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Experimental Analysis and CFD Simulation of Pressure Drop of Carbon Dioxide in Horizontal Micro Tube Heat Exchangers

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Abstract

The condensation of carbon dioxide in single horizontal micro tube condensers of inner diameter ranged from 0.6 mm up to 1.6 mm over mass flow rates from $2.5*10^{-5}$ to $17*10^{-5}$ kg/s and vapor qualities from 0.0 to 1.0 is studied experimentally. The inlet condensing pressure is changed from 33.5 to 45 bars. The saturation temperature ranged from -1.5 ^{o}C up to 10 ^{o}C.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 flow rates and different saturation temperatures. The values of pressure drop obtained from the simulation of carbon dioxide are compared with three different prediction correlations and the pressure drop experimental data resulted from this study. The Muller method gave the best fit to the experimental results with average standard deviations of 6.4%, followed by CFD model with 8.9% then Friedel and Gronnerud methods with 12.1% and 17.3%, respectively.

Keyword: Refrigerant, Frictional Pressure drop, Micro tubes, Carbon Dioxide, Condensers, CFD, Two-phase flow.

INTRODUCTION

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. Also as a main type of the pressure, the prediction of the two-phase frictional pressure drop during the condensation of two-phase flow refrigerants is important for accurate design and optimization of refrigeration, air-conditioning systems.

The two phase flow in forced convective condensation of refrigerants is affected by inertial, viscous and pressure forces. In addition it is also affected by interfacial tension, liquid wetting the tube wall and the momentum exchange between the liquid and vapor 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.

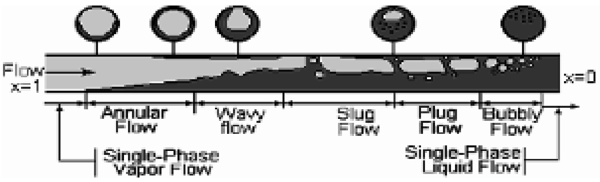

Figure 1. Flow patterns for horizontal condenser

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. Associated with the compactness resulted from the use of CO_2 as working fluid, the use of the micro tube technology (tubes having diameter of less than 3 mm) in the heat exchangers design yields a very compact and lightweight equipments. The high heat transfer coefficients and significant potential in decreasing the heat exchanger surface area are the major advantages of using this kind of geometry. For these reasons micro tube heat exchangers have been used in bioengineering and microelectronics as well as in evaporators and condensers of refrigeration systems. The optimal use of the two-phase pressure drop during the condensation of refrigerants to obtain the maximum heat transfer performance is one of the primary design goals. Also, the accurate prediction of two-phase pressure drops is a particularly important aspect of the first and second law optimizations of these systems. Yun and Kim [1] investigated two-phase pressure drops of CO_2 in mini tubes with inner diameters of 2.0 and 0.98 mm and in micro channels with hydraulic diameters from 1.08 to 1.54 mm. The pressure drop of CO_2 in the mini tubes shows very similar trends with those in large diameter tubes. Huai et al. [2] presented experimentally a study of boiling heat transfer and pressure drop of CO_2 flowing in a multi-port extruded aluminum test section, which had 10 circular channels, each with an inner diameter of 1.31 mm. The results indicated that pressure drop along the test section is very small. Ould Didi et al [3] studied the prediction of two-phase pressure gradients of refrigerants during evaporation in horizontal tubes of more than 10 mm diameter for different mass velocities and different vapor qualities. The resulted experimental data have then been compared against seven two-phase frictional pressure drop prediction methods. Kattan et al [4] studied the flow boiling in horizontal tubes through the development of an adiabatic two phase flow pattern map. Moreno Quibén. J., Thome. J. R.[5] presented a flow pattern based two-phase frictional pressure drop model for horizontal tubes through an adiabatic experimental study. In the present study, experimental test data resulted from the condensation of CO2 in horizontal copper micro tubes under the effect of free convection inside a chest freezer have been compared to the following three widely quoted prediction methods for the frictional pressure drop in two-phase flows: Friedel [6], Gronnerud [7], and Muller-Steinhagen and Heck [8].The two-phase pressure drop tests cover three different tube diameters (0.6 ,1 and 1.6 mm) with total length of 29.72 m over mass flow rates from 2.5*10-5 to 17*10-5 kg/s for saturation pressures ranging from 33.5 to 45 bars and saturation temperature ranged from (-1.5 oC up to 10 oC).The variation of the vapor quality with the frictional pressure drop were plotted for different operating parameters.

Two Phase Pressure Drop

The total two-phase pressure drop for flows inside micro tubes is consisting of three components: the static pressure drop (ΔP_{static}); the momentum pressure drop (ΔP_{mom}); and the frictional pressure drop (ΔP_{frict}) and it is represented as follows:

$$\Delta P_{total} = \Delta P_{static} + \Delta P_{mom} + \Delta P_{frict} \quad (1)$$

Because ΔP_{static}= 0 for horizontal tube, The momentum pressure drop represents the change in kinetic energy of the flow and is for the present case given by: 〖ΔP〗_mom=

$$\dot{m}^2 total\left\{\left[\frac{(1-x)^2}{\rho_L(1-\varepsilon)}+\frac{x^2}{\rho G^{\varepsilon}}\right]_{out} - \left[\frac{(1-x)^2}{\rho_L(1-\varepsilon)}+\frac{x^2}{\rho G^{\varepsilon}}\right]_{in}\right\} \quad (2)$$

Where: m_{total} is the total mass velocity of liquid plus vapor and x is the vapor quality. In the present study, the void function ε is obtained from Ould Didi et al [3] version of the drift flux model of Rouhani and Axelsson [9] for horizontal tubes:

$$\varepsilon = \frac{x}{\rho_G}\left[(1+0.12(1-x))\left(\frac{x}{\rho_G}+\frac{1-x}{\rho_L}\right)+\frac{1.18(1-x)[g\sigma(\rho_{L-\rho_G})]^{0.25}}{\dot{m}_{total}^2\ \rho_L^{0.5}}\right]^{-1}$$

(3)

Hence, the experimental two-phase frictional pressure drop is obtainable from equation (1) by Subtracted the calculated momentum pressure drop from the measured total pressure drop.

Comparison of CFD and Experimental Pressure Drop Data with Literature Correlations

The following three different literature two-phase frictional pressure drop correlations are compared to the present CFD and experimental data.

Friedel correlation

The two phase frictional pressure drop according to Friedel [6] is calculated by using the following equation:

$$\Delta p_{frict} = \Delta p_L\ \varphi_{L_o}^2 \quad (4)$$

Where ΔP_L is the liquid-phase pressure drop at which it is calculated according to the following equation:

$$\Delta p_L = 4 f_L (L/D_i)\ \dot{m}_{total}^2 (1-x)^2 (1/2_{\rho_L}) \quad (5)$$

Where the liquid friction factor is calculated as follows:

$f = \frac{0.079}{\mathrm{Re}^{0.25}}$, and liquid Reynolds number is obtained from:

$$\mathrm{Re}=\frac{\dot{m}_{total} Di}{\mu}$$

And the vapor quality (x) is in the range of: $0 \le x < 1$

Friedel two-phase multiplier using the liquid dynamic viscosity (μ_L) is correlated as follows:

$$\varphi_{L_o}^2 = E + \frac{3.24FH}{Fr_h^{0.045} We_L^{0.035}} \quad (6)$$

The recommended value for the ratio of (μ_L/μ_G) for Friedel's method is typically less than 1000.

Where Fr_h, E, F and H are as follows:

$$Fr_h = \frac{\dot{m}_{total}^2}{g d_i \rho_h^2} \quad (7)$$

$$E = (1-x)^2 + x^2 \frac{\rho_L f_G}{\rho_G f_L} \quad (8)$$

$$F = x^{0.78} (1-x)^{0.224} \quad (9)$$

$$H = \left(\frac{\rho_L}{\rho_G}\right)^{0.91} \left(\frac{\mu_G}{\mu_L}\right)^{.019} \left(1-\frac{\mu_G}{\mu_L}\right)^{0.7} \quad (10)$$

The liquid Weber (We_L) is defined as:

$$We_L = \frac{\dot{m}_{total}^2\ D_i}{\sigma_{\rho_h}} \quad (11)$$

And the homogeneous density ρ_h is used:

$$\rho_h = \left(\frac{x}{\rho_G} + \frac{1-x}{\rho_L}\right)^{-1} \quad (12)$$

Gronnerud correlation

The two phase frictional pressure drop according to Gronnerud [7] is calculated as follows:

$$\Delta p_{frict} = \varphi_{gd}\ \Delta p_L \quad (13)$$

And;

$$\varphi_{gd} = 1 + \left(\frac{dp}{dz}\right)_{Fr} \left[\frac{\left(\frac{\rho_L}{\rho_G}\right)}{\left(\frac{\mu_L}{\mu_G}\right)^{0.25}} - 1\right] \quad (14)$$

Where Eq.(5) is used for Δp_L and his two phase multiplier is a function of:

$$\left(\frac{dp}{dz}\right)_{Fr} = f_{Fr}[x + 4(x^{1.8} - x^{10} f_{Fr}^{0.5})] \quad (15)$$

the vapor quality (x) is in the range of: $0 \leq x \leq 1$

If the liquid Froude number Fr_L is greater than or equal to 1, then the friction factor f_{Fr} is set to 1.0; if Fr_L is less than 1, then:

$$f_{Fr} = Fr_L^{0.3} + 0.0055\left(ln\frac{1}{Fr_L}\right)^2 \quad (16)$$

Where:

$$Fr_L = \frac{\dot{m}_{total}^2}{g d_i \rho_L^2} \quad (17)$$

Muller-Steinhagen and Heck correlation

This two-phase frictional pressure gradient according to Muller-Steinhagen and Heck correlation [8] is represented as follows:

$$\left(\frac{dp}{dz}\right)_{frict} = G(1-x)^{1/3} + bx^3 \quad (18)$$

Where the factor G is:

$$G = a + 2(b-a)\,x \quad (19)$$

Where a and b are the frictional pressure gradients for all the flow liquid $(dp/dz)_{Lo}$ and all the flow vapor $(dp/dz)_{Go}$ which are obtained respectively from the following two equations:

$$\left(\frac{dp}{dz}\right)_{Lo} = f_L \frac{2\dot{m}_{total}^2}{d_i \rho_L} \quad (20)$$

$$\left(\frac{dp}{dz}\right)_{Go} = f_G \frac{2\dot{m}_{total}^2}{d_i \rho_G} \quad (21)$$

Experimental Work

The experimental test rig is built according to following terms and conditions:

Experimental conditions

A selected single micro tube heat exchangers were fabricated according to the specifications and the experimental conditions listed in Table 1 and the condensation process of CO_2 gas is performed inside it.

Table1. Experimental conditions

Test section	Micro tube condenser
Process	Condensation inside a chest freezer of -28 °C
Working fluid	CO2
Inner tube diameter (mm)	0.6, 1.0,1.6
Total tube length (m)	29.72
Test section inlet pressure (kPa)	3350, 3600, 4000, 4500
Saturation temperature (oC)	-1.5, 1.23, 5.30, 9.98
Mass flow rate	(2.66 ,5.32, 8,10.6)*10-5 kg/s

Experiment set-up and Test Rig

The experimental set-up consists mainly of the condenser and the evaporator, chest freezer, the pressurized carbon dioxide gas cylinder as a main source of carbon dioxide gas, pressure regulating valve with built-in gas cylinder pressure gauges, sight glasses, pressure transducers, high pressure cutoff and isolating valves and a volume flow meter for measuring the mass flow rate of the gas. Data acquisition system is used to measure the temperatures and pressures at different locations of the condenser .Computer and printer are used to monitor the experimental data as indicated in figure 2.

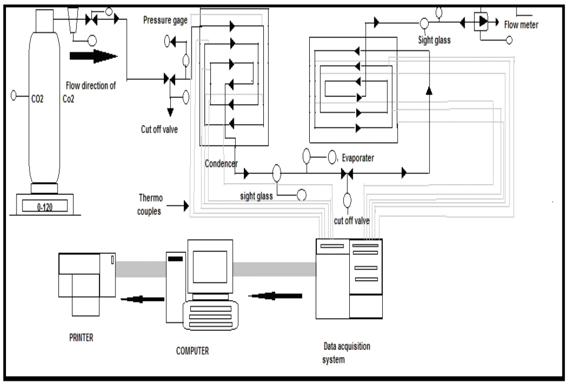

Figure 2. Test Rig

Test sections and measurement methods

A horizontal copper micro tube condenser of 29.72 m length which is settled inside in a chest freezer with inside temperature of -28 °C is used throughout the experimental work. A K-Type thermocouples located at 32 points distributed along the tube were used to measure the outside surface temperatures of the condenser. The two phase pressure drops were measured by using 10 differential calibrated pressure transducers located along the micro tube each had an accuracy of 0.3%. Data Acquisition System with computer display was used to record and monitor the surface temperature and the local pressure readings at different positions along the condenser after a steady state conditions are achieved. The inlet and outlet pressures of the test sections were measured by using differential pressure transducers. A calibrated Coriolis flow meter with accuracy of 0.3% of the reading is used to measure the flow rate of the super heated refrigerant at the exit of the evaporator. Two sight glasses are used to monitor the presence of vapor (χ= 0) and liquid (χ =1) at the inlet and the out let of the condenser. Another third sight glass is located at the exit of the evaporator to monitor the presence of only vapor without any droplets of the refrigerant. The physical properties of the refrigerant were obtained using the REFPROP [11].The local qualities of the vapor were obtained from energy balance inside the micro tube depending on the experimental local pressures which are determined from the readings of the pressure transducers. Licensed LABVIEW software is used to analyze the data during the experimental tests.

CFD Analysis

The micro tube condenser with its specified dimensions and experimental conditions (Table 1) is modeled and steady state simulations are carried out in FLUENT SOFTWARE. The pressure drop is evaluated as a function of the two phase refrigerant quality. The pressure drop CFD analysis is performed under adiabatic conditions for turbulent flow as the Reynolds Number, based on the average properties, exceeds 2300 for all flow rates considered [12].

RESULTS AND DISCUSSIONS

The effect of mass flow rate, pipe diameter and vapor quality on the condensation frictional pressure drop gradients is comparatively studied according to the following subtitles:

Variation of the total and frictional pressure gradients with mass flow rate and the microtube diameter

Figures (3 and 4) depict the variation of the total condenser pressure drop with the mass flow rate for inlet condenser pressure (P_{in} = 33500 kPa) and different internal diameters. It can be noticed from these figures that the pressure drop increases as the mass flow rate increases and it decreases as the internal micro tube diameter increases because of the increase of liquid viscosity and the decrease of vapor density.

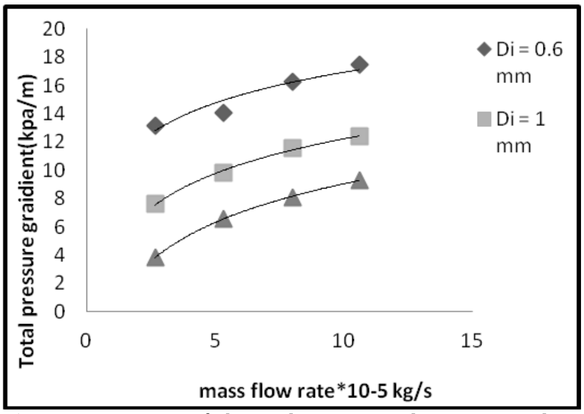

Figure 3. Variation of the total experimental pressure gradient with mass flow rate

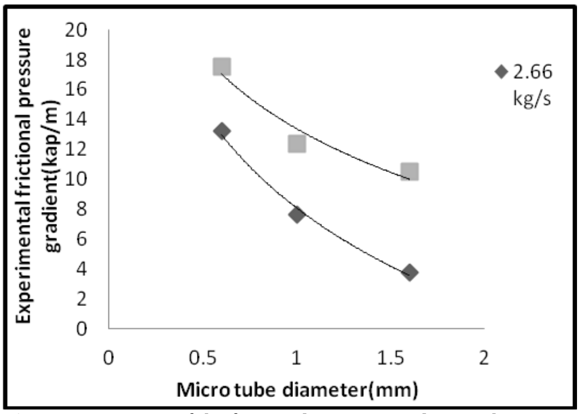

Figure 4. Variation of the frictional pressure gradient with micro tube diameter

Variation of the frictional pressure gradient with vapor quality and micro tube diameter

Equations (1, 2 and 3) are used to find the two phase experimental frictional pressure gradients for all test sections and experimental conditions depending on the experimental local pressure readings and the associated vapor qualities along the test sections. The experimental frictional pressure drops have been compared to all four methods described earlier. The predicted frictional pressure gradient for different experimental situations is depicted graphically as show below.

Figures (5) depicts the CO_2 data in the (0.6 mm) micro tubes at mass flow rate of $2.66*10^{-5}$ kg/s and inlet pressure of 3350 kPa for the four different prediction methods. It can be noticed from these figures that the predicted values of the frictional pressure gradient go through a maximum at a vapor quality of 0.8, which corresponds to the transition from annular flow to annular flow with partial dry out (i.e. annular flow to stratified-wavy flow transition) predicted by the Kattan et al [4] flow pattern map. Figure 8 shows that as the vapor quality increases the predicted frictional pressure gradient increases.

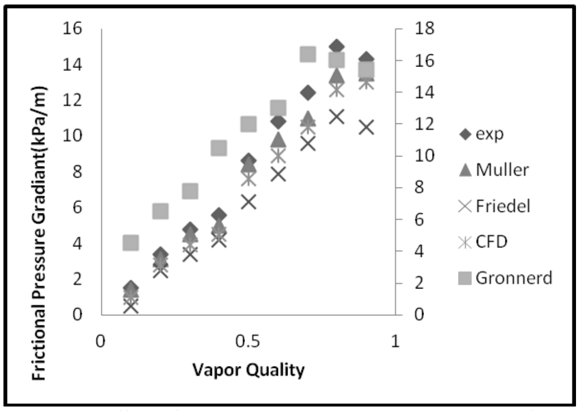

Figure 5. Effect of vapor quality on the pressure gradient for Di =0.6 mm and mass flow rate =$2.66*10^{-5}$ kg/s

Figure (6) depict the CO_2 data in the (1 mm) diameter micro tube and same inlet conditions as in figure (5). The figure compares the experimental frictional pressure gradients with predicted values for the four different prediction methods. Figure (6) clarify that Muller's method

[8] gives the best prediction of the frictional pressure gradient. CFD method comes secondly in fitting the experimental frictional pressure gradient.

Figure (7) shows the variation of the experimental frictional pressure gradient with the vapor quality for the same conditions as in figures (5) and (6). It is clear from this figure that the experimental values of the frictional pressure gradient go through a maximum at a vapor quality of 0.8, the same as that of the predicted values and also it is clear from this figure that the maximum value of the experimental frictional pressure drop is reached at D_i = 0.6 mm and vapor quality of 0.8.

Figure (8) depicts the CO_2 data for D_i = 0.6 mm at mass flow rate of $8*10^{-5}$ kg/s and inlet pressure of 3350 kPa. It can be noticed from this figure that the experimental and the predicted pressure gradient are sharply increased due to the increase in mass flow rate in comparison with its values in figures (5, 6, 7) at which the mass flow rate is $2.66*10^{-5}$ kg/s. Muller's method is still the best fit to the experimental frictional pressure drop values with maximum pressure drop at about 0.8 vapor quality.

Figure (9) depicts the CO_2 data for D_i = 1.6 mm at mass flow rate of $2.66*10^{-5}$ kg/s and inlet pressure of 3350 kPa. This figure clarify that the experimental and the predicted pressure gradients are sharply decreased due to the increase in micro tube diameter in comparison with its values in figure (5) at which the micro tube diameter (D_i) is 0.6 mm. Also Muller's method is still the best fit to the experimental frictional pressure drop values with maximum pressure drop at about 0.8 vapor quality.

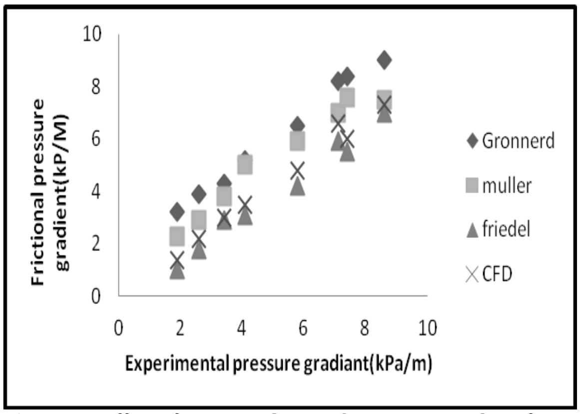

Figure 6. Effect of vapor quality on the pressure gradient for Di =1 mm and mass flow rate =$2.66*10^{-5}$ kg/s and P_i = 3350 kPa

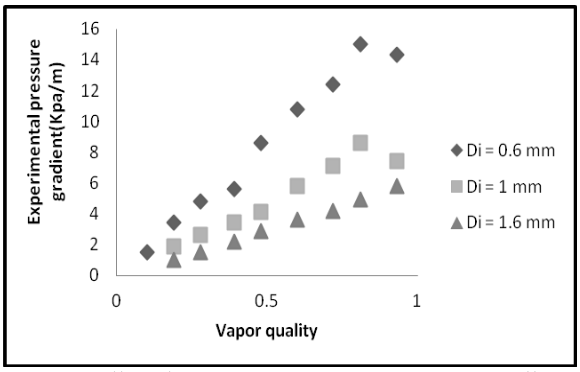

Figure 7. Effect of vapor quality on the pressure gradient different D_i, mass flow rate =$2.66*10^{-5}$ kg/s and Pi = 3350 kPa

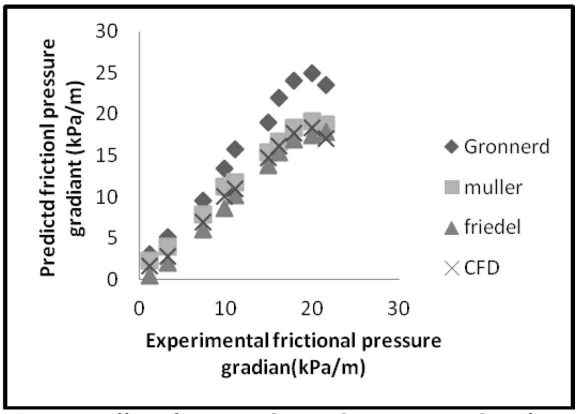

Figure 8. Effect of vapor quality on the pressure gradient for Di =0.6 mm and mass flow rate =$8*10^{-5}$ kg/s and P_i = 3350 kPa

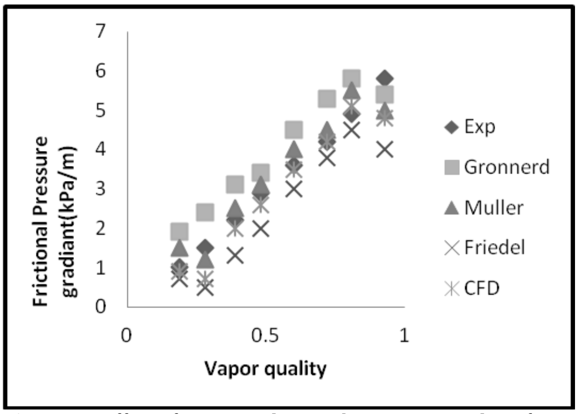

Figure 9. Effect of vapor quality on the pressure gradient for Di =1.6 mm and mass flow rate =$2.66*10^{-5}$ kg/s and P_i = 3350 kPa

In figures (10, 11, 12), the predicted frictional pressure gradients are plotted against the experimental frictional pressure gradients for different mass flow rates, different tube diameters and for inlet pressure of 3350 kPa. It is clear from these three figures that the four prediction methods fit the experimental data but with different average standard deviation ranged from 6.4% to Muller method till 8.9% to CFD method, 12.1% to Gronner method and then17.3% to Friedel method.

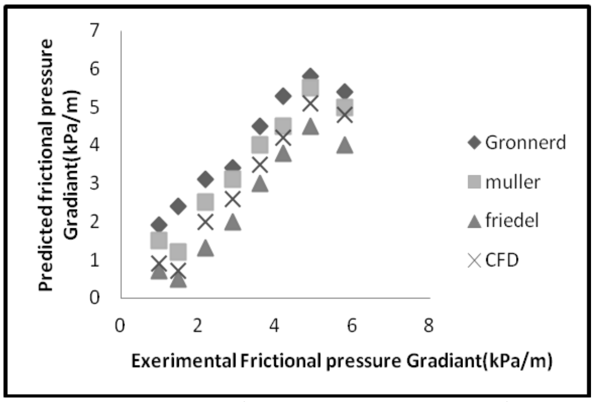

Figure10. Experimental frictional versus predicted frictional pressure gradients for different methods, mass flow rate = $2.66*10^{-5}$ kg/s, Di = 1.6 mm and P_i = 3350 kPa

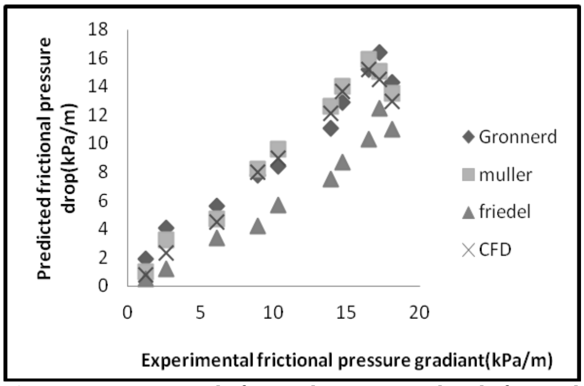

Figure11. Experimental frictional versus predicted frictional pressure gradients for different methods, mass flow rate = $5.32*10^{-5}$ kg/s, Di = 0.6 mm and P_i = 3350 kPa

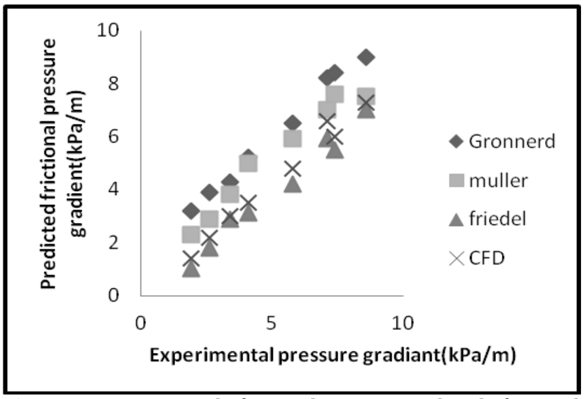

Figure12. Experimental frictional versus predicted frictional pressure gradients for different methods, mass flow rate = $2.66*10^{-5}$ kg/s .Di = 1 mm and P_i = 3350 kPa

CONCLUSIONS

1. Four different prediction methods based on the vapor quality of the refrigerant were used to predict the frictional pressure drop during the condensation of carbon dioxide in micro tubes. The Muller method gave the best fit, CFD gave the second best fit while Gronnerd and Friedel methods gave the third and the fourth best fit with average standard deviations of 6.4%, 8.9%, 12.1% and 17.3% respectively.

2. The peak two phase frictional pressure gradient of CO_2 was observed at high vapor qualities.

3. The two phase frictional pressure gradient of CO_2 increased as the micro tube diameter decreased and it increased as the mass flow rate increased.

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