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FLOW BOILING HEAT TRANSFER CHARACTERISTICS OF R-134A IN HORIZONTAL AND VERTICAL MINI-CHANNELS

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ABSTRACT

Heat transfer experiments are carried out to obtain the data of R-134a during flow boiling in a circular mini-channel having a diameter of 1.75 mm and a length of 600 mm. The DC power supply generating 80 A at 12 V is used to apply heat to the test section. T-type thermocouples are installed at the inlet and outlet of the test section to measure the system temperature. The 10 thermocouples are installed on the top and bottom sides at equal distances along the tube to measure the wall temperature. Variable area type flow meters calibrated specially in the range 0.02-0.2 LPM for R-134a by the manufacturer are used to measure the refrigerant flow rate. A transparent tube is installed to match up with the test section outlet and to serve as a viewing window for visualisation. Two sets of the horizontal flow and vertical upward flow data are analysed. Flow regime map and heat transfer coefficient tend to be dependent with flow direction under the certain experimental conditions.

INTRODUCTION

Flow boiling heat transfer in micro/mini-channel systems is attractive to researchers worldwide because of the utilization of latent heat of vaporization in their compact sizes. The advantages of using these small channels are also summarized in Ribatski et al. [1]. Definitions for various channels have been given by several investigators. For instance, arbitrary channel classifications associated with the hydraulic diameter D_h have been proposed. Mehendale et al. [2] employed the hydraulic diameter as an important parameter for defining heat exchangers and Kandlikar [3] proposed criteria for small flow channels used in engineering applications. However, such criteria cannot relate the channel diameter to heat transfer and fluid flow behaviors. Regarding the experimental data for two-phase gas-liquid flow through small channels, Chung and Kawaji [4] found from compiling flow characteristics including flow pattern, void fraction and pressure drop that diameters between 100 and 250 μm seemed to be in the range for mini-to-micro-scale threshold diameter. Their findings were also confirmed by Saisorn and Wongwises [5]. According to existing criteria which are unclear, further exhaustive experimental investigations are still required to meet a more general definition dealing with channel classification.

Compared with the reported two-phase flow and heat transfer characteristics in ordinarily sized channels, which are available in a relatively large number of publications, flow boiling phenomena in mini- and micro-channels tend to show different behaviors due to the effects of the limited and confined space. Although some relevant information is currently available in the literature, a complete understanding has not yet been clarified with regard to the trends and parameters dominating the phase-change heat transfer mechanism in these small-scale channels.

In what follows, recent studies associated with flow boiling heat transfer in small channels are briefly outlined.

Huo et al. [6] studied experimentally boiling heat transfer of R-134a fluid in small vertical tubes of 2.01 and 4.26 mm in diameter. In the low vapour quality range, the heat transfer coefficient in both tubes increased with increasing heat flux and saturated pressure but was independent of vapour quality. These results were attributed to nucleate boiling being the dominant heat transfer mode. In other ranges of vapour quality, however, the dominant heat transfer mode was not addressed as a result of inconsistency in the experimental data. Under the same controlled conditions, they found that the nucleate boiling heat transfer coefficient was higher for the 2.01 mm tube than for the 4.26 mm tube.

Heat transfer of R-134a refrigerant during flow boiling in horizontal tubes with different diameters including 0.51, 1.12 and 3.1 mm was studied experimentally by Saitoh et al. [7]. Nucleate boiling was reported in the low vapour quality region whereas convective evaporation was dominant in the high vapour quality region. The latter mechanism was found to be less dominant as the tube diameter decreases. The smaller the tube diameter, the higher is the effect of saturation temperature on the heat transfer coefficient. The flow instability was also discussed in this work.

Choi et al. [8] reported the heat transfer characteristics of CO_2 through horizontal mini-channels having diameters of 1.5 and 3 mm. They indicated that nucleate boiling was predominant in the low vapour quality region and a convective boiling heat transfer contribution appeared in moderate and high vapour quality regions. The variation of local heat transfer coefficient with heat flux, mass flux, vapour quality and saturation temperature was discussed. A more active nucleate boiling was addressed when the tube with smaller diameter was used. Flow boiling heat transfer of different refrigerants for horizontal flow was continually carried out by Choi et al. [9]. They indicated that the use of CO_2 caused the heat transfer coefficient to be higher than that of R-134a and R-22.

Shiferaw et al. [10] compared their flow boiling data with existing correlations. The data points were obtained from experiments with R-134a flowing through vertical small tubes having diameters of 4.26 and 2.01 mm. The comparisons revealed that the existing correlations did not predict well their data. Comments and suggestions were provided by the authors for further development of the prediction. Similar experiments were conducted by Shiferaw et al. [11] to compare the results obtained from a vertical tube having a diameter of 1.1 mm with the three-zone flow boiling model developed by Thome et al. [12] and Dupont et al. [13]. Generally, the model predicted well the experimental data, especially at relatively low pressure. Regarding the heat transfer characteristics, an insignificant influence of mass flux and vapour quality was observed while the heat transfer coefficient increased with increasing heat flux and saturation pressure.

Three different refrigerants, R-134a, R-236fa and R-245fa, flowing in horizontal direction were tested for flow boiling in a 1.03 mm diameter tube by Ong and Thome [14]. The trends of the data were investigated, showing that the heat transfer coefficient depended on heat flux at low vapour qualities and on mass flux at high vapour qualities. Regarding the refrigerants tested at low vapour qualities, R-134a gave the highest heat transfer coefficient followed by R-236fa and R-245fa, respectively.

The recent studies, previously mentioned, mainly explored the heat transfer mechanisms in small channels with various diameters. Different heat transfer behaviors were obtained based on that the experimental conditions were varied except the channel orientation in which the important information is still lacking. The present experimental work is therefore carried out to compare the heat transfer results obtained from the horizontal channel with those from the vertical upward flow under the same test conditions. The two-phase flow patterns and heat transfer coefficients are also presented in this paper.

NOMENCLATURE

Bo	=	Bond number
Co	=	confinement number
D_b	=	nominal bubble size (m)
D_h	=	hydraulic diameter (m)
G	=	mass flux (kg/m^{2}s)
g	=	gravitational acceleration (m/s^2)
h	=	heat transfer coefficient (W/m^{2}K)
q	=	heat flux (W/m^2)
Re	=	Reynolds number
T	=	temperature (^{o}C)
x	=	vapour quality
Greek symbols		
μ	=	dynamic viscosity (Ns/m^2)
ρ	=	density (kg/m^3)
σ	=	surface tension (N/m)
Subscripts		
avg	=	average
G	=	vapour phase
L	=	liquid phase
sat	=	saturation
wall,in	=	inner wall

EXPERIMENTAL APPARATUS AND PROCEDURE

The experimental apparatus, developed by Kaew-On and Wongwises [15] as shown schematically in Fig. 1, is used to investigate flow boiling of R-134a in a circular mini-channel. The test section is attached to the rigid frame having adjustable connector which is served for channel orientation purpose. Horizontal flow and vertical upward flow are conducted in current work. The main components of the system include a test section, refrigerant loop, sub-cooling loop, and a data acquisition system

For the refrigerant circulating loop, as seen in Fig. 1, liquid refrigerant is pumped by a gear pump which can be regulated by means of an inverter. The refrigerant then passes in series through a filter/dryer, a refrigerant flow meter, pre-heater, sight glass tube, and enters the test section. The inlet quality before

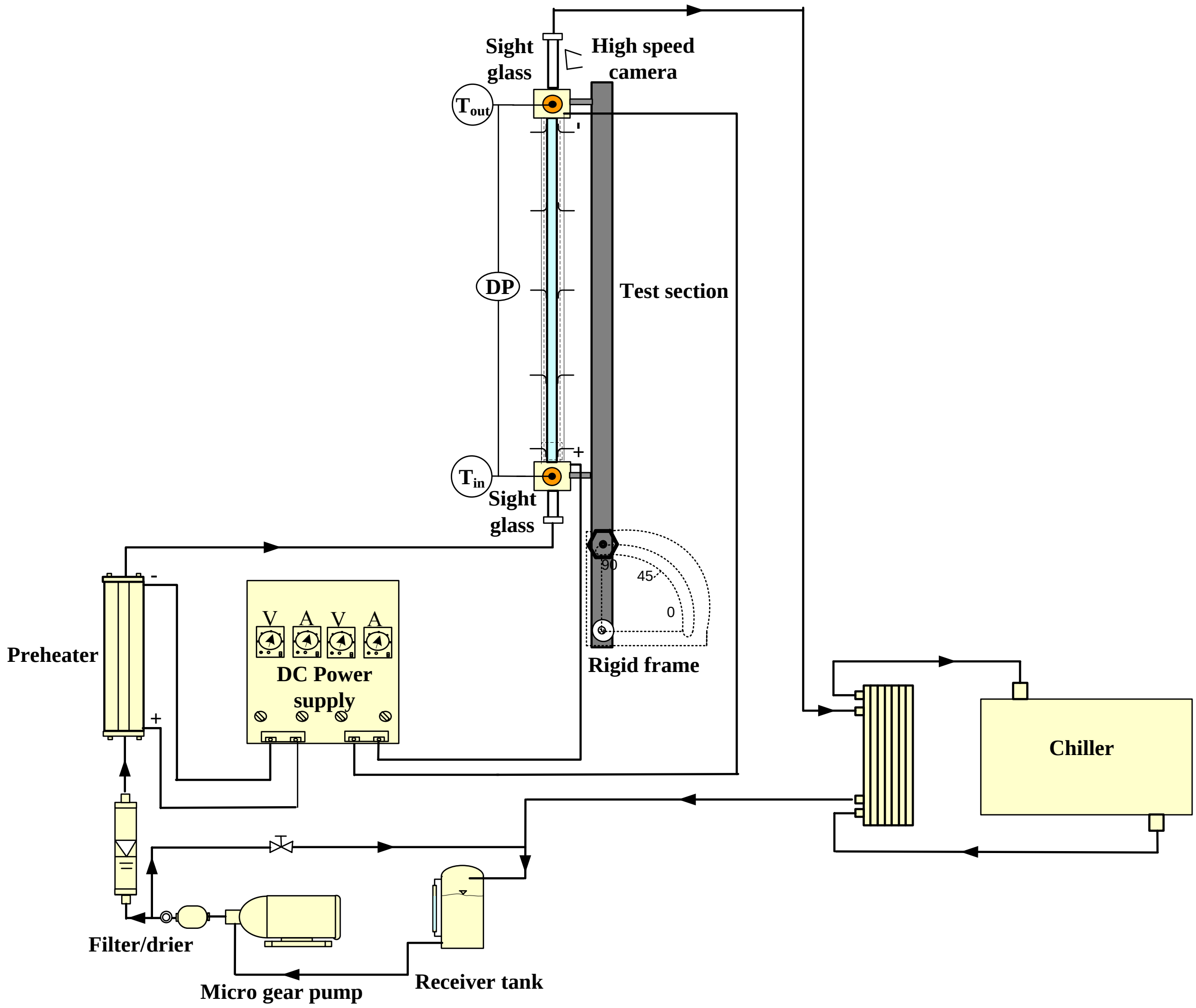

Fig. 1 Schematic diagram of experimental apparatus

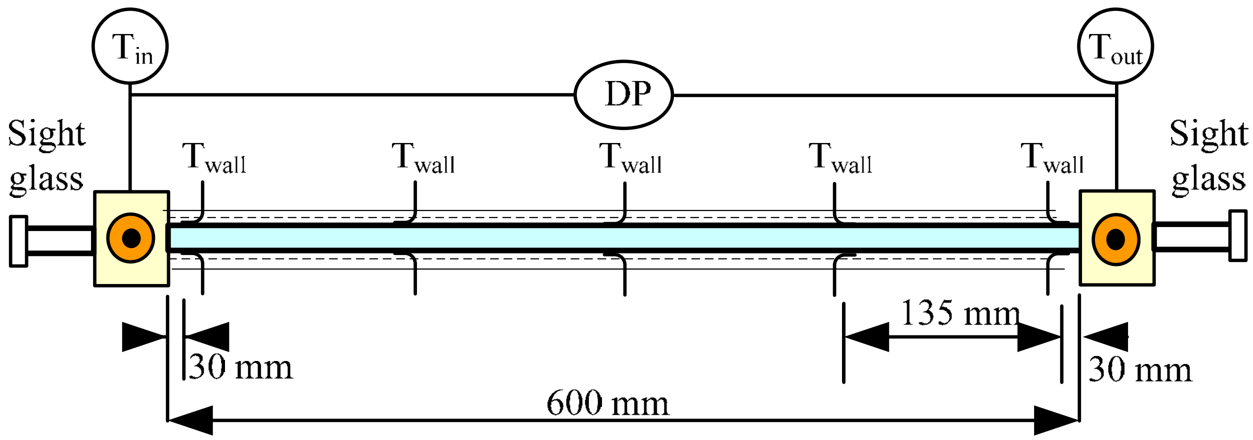

Fig. 2 Schematic diagram of test section

entering the test section is controlled by the pre-heater with a DC power supply used to apply heat, and the two-phase condition at the test section inlet can be observed through the well installed sight glass tube. Leaving the test section, the refrigerant vapour subsequently condenses in a sub-cooler and then is collected in a receiver; it eventually returns to the refrigerant pump to complete the cycle. Instrumentation is installed at various positions, as shown in Fig. 1, to monitor the state of the refrigerant. All the signals from the thermocouples and the differential pressure transducer are recorded by a data logger.

Fig. 2 shows a schematic diagram of the test section, a stainless steel tube which is 600 mm long and 1.75 mm in diameter. The regulated DC power supply generating 80 A at 12 V is used to apply heat to the test section. The voltage and electric current values are measured by a multimeter (Fluke 336 meter) which has an uncertainty of ±2% for voltage and ±1% for current. T-type thermocouples are installed at the inlet and outlet of the test section to measure the refrigerant temperatures. Since the two-phase conditions for flow boiling are provided in the test section during the experiments, the saturation pressures at the test section inlet and outlet are obtained based on the corresponding saturation temperatures. The 10 thermocouples are installed on the top and bottom sides at equal distances along the tube to measure the wall temperature. All thermocouples on the tube surface are fixed with special glue. The test section is well insulated by using rubber foam with a thermal conductivity of 0.04 W/mK. A variable area type flow meter is specially calibrated in the range 0.02–0.2 LPM for R-134a by the manufacturer. All thermocouples, differential pressure transducer and relevant instruments installed in the experimental apparatus are well calibrated.

In this work, the experiments are conducted in such a way that the heat applied to the test section is varied by small increments, while the refrigerant flow rate, system pressure, and inlet vapour quality in the test section are kept constant at the desired value. The system is allowed to approach a steady state before the flow pattern and relevant data are recorded. During the experiment, the temperature and pressure drop are continuously recorded along the test section by the data logger. The profiles of the temperature help us to know the limit of useful data because a large increase in wall temperature and outlet refrigerant temperature is observed when dry-out occurs. The experiments are performed in the range of 200-700 kg/m^{2}s for mass flux, 1-80 kW/m^2 for heat flux, and 8-10 bar for saturation pressure.

The heat transfer coefficient for R-134a during flow boiling in the test section is calculated by

$$h = \frac{q}{\left(T_{wall,in,avg} - T_{sat,avg}\right)} \tag{1}$$

For each position where the thermocouple is installed on the tube surface, the inner wall temperature at a given position is determined using the equations for steady-state one-dimensional heat conduction through the tube wall with internal heat generation. There are ten positions along the tube, at which the inner wall temperatures are determined and, hence, the average value of the inner wall surface temperature ($T_{wall,in,avg}$) of the test section is obtained using the arithmetic mean of the temperatures along the tube. $T_{sat,avg}$ represents the average temperature of the refrigerant at the test section inlet and outlet.

RESULTS AND DISCUSSION

In this section, the two-phase flow regime is presented at first and the heat transfer coefficient will be subsequently discussed.

The visual observation of flow patterns is carried out through a viewing window which is a transparent tube, installed to match up with the test section outlet. The flow regime map is developed based on the observed flow patterns which are registered by a high quality camera (Fujifilm FinePix S7000) having shutter speeds of 1/15 to 1/10000 s. The camera together with an adjustable light source, comprising mainly of 150 W halogen lamp and dimmer, are aligned across the viewing section. The alignment of the flow visualization system is normal to the viewing section.

Regarding the visual observation, the observed flow patterns include slug flow, throat-annular flow, churn flow, annular flow and annular-rivulet flow. Throat-annular flow and annular-rivulet flow are also indicated by Saisorn and Wongwises [16], and have never been observed in ordinarily sized channels. Fig. 3 illustrates the flow regime map obtained from the present data. As shown in the figure, the vertical upward flow results are compared with the transition lines for the horizontal flow data. The comparisons show that, at a saturation pressure of 8 bar, the channel orientation tends to have no significant effect on the two-phase flow regime. With a saturation pressure of 10 bar, however, it is found that the flow regime for the vertical upward flow is no longer compatible with that for the horizontal one. Since, as addressed in Kandlikar [17], flow boiling phenomena are affected by interactions among five major forces including surface tension, inertia, gravity, viscous and evaporation momentum forces, the relative effects of these forces under different saturation pressure may cause discrepancy in the boiling process. As the saturation pressure increases, the reduced surface tension force and the increase in evaporation momentum force, which is due to the lower latent heat of vapourization, accompanying with the gravitational effects, may result in the difference between the flow regime maps for such different channel orientations

In addition to flow regime results, the heat transfer coefficient data for corresponding channel orientations and flow patterns are presented in Figs. 4 and 5. For a saturation pressure of 8 bar as illustrated in Fig. 4, the flow patterns seem to influence the flow boiling heat transfer process. The figure indicates that heat transfer coefficient is relatively low for slug flow regime and continuously increases to the value corresponding to annular flow, but decreases thereafter when the annular-rivulet flow, which corresponds to local dry-out region, is established. According to the channel orientation as also presented in the figure, the heat transfer coefficients with regard to the vertical upward flow are higher than those for horizontal flow under a mass flux of 495 kg/m^2s, which associates with turbulent liquid region. It is noted from the

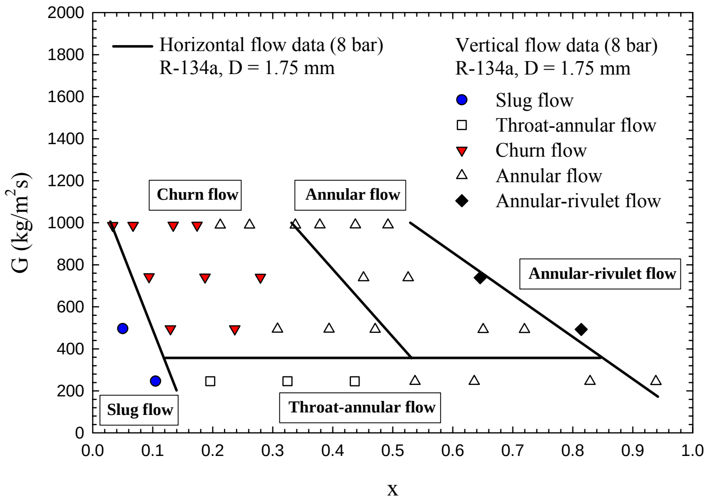

Fig. 3 Flow regime map

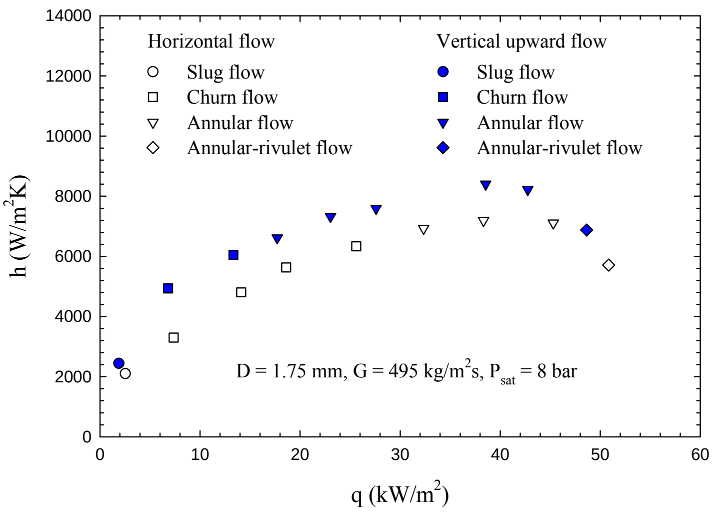

Fig. 4 Heat transfer coefficients for horizontal and vertical upward flows (8 bar)

experimental data covering mass flux range of 246-743 kg/m^2s that the heat transfer coefficients are almost in equal values in the two channel orientations under the laminar liquid flow taking place at a mass flux of 246 kg/m^2s. Nevertheless, the deviation is found to increase with mass flux. As mass flux increases, the increased inertia force contributes to reaching the macro-scale behavior for which the channel orientation becomes important. These observed characteristics should be examined by the macro-to-mini-channel criteria.

For boiling process, the confinement number was recommended by Kew and Cornwell [18] to be used as the
transition criterion for checking whether the micro-scale effects are important. The confinement number is defined as

$$Co = \frac{D_b}{D_h} \tag{2}$$

where D_h is hydraulic diameter and D_b is the nominal bubble size or Laplace constant which is expressed by

$$D_b = \sqrt{\frac{\sigma}{g(\rho_L - \rho_G)}} \tag{3}$$

The confinement number below 0.5 stands for ordinarily sized channel. In the present work, the confinement number for all mass flux values under a saturation pressure of 8 bar is slightly above 0.5, presumably indicating a mini-channel flow where gravitational effects should be insignificant. This criterion only agrees well with the results obtained at a mass flux of 246 kg/m^2s.

Interestingly, the macro-scale behavior, which is observed at mass flux beyond 246 kg/m^2s, are exactly indicated by the criterion recently developed by Li and Wu [19]. In addition to surface tension and gravitational forces taken into account in the Kew and Cornwell's criterion [18], the viscous and inertia forces were considered by Li and Wu [19]. The proposed criterion, which is the combined non-dimensional parameter ($Bo\,Re_L^{0.5}$), was established by compiling the database extracted from the literature. Bo stands for Bond number which is determined by

$$Bo = \frac{g(\rho_L - \rho_G)D_h^2}{\sigma} \tag{4}$$

The liquid Reynolds number, Re_L is expressed as

$$Re_L = \frac{G(1-x)D_h}{\mu_L} \tag{5}$$

When $Bo\,Re_L^{0.5} \le 200$, the micro-scale effects dominate whereas the macro-scale phenomena take place in the region $Bo\,Re_L^{0.5} > 200$. The values of $Bo\,Re_L^{0.5}$ are 162.02, 235.98 and 303.76 for mass flux values of 246, 495 and 743 kg/m^2s, respectively, which is in agreement with the present results.

Fig. 5 presents the experimental results for a saturation pressure of 10 bar. Compared with the results shown in Fig. 4, the heat transfer coefficient is lower than that for the saturation pressure of 8 bar across the experimental range of heat flux. This may be attributed to the fact that, with an increase in saturation pressure, which corresponds to the smaller latent heat of vaporization, the lower liquid viscosity can contribute the thinner liquid film on the tube wall to become easily broken

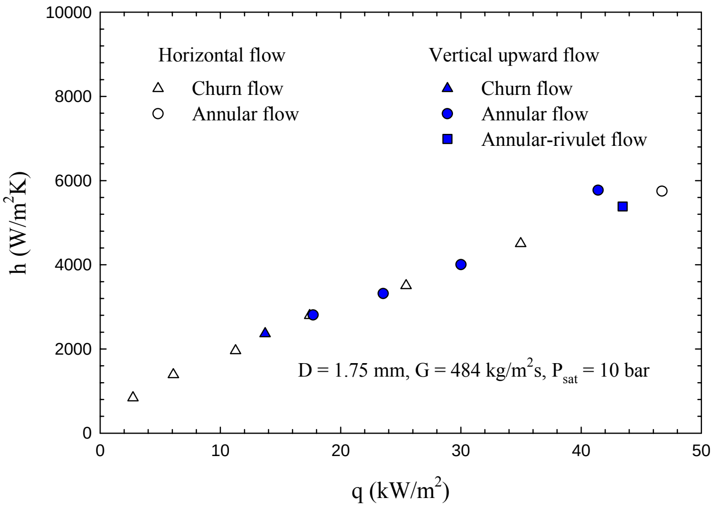

Fig. 5 Heat transfer coefficients for horizontal and vertical upward flows (10 bar)

and, hence, a decrease in heat transfer coefficient is established. Choi et al. [8] and Kaew-On and Wongwises [15] also reported the similar trend regarding the effect of saturation pressure on heat transfer coefficient. Such phenomena seems to dominate over the inertia and gravitational effects, leading to that the vertical upward flow results are almost identical to those for the horizontal flow even at high mass flux. At this saturation pressure, the criterion proposed by Kew and Cornwell [18], and by Li and Wu [19] indicate the macro-scale flow region in which the values of $\mathrm{Bo}\,\mathrm{Re}_L^{0.5}$ are larger than 200 for all mass flux values.

CONCLUSION

Flow pattern and heat transfer coefficient data related to heat transfer of R-134a during flow boiling in mini-channels are obtained from the present study. The test section is made from a 1.75 mm diameter stainless steel tube having a length of 600 mm. The DC power supply is connected to the test section for providing constant heat flux conditions. The flows of R-134a through horizontal and vertical upward directions are examined for channel orientation effects on the heat transfer characteristics. The flow pattern maps and heat transfer coefficients for corresponding directions are presented, indicating that the gravitational effects tend to be less dominant when the refrigerant experiences low mass flux and saturation pressure conditions.

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REFERENCES

[1] Ribatski, G., Wojtan, L., and Thome, J.R., 2006, "An Analysis of Experimental Data and Prediction Methods for Two-Phase Frictional Pressure Drop and Flow Boiling Heat Transfer in Micro-Scale Channels," Exp. Therm. Fluid Sci., 31, pp. 1-19.

[2] Mehendale, S.S., Jacobi, A.M., and Ahah, R.K., 2000, "Fluid Flow and Heat Transfer at Micro- and Meso-Scales with Application to Heat Exchanger Design," Appl. Mech. Rev., 53, pp. 175-193.

[3] Kandlikar, S.G., 2002, "Fundamental Issues Related to Flow Boiling in Mini-Channels and Micro-channels," Exp. Therm. Fluid Sci., 26, pp. 389-407.

[4] Chung, P.M.-Y., and Kawaji, M., 2004, "The Effect of Channel Diameter on Adiabatic Two-Phase Flow Characteristics in Microchannels," Int. J. Multiphase Flow, 30, pp.735-761.

[5] Saisorn, S., and Wongwises, S., 2010, "The Effects of Channel Diameter on Flow Pattern, Void Fraction and Pressure Drop of Two-Phase Air-Water Flow in Circular Micro-Channels," Exp. Therm. Fluid Sci., 34, pp. 454-462.

[6] Huo, X., Chen, L., Tian, Y.S., and Karayiannis, T.G., 2004, "Flow Boiling and Flow Regimes in Small Diameter Tubes," Appl. Therm. Eng., 24, pp. 1225-1239.

[7] Saitoh, S., Daiguji, H., and Hihara, E., 2005, "Effect of Tube Diameter on Boiling Heat Transfer of R-134a in Horizontal Small-Diameter Tubes," Int. J. Heat Mass Transfer, 48, pp. 4973-4984.

[8] Choi, K.I., Pamitran, A.S., and Oh, J.T., 2007, "Two-Phase Flow Heat Transfer of CO_2 Vaporization in Smooth Horizontal Minichannels," Int. J. Refireration, 30, pp. 767-777.

[9] Choi, K.I., Pamitran, A.S., Oh, C.Y., and Oh, J.T., 2007, "Boiling Heat Transfer of R-22, R-134a, and CO_2 in Horizontal Smooth Minichannels," Int. J. Refrigeration, 30, pp. 1336-1346.

[10] Shiferaw, D., Huo, X., Karayiannis, T.G., and Kenning, D.B.R., 2007, "Examination of Heat Transfer Correlations and a Model for Flow Boiling of R134a in Small Diameter Tubes," Int. J. Heat Mass Transfer, 50, pp. 5177-5193.

[11] Shiferaw, D., Karayiannis, T.G., and Kenning, D.B.R., 2009, "Flow Boiling in a 1.1 mm Tube with R134a: Experimental Results and Comparison with Model," Int. J. Therm. Sci., 48, pp. 331-341.

[12] Thome, J.R., Dupont, V., and Jacobi, A.M., 2004, "Heat Transfer Model for Evaporation in Microchannels. Part I: Presentation of the Model," Int. J. Heat Mass Transfer, 47, pp. 3375-3385.
[13] Dupont, V., Thome, J.R., and Jacobi, A.M., 2004, "Heat Transfer Model for Evaporation in Microchannels. Part II: Comparison with the Database," Int. J. Heat Mass Transfer, 47, pp. 3387-3401.
[14] Ong, C.L., and Thome, J.R., 2009, "Flow Boiling Heat Transfer of R134a, R236fa and R245fa in a Horizontal 1.030 mm Circular Channel," Exp. Therm. Fluid Sci., 33, pp. 651-663.
[15] Kaew-On, J., Wongwises, S., 2009, "Experimental Investigation of Evaporation Heat Transfer Coefficient and Pressure Drop of R-410A in a Multiport Mini-Channel," Int. J. Refrigeration, 32, pp. 124-137.
[16] Saisorn, S., and Wongwises, S., 2008, "Flow Pattern, Void Fraction and Pressure Drop of Two-Phase Air-Water Flow in a Horizontal Circular Micro-Channel," Exp. Therm. Fluid Sci., 32, pp. 748-760.
[17] Kandlikar, S.G., 2010, "Scale Effects on Flow Boiling Heat Transfer in Microchannels: A Fundamental Perspective," Int. J. Therm. Sci., doi:10.1016/j.ijthermalsci.2009.12.016.
[18] Kew, P.A., and Cornwell, K., 1997, "Correlations for the Prediction of Boiling Heat Transfer in Small-Diameter Channels," Appl. Therm. Eng., 17, pp. 705-715.
[19] Li, W., and Wu, Z., 2010, "A general Criterion for Evaporative Heat Transfer in Micro/Mini-Channels," Int. J. Heat Mass Transfer, 53, pp. 1967-1976.