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Two Phase Flow Pressure Drop of Pure and Mixed Refrigerants in Flow Boiling

Arijit Kundu[1] and Ravi Kumar[2]
[1] Jalpaiguri Government Engineering College, Jalpaiguri, India
Email: akundu261@gmail.com
[2]Indian Institute of Technology Roorkee, Uttarakhand, India
Email: ravikfme@iitr.ac.in

***Abstract*—The present study explores the two-phase flow pressure drop characteristics and flow visualizations for pure refrigerants R600a, and refrigerant mixture R410A in a smooth horizontal tube (7 mm internal diameter) uniformly heated by the resistance heating effect. The investigations were carried out at the inlet temperature range of 5ºC-9ºC varying the refrigerant mass flux within the range 100-400 kg/m^{2}s, the heat fluxes within the range 3.0-10.0 kW/m^2; the average qualities of boiling refrigerant were between 0.10 and 0.95. The influence of these parameters and fluid thermo-physical properties of the two-phase pressure drop were analyzed. Moreover, the pressure drop predicted by some available models and empirical correlations in the open literature were compared with the present data.**

***Index Terms*— Two phase flow boiling, horizontal pressure drop, R600a, R410A.**

I. INTRODUCTION

Prediction of pressure drops in flow boiling inside small diameter tubes of environmentally friendly refrigerants (demanded by the international protocols [1, 2]) is not only significant to design an efficient direct expansion evaporator in optimized refrigeration system, but the required teat data for the refrigerants also necessary for the purpose of refrigeration industry as the equivalent saturation temperature loss in the evaporator line corresponding to the two-phase pressure drop may limit the design considerations due to improper prediction of the pressure drop by the existing methods leading to about 100% deviation from the actual [3]. The use and production of hydrocholofluorocarbon (HCFC) refrigerants have been prohibited [1, 2] because of their high ozone depletion potential (ODP) and total equivalent warming impact (TEWI). And for this, researches for a suitable replacement have been escalating in recent decades, but there is no such single component refrigerant which has a thermodynamic efficiency close to the most used working HCFC R22 in refrigeration field worldwide and fulfils international amendment criteria in climatic sanctuary aspects as well. Atmospheric Research and Environment Program (AREP) has initiated to recognize the substitution of HCFCs by publishing an updated list of alternative refrigerants in which some pure refrigerants like R600a and some refrigerant mixtures like R410A came forth as the first rate choices.

Unlike the hefty number of data for mini and large tubes, there are relatively very few published researches on two-phase flow boiling pressure drop and flow patterns of refrigerants in small tubes. High heat transfer coefficient and low required fluid mass with optimizing the size of compact heat exchangers by reducing the

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tube wall friction loss are the substantial requirements to design an efficient evaporator which in turn defines the performance of a good refrigeration system. Reducing cost, charge inventory and intensifying heat transfer performance are the favourable aspects of utilizing mini tubes and sometimes with extended surfaces. But the high pressure loss caused by the flow restriction in such flow contours increases the energy consumption in the driving devices [4]. Large tubes indeed sacrifices heat transfer rate on behalf of lowering the pressure loss during the flow inside. Kew and Cornwell [5] described the significance of the flow restriction in the small size channels. According to the definition of Mehendale et al. [6], the distinction between small diameter channels and normal size channels is 6 mm. Kandlikar [7] differentiated the small diameter and normal size channels by 3 mm. A hydraulic diameter of 6 mm as the criterion of small diameter was considered by Wolk et al. [8] in their study. Wongwises et al. [9] have described a hydraulic diameter of 7.5 mm as a typical standard of small diameter tubes.

There are different predicting methods [10-14] comparing the experimental pressure drop data, especially the frictional pressure drop of boiling inside so called small channels over the last few decades. In fact, there is a huge gap in literature illustrating and comparing the pressure drop variations refrigerants in flow boiling through small tubes. Present study evaluate experimentally and compares the two-phase flow pressure drop gradients of R600a and R410A in a electrical resistive heated 7 mm inside diameter smooth circular horizontal tube to acquire data useful for refrigerant industry.

A. Existing two-phase pressure drop models

The prediction of two phase pressure drop gradient is significant in the design of heat exchanger for the refrigeration and process industries. The total pressure drop for flows inside the tubes concludes from the potential and kinetic energy loss with the frictional energy dissipation at the channel wall of fluid flow (Eq. 1).

$$\Delta P_{TP} = \Delta P_{Static} + \Delta P_{Mom} + \Delta P_{Fric} \tag{1}$$

Evaluation of the two phase pressure drop across the length of the heated tube is very much multifaceted because of the existence of various flow regimes with wide-ranging physical conditions along the tube.

The static pressure drop for two phase flow is given by Eq. 2.

$$\Delta P_{Static} = \rho_{TP} \cdot g \cdot H \sin\alpha \tag{2}$$

Present study for horizontal tube includes $\Delta P_{Static} = 0$ because of $\alpha = 0^o$. The momentum pressure drop due to the change in kinetic energy can be expressed as:

$$\Delta P_{Mom} = G^2 \left\{ \left[\frac{(1-x)^2}{(1-\varepsilon)\rho_f} + \frac{x^2}{\varepsilon\rho_g} \right]_{out} - \left[\frac{(1-x)^2}{(1-\varepsilon)\rho_f} + \frac{x^2}{\varepsilon\rho_g} \right]_{in} \right\} \tag{3}$$

Accurate prediction of void fraction ε is necessary and numerous methods are available for predicting the void fraction which has been quoted in Table I.

TABLE I. VOID FRACTION CORRELATIONS

Authors	Correlations
Homogeneous	$\varepsilon_H = \left[1 + \left(\frac{1-x}{x}\right) \cdot \left(\frac{\rho_g}{\rho_f}\right) \cdot \left(\frac{\mu_f}{\mu_g}\right)\right]^{-1}$
Lockhart and Martinelli [16]	$\varepsilon = \left[1 + 0.28\left(\frac{1-x}{x}\right)^{0.64} \cdot \left(\frac{\rho_g}{\rho_f}\right)^{0.36} \cdot \left(\frac{\mu_f}{\mu_g}\right)^{0.07}\right]^{-1}$
Chisholm [17]	$\varepsilon = \left[1 + \sqrt{1 - x\left(1 - \frac{\rho_f}{\rho_g}\right)}\left(\frac{1-x}{x}\right) \cdot \left(\frac{\rho_g}{\rho_f}\right)\right]^{-1}$
Chen [18]	$\varepsilon = \left[1 + 0.18\left(\frac{1-x}{x}\right)^{0.60} \cdot \left(\frac{\rho_g}{\rho_f}\right)^{0.33} \cdot \left(\frac{\mu_f}{\mu_g}\right)^{0.07}\right]^{-1}$
Zhao et al. [19]	$\varepsilon = \left[1 + \varepsilon^{-0.125}\left(\frac{1-x}{x}\right)^{0.875} \cdot \left(\frac{\rho_g}{\rho_f}\right)^{0.875} \cdot \left(\frac{\mu_f}{\mu_g}\right)^{0.875}\right]^{-1}$
Steiner [20]	$\varepsilon = \frac{x}{\rho_g}\left[(1 + 0.12(1-x)) \cdot \left(\frac{x}{\rho_g} + \frac{1-x}{\rho_f}\right) + \frac{1.18}{G} \cdot \left(\frac{\sigma g(\rho_f - \rho_g)}{\rho_f^2}\right)^{0.25}(1-x)\right]^{-1}$

The frictional pressure loss in two phase flow through tubes and annuli is typically predicted by separated flow models. Some common methods and predicted correlations are tabulated in Table II. Whalley [21] presented an extensive comparison between various in print correlations with around 25000 data points and their recommendations was Friedel [22] correlation to be preferred for all fluids. In very recent time, Ould-Didi et al. [3] have expressed favor over Grönnerud [23] correlations equally impressive to those of Friedel [22] for their comparison of refrigerants flow boiling data in 10.92 mm and 12 mm internal diameter tube with mass velocities from 100 to 500 kg/m^{2}s at vapor qualities between 0.04 and 0.99.

TABLE II. TWO PHASE FRICTIONAL PRESSURE DROP CORRELATIONS

Authors	Correlations
Cicchitti et al. [15]	$\left(\frac{dP}{dz}\right)_{Fr} = \frac{2f_{TP}G^2}{D_i\rho_{TP}}$ $f_{TP} = 0.079/Re_{TP}^{0.25}$ $Re_{TP} = GD_i/\mu_{TP}$ $\mu_{TP} = x\mu_g + (1-x)\mu_f$ $\rho_{TP} = \left(x/\rho_g + (1-x)/\rho_f\right)^{-1}$
Lockhart and Martinelli [16]	$\Delta P_{Fric} = \phi_{ftt}^2 \Delta P_f$ $\phi_{ftt}^2 = 1 + \frac{C}{X_{tt}} + \frac{1}{X_{tt}^2}$ for $Re_f > 4000$ $\Delta P_{Fric} = \phi_{gtt}^2 \Delta P_g$ $\phi_{ftt}^2 = 1 + \frac{C}{X_{tt}} + \frac{1}{X_{tt}^2}$ for $Re_f < 4000$
Chisholm [17]	$\left(\frac{dP}{dz}\right)_{Fric} = \left(\frac{dP}{dz}\right)_f \phi_{Ch}^2$ $\left(\frac{dP}{dz}\right)_f = f_f\frac{2G^2}{D_i\rho_f}$ $f_f = \frac{16}{Re_f}$ $\phi_{Ch}^2 = 1 + (Y^2-1)\left\{Bx^{(2-n)/2}(1-x)^{(2-n)/2} + x^{(2-n)}\right\}$ $Y^2 = \frac{(dP/dz)_g}{(dP/dz)_f}$ $n = 0.25$
Friedel [22]	$\Delta P_{Fric} = \phi_{fr}^2 \Delta P_f$ $\phi_{fr}^2 = E + \frac{3.24\, F \cdot H}{Fr_H^{0.045} We_f^{0.035}}$ $Fr_H = \frac{G^2}{gD_i\rho_H^2}$ $E = (1-x)^2 + x^2\frac{\rho_f f_g}{\rho_g f_f}$ $F = x^{0.78}(1-x)^{0.224}$ $\rho_H = \left(\frac{x}{\rho_g} + \frac{1-x}{\rho_f}\right)^{-1}$ $H = \left(\frac{\rho_f}{\rho_g}\right)^{0.91} \cdot \left(\frac{\mu_g}{\mu_f}\right)^{0.19} \cdot \left(1 - \frac{\mu_g}{\mu_f}\right)^{0.7}$
Grönnerud [23]	$\Delta P_{Fric} = \phi_{Gd}^2 \Delta P_f$ $\phi_{Gd} = 1 + \left(\frac{dP}{dz}\right)_{Fr}\left[\frac{(\rho_f/\rho_g)}{(\mu_f/\mu_g)^{0.25}} - 1\right]$ $\left(\frac{dP}{dz}\right)_{Fr} = f_{Fr}\{x + 4(x^{1.8} - x^{10}f_{Fr}^{0.5})\}$ $Fr_f = \frac{G^2}{gD_i\rho_f^2}$

II. EXPERIMENTAL ASPECTS

A. *Test Facility and Methods*

Fig. 1 shows the schematic illustration of the experimental facility to provide the two phase pressure drop during flow boiling of the test refrigerants under different ranges of test parameters. The test setup consists of a semi-hermetic compressor integrated with oil separator, water-cooled condenser, sub-cooler, thermostatic expansion device, pre-evaporator, post-evaporator and test-evaporator. The sub-cooler after the condenser was included for ensuring liquid refrigerant to enter to the Coriolis effect mass flow meter measuring the accurate mass flow rate of the refrigerant. A filter-dryer was fitted after the flow meter to entrap foreign particles and moisture in refrigerant. A suitable pre-evaporator had been designed and accommodated to control the vapour quality in the test evaporator. By regulating the power supply with an AC variac, heat input to the pre-evaporator and test evaporator was controlled. An accumulator was installed upstream of the compressor suction to ensure the refrigerant vapour entering to the compressor.

For proper revelation of pressure state at the time of the experiments, bourdon type pressure gauges were fitted at each section between the components in the main circuit of refrigerant flow. Sight glasses were fitted at the entry and exit to the test evaporator to check the phase of the refrigerant at the time of flow through the refrigeration circuit; whereas a proper view section of high tempered borosilicate transparent tube was incorporated just ahead of the test evaporator outlet for flow pattern visualization. Piezoelectric transducers with a range of 0-300 psi (uncertainty declared by the manufacturer is of ±0.05% of full scale) have been

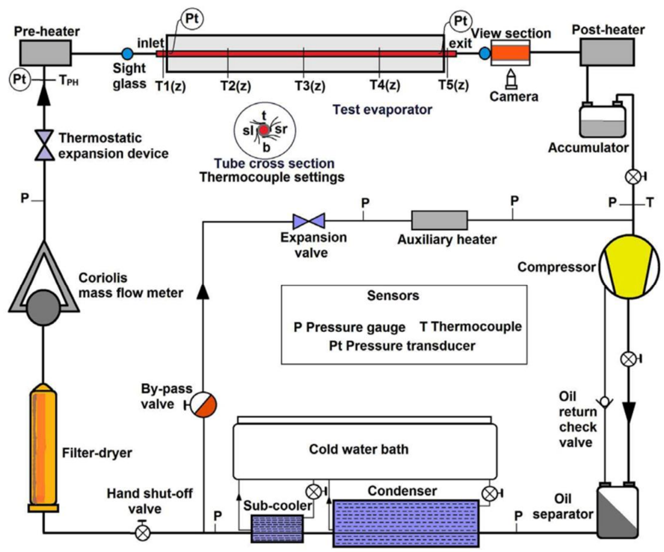

Fig. 1. Schematic of test arrangement

introduced upstream of the pre-evaporator and at the inlet and outlet of the test evaporator to measure the absolute pressures. The test section was made of smooth copper tube with inner diameter of 7 mm and outside diameter of 9.57 mm. Test evaporator tube was heated by flexible Nichrome (Nickel-Chromium 80-20 percent by weight) heater wire (2 kW capacity, with calibrated accuracy of 5W) wrapped over mica thin tape which was wound around the tube over full test length to take care from electrical leakage. Electrical heat inputs to the test section, pre-heater and post-heater by the variac are determined by the product of the voltage drop across the section and the current intensity. The digital multimeter (range 750 V, 20 A; accuracy ±0.05% of the reading ±5 digits; resolution 0.01 A; frequency 40-400 Hz) was used to measure the corresponding readings.

The experiments were anticipated to provide information about the performance of the refrigeration circuit running with different refrigerants. The pressure measurements were recorded through data acquisition system (input +/-10V, 250 kS/s, sampling rate of 0.1-0.2 Hz) and LABVIEW 10.0 software programs. The test parameters were measured or recorded when steady state conditions approaches which were signified for temperature changes nearly ±0.02°C and for pressures of about ±5 kPa.

B. Data Reduction

To acquire data in two-phase flow of the refrigerants, it was insured that the refrigerant must enter to the test evaporator in saturated condition by the energy balance in the pre-evaporator and first determining the axial position (z_{Sat}) where the bubble forming starts as:

$$z_{Sat} = \frac{\dot{m}C_P\left(T_{Sat} - T_{in,PH}\right)}{\pi D_i \dot{q}_{PH}} \tag{4}$$

The resistance heat flow, Q and heat flux, $\dot{q}$ to the refrigerant from outside of the tube (pre-evaporator and test section) has been calculated by using following Eq. (5):

$$Q = \eta \cdot V \cdot I \quad \text{where} \quad \dot{q} = Q/\pi D_o L \tag{5}$$

To evaluate the heat loss to the ambient in the test section and pre-heater, single phase tests were carried out to find the heating coefficient, η (typically around 0.935) was experimentally determined. For determining vapour quality at any local axial position along the evaporator tube, is defined by the local heat transfer to the fluid as:

$$x_{th}(z) = \frac{\pi D_i (z - z_{Sat})\dot{q}}{\dot{m} h_{fg}} \tag{6}$$

The experimental two-phase boiling frictional pressure drop gradient of flow inside the horizontal test section can be obtained by combining Eq. (1-3) as:

$$\left(\frac{dP}{dz}\right)_{fric} = \left(\frac{dP}{dz}\right)_{TP} - G^2 \frac{d}{dz}\left(\frac{(1-x)^2}{(1-\varepsilon)\rho_f} + \frac{x^2}{\varepsilon \rho_g}\right) \tag{7}$$

For void fraction (ε) data, Steiner [22] correlations was used.

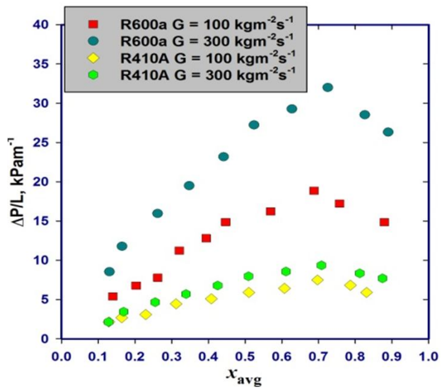

Fig. 2. Comparison of pressure drop gradient varying vapour quality

C. Results and Discussion

A highly dependence of the two-phase pressure drop gradient of the refrigerant mass velocity and mean vapour quality can be observed in Fig. 2. Almost same behaviour of the pressure drop variation is present for both the test refrigerants. Indeed, at the low reduced pressures ($P_{reduced}|_{R600a}$=0.0526-0.0680; $P_{reduced}|_{R410A}$=0.1954-0.2469) the lowerρ_l/ρ_gratio causes a stronger variation of the two-phase density and thus, a large dependence on the mass velocity [23] which can be seen from Figure 2. It can be seen that the pressure gradients of all refrigerants confront with the typical acclivity with vapour quality due to the liberal increase of the two-phase mixture velocity; then those reach a crest at high vapour qualities and finally fall for each mass velocity.

In general, the two-phase pressure drop is primarily sourced by the frictional pressure drop for the horizontal condition of the tube. The variation of two-phase frictional pressure drops of the test refrigerants has been depicted in Fig. 16 as a function of liquid Reynolds number. The frictional pressure drop is a change of pressure ensuing from the energy dissipated in the flow by friction, eddying etc. [24]. As the flow goes downstream and vaporization takes place, the density of the liquid-vapour mixture decreases. As a result, the flow accelerates much more and the frictional pressure drops increases. Two-phase frictional pressure drops of R600a are very much higher than those of R410A.The difference increases with the increase in mass velocity due to the growth and transition of flow patterns. Considering all the experiments conducted, it can be observed that for the variation of the mass velocity from 100 to 300 kg/m^{2}s, the R410A two-phase pressure drops lower by a factor ranging from152%-240%, with respect to those for R600a; where the R600a frictional pressure drops are more than those of R410A in the range of 155%-290% for a change in the mass velocity from 200 to 400 kg/m^{2}s.

III. Conclusions

Two-phase pressure drop strongly depends on mass velocity and merely on imposing heat flux for all test fluids. For the variation of the mass velocity from 200 to 400 kg/m^{2}s, R600a frictional pressure drops are more than those of R410A in the range of 155%-290%. When in comparison for R410A, the relative performance index of R600a decreases with the increase in mass velocity and varies merely with vapour quality; R410A performs better than R600a. Intermittent to annular flow pattern transition for R410A has much delayed and the transition from annular to wavy flow occurs at respectively lower mass velocities than R600a at nearly the same saturation temperature.

REFERENCES

[1] The Copenhagen Amendment, The amendment to the Montreal Protocol agreed by the Fourth Meeting of the Parties, 1992, http://ozone.unep.org.

[2] UNFCCC, United Nations Framework Convention on Climate Change, National Submissions of Greenhouse Gas Emissions on the Common Reporting Format, UNFCCC Secretariat, Bonn, Germany, 2003, http://unfccc.int.

[3] M.B. Ould Didi, N. Kattan, J.R. Thome, Prediction of two-phase pressure gradients of refrigerants in horizontal tubes, Int. J. Refrigeration 25 (2002) 935-947.

[4] R. Kouhikamali, A.S. Kojidi, M. Asgari, F. Alamolhoda, The effect of condensation and evaporation pressure drop on specific heat transfer surface area and energy consumption in MED-TVC plants, Desalination and Water Treatment 46 (2012) 68-74.

[5] P.A. Kew, K. Cornwell, Correlations for the prediction of boiling heat transfer in small diameter channels, Appl. Therm. Eng. 17 (1997) 705-715.

[6] S.S. Mehendale, A.M. Jacobi, R.K. Ahah, Fluid flow and heat transfer at micro- and meso-scales with application to heat exchanger design, Appl. Mech. Rev. 53 (2000) 175-193.

[7] S.G. Kandlikar, Fundamental issues related to flow boiling in mini channels and micro channels, Exp. Therm. Fluid Sci. 26 (2002) 38-47.

[8] G. Wolk, M. Dreyer, H.J. Rath, Flow patterns in small diameter vertical non-circular channels, Int. J. Multiphase Flow 26 (2000) 1037-1061.

[9] S. Wongwises, S. Disawas, J. Kaewon, C. Onurai, Two-phase evaporative heat transfer coefficients of refrigerant HFC-134a under forced flow conditions in a small horizontal tube, Int. Commun. Heat Mass Transfer 27 (2000) 35-48.

[10] C.S. Kuo, C.C. Wang, Horizontal flow boiling of R22 and R407C in a 9.52 mm micro-fin tube, Appl. Therm. Eng. 16 (1996) 719-731.

[11] C. Wang, C. Chiang, Two-phase heat transfer characteristics for R-22/R-407C in a 6.5 mm smooth tube, Int. J. Heat Fluid Flow 18 (1997) 550-558.

[12] P. Haberschill, C. Branescu, M. Lallemand, Pressure drop of micro-finned tubes during boiling of R22 and R407C, In Proceedings: Heat and Mass Transfer of Refrigeration Machines and Heat Pumps, Valencia (Spain), 31 March–02 April (2003) 75-79.

[13] J. Wongsa-ngam, T. Nualboonrueng, S. Wongwises, Performance of smooth and micro-fin tubes in high mass flux region of R-134a during evaporation, Heat Mass Transfer 40 (2004) 425-435.

[14] A. Greco, G.P. Vanoli, Experimental two-phase pressure gradients during evaporation of pure and mixed refrigerants in a smooth horizontal tube. Comparison with correlations, Int. J. Heat Mass Transfer 42 (2006) 709-725.

[15] Cicchitti, C. Lombardi, M. Silvestri, G. Soldaini, R. Zavattarelli, Two-phase cooling experiments - pressure drop, heat transfer and burnout measurements, Energia Nucleare 7 (6) (1960) 407-425.

[16] R.C. Lockhart, R.W. Martinelli, Proposed correlation of data for isothermal two phase, two component flow in pipes, Chem. Eng. Proc. 45 (1949) 39-48.

[17] D. Chisholm, Pressure gradients due to friction during the flow of evaporating two-phase mixtures in smooth tubes and channels, Int. J. Heat Mass Transfer 16 (1973) 347-348.

[18] J.J. Chen, A further examination of void-fraction in annular two-phase flow, Int. J. Heat Mass Trans. 29 (1986) 1760-1763.

[19] J.F. Zhao, On the void fraction matched model for the slug to annular transition at microgravity, J. Basic Sci. Eng. 8 (2000) 394-397.

[20] D. Steiner, Heat transfer to boiling saturated liquids, in: VDI-Wärmeatlas (VDI Heat Atlas), Verein Deutscher Ingenieure, VDI-Gessellschaft Verfahrenstechnik und Chemie-ingenieurwesen (GCV), Dusseldorf (1993) (Translator: J.W. Fullarton).

[21] P.B. Whalley, Multiphase Flow and Pressure Drop, Hewitt GF 1983 Heat Exchanger Design Handbook, Hemisphere, Washington, DC, 2 (1980) 3-11.

[22] L. Friedel, Improved friction pressure drop correlations for horizontal and vertical two-phase pipe flow, In Proceeding: European Two-Phase Flow Group Meeting, Ispra, Italy (1979).

[23] R. Grönnerud, Investigation of liquid hold-up, flow-resistance and heat transfer in circulation type evaporators, part IV: two-phase flow resistance in boiling refrigerants, Bulletin de l'Inst. du Froid, Annexe 1972-1 (1972).
[24] J.G. Collier, J.R. Thome, Convective boiling and condensation, 3rd ed., Oxford University Press, Oxford (1994).