VOL. 4, NO. 9, NOVEMBER 2009 ISSN 1819-6608
ARPN Journal of Engineering and Applied Sciences

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PREDICTION OF PRESSURE DROP OF REFRIGERANTS FOR TWO-PHASE FLOW INSIDE A HORIZONTAL TUBE USING CFD ANALYSIS

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ABSTRACT

Two phase flow in a horizontal tube has widespread applications, particularly in the condensers of refrigeration and air conditioning systems. Pressure drop prediction is especially important for condensers because the local condensing temperature is a function of local pressure, affecting the mean temperature difference in the heat exchanger. In the present analysis, two phase flow is treated as a single phase pseudo fluid with average properties of liquid and vapor using homogeneous model. CFD analysis of two phase flow of refrigerants inside a smooth horizontal tube is carried out under adiabatic conditions using commercial CFD software, FLUENT for different mass fluxes ranging from 100 to 1000kg/m^{2}s and at different saturation temperatures of 40^{0}C, 50^{0}C and 60^{0}C. The values of pressure drop obtained from the simulations for refrigerants, R22, R134a and R407C are compared with correlations and experimental data available in literature.

Keywords: Adiabatic two phase flow, homogeneous model, liquid–vapor flow, CFD analysis.

INTRODUCTION

In forced convective condensation of refrigerants, the vapour and liquid flow inside a tube simultaneously. In addition to inertial, viscous and pressure forces, two phase flow is affected by interfacial tension, liquid wetting the tube wall and the momentum exchange between the liquid and vapour phases. Hence the morphology of two phase flow changes with geometry and orientation. The flow regimes for a horizontal tube are shown in Figure-1.

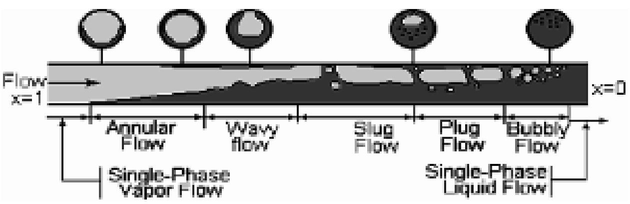

Figure-1. Flow patterns for horizontal co-current flow for condensation.

Pressure drop prediction is especially important for condensers because the local condensing temperature is a function of local pressure, affecting the mean temperature difference in the heat exchanger. In industry, condenser pressure drop should not be greater than ±10% of the operating pressure in order to prevent significant decrease in mean temperature difference due to pressure drop.

Pressure drop during condensation inside a tube of constant cross sectional area, is made up of terms involving wall friction, momentum transfer (flow acceleration) and gravity, as given by Eq. (1) based on separated flow model [1].

$$-\left[\frac{dp}{dz}\right]=-\left[\frac{dp}{dz}\right]_F-\left[\frac{dp}{dz}\right]_a-\left[\frac{dp}{dz}\right]_g \quad (1)$$

The gravity pressure drop $[dp/dz]_g$ becomes zero for horizontal tubes. During condensation the momentum transfer term, $[dp/dz]_a$ contributes to the overall pressure drop due to the mass transfer that occurs at the liquid-vapor interface. However, for a condensing flow the kinetic energy of outgoing flow is smaller than that of incoming flow. Hence the momentum pressure head results in an increase in the pressure at the exit than at the inlet, i.e. a pressure recovery. For condensing flows, it is common to ignore the momentum recovery as only some of it may actually be realized in the flow and ignoring it provides some conservatism in the design. Correlations of frictional pressure gradient, $[dp/dz]_F$ developed using the two-phase frictional multiplier approach, ϕ^2 are presented as follows.

Lockhart and Martinelli [2] performed pioneering work to evaluate two phase friction pressure gradient using two phase multipliers for adiabatic air-water mixtures at atmospheric pressure. Their correlations were modified for diabatic flows by Martnelli and Nelson [3] where, ϕ_{lo}, defined by Lockhart-Martinelli is corrected as, $\phi_{lo}=\phi_l\left(1-x\right)^{1.75}$. These multipliers are functions of Martinelli parameter, X_{tt} defined as a dimensionless combination of the physical properties. Subsequently, X_{tt} is being used in several convective condensation and boiling correlations as one of the flow governing parameters. The generality of Lockhart and Martinelli multipliers is thus well acclaimed in two phase studies. Grönnerud correlation [4] is developed for refrigerants. Chisholm method [5] is recommended for fluids with property index, $\left[\left(\mu_l/\mu_v\right)^{0.2}/\left(\rho_l/\rho_v\right)\right]>0.01$. Friedel [6] developed a correlation for two phase multiplier for vertical upward and horizontal flow in round tubes. The correlation is recommended for fluids with

$(\mu_l / \mu_v) < 1000 kg/m^2 s$. Müller-Steinhagen and Heck [7] proposed an empirical interpolation between all liquid and all vapor flow. These correlations of pressure drop are presented in Table-1.

Tribbe and Müller-Steinhagen [9] reported an extensive comparison of 35 two phase pressure drop predictive methods using a large database of air-oil, cryogenics, steam-water, air-water fluid combinations and several refrigerants. They observed that statistically Müller-Steinhagen and Heck correlation predicted the pressure drop well compared to other correlations. Didi *et al* [11] mapped their experimental data using Kattan, Thome and Farvat's flow pattern map [8] and observed that the Müller-Steinhagen and Heck correlation for annular flow and Grönnerud correlation for intermittent and stratified wavy flow is in good agreement.

In a recent paper Quiben and Thome [12, 13] compared their experimental data with the correlations of Müller-Steinhagen and Heck and Grönnerud for R22, R410A and R134a for horizontal tubes of different diameters. They observed that only 40% of the data were captured within ±20% with Grönnerud correlation and 50% of data within ±20% with Müller-Steinhagen and Heck correlation. In this regard, an attempt is made in the present study to predict the pressure drop using CFD analysis of two-phase flow of refrigerants inside a horizontal tube.

The CFD analysis is performed for the flow of refrigerants, R134a, R22 and R407C inside a tube of length 1600 mm and internal diameter, 8mm using commercial CFD software, FLUENT. The CFD analysis is performed under adiabatic conditions as the pressure drop is not affected by heat transfer. The properties of refrigerants are obtained from the refrigerant property data base, REFPROP, version 6.01. The average properties for two phase fluid are evaluated using homogeneous model. A brief review of homogeneous model is presented as follows:

HOMOGENEOUS MODEL

Basic equations of two phase flow are developed considering the flow to be steady and one dimensional in the sense that all dependent variables are idealized as being constant over any cross section of the tube or duct, varying only in axial direction. Homogeneous model is a special case of separated flow model [1] where each phase is assumed to travel with constant and equal velocities with thermal equilibrium between two phases. The homogeneous model also known as 'Friction Factor' or 'Fog Flow' model, considers two phases to flow as a single phase possessing mean fluid properties of liquid and vapor. The model finds its applications in steam generation, petroleum and refrigeration industries.

For steady homogeneous flow model, the basic equations for condensation inside a horizontal tube are reduced to the following form [1]:

Continuity Equation: $\dot{m} = \bar{\rho}\bar{u}A$ (2)

Momentum Equation:

$$-Adp - d\bar{F} - A\bar{\rho}gdz = \dot{m}d\bar{u} \quad (3)$$

Where the average wall friction, $d\bar{F}$ in terms of wall shear stress, τ_w acting over the inside area of the tube can be expressed as:

$$d\bar{F} = \tau_w (Pdz) \quad (4)$$

The frictional pressure gradient can be obtained using the Fanning Equation, Eq. (5).

$$-\left[\frac{dp}{dz}\right]_F = \frac{1}{A}\left[\frac{d\bar{F}}{dz}\right] = \tau_w\left[\frac{4}{d}\right] \quad (5)$$

The average properties for homogeneous pseudo fluid are developed from the fundamentals as mentioned elaborately by Collier [1]. The average fluid density obtained by equating liquid and vapor velocities is,

$$\frac{1}{\bar{\rho}} = \left[\frac{x}{\rho_v} + \left(\frac{1-x}{\rho_l}\right)\right] \quad (6)$$

Possible forms of relationships for mean two phase viscosity, $\bar{\mu}$ based on limiting conditions, at x=0, $\bar{\mu} = \mu_l$ and at x=1, $\bar{\mu} = \mu_v$, are represented by Eqs. (7) to (9).

$$\frac{1}{\bar{\mu}} = \left[\frac{x}{\mu_v} + \left(\frac{1-x}{\mu_l}\right)\right] \quad \text{(McAdams model)} \quad (7)$$

$$\bar{\mu} = x\mu_v + (1-x)\mu_l \quad \text{(Cicchitti model)} \quad (8)$$

$$\bar{\mu} = \bar{\rho}\left[\frac{x\mu_v}{\rho_v} + \frac{(1-x)\mu_l}{\rho_l}\right] \quad \text{(Dukler model)} \quad (9)$$

CFD ANALYSIS

The tube with dimensions, ϕ 8 mm X 1600 mm is modeled and volume mesh is generated in GAMBIT and steady state simulations are carried out in FLUENT for three different types of grids with total number of volumes for grid-1 as 51120, grid-2 as 98048 and grid-3 as 150024. The pressure drop is evaluated at each quality using three types of grid and Figure-2 shows that the variation of pressure drop for grid-2 and grid-3 is almost negligible. The pressure drop results presented in the present CFD analysis are obtained with grid-2. The analysis is performed under adiabatic conditions for turbulent flow as the Reynolds Number, Re based on the average properties exceeds 2300 for all flow rates considered.

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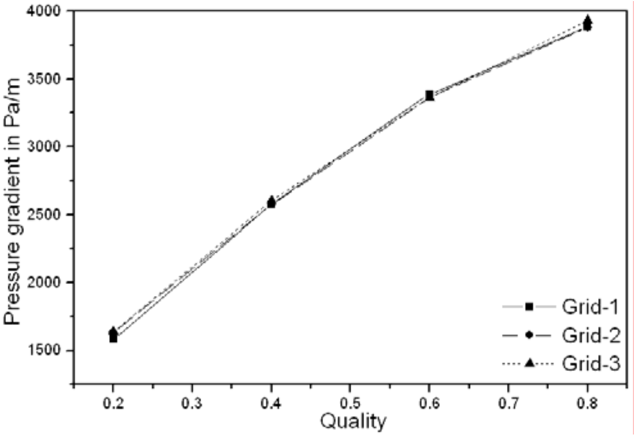

Figure-2. Variation of pressure gradient with grid.

The material properties are evaluated using Eqs. (6) to (9) at each quality, x for different saturation pressures. Simulations are performed for three average viscosity models given by Eqs. (7) to (9). Flow is considered fully developed with specified mass flux at the inlet and outflow condition at the outlet. The area weighted average of wall shear stress is reported and the pressure gradient at a given quality for the tube is obtained from the wall shear stress using Eq. (5).

Graphs are drawn for the variation of pressure gradient with quality for different mass fluxes. The so obtained pressure drop data is compared with the separated flow correlations presented in Table-1 and experimental data of Cavallini *et al.* [10].

RESULTS AND DISCUSSIONS

The pressure drop of refrigerants under two phase flow conditions inside the tube are evaluated using three different models of dynamic viscosity.

Selection of average dynamic viscosity model

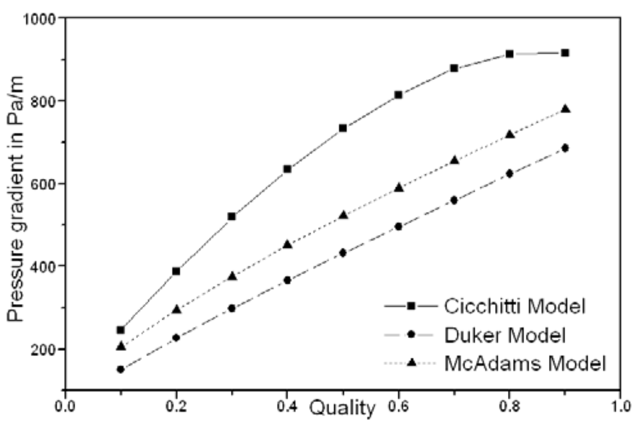

Figure-3. Pressure gradient with different models of $\bar{\mu}$ for R134a at T_s= 40^0C at G= 176 kg/m^{2}s.

Figure-3 shows that there is a noticeable variation in the pressure gradient calculated from the three models of average viscosity at a particular quality at low mass flux and the variation decreasing with increase of mass flux as shown in Figure-4. It is clear that pressure gradient obtained by all the three models particularly that of McAdam's and Dukler's models tend to converge with the increase of mass flux as shown in Figure-4. This is due to the effect of mass flux being dominant compared to that of average viscosity on pressure gradient at high mass flux.

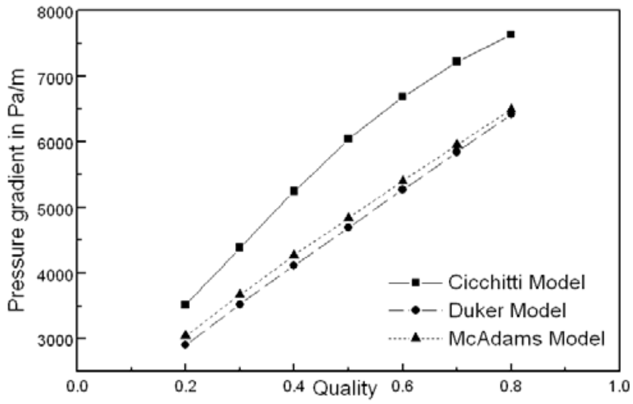

Figure-4. Pressure gradient with different models of $\bar{\mu}$ for R134a at T_s= 40^0C at G= 750 kg/m^{2}s.

Further, the difference in the pressure gradient calculated from the three models of viscosity reduces for high saturation pressure compared to that of low saturation pressure of same refrigerant, R134a as represented by Figures 3 and 5. This is due to the average dynamic viscosity obtained using Dukler's and McAdams models are nearly same particularly at high saturation temperatures and for high pressure refrigerants, R22 and R407C even at low condensation temperatures.

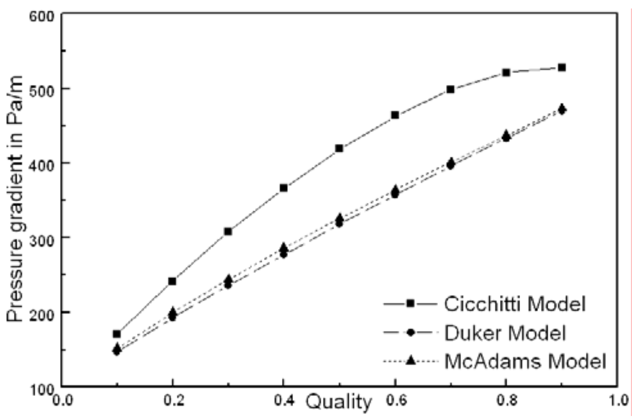

Figure-5. Pressure gradient with different models of $\bar{\mu}$ for R134a at T_s= 60^0C at G= 176 kg/m^{2}s.

Figures 3 and 5 show that pressure drop obtained using Cicchitti linear model of $\bar{\mu}$ is higher at any given quality, hence the same is selected for conservative analysis for further simulations in CFD for all the refrigerants considered at different saturation temperatures and mass fluxes.

VOL. 4, NO. 9, NOVEMBER 2009 ISSN 1819-6608
ARPN Journal of Engineering and Applied Sciences

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Comparison of CFD pressure drop data with correlations

Table-1. Correlations of comparison with CFD simulations.

$-\left[\frac{dp}{dz}\right]_F = -\phi_{lo}^2\left[\frac{dp}{dz}\right]_{lo}$ Where $\phi_{lo}{}^2$ is the two phase multiplier given by different correlations as follows:
where $\left[\frac{dp}{dz}\right]_{lo} = \frac{\left[2 f_l G^2 x^2\right]}{\rho_l d}$

#	Description	Correlation
1	Chisholm Correlation [1973] **Flow Regime:** Adiabatic two-phase flow-annular **Range:** (μ_l/μ_v) >1000 & G > 100 kg/m^{2}s	$\phi_{Ch}^2 = 1+\left[Y^2-1\right]\left[B(x(1-x))^{(2-n)/2}+x^{2-n}\right]$ where 'n' is the exponent from the friction factor expression of Blasius (n=0.25) and Where $Y^2 = (dp/dz)_{vo}/(dp/dz)_{lo}$ For $0<Y<9.5$, Chisholm's parameter B is calculated as: $B = 55/G^{0.5}$ for $G \geq 1900 kg/m^2s$ $B = 2400/G$ for $500<G<1900kg/m^2s$ $B = 4.8$ for $G<500kg/m^2s$ For $9.5<Y<28$, B is: $B = 520/\left[YG^{0.5}\right]$ for $G \leq 600kg/m^2s$ $B = 21/Y$ for $G>600kg/m^2s$ For $Y>28$, $B = 15000/\left[Y^2G^{0.5}\right]$
2	Grönnerud Correlation [1979]	$\phi_{gd}^2 = 1+\left[\frac{dp}{dz}\right]_{Fr}\left[\frac{(\rho_l/\rho_v)}{(\mu_l/\mu_v)^{0.25}}-1\right]$ $\left[\frac{dp}{dz}\right]_{Fr} = f_{Fr}\left[x+4\left(x^{1.8}-x^{10}f_{Fr}^{0.5}\right)\right]$ If $Fr_l \geq 1$, $f_{Fr}=1$ or if $Fr_l<1$, $f_{Fr} = Fr_l^{0.3}+0.0055\left[\ln\frac{1}{Fr_l}\right]$ where $Fr_l = \frac{G^2}{gd\rho_l^2}$
3	Friedel Correlation [1979] **Flow Regime:** Adiabatic two-phase flow-annular **Range:** (μ_l/μ_v) <1000	$\phi_{fr}^2 = E+3.24FH/\left[Fr_H^{0.045}We_l^{0.035}\right]$ where $Fr_H = G^2/(gd\rho_H)$ and 'ρ_H' is homogeneous density obtained from: $\rho_H = \left[(x/\rho_v)+(1-x)/\rho_l\right]^{-1}$ $E = [1-x]^2 + x^2\left[\rho_l/\rho_v\right]\left[f_{vo}/f_{lo}\right]$ and $F = x^{0.78}[1-x]^{0.224}$ $H = \left[\rho_l/\rho_v\right]^{0.91}\left[\mu_v/\mu_l\right]^{0.19}\left[1-(\mu_v/\mu_l)\right]^{0.7}$ The liquid Weber, 'We_l' is defined as, $We_l = (G^2d)/(\sigma\rho_H)$
4	Müller-Steinhagen and Heck Correlation [1986]	$[dp/dz]_F = G[1-x]^{1/3}+Bx^3$ Where the factor G is, $G = A+2[B-A]x$ A And B are the frictional pressure gradients for all the flow liquid flow, $[dp/dz]_{lo}$ and for all vapor flow, $[dp/dz]_{vo}$.

The selection of correlations is based on the statistical study of two phase correlations in the recent literature [9, 11, 12 and 13]. The correlations selected for comparison are given in Table-1.

The average deviation of CFD result from the pressure gradient obtained from different correlations is given in Table-2 for low, medium and high mass flux considered in the analysis for R134a and R407C. From Table-2, it is clear that the average deviation of pressure gradient obtained from CFD analysis from that of correlations is generally less for R134a compared to R407C. Grönnerud and Lockhart-Martinelli correlations

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show maximum deviation in the range of 75-80% and more than 100% respectively from the pressure drop obtained using CFD analysis particularly at high mass flux. The deviation of CFD result from Grönnerud correlation is in the range of less than 20% at low and medium qualities for a low mass flux which matches with result mentioned by Didi *et al.* [12] in their experimental study.

Table-2. Average deviation of CFD data with the separated flow correlations of pressure drop.

	Mass flux G (kg/m^{2}s)	**Grönnerud correlation**	**Friedel correlation**	**Lockhart-martinelli correlation**	**Chisholm correlation**	**Müllersteinhagen and Heck correlation**
R134a at 40^{0}C	1058	75%	13%	99%	12%	14%
	528	53.5%	4.4%	68%	44%	14.4%
	176	37%	28%	81%	64%	20%
R134a at 50^{0}C	1058	74%	20%	142%	11%	13%
	528	54.73%	6.2%	101%	47%	16.8%
	176	39%	35%	121%	60%	19%
R407C at 40^{0}C	1058	80.6%	23.3%	140%	16%	9.8%
	528	55%	10.9%	107%	49%	10.8%
	176	38%	36%	119%	70%	15.7%
R407C at 50^{0}C	1058	75.4%	22.3%	155%	12%	12%
	528	51%	7.4%	108%	49%	13.53%
	176	40%	37%	131%	60%	17%

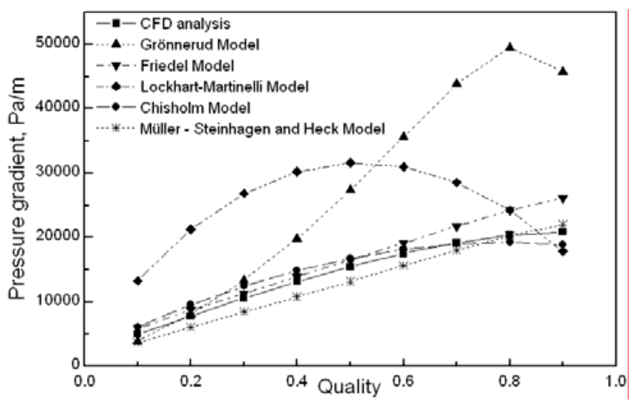

Figure-6. Pressure gradient of R134a at 40^{0}C for G = 1058 kg/m^{2}s.

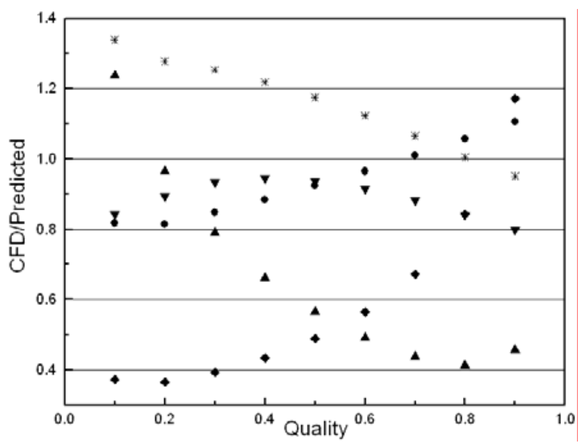

Figure-7. Comparison of CFD result with that of predicted for R134a at 40^{0}C for G = 1058 kg/m^{2}s.

CFD result closely follows the predictions of Friedel correlation with deviation in the range of 10% particularly for medium mass flux for low and high pressure refrigerants, in the range of 20% at high mass flux as shown in Figures 6 to 9 and in the range of 33% at low mass flux as represented in Figures 10 to 13.

VOL. 4, NO. 9, NOVEMBER 2009 ISSN 1819-6608
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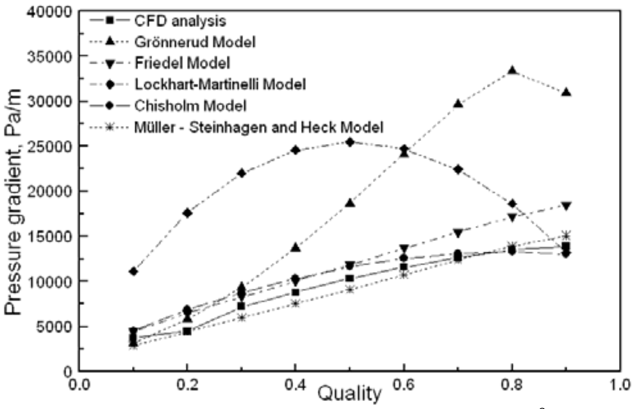

Figure-8. Pressure gradient of R407C at 40^0C for G = 1058 kg/m^{2}s.

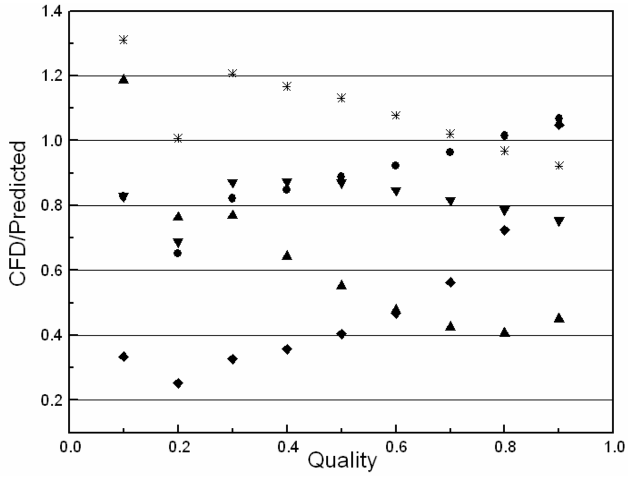

Figure-9. Comparison of CFD result with that of pedicted for R407C at 40^0C for G = 1058 kg/m^{2}s.

The deviation of CFD result with Chisholm correlation is in the range of 11% - 16% for high mass flux for all the refrigerants considered as represented in Table-2 and the same is shown in Figures, 7 and 9. The Chisholm, Friedel and Müller-Steinhagen and Heck correlations scatter very near to CFD data at high mass flux as shown in Figures 7 and 9.

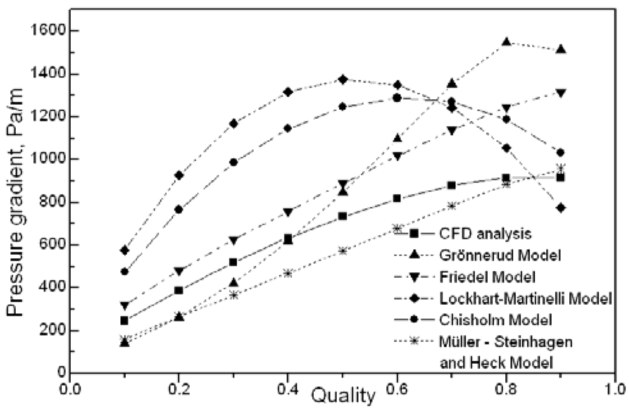

Figure-10. Pressure gradient of R134a at 40^0C for G = 176 kg/m^{2}s.

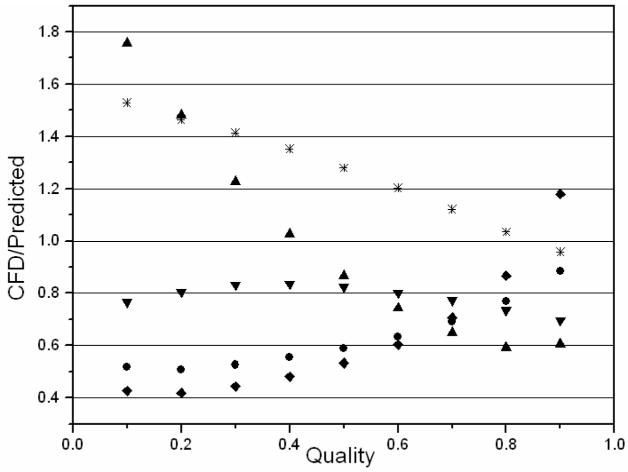

Figure-11. Comparison of CFD result with that of pedicted fr R134a at 40^0C for G = 176 kg/m^{2}s.

Table-2 shows that the deviation of CFD result from that of Müller-Steinhagen and Heck correlation is less than 20% for all the flow rates and refrigerants considered. This can be clearly observed in Figures 7 and 9. This result is in concurrence with the statistical study of Tribbe and Müller-Steinhagen [9], according to which statistically Müller-Steinhagen and Heck correlation predicts the pressure drop well for high mass flux where the flow regime is predominantly annular. The predictions of Müller-Steinhagen and Heck correlation differ from that of CFD result with as less as 15% deviation particularly for medium and high mass flux. At low mass flux, CFD result is within 20% deviation from Friedel Correlation as shown in Figures 11 and 13.

These results show that the results of CFD simulations are in good agreement with Chisholm, Friedel and Müller-Steinhagen and Heck correlations with a deviation less than 20%. This result justifies the use of Cicchitti Model of average viscosity for pressure drop predictions for the range of mass flux considered and for refrigerants, R134a and R407C. Similar results are obtained for R22 at all saturation temperatures and mass flux considered.

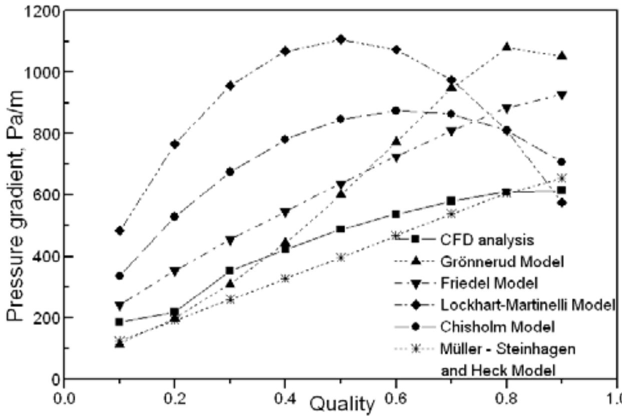

Figure-12. Pressure gradient of R407C at 40^0C for G = 176 kg/m^{2}s.

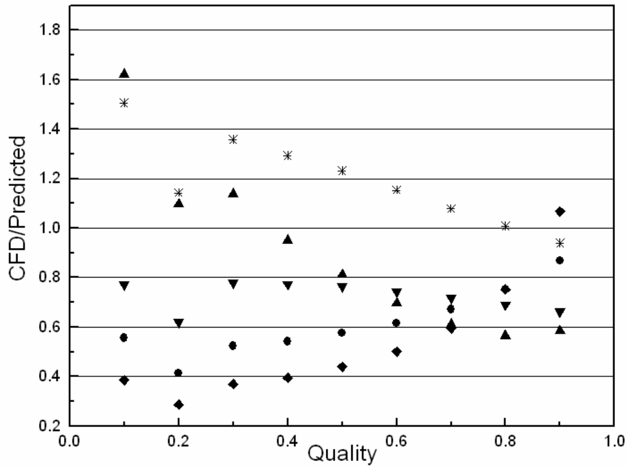

Figure-13. Comparison of CFD result with that of predicted fr R134a at 40^0C for G = 176 kg/m^{2}s.

Comparison of CFD results with experimental data

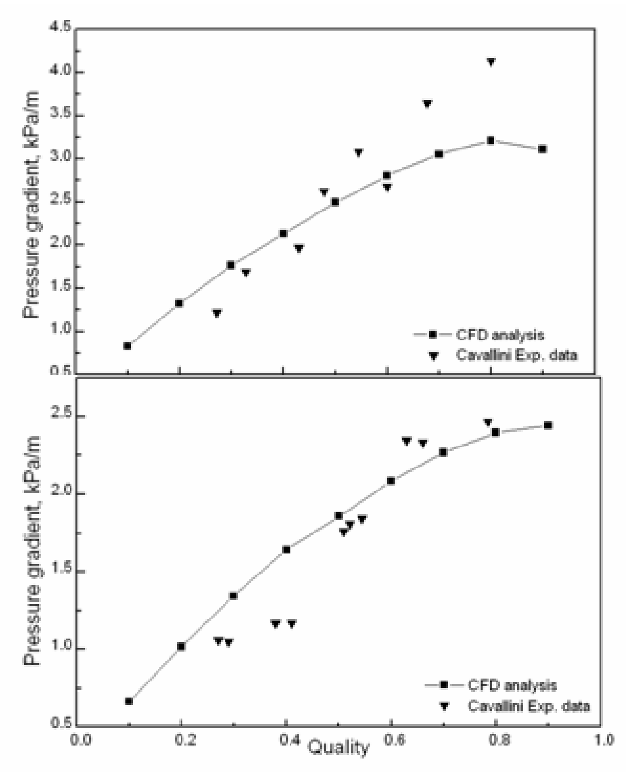

Figure-14. Comparison of CFD result with experimental values of R134a and R22 at Ts $= 40^0 C$, G = 400kg/m^{2}s

The CFD result is also compared with the experimental data of Cavallini et al. [10]. Figure-14 shows the comparison of pressure drop predicted by CFD result and experimental data of Cavallini for a mass flux of 400 kg/m^{2}s for refrigerants, R134a and R22. The deviation of CFD result from that of experimental data of Cavallini is found be higher at high qualities for low pressure refrigerant, R134a compared to that of high pressure refrigerant, R22 as shown in Figures 15 and 16, as the homogeneous model is developed for high pressure fluids where the liquid and vapor velocities are nearly same. Figure-16 shows that the experimental data matches well with CFD result for medium and high qualities for R22.

Nomenclature

A Cross sectional area of the tube, m^2
d Inside diameter of the tube, m
f Friction factor
$\overline{dF}$ Average wall friction, N
g Acceleration due to gravity, m/s^2
G Mass flux, $kg/m^2 s$
$\dot{m}$ Mass flow rate, kg/s
p Pressure acting on the control volume, Pa
P Perimeter of the tube, m
Re Reynolds Number based on average properties, $\frac{Gd}{\bar{\mu}}$
$\bar{u}$ Average z direction velocity of pseudo fluid
x Quality of the refrigerant
z Flow direction along length of the tube

Greek symbols

μ Viscosity of fluid, $Pa-s$
ρ Density of fluid, kg/m^3
τ Shear stress, N/m^2
ϕ^2 Two phase friction multiplier

Subscripts

F Friction
i Interface
l Liquid phase
s Saturation
TP Two phase
v Vapor phase
w Wall

REFERENCES

[1] John G. Collier. 1972. Convective Boiling and Condensation. Mc Graw Hill.

[2] Lockhart R. W., Martinelli R. C. 1947. Proposed correlation of data for isothermal two phase, two component flows in pipes. Chemical Engineering Proceedings. 45(1): 39-48.

[3] Martinelli R.C., Nelson D. B. 1948. Prediction of pressure drop during forced-circulation boiling of water. Trans. ASME. 70: 695-702.

[4] Grönnerud R. 1979. Investigation of liquid hold-up, flow-resistance and heat transfer in circulation type of evaporators, part IV: two-phase flow resistance in boiling refrigerants. In: Annexe1972-1, Bull. De l'Inst. du Froid.

[5] Chisholm D. 1973. Pressure gradients due to friction during the flow of evaporating two phase mixtures in smooth tubes and channels. Int. J. Heat and Mass Transfer. (16): 347-358.

[6] Friedel L. 1979. Improved friction pressure drop correlations for horizontal and vertical two phase pipe flow. Paper E2, European Two Phase Flow Group Meeting, Ispra, Italy.

[7] Müller-Steinhagen. H., Heck. K. 1986. A Simple Friction Pressure Drop Correlation for Two-Phase Flow in Pipes. Chem. Eng. Process. 20: 297-308.

[8] Kattan N., Thome J. R., Favrat D. 1998. Flow boiling in horizontal tubes: Part I- Development of diabatic two phase flow pattern map. J. of Heat Transfer. 120: 140-147.

[9] Tribbe. C., Müller-Steinhagen. H. 2000. An Evaluation of the Performance of Phenomenological models for Predicting Pressure Gradient during Gas-Liquid Flow in Horizontal Pipelines. Int. J. Multiphase Flow. 26: 1019-1036.

[10] Cavallini A., Censi G., Del Col D., Doretti L., Longo G.A., Rossetto L. 2001. Expérimental Investigations on condensation heat transfer and pressure drop of new HFC refrigerants. Int. J. of Refrigeration. 24: 73-87.

[11] Ould Didi. M. B, Kattan. N, Thome. J. R. 2002. Prediction of Two Phase Pressure Gradients of Refrigerants in Horizontal Tubes. Int. J. Refrigeration. 25: 935-947.

[12] Moreno Quibén. J., Thome. J. R. 2007. Flow Pattern Based Two-Phase Frictional Pressure Drop Model for Horizontal Tubes, Part I: Diabatic and Adiabatic Experimental Study. Int. J. Heat Fluid Flow. 28(5): 1049-1059.

[13] Moreno Quibén. J., Thome. J. R. 2007. Flow Pattern Based Two-Phase Frictional Pressure Drop Model for Horizontal Tubes, Part II: New Phenomenological Model. Int. J. Heat Fluid Flow. 28(5): 1060-1072.