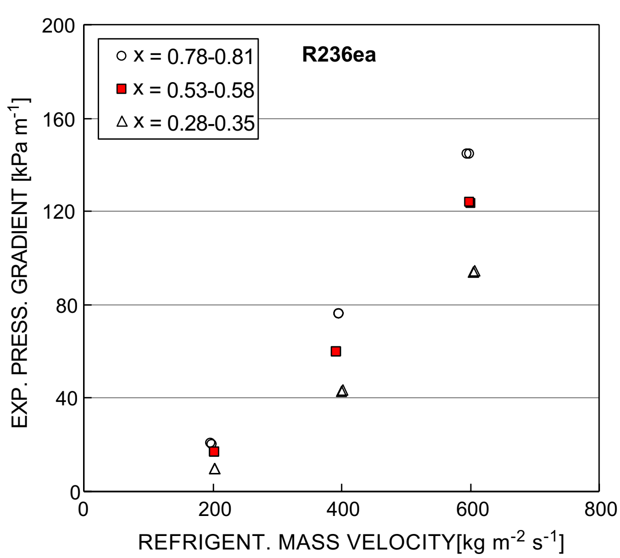

Fig. 7. R236ea experimental frictional pressure gradient vs mass velocity.

vapour quality ($x = 0.58$), R236ea presents a pressure gradient of 123.6 kPa/m. It comes out that the pressure drop behaviour keeps from using R236ea (low pressure fluid) at high mass velocity in a multiport minichannel application. For this reason, the maximum value of mass velocity in the R236ea tests was chosen equal to 600 kg/(m^2 s).

With regard to the other two fluids, at 1000 kg/(m^2 s) and 0.5 vapour quality, the frictional pressure gradient of R410A and R134a are 33.3 kPa/m and 112.5 kPa/m, respectively. This difference is mainly due to the different values of the vapour phase density of the two fluids: at 40 °C saturation temperature, the R410A vapour density is more than double as compared to R134a. The lower frictional pressure drops measured for R410A should be exploited in the design process of a condenser to get higher performance by increasing the mass velocity in the tubes. Of course, the comparison among the performances of the fluids should also account for their different heat transfer characteristics.

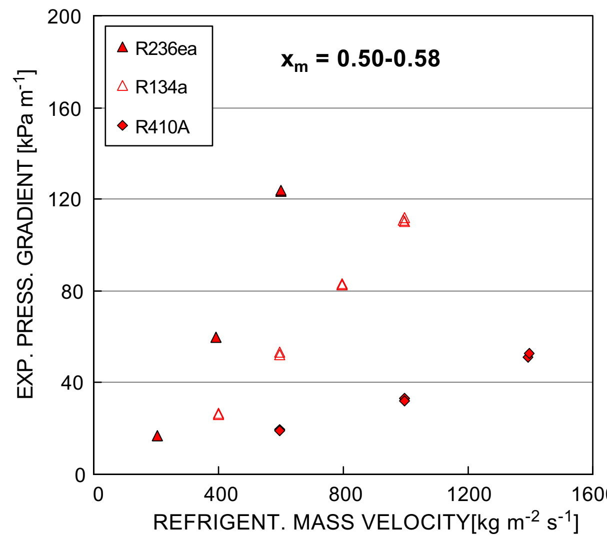

Fig. 8. Experimental frictional pressure gradient vs mass velocity at constant vapour quality for the three refrigerants.

5. Comparison with models

Experimental data are compared against models available in the literature for predicting two-phase frictional pressure gradient. Some of the models are well-known and were developed for macroscale channels, some others were specifically developed from data taken inside minichannels and small diameter tubes.

The first comparison shows the predicted values given by the Lockhart and Martinelli correlation vs the experimental values (Fig. 9). The prediction is clearly not satisfactory for all fluids.

An interesting tip about correlating two-phase pressure drop can be seen from Figs. 10–12, where the two-phase multiplier Φ_L^2 is plotted against the Martinelli parameter X. For R134a the friction multiplier collapse into a single curve for all experimental conditions. The Martinelli–Chisholm method suggests using $C = 12$–20 for R134a data points, while the experimental values lie between curves for $C = 5$ and 10. When looking at R236ea two-phase multiplier, one can see that different curves can be plotted corresponding to the different values of mass velocity. With reference to Eq. (3), R236ea data points call for a higher value of the constant C when increasing mass velocity: the experimental values lie between the curves $C = 5$ and 20, while according

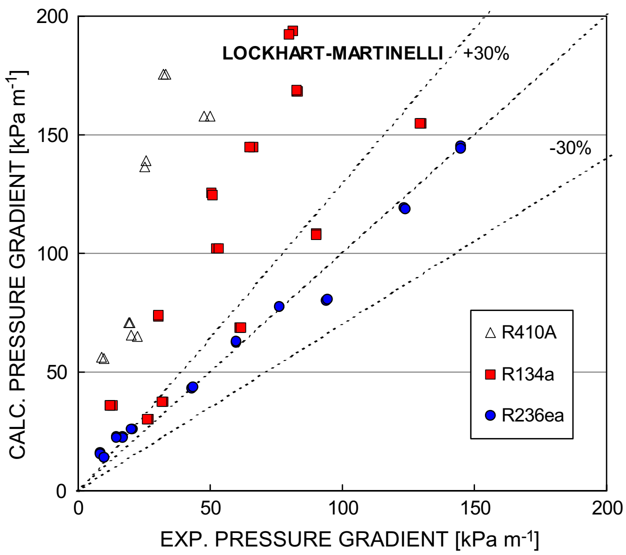

Fig. 9. Calculated vs experimental pressure gradient: the Lockhart and Martinelli correlation.

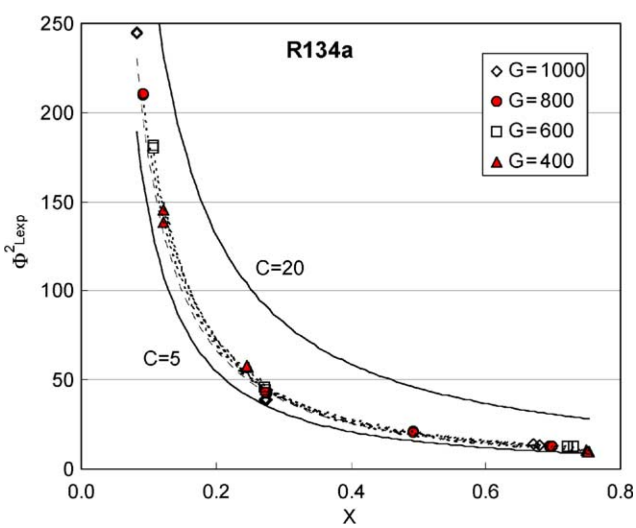

Fig. 10. R134a experimental two-phase multiplier vs Martinelli parameter.

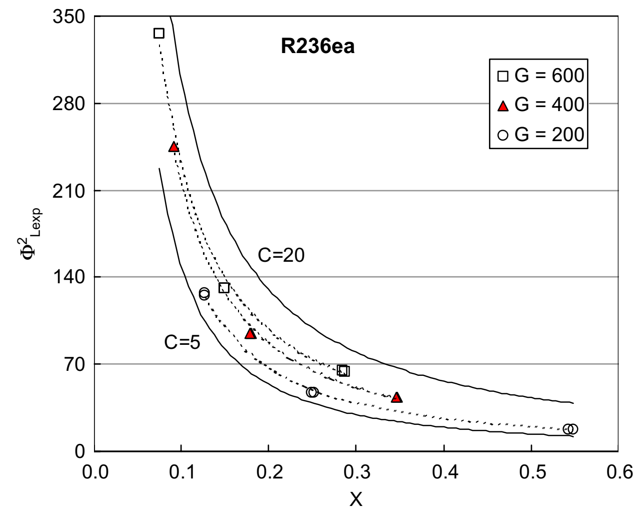

Fig. 11. R236ea experimental two-phase multiplier vs Martinelli parameter.

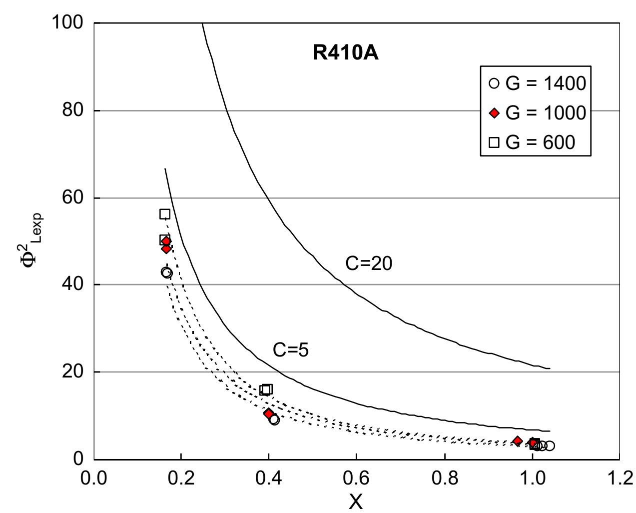

Fig. 12. R410A experimental two-phase multiplier vs Martinelli parameter.

to Chisholm *C* should be to 12. With regard to mass velocity, R410A presents an opposite trend to the one of R236ea. R410A data call for *C* values below 5, while the correlation suggests using 20. The above graphs suggest that the *C*-value in the correlation for pressure drop in minichannels should more effectively account for the reduced pressure.

The comparison with the Friedel correlation is presented in Fig. 13. This model is able to predict the R134a and R236ea data with satisfactory agreement. Recently, Koyama et al. [10] found a similar result when comparing their R134a data to the Friedel correlation. On the other hand, this model does not reproduce the experimental trend of the R410A data, giving a severe overprediction of the experimental values.

Chen et al. model underestimates R134a and R236ea data while it overpredicts R410A data (Fig. 14). All three sets of data are overpredicted by the Tran et al. correlation (Fig. 15), which was originally developed for low values of the reduced pressure ($p_R < 0.2$).

Fig. 16 shows the comparison of experimental pressure gradients to the Mueller-Steinhagen and Heck [12] correlation: the model is able to catch the experimental trend for the R134a and R236ea data sets, while it severely overpredicts R410A data.

The Zhang and Webb correlation was developed for small diameter tubes, as already stated. This correlation is in good agreement with R134a experimental pressure gradients (Fig. 17), with a relative deviation of −1.8% and a standard deviation of 11.5%. With regard to the R410A data, the experimental values are 30%

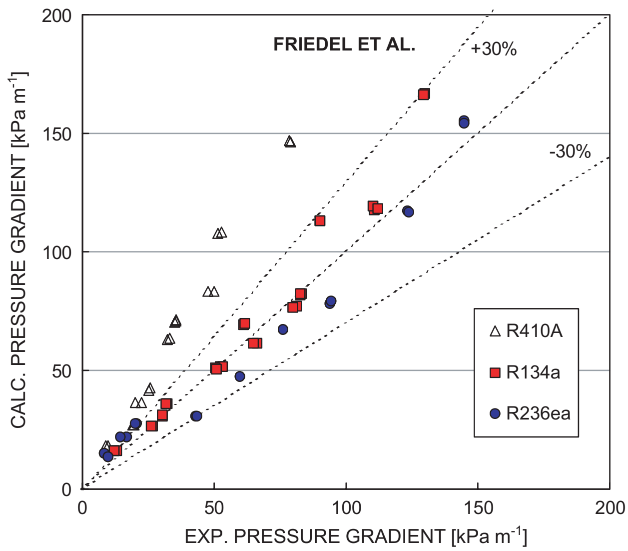

Fig. 13. Calculated vs experimental pressure gradient: the Friedel et al. correlation.

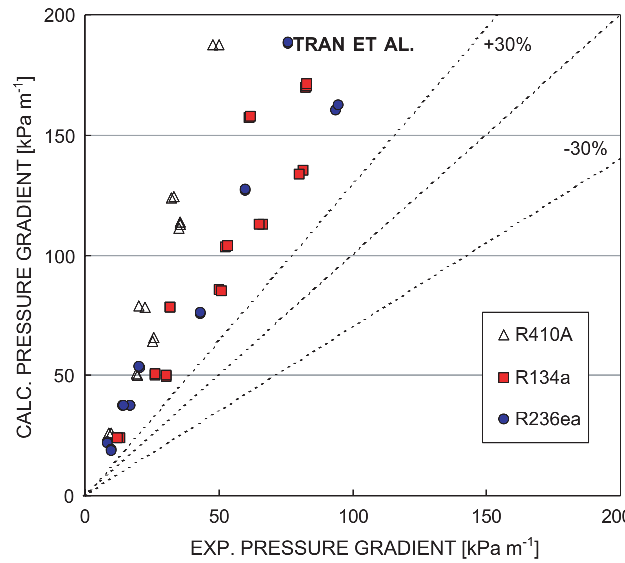

Fig. 15. Calculated vs experimental pressure gradient: the Tran et al. correlation.

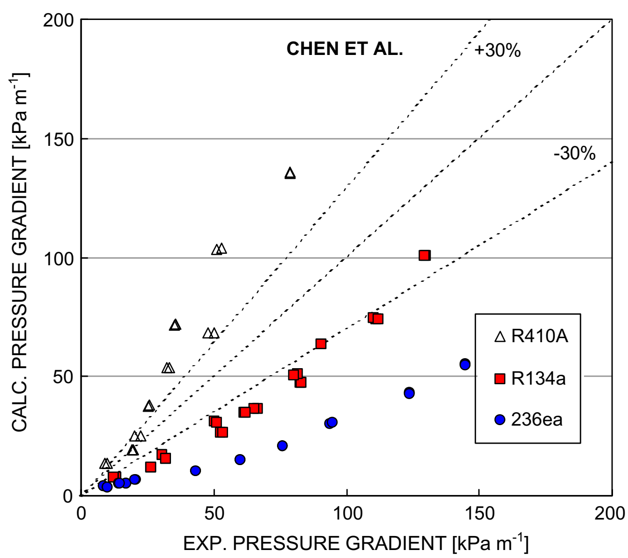

Fig. 14. Calculated vs experimental pressure gradient: the Chen et al. correlation.

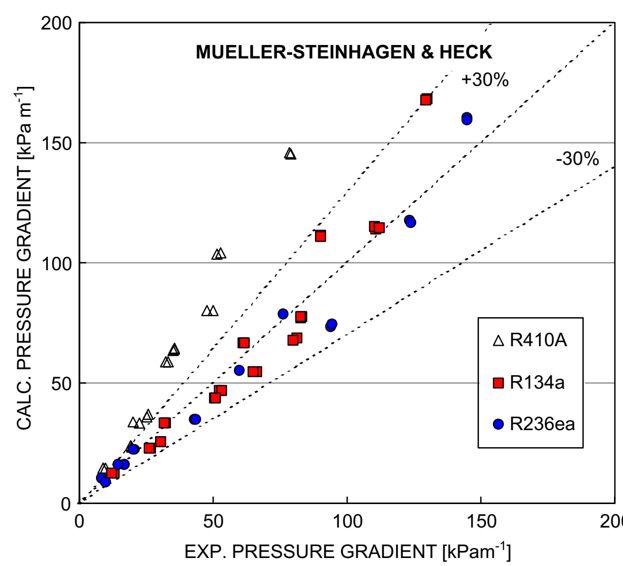

Fig. 16. Calculated vs experimental pressure gradient: the Mueller-Steinhagen and Heck correlation.

overpredicted by the model. R236ea data are also overpredicted by this model, which was originally developed for reduced pressure values higher than 0.2, thus excluding R236ea data points.

The correlation by Mishima and Hibiki is compared to data in Fig. 18, showing its ability to catch very well the experimental trend of R134a data. The R410A data set is overpredicted. In the Mishima and Hibiki method, the frictional pressure gradient is obtained by means of Eq. (3) where the constant C is given as a function of the tube hydraulic diameter only. No dependence on the reduced pressure is accounted for in the value of C.

In the comparison with the Yan and Lin [16] correlation, in Fig. 19, the experimental pressure gradient is well predicted at high mass velocity for R134a and R410A, but it is generally overpredicted at lower values of mass velocity. This model does not catch the experimental trend for the tested fluids.

A summary of the comparison of experimental data with models is given in Table 1. As one can see from the table and from the graphs, no model shows to be able to predict experimental data points with satisfactory agreement for all the three fluids.

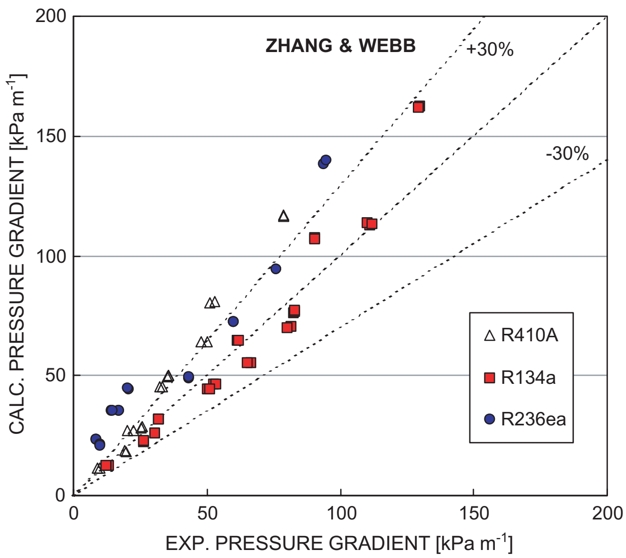

Fig. 17. Calculated vs experimental pressure gradient: the Zhang and Webb correlation.

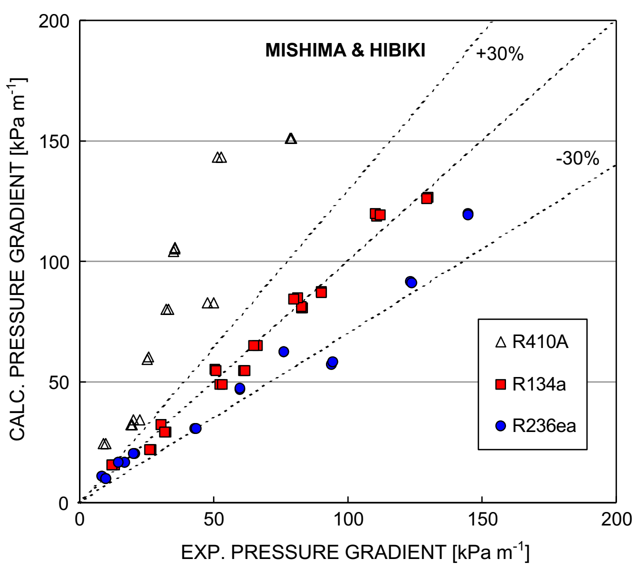

Fig. 18. Calculated vs experimental pressure gradient: the Mishima and Hibiki correlation.

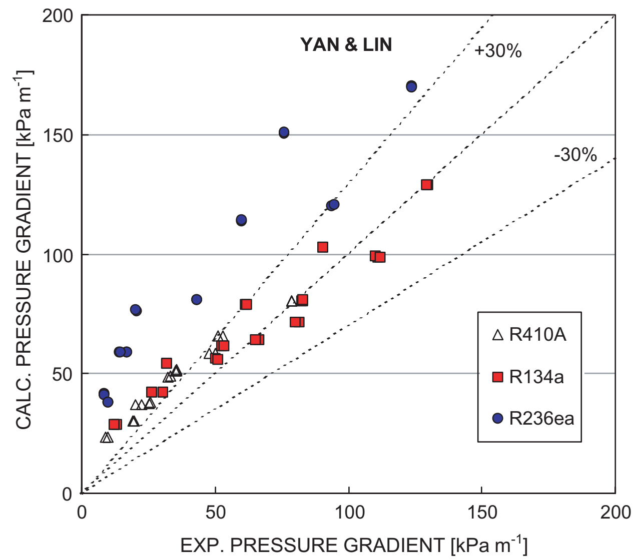

Fig. 19. Calculated vs experimental pressure gradient: the Yan and Lin correlation.

6. Conclusions

The frictional pressure drop in a multiport minichannel tube is measured during adiabatic two-phase flow of R134a, R236ea and R410A. The choice of these three fluids allows to vary an important parameter such as the reduced pressure in the range 0.1–0.5. As expected, R410A presents a significantly lower pressure drop in comparison with R134a and R236ea at the same operating conditions. The low pressure fluid R236ea shows the highest pressure gradient among the three fluids.

The experimental data are compared against several models available in the literature, finding that the correlations by Friedel et al., Zhang and Webb, Mishima and Hibiki, and Mueller-Steinhagen and Heck are in good agreement with the R134a experimental data. The R236ea data are in good agreement with the predictions by Mueller-Steinhagen & Heck, showing the lowest deviations, although the model is able to catch the

Table 1
Average deviation e_R and standard deviation σ_N calculated for the different models with R134a, R410A and R236ea data

Fluid model	R236ea		R134a		R410A	
	e_R (%)	σ_N (%)	e_R (%)	σ_N (%)	e_R (%)	σ_N (%)
Friedel et al.	14.9	34.7	8.3	12.5	82.6	21.4
Zhang and Webb	85.0	55.4	−2.4	12.3	31.1	18.0
Mishima and Hibiki	−9.3	20.8	1.2	11.6	127.1	51.8
Yan and Lin	175.7	130.5	28.0	44.0	52.9	42.6
Chen et al.	−66.3	6.9	−40.2	8.4	57.0	34.6
Tran et al.	128.2	35.5	113.6	44.6	243.7	64.3
Mueller-Steinhagen and Heck	−0.1	13.9	−1.7	14.1	66.8	22.7
Lockhart-Martinelli	22.6	32.0	93.3	61.9	392.3	142.1

$$e_i = \frac{\left(\frac{dp}{dz}_{CALCi} - \frac{dp}{dz}_{EXPi}\right)}{\frac{dp}{dz}_{EXPi}} \cdot 100, \quad e_R = \frac{1}{Np} \cdot \sum_{i=1}^{Np} \frac{\left(\frac{dp}{dz}_{CALCi} - \frac{dp}{dz}_{EXPi}\right)}{\frac{dp}{dz}_{EXPi}} \cdot 100, \quad \sigma_N = \sqrt{\frac{\sum_{i=1}^{Np}(e_i - eR)^2}{Np-1}}.$$

experimental trend only for average values of vapour quality.

The above models show not to be not accurate when they are compared against R410A data: all the correlations tend to overpredict the experimental values. Among the models, the correlation by Zhang and Webb presents the lowest deviation, with an average deviation of 32% and a standard deviation of 18%.

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