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CFD Analysis of Two Phase Flow in a Horizontal Pipe – Prediction of Pressure Drop

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Abstract**—**In designing of condensers, the prediction of pressure drop is as important as the prediction of heat transfer coefficient. Modeling of two phase flow, particularly liquid – vapor flow under diabatic conditions inside a horizontal tube using CFD analysis is difficult with the available two phase models in FLUENT due to continuously changing flow patterns. In the present analysis, CFD analysis of two phase flow of refrigerants inside a horizontal tube of inner diameter, 0.0085 m and 1.2 m length is carried out using homogeneous model under adiabatic conditions. The refrigerants considered are R22, R134a and R407C. The analysis is performed at different saturation temperatures and at different flow rates to evaluate the local frictional pressure drop. Using Homogeneous model, average properties are obtained for each of the refrigerants that is considered as single phase pseudo fluid. The so obtained pressure drop data is compared with the separated flow models available in literature.

Keywords**—**Adiabatic conditions, CFD analysis, Homogeneous model and Liquid – Vapor flow.

I. Introduction

IN internal condensation, the vapor and liquid flow simultaneously inside the channel or pipe. The resulting two phase flow is more complicated physically than single phase flow. In addition to the usual inertia, viscous and pressure forces present in single phase flow, two phase flows are also affected by interfacial tension forces, the wetting characteristics of the liquid on the tube wall and the exchange of momentum between the liquid and vapor phases in the flow. Because of these effects, the morphology of two phase flow patterns varies for different geometries of channels or tubes and their orientations. Condensation inside horizontal tubes is governed by a combination of gravity forces and interfacial shear stresses, the relative contribution of which change with geometry and fluid flow conditions. Generalized analytical treatment of the vapor – liquid flow is extremely difficult due to this.

Basic one dimensional model of single component two phase flow is developed for the stratified flow where each phase is in contact with the channel and has a common interface. The resulting momentum and energy equations are further solved for pressure drop using different models like Homogeneous flow model and Separated flow model.

Prediction of two phase pressure drop inside a tube is of paramount importance to the design and optimization of refrigeration, air-conditioning and heat pump systems. From the separated flow model, the frictional pressure drop for two phase, two component, isothermal flow in horizontal tubes was initially developed by Lockhart and Martinelli in 1944 [1] using the two phase multiplier. A later extension of their work to cover the accelerative component resulted into well known Martinelli-Nelson correlation for the prediction of pressure drop for forced circulation boiling and condensation. Later, the calculation methodology for two phase friction multiplier was developed by Thom [1], Baroczy [1] and Chisholm [1].

In the recent literature, Tribbe and Muller-Steinhagen [2] presented an extensive comparison of 35 two phase pressure drop predictive methods compared to a large database for the following fluid combinations: air-oil, cryogenics, steam-water, air-water and several refrigerants. They made a statistical comparison for this large database also segregating the data by fluid. They found that statistically the method of Muller-Steinhagen and Heck [3] gave the best and most reliable results. In the revised edition, Chapter 13 of Engineering Data book III of the Wolverine Tube Inc., [4] it is mentioned that overall, the Gronnerud and the Muller-Steinhagen and Heck methods to be equally the best, while the Friedel method was the third best in a comparison of seven leading predictive methods. In addition, the following recommendations were made:

- $\left(\mu_l / \mu_g\right) < 1000$ and $G < 2000$ kg/m^{2}s, Friedel correlation should be used.
- $\left(\mu_l / \mu_g\right) > 1000$ and $G > 100$ kg/m^{2}s, Chisholm correlation should be used.
- $\left(\mu_l / \mu_g\right) > 1000$ and $G < 100$ kg/m^{2}s, Lockhart – Martinelli correlation should be used.

Kattan et al. [5] segregated the data by flow regimes using the flow pattern map and the authors found that predictive methods work differently with varying the flow regime, since the models are not able to capture completely the effects of the variations in flow structure. Recently, Moreno Quiben and Thome [6,7] published a work in which they made a comprehensive study to run accurate experiments. Then using

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a new flow pattern map by Wojtan et al. [8], they built a flow pattern based model for predicting pressure drops. However, there is not much data reported in the literature on modeling and analysis of two phase flow using CFD software.

II. Separated Flow Model

The basic equations for the separated flow model are not dependent on the particular flow configuration. It is assumed that the velocities of each phase are constant.

Pressure drop during In-Tube condensation can be obtained from the two phase flow momentum equation or energy equation [1] based on the separated flow model. It is assumed that velocities of each phase are constant, in any given cross section within the zone occupied by the phase. From the two phase flow momentum equation, the pressure drop equation for condensation inside horizontal tube is developed as:

$$-\left(\frac{dp}{dz}\right) = -\left(\frac{dp}{dz}\right)_f - \left(\frac{dp}{dz}\right)_a - \left(\frac{dp}{dz}\right)_z \tag{1}$$

$$-\left(\frac{dp}{dz}\right)_z = g\left[\varepsilon\rho_g + (1-\varepsilon)\rho_l\right] \tag{2}$$

$$-\left(\frac{dp}{dz}\right)_a = G^2 d\left\{\left(\frac{x^2}{\rho_g\varepsilon}\right) + \left(\frac{(1-x)^2}{(\rho_l(1-\varepsilon))}\right)\right\}/dz \tag{3}$$

where $-\left(\frac{dp}{dz}\right)_z$ is the gravity pressure drop component, $-\left(\frac{dp}{dz}\right)_a$ is the acceleration pressure gradient and $-\left(\frac{dp}{dz}\right)_f$ is frictional pressure gradient.

The gravity pressure gradient is relevant only for long vertical tubes, while the momentum pressure drop results in an increase in the pressure at the exit than at the inlet, as for condensing flows, the kinetic energy of outgoing flow is smaller than that of incoming flow. Hence, it is common practice 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.

In addition, evaluation of momentum and gravity pressure drop requires the void fraction data. In condenser, due to high vapor density which is the result of high pressure on condenser side, at a given mass flux and quality, the vapor velocity is slower than that of evaporator. Lower velocities bring the flow closer to stratified flow regime. In this region, void fraction predictions by any of the common void fraction models are inaccurate resulting into inaccurate pressure drop predictions. Hence in the present study, only frictional pressure drop is evaluated using CFD analysis and compared with the correlations based on separated flow model available in the literature.

The values of experimental condensation heat transfer coefficients and pressure drop values reported in recent literature are quasi local, obtained when small quality variations occur in the test tube. This experimental procedure helps in understanding of physical phenomenon of condensation. In this procedure, part of condensation is performed within a test section. The experimental data is reported for the average quality across the test section. In the present study, to obtain the local pressure drop data, the two phase flow is modeled for a particular quality under adiabatic conditions using CFD software.

A. Working Fluid

The working fluids considered are refrigerants, R22, R134a and R407C.

B. Dimensions of the Tube

ID of the tube = 8.5mm
Length of the tube = 1.2 m

III. Mathematical Formulation

A homogeneous flow model, a special case of separated flow analysis in which vapor and liquid velocities are assumed to be constant and equal. In this model, two-phase flow is treated as single phase pseudo fluid with suitably averaged properties of the liquid and vapor phase. For steady homogeneous flow model, the basic equations for condensation inside a horizontal tube are reduced to the following form:

$$\text{Continuity Equation: } \dot{m} = \bar{\rho}\bar{u}A \tag{4}$$

$$\text{Momentum Equation: } -Adp - d\bar{F} - A\bar{\rho}gdz = \dot{m}d\bar{u} \tag{5}$$

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

$$d\bar{F} = \tau_w(Pdz) \tag{6}$$

This frictional pressure drop equation is the Fanning Equation, in which the two phase friction factor can be calculated by the Blasius equation using the average properties.

$$f_{TP} = 0.079\left[\frac{Gd}{\bar{\mu}}\right]^{-0.25} \tag{7}$$

The average properties for homogeneous pseudo fluid are developed from the fundamentals as mentioned elaborately in Collier [1].

The average fluid density is given by:

$$\frac{1}{\bar{\rho}} = \left[\frac{x}{\rho_g} + \left(\frac{1-x}{\rho_l}\right)\right] \tag{8}$$

Possible forms of relationships for mean two phase viscosity, $\bar{\mu}$ based on limiting conditions, at x =0, $\mu_l = \bar{\mu}$ and at x =1, $\mu_g = \bar{\mu}$, are:

$$\frac{1}{\bar{\mu}} = \left[\frac{x}{\mu_g} + \left(\frac{1-x}{\mu_l}\right)\right] \quad \text{(McAdams)} \tag{9}$$

$$\bar{\mu} = x\mu_g + (1-x)\mu_l \quad \text{(Cicchitti)} \tag{10}$$

$$\bar{\mu} = \bar{\rho}\left[\frac{x\mu_g}{\rho_g} + \frac{(1-x)\mu_l}{\rho_l}\right] \quad \text{(Duker)} \qquad (11)$$

Based on the above average properties, two phase frictional pressure drop for horizontal tube of internal diameter, d is calculated as:

$$\Delta P = \frac{2 f_{TP} G^2 L}{\bar{\rho} d} \qquad (12)$$

IV. CFD Analysis

The working fluids considered in the present case are refrigerants as in tube condensation of refrigerants has lot of applications in refrigeration and air conditioning industry. The refrigerants are selected such that they cover the wide pressure range and variety. R407C is selected as it is high pressure mixture with large temperature glide. To calculate the average properties, the condensation temperatures considered are 40^0C, 50^0C and 60^0C.

A. Modeling of the Tube

3-D tube is modeled in GAMBIT and exported to FLUENT. Analysis is done for grid independent study and the optimum mesh size is selected as 24X80. The meshed model of pipe is shown in Figs. 1 and 2.

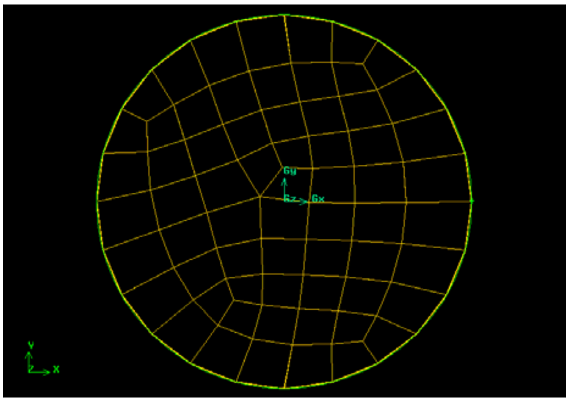

Fig. 1 Front view of the 3-D mesh model

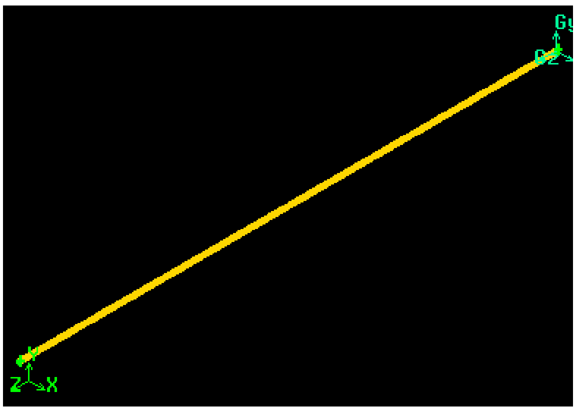

Fig. 2 Isometric view of the meshed 3-D pipe

B. Selection of Solver

3-D Segregated model with implicit formulation is selected. The solver conditions are set for steady flow with enhanced wall treatment. Adiabatic conditions are considered at the wall.

C. Selection of Viscous Model

Since for the entire range of flow rates, the Reynolds Number based on the average properties exceeds 2300, $k-\varepsilon$ model is considered.

D. Material Properties

With these solver conditions set, for each quality, x average properties are calculated for each refrigerant and material properties are given. Simulations are performed for different average kinematic viscosity values calculated using equations, (9) – (11).

E. Setting of Operating Conditions

The operating pressure is set as per the saturation pressure considered for each refrigerant. Since the flow rates considered fall under turbulent – turbulent case, the gravity is not considered in the flow analysis.

F. Applying the Boundary Conditions

- Mass flow inlet: Mass flow rates of 0.01 kg/s to 0.06 kg/s is considered. The flow rates selected are to cover the entire flow regimes for a horizontal tube.
- Outflow condition: At the outlet, outflow condition is given as fluid is not left to atmosphere.
- Wall: No slip condition is given at the wall.

G. Solution Controls

SIMPLE algorithm is selected for Pressure – Velocity coupling. Second order upwind method is selected for discretization of momentum, turbulence K.E and turbulence dissipation rate. For pressure, standard discretization scheme is selected.

H. Convergence

Convergence criteria of 0.001 is considered.

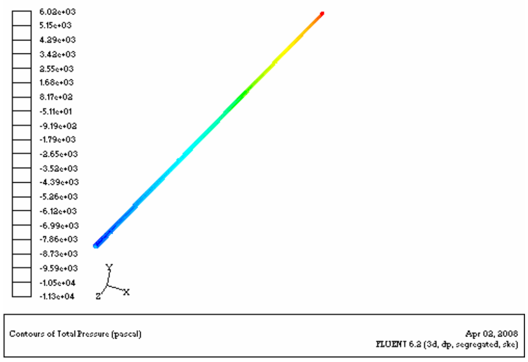

Fig. 3 Variation of total pressure for R134a at 40^0C and G- 1058 kg/m^2s at a quality of 0.5

I. Post Processing

Thus for every quality, x and for every refrigerant at each flow rate, the model is simulated to get the pressure drop data. The pressure drop data is obtained from area weighted value of shear stress for each case.

V. Results and Discussions

For all the refrigerants, the pressure drop is evaluated using the different models of kinematic viscosity as given by equations (9) to (11) at saturation temperatures of 40^0C, 50^0C and 60^0C. The variation of pressure drop obtained using different models of $\bar{\mu}$ with quality for R134a at saturation temperatures of 40^0C and 60^0C are presented in Figs. 4 - 7.

Fig. 4 and 5 show that there is a noticeable variation in the pressure drop values calculated from the three models at a particular quality at low saturation pressure corresponding to T_s = 40^0C, the variation decreasing with the increase of flow rate.

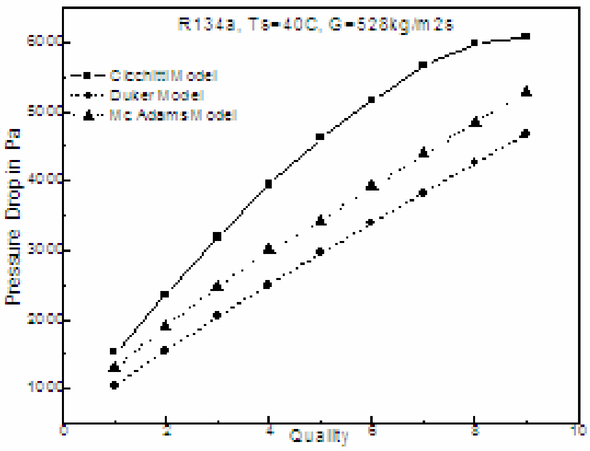

Fig. 4 Pressure drop with different models of $\bar{\mu}$ for R134a at T_s= 40^0C at G= 528 kg/m^{2}s

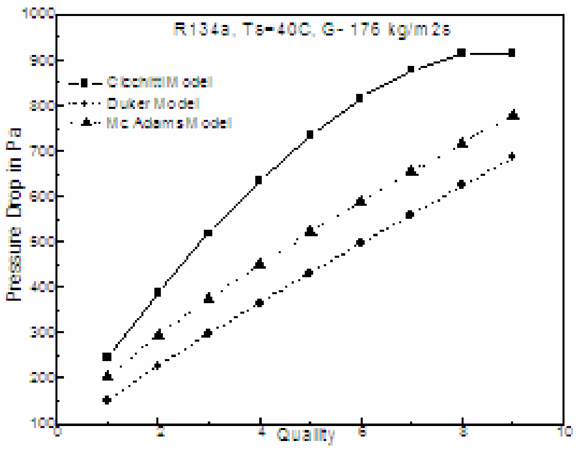

Fig. 5 Pressure drop with different models of $\bar{\mu}$ for R134a at T_s= 40^0C at G= 176 kg/m^{2}s

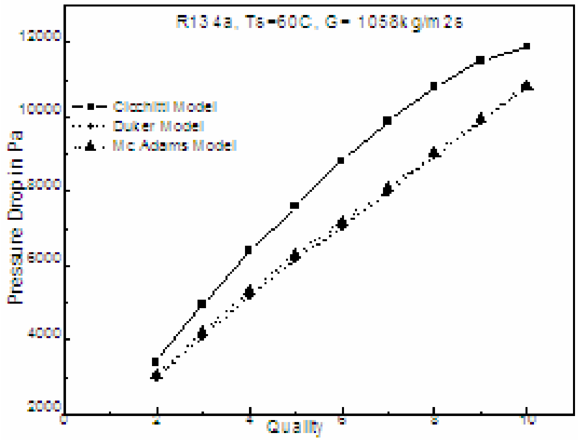

Fig. 6 Pressure drop with different models of $\bar{\mu}$ for R134a at T_s= 60^0C at G= 1058 kg/m^{2}s

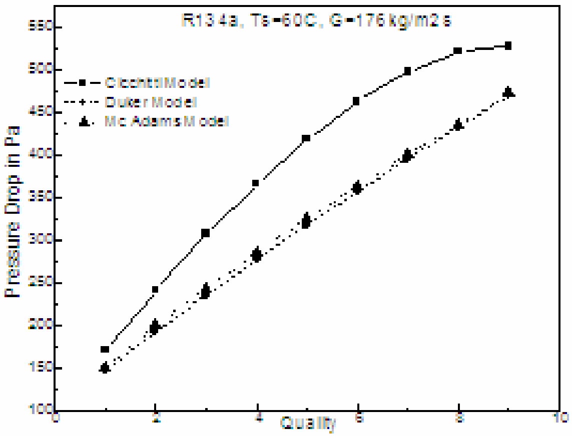

Fig. 7 Pressure drop with different models of $\bar{\mu}$ for R134a at T_s= 60^0C at G= 176 kg/m^{2}s

For the same Refrigerant R134a, the variation in pressure drop data obtained from different models at a given quality is reduced at the high pressure corresponding to the saturation temperature of 60^0C particularly for the Duker's and McAdam's models as shown in Figs. 6 and 7. The reason can be explained by means of the average kinematic viscosity data for R134a at saturation temperatures of 40^0C and at 60^0C presented in Table I and Table II.

TABLE I
Average Properties for R134a at 40^0C

X	$\bar{\rho}$	$\bar{\mu}_C X10^{-6}$	$\bar{\mu}_{MA} X10^{-6}$	$\bar{\mu}_D X10^{-6}$
0.1	358.44	148.67	74.24	55.08
0.2	212.41	133.55	48.00	34.95
0.3	150.92	118.42	35.47	26.47
0.4	117.04	103.3	28.12	21.80
0.5	95.58	88.17	23.30	18.84
0.6	80.78	73.04	19.88	16.80
0.7	69.94	57.92	17.35	15.31
0.8	61.67	42.79	15.38	14.17
0.9	55.14	27.68	13.81	13.27

TABLE II
AVERAGE PROPERTIES FOR R134A AT 60^0C

X	$\bar{\rho}$	$\bar{\mu}_C X10^{-6}$	$\bar{\mu}_{MA} X10^{-6}$	$\bar{\mu}_D X10^{-6}$
0.1	499.24	113.43	69.03	60.98
0.2	327.08	102.36	47.76	41.27
0.3	243.21	91.28	36.51	31.66
0.4	193.58	80.21	29.54	25.98
0.5	160.77	69.14	24.81	22.22
0.6	137.47	58.07	21.39	19.56
0.7	120.07	47.00	18.79	17.56
0.8	106.58	35.92	16.76	16.02
0.9	95.81	24.85	15.13	14.79

From Table I and Table II, it is clear that at higher saturation pressures, the average kinematic viscosity values calculated using Duker and McAdam's models are nearly same resulting the similar variation of pressure drop data obtained using these two models.

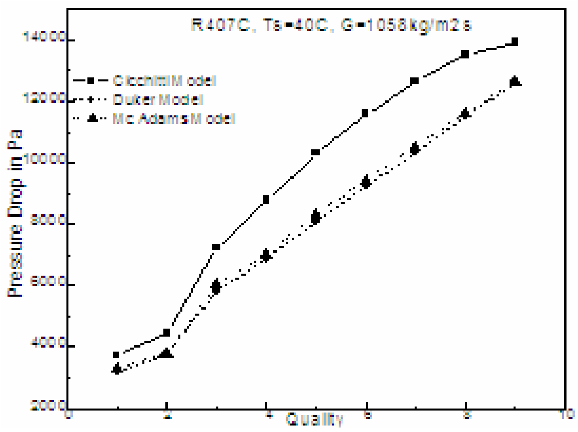

Fig. 8 Variation of Pr. Drop for different models of $\bar{\mu}$ for R407C at 40^0C

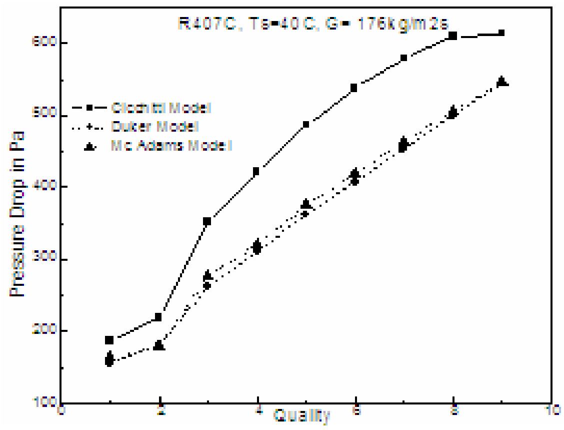

Fig. 9 Variation of Pr. Drop for different models of $\bar{\mu}$ for R407C at 40^0C

Figs. 8 and 9 represent the variation of pressure drop calculated from three models of average kinematic viscosity with quality for refrigerant mixture, R407C. Since R407C is a high pressure refrigerant, the pressure drop data predicted using different models of $\bar{\mu}$ matches particularly at high flow rates. The pressure drop variation for R22 is similar to that of R407C.

From the Figs. 4- 7 and from Figs. 8 and 9, it is clear that all the three models tend to converge with the increase of mass flow rate since the effect of $\bar{\mu}$ is not dominant at high flow rates in the calculation of friction factor and pressure drop.

Figs. 4- 9 show that pressure drop obtained with Cicchitti model of $\bar{\mu}$ is higher at any given average quality along the length of the tube. Hence Cicchitti linear model of average kinematic viscosity is selected for further simulation in CFD for all the refrigerants at different saturation temperatures and flow rates.

Using Cicchitti model of $\bar{\mu}$, the pressure drop data is obtained from CFD analysis and the results are presented in the Fig.7. The adiabatic pressure drop increases with increase of average quality along the tube for a particular flow rate. This is due to average density and average viscosity of any refrigerant approaching the density and viscosity of refrigerant gas. This reduction in the average properties with quality increases the Reynolds Number, results in increase of friction factor and pressure drop. The so obtained adiabatic pressure drop values at each quality for a particular flow rate and refrigerant are compared with the pressure drop correlations available in the literature.

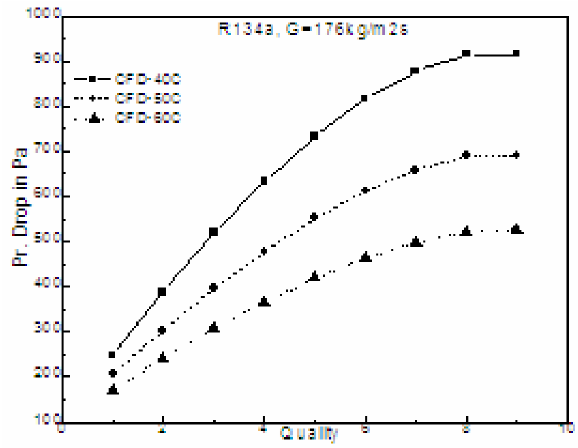

Fig. 10 CFD result of Pr. Drop of R134a at G=176 kg/m^2s for different saturation temperatures

VI. COMPARISON OF PRESSURE DROP DATA

The frictional pressure drop predictions based on separated flow model are considered for comparison as the CFD analysis is performed based on the special case of separated flow model.

The correlations used for comparison are:

1. Homogeneous flow model
2. Gronnerud Correlation
3. Friedel Correlation
4. Lockhart – Martinelli Correlation
5. Chisholm Correlation
6. Muller – Steinhagen and Heck Correlation

For different flow rates, the graphs of comparison are presented below:

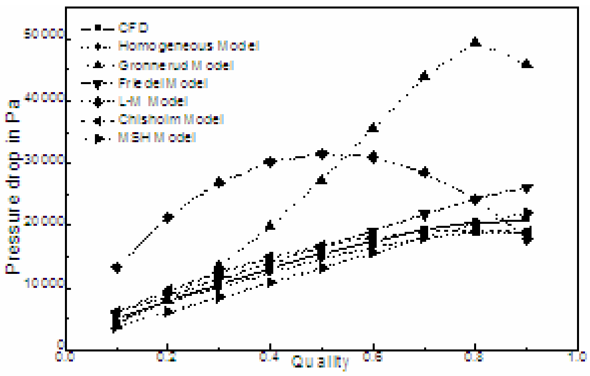

Fig. 11 Pressure drop of R134a at 40^0C with G= 1058 kg/m^{2}s

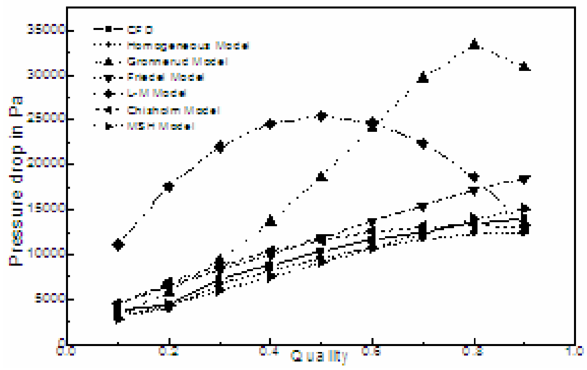

Fig. 12 Pressure drop of R407C at 40^0C with G= 1058 kg/m^{2}s

At high flow rates, for the same saturation temperature, low pressure refrigerant, R134a and high pressure refrigerant mixture, R407C show the same trend of pressure drop with quality for different correlations and from CFD analysis as shown in Figs. 11 and 12. It can be observed that the pressure drop penalty is higher for R134a compared to R407C.

Figs. 11 and 12 show that all the correlations merge except Lockhart - Martinelli and Gronnerud Correlations at high flow rates. Similar results are obtained at high flow rate with R134a, R22 and R407C at other saturation temperatures.

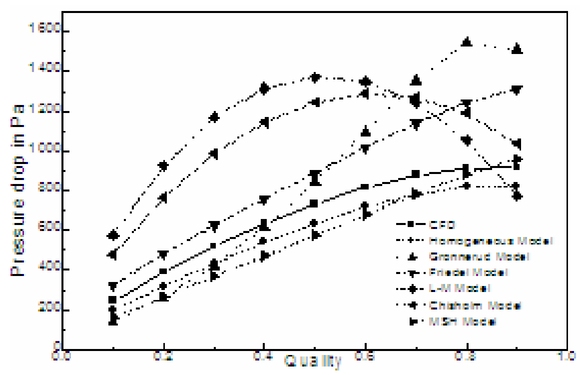

Fig. 13 Pressure drop of R134a at 40^0C with G= 176 kg/m^{2}s

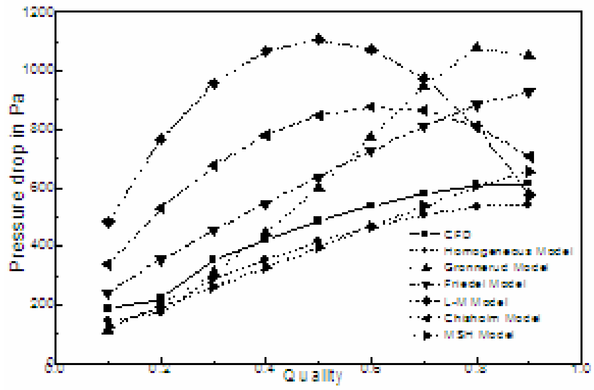

Fig. 14 Pressure drop of R407C at 40^0C with G= 176 kg/m^{2}s

At a low mass flux of 176 kg/m^{2}s, all the correlations merge well for both the refrigerants as shown in Figs. 13 and 14. The deviation is less pronounced for R134a compared to R407C at a saturation temperature of 40^0C. The average deviation at highest and lowest mass flux for R134a and R407C is given in Table III.

TABLE III
AVERAGE DEVIATION OF CFD DATA WITH THE SEPARATED FLOW CORRELATIONS

	G in kg/m^2 s	dp-Gr	dp-Fr	dp-L-M	dp-Ch	M-S-H
R134a at 40^0C	1058	75%	13%	99%	12%	14%
	176	37%	28%	81%	64%	20%
R134a at 60^0C	1058	74%	20%	142%	11%	13%
	176	39%	35%	121%	60%	19%
R407C at 40^0C	1058	80.6%	23.3%	140%	16%	9.8%
	176	38%	36%	119%	70%	15.7%
R407C at 50^0C	1058	75.4%	22.3%	155%	12%	12%
	176	40%	37%	131%	60.%	17%

dp-Gr : Deviation of CFD pressure drop - Gronnerud Correlation
dp-Fr : Deviation of CFD pressure drop - Friedel Correlation
dp-L-M: Deviation of CFD pressure drop - Lockhart – Martinelli
dp-Ch : Deviation of CFD pressure drop - Chisholm Correlation
dp-MSH: Deviation of CFD pressure drop- Muller – Steinhagen and Heck

The average deviation pressure drop obtained from CFD analysis from all the correlations is considerably less for R134a compared to R407C. Further, the deviation is less at 40^0C compared to that of 50^0C. Gronnerud and Lockhart – Martinelli correlations show maximum deviation from the pressure drop obtained using CFD analysis particularly at high mass flux. The deviation of Gronnerud correlation from CFD results is less for low mass flux, in the range of 40%.

Friedel correlation predicts the pressure drop data of CFD analysis well with less deviation at high mass flux in comparison with that at low mass flux.

The pressure drop predictions using Chisholm correlation shows less deviation at high mass flux and

Table III shows that Muller – Steinhagen and Heck predicts the CFD results well for both low pressure refrigerant, R134a and high pressure mixture, R407C for all flow rates and saturation temperatures.

VII. Conclusion

CFD analysis of two phase, single component tube flow is modeled using FLUENT. Average properties of the refrigerants, R134a, R22 and R407C are evaluated using Homogeneous model for each quality. A pseudo single phase fluid is thus considered in the CFD analysis. The resulting pressure drop data obtained at adiabatic conditions match well by separated flow correlations. The CFD results match well with Muller – Steinhagen and Heck correlation.

VIII. Nomenclature

$\dot{m}$ Mass flow rate in kg/s
G Mass flux, $kg/m^2 s$
x Quality of the refrigerant
T_s Condensing temperature
$\bar{u}$ Average z direction velocity of pseudo fluid
A Cross sectional area of the tube in m^2
g Acceleration due to gravity
d Inside diameter of the tube in m
f_{TP} Two phase friction factor based on average properties
ΔP Pressure drop across the tube

Greek Symbols

τ_w Wall shear stress
μ Dynamic viscosity in $kg\ m/s$
ρ Density in kg/m^3
$\bar{\mu}$ Average viscosity of the pseudo fluid $kg\ m/s$
$\bar{\rho}$ Average density of pseudo fluid in kg/m^3

Subscripts

l Liquid phase
g Gaseous phase
w Wall
s Saturation
TP Two phase

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