Journal of Mechanical Engineering Research Vol. 2(4), pp. 71-84, September 2010
Available online at http://www.academicjournals.org/jmer
ISSN 2141 – 2383 © 2010 Academic Journals

Full Length Research Paper

A new experimental technique to determine heat transfer coefficient and pressure drop in smooth and micro-fin tube

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Accepted 13 April, 2010

An experimental test facility is designed and built to calculate condensation heat transfer coefficients and pressure drops for HFC-134a in a 10.21 mm ID smooth and 8.56 mm ID micro-fin tube. The main objective of the experimentation is to investigate the enhancement in condensation heat transfer coefficient and increase in pressure drop using micro-fin tube for different condensing temperatures and further develop an empirical correlation for heat transfer coefficient and pressure drop, which takes into account, variation of condensing temperature and mass flux of refrigerant. The experimental setup has a facility to vary the different operating parameters such as condensing temperature, cooling water temperature, flow rate of refrigerant and cooling water etc. and study their effect on heat transfer coefficients and pressure drops. The hermetically sealed reciprocating compressor is used in the system, thus the effect of lubricating oil on the heat transfer coefficient is taken in to account. This paper reports the detailed description of design and development of the test apparatus, control devices, instrumentation, experimental procedure and data reduction technique. It also covers the comparative study of experimental apparatus with the existing one from the available literature survey. The condensation and pressure drop of HFC-134a in a smooth tube are measured and the values of condensation heat transfer coefficients for different mass flux and condensing temperatures were obtained using modified Wilson plot technique with correlation coefficient above 0.9. The condensation heat transfer coefficient and pressure drop increases with increasing mass flux and decreases with increasing condensing temperature. The results are compared with existing available correlations for validation of test facility. The experimental data points have good association with few available correlations. The condensation and pressure drop of HFC-134a in a micro-fin tube are also measured and the values of condensation heat transfer coefficients obtained. The enhancement and penalty factors of HFC-134a are 1.24 - 2.42 and 1 - 1.77 respectively.

Key words: Experimental technique, micro-fin tube, condensation heat transfer, pressure drop, heat transfer enhancement.

INTRODUCTION

In-tube condensation is quite common in refrigeration and air-conditioning applications. It is the binding choice for air-cooled and evaporative condensers. In-tube condensation is often thought of as a process of film-wise condensation (less effective than drop-wise condensation) (Kern, 2003) of vapor inside a tube, hence air-cooled condensers are less effective. Another draw back of air-cooled condenser is that it operates at a greater condensing temperature than water-cooled condenser; hence the compressor (and the refrigeration system) delivers 15 to 20% lower capacity (Arora, 2004). Therefore one has to use a larger compressor to meet the requirement. At the same time, the compressor consumes

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greater power. Hence the air-cooled system has a lower ratio of overall energy efficiency. The augmentation of In-tube evaporation and condensation heat transfer can result in smaller and more efficient evaporators and condensers. Micro-fin tubes (Figure 3) have been successfully implemented in the air-conditioning and refrigeration industries for effectively improving tube-side performance. This success is because of their ability to significantly improve heat transfer coefficient with only moderate increase in pressure drop; hence this augmentation technique shows great potential as an energy saving technique. An experimental program designed to investigate potential augmentation technique has been carried out worldwide as part of a large study of In-tube condensation.

The range of operating parameters used in experimental test facilities developed by different researchers is given in Table 1. It is found that in many test setups, refrigerant pump is used as a circulating device instead of compressor and used for small range of operating conditions. No study found higher condensing temperatures such as 55 - 60°C. Also in very few studies new refrigerants are used. The present test facility overcomes these deficits of the literature survey and achieved the following range:

Mass flux (Gr) = 50 - 800 kg/s.m^2
Condensing pressure (Pd) = 7.5 - 16.5 bar (gauge)
Condensing temperature (Th) = 35 - 60°C
Cooling water temperature (Tci) = 2 - 40°C

As for cooling water supply for test, condenser evaporator tank is utilized, no separate chilled water plant is required and heating is achieved with the help of 8 kW capacity heaters which are immersed in the evaporator tank. The test is carried out with HFC-134a refrigerant.

DESCRIPTION OF THE TEST APPARATUS

The test apparatus, as shown schematically in Figure 1 consist of four circuits namely, refrigerant main, auxiliary, cooling water and chilled water circuit. Details of these circuits are given below.

The refrigerant main circuit links compressor to main condenser to expansion valve to evaporator and back to compressor. Compressor used is of hermetically sealed reciprocating type with a cooling capacity of 7.6 kW and suitable for HFC-134a, R-404A, R-407C, R-507A refrigerants. Main condenser is shell and tube type with refrigerant through shell and cooling water through tubes. Thermostatic expansion valve is used as an expansion device. The evaporator is of tank and coil type; with refrigerant flowing through coil and surrounded by water in the tank, heaters are immersed in the tank to provide heat source for evaporator as well as maintain desired water temperature in the tank.

The refrigerant auxiliary circuit links compressor to test condenser to expansion valve to evaporator and back to compressor. All the devices in this circuit are common with main circuit except test condenser. The test condenser is a shell and U bend tube exchanger with the refrigerant flowing inside the inner tube (di = 10.21 mm) and chilled water flowing through the shell of diameter 50.8 mm. Table 2 provides the dimensions of smooth and micro-fin tube. In order to induce turbulence and direct the water flow outside the tubes, baffles are employed. The center to center distance between baffles is called baffle spacing (B). The baffle spacing is not usually greater than shell ID and not less than one-fifth the shell ID. For desired effect it is generally taken as 0.2 Ds or 2 inches whichever is greater. Considering that (B = 2 inches = 50.8 mm) (Kern, 2003), baffles will be of segmental type, also known as 25% cut baffles. The test condenser is designed for maximum loading capacity. The maximum loading condition occurs for 35°C condensing temperature with mass flux of 800 kg/m^2.s.

The chilled water is used in test rig which flows in close cycle between evaporator and test condenser. The circuit mainly joins components such as, pump, Rota meter, test condenser evaporator and back to pump. This circuit allows increasing or decreasing the chilled water flow rate with the help of valve according to cooling required in test condenser. The heat absorbed in test condenser is rejected at evaporator. To match the cooling capacity of refrigeration unit extra arrangement of heaters are used. The pump is selected on the basis of maximum flow rate and maximum pressure drop. The pump selected to meet the requirements is 3000 Lph and 28 m head.

The cooling water circuit as shown in Figure 2 is used to cool water circulating from the main condenser; the heat absorbed in the main condenser by cooling water is ejected in the force drought cooling tower and circulated back from the main condenser with the help of pump of capacity 1500 Lph and 2 m head. Plate type valves are used in lines to regulate the flow of refrigerant and water.

Instrumentation

The measurements taken in the system are pressure, temperature and flow at various locations in the apparatus. These measurement points are as follows.

Temperature measurements

1. Before and after the test condenser (refrigerant circuit), in order to measure the degree of superheating and sub cooling during condensation process.
2. Before and after the test condenser (chilled water circuit), to measure chilled water temperatures used for the calculation of heat absorbed by water.
3. To measure the temperature of chilled water in the evaporator thus monitoring the steady state.
4. Before and after evaporator, to measure the refrigerant temperatures, to ensure state of refrigerant.

Pressure measurements

1. At the inlet and outlet of the test condenser, to measure the refrigerant pressures required to calculate the pressure drop across the test condenser, consequently used to calculate the friction factor.
2. At the inlet of compressor, to measure the suction pressure required during analyzing system performance.
3. Mounted on main condenser, to measure condenser pressure, monitor the condensing temperature and to ensure the system balancing when the refrigerant flow rate is changed.

Flow measurements

1. In the auxiliary refrigerant circuit, to measure the refrigerant flow rate in the test condenser, required to calculate Reynolds number and heat rejected by refrigerant.

Table 1. Range of operating parameters used in various test facilities.

S. No	Authors (Year)	Range of experimental parameters covered	Working fluids	Circulating device
1	Yirong Jiang, Srinivas Garimella (2003)	Gr: 200 - 500 kg/m^2.s Pd: 7.5 - 10.5 bar Tci: not given	R-404A, water coolant, steam	Refrigerant pump
2	L.M.Schlager, M. B. Pate, Bergles (1990)	Gr: 75 - 400 kg/m^2.s Pd: 15 - 16 bar Tci: not given	R-22, water-glycol, water	Refrigerant pump
3	J. C. Khanpara, Bergles (1986)	Gr: 197 - 594 kg/m^2.s Pd: fixed pressure 2.41 bar Tci: not given	Refrigerant, water, coolant	Refrigerant pump
5	Wang Fazio (1985)	Gr: 17.14 - 85.55 kg/m^2.s Pd: -6.8 - 11.4 bar Tci: city water at constant temperature	R-12,R-22,cold water, hot water	Open type reciprocating compressor
6	Said and Azer (1982)	Gr: 14.14 - 305.89 kg/m^2.s Pd: 1.32 - 3.05 bar Tci: 11.7 - 35.9℃	R-113, water	Refrigerant pump
7	Stoecker and Kornota (1985)	Gr: fixed flow rate of 0.023 kg/s was maintained. Pd: 4.78 - 6.09 bar Tci: city water at constant temperature	R-114,R-12, cooling water	Refrigerant pump
8	Tichy, Macken and Duval (1985)	Gr: 94.44 - 944.44 kg/m^2.s Pd: 4.8 - 9.3 bar Tci: city water at constant temperature.	R-12, cooling water	Open type reciprocating compressor
9	Keumnam and Sang-Jin Tae (2000)	Gr: 100 - 400 kg/m^2.s Pd: fixed pressure Tci: 11.7 - 35.9℃	R-407C, R-12,	Refrigerant pump
10	Steve J. Eckels and Brian A. Tesene (1999)	Gr: 125 - 600 kg/m^2.s Pd: 8.8 - 11.6 bar Tci: contant temperature water	R-22, R-134a, R-410a	Refrigerant pump
11	Minh Luu And Bergles (1980)	Gr: 86 - 760 kg/m^2.s Pd: 2.41 - 6.55 bar Tci: 10 - 104℃	R-113, water, steam	Refrigerant pump
12	Smit and Meyer (2002)	Gr: 100 - 600 kg/m^2.s Pd: fixed pressure of 24.3 bar Tci: 10 - 85℃	R-22, water-glycol, water	Open type reciprocating compressor
13	Tandon,varma and Gupta (1985)	Gr: 175 - 560 kg/m^2.s Pd: 1.4 - 8 bar Tci: fixed temperature water	R-22, water-glycol, water	Open type compressor
14	Steve J. Eckels Doerr and Pate Brian A. Tesene (1994)	Gr: 86 - 375 kg/m^2.s Pd: fixed 8.3 bar Tci: not given	R-134a	Refrigerant pump

Table 1. Contd.

15	Eckels and Pate (1991)	Gr:130 - 400 kg/m^2.s Pd: 6.2 - 11.5 bar Tci: not given	HFC-134a, CFC-12, water-glycol mixture	Refrigerant pump
16	Agrawal,Kumar and Varma (2004)	Gr: 210 - 372 kg/m^2.s Pd: 14.4 - 21.9 bar Tci: 20 - 30°C	R-22, water	Open type compressor
17	Chato and Dobson (1998)	Gr: 25 - 800 kg/m^2.s Pd: 7.5 - 10.5 bar Tci: constant temperature water	R-134a, R-22, R-32/R-125	-

Gr: mass flux of refrigerant; Pd: condensing pressure; Tci: temperature of cooling water used in condenser.

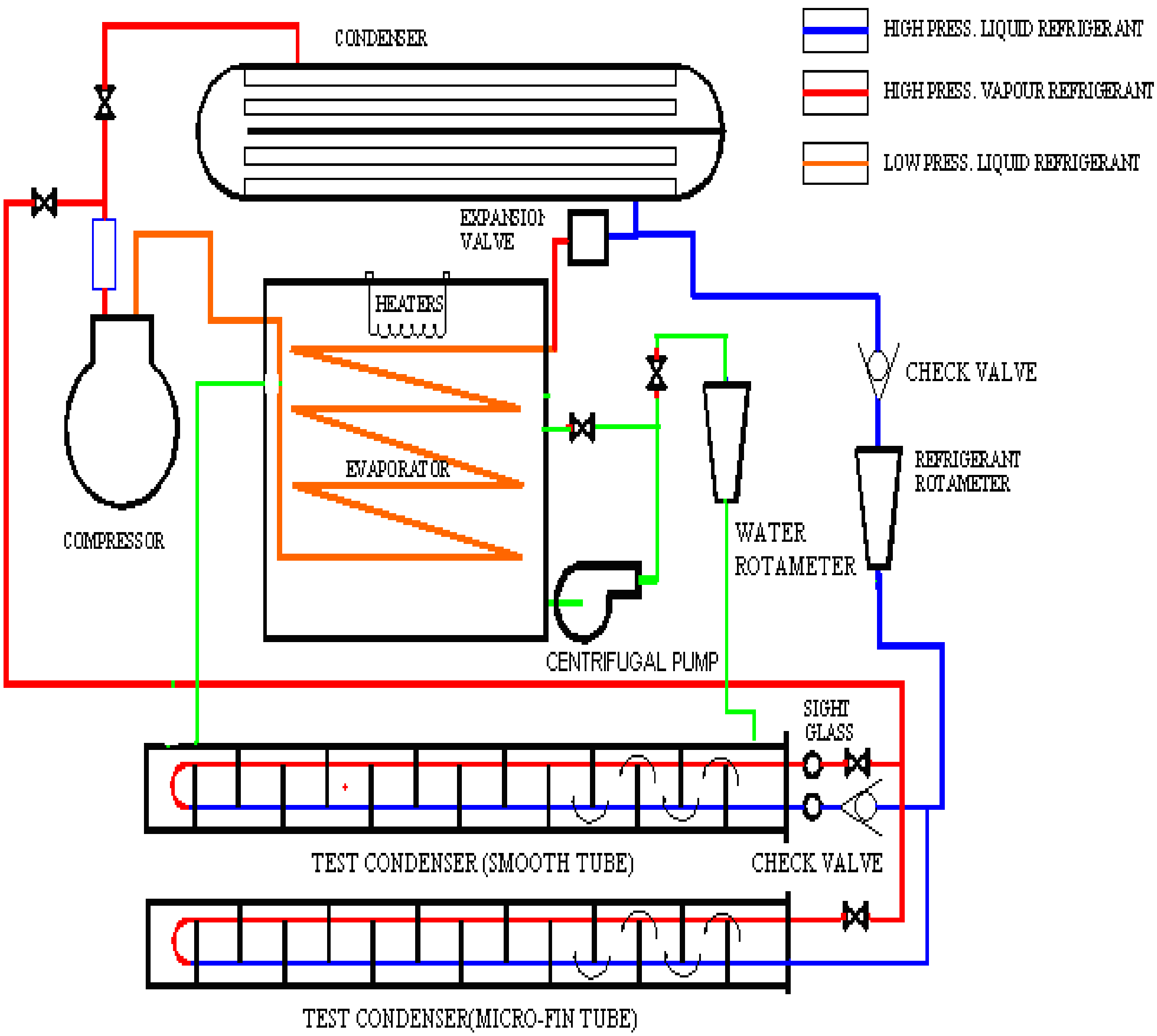

Figure 1. Experimental test facility.

Table 2. Smooth and micro-fin tube dimensions.

Parameter	Smooth tube	Micro-fin tube
Outside diameter, do (mm)	9.42	9.52
Bottom thickness, t (mm)	0.64	0.28
Number of fins, N	--------	60
Spiral angle, β , degree	--------	18
Apex angle, γ, degree	--------	45
Fin height, ef (mm)	--------	0.2
Fin tip diameter, dt (mm)	--------	8.56
Max. inside diameter, di (mm)	8.14	8.96
Length of tube, L (m)	4.5	4.5
Cross sectional area, Ac (mm^2)	52.04	63.053

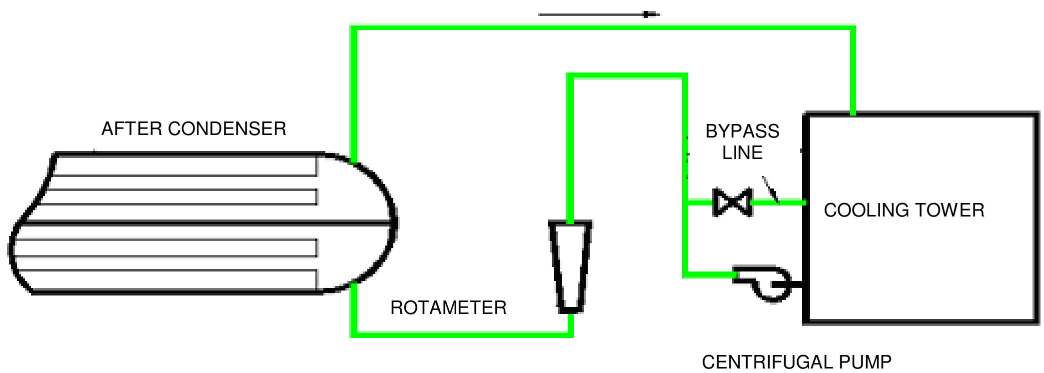

Figure 2. Cooling water circuit for main refrigerant circuit.

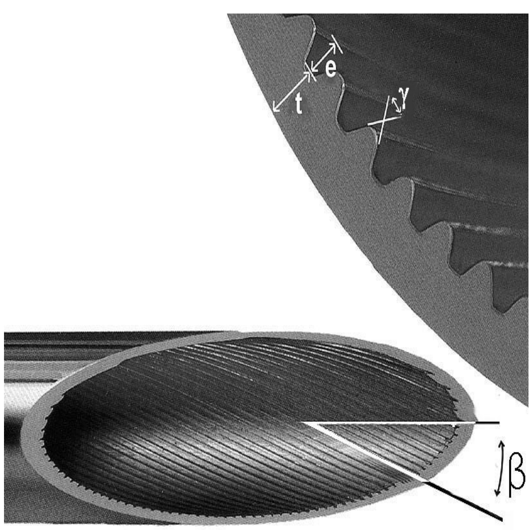

Figure 3. Micro-fin tube.

2. In the chilled water circuit, to measure the water flow rate in the test condenser, required to calculate the heat absorbed by chilled water in the test condenser.
3. In the cooling water circuit to measure the water flow rate, used during analyzing system performance.

PT100 (Resistance Temperature Detector made of platinum with a base of 100 Ω at 0℃) with 1% accuracy is used for temperature measurements. Pressure transmitters with 0.25% accuracy and 13% uncertainties are used to measure pressure difference across the test condenser, while Bourdon pressure gauges are used in other locations. Rota meters with 1% accuracy are used to measure all flow rates. All measuring instruments are calibrated from recognized calibration centers.

EXPERIMENTAL PROCEDURE

The experimentation is carried out for different mass flow rate and different condensing temperature of refrigerant. One particular condensation process (for a particular mass flow rate and condensing temperature) is also achieved for different flow rate and temperature of chilled water.

The following are steps for carrying out experimentation for 100 Lph (refrigerant) flow and 40℃ condensing temperature:

1. Start refrigerant main and cooling water circuit, auxiliary circuit remains closed.
2. Reduce the temperature of water in the evaporator to 5℃.
3. Adjust the cooling water flow to achieve 40℃ condensing temperature in main circuit.
4. Start the chilled water pump and allow the water to flow through test condenser, set the flow rate of chilled water at 1000 Lph.

5. Gradually open the valve of auxiliary circuit until the mass flow rate of refrigerant reaches 100 Lph.
6. Adjust the flow rate of chilled water (say to 700 Lph) to adjust condensing temperature 40°C and achieve the condensation with 10°C sub cooling.
7. Allow the system to stabilize, and record all readings such as test condenser inlet, outlet temperatures of chilled water and refrigerant etc. after steady state.
8. Increase the temperature of water in the evaporator by 5°C with the help of heater.
9. Repeat steps 6 to 8 for different chilled water inlet temperatures say 10, 15, 20, 25 and 30°C respectively.
10. Repeat steps 1 to 9 for mass flow rate of 20, 40, 60, 80,120, 140 and 160 Lph.

Data reduction

The data analysis procedure determines the average convective heat transfer coefficient of pure refrigerant, which also takes into account oil present in the refrigerant. In addition, the data analysis determines the correlation constants required for average convective heat transfer coefficient of water and refrigerant side using modified Wilson plot technique. The following is a brief description of the data reduction equations.

The equations to find rate of heat rejected by refrigerant and rate of heat absorbed by cooling water are as follows. The variation between the heat rejected by refrigerant and heat absorbed by water is within 5%.

$$Q_r = Q_{sv} + Q_c + Q_{sl} \tag{1}$$

$$Q_{sv} = m_r c_{p_v} (T_{r_i} - T_{h_i}) \tag{2}$$

$$Q_{sl} = m_r c_{p_l} (T_{h_o} - T_{r_o}) \tag{3}$$

$$Q_c = m_r (h_{t_i} - h_{t_o}) \tag{4}$$

$$Q_w = m_w c_{p_w} (T_{w_o} - T_{w_i}) \tag{5}$$

The average LMTD value is obtained by using following equations indicated in (Kern, 2003)

$$T_{w_d} = T_{w_i} + \frac{Q_{sv}}{m_w c_{p_w}} \tag{6}$$

$$T_{w_c} = T_{w_d} + \frac{Q_c}{m_w c_{p_w}} \tag{7}$$

$$LMTD_d = \frac{(T_{ri} - T_{wi}) - (T_{hi} - T_{wd})}{\ln \frac{(T_{ri} - T_{wi})}{(T_{hi} - T_{wd})}} \tag{8}$$

$$LMTDc = \frac{(T_{hi} - T_{wc}) - (T_{ho} - T_{wd})}{\ln \frac{(T_{hi} - T_{wc})}{(T_{ho} - T_{wd})}} \tag{9}$$

$$LMTDs = \frac{(T_{ho} - T_{wo}) - (T_{ro} - T_{wc})}{\ln \frac{(T_{ho} - T_{wo})}{(T_{ho} - T_{wc})}} \tag{10}$$

$$LMTD = \frac{Q_r}{\sum \frac{Q}{LMTD}} = \frac{Q_r}{\sum \left(\frac{Q_d}{LMTD_d} + \frac{Q_c}{LMTD_c} + \frac{Q_s}{LMTD_s} \right)} \tag{11}$$

The overall HTC is determined by using:

$$Uo = \frac{Q_r}{A_o LMTD} \tag{12}$$

The overall thermal resistance of the condensation process in shell and tube condensers (Rov) can be expressed as the sum of the thermal resistances corresponding to external convection (Ro), internal convection (Ri) and the tube wall (Rt) as shown in Eq. (13)

$$Rov = Ri + Ro + Rt \tag{13}$$

The individual resistances can be obtained by using following expressions:

$$R_{ov} = \frac{1}{U_o A_o} \tag{14}$$

$$R_i = \frac{1}{h_i A_i} \tag{15}$$

$$R_o = \frac{1}{h_o A_o} \tag{16}$$

$$R_t = \frac{\ln(\frac{d_o}{d_i})}{2\pi L k_t} \tag{17}$$

For a specific condition of the condensation process (particular condensing pressure and refrigerant flow rate), with different flow rate of cooling water, the overall thermal resistance is varied mainly due to the variation in outside heat transfer coefficient; meanwhile the remaining thermal resistances stay nearly constant. Therefore the thermal resistances due to internal convection and tube wall can be considered constant as indicated in Eq. (18).

$$C1 = Ri + Rt \tag{18}$$

The average heat transfer coefficient for flow across cylinders can be expressed as:

$$ho = CRewmPrw0.33 \left(\frac{k_w}{d_o} \right) \tag{19}$$

Where,

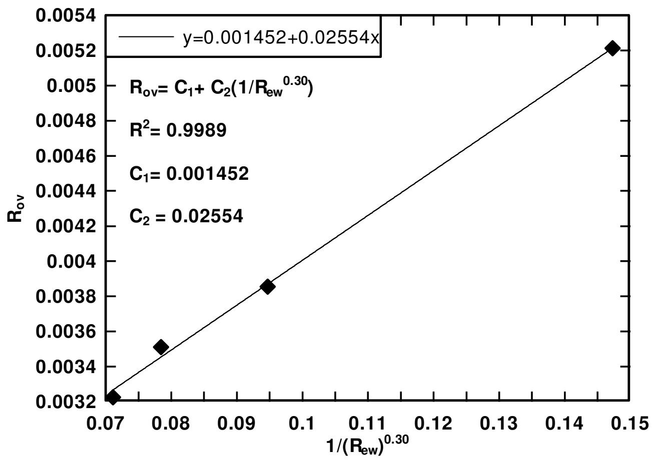

Figure 4. Modified Wilson plot 1.

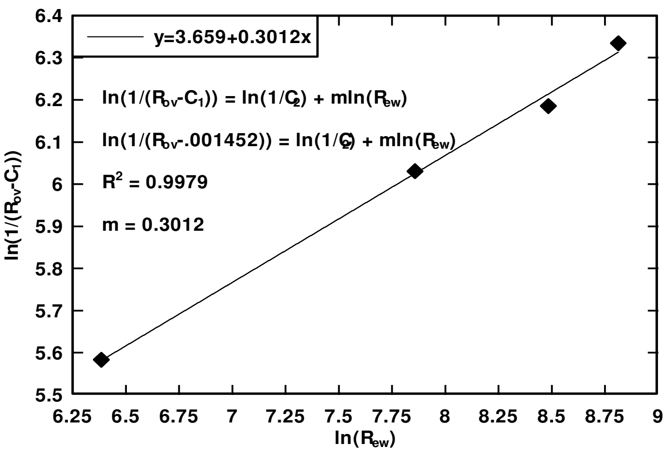

Figure 5. Modified Wilson plot 2.

$$\mathrm{Re}_w = \frac{GD_e}{\mu_w} \tag{20}$$

$$\mathrm{Pr}_w = \frac{\mu_w C_{pw}}{k_w} \tag{21}$$

Putting Eq. (19) in Eq. (16), we have

$$\mathrm{Ro} = \mathrm{C2}\ \frac{1}{\mathrm{Re}_w^{\ m}} \tag{22}$$

Where,

$$\mathrm{C2} = \frac{1}{C}\left(\frac{1}{\mathrm{Pr}_w^{0.33}}\right)\left(\frac{do}{kw}\right)\left(\frac{1}{A_o}\right) \tag{23}$$

Putting Eq.(18) and (22) in Eq.(13), we have

$$\mathrm{Rov} = \mathrm{C1} + \mathrm{C2}\ \frac{1}{\mathrm{Re}_w^m} \tag{24}$$

$$\ln\left(\frac{1}{R_{ov}-C_1}\right) = \ln\left(\frac{1}{C_2}\right) + m\ln(\mathrm{Re}) \tag{25}$$

$$\Delta P_{frict} = \Delta P_{total} + \Delta P_{mom} - \Delta P_l - \Delta P_g \tag{26}$$

The values of constants C1 and C2 are obtained according to Eq. (24) using least square technique initially by assuming the value of m and plotting graph as shown in Figure 4. Put the value of C1 in Eq. (25) and determine the value of 'm' again by using the same technique (from plot as shown in Figure 5.) If the value of 'm' obtained is equal to the value initially assumed, then the process is finished and the value of exponent is determined. Otherwise, the iteration process is repeated by assuming new 'm' value. Moreover, the coefficient C and the exponent 'm' of the general dimensionless correlation as indicated in Eq. (19) are also obtained, thus the general correlation is determined assuming only the value of the exponent of the Prantdl number. This technique is known as modified Wilson plot technique (Jose et al., 2005). Obtain the values of ho, Ro and Rt using Eq. (19), (16) and (17) respectively. Putting these values in eq. (13) to determine Ri; consequently determine hi using Eq. (15).

RESULTS AND DISCUSSION

The heat transfer coefficients and pressure drops of HFC-134a are measured in smooth and micro-fin tubes at different condensing temperatures of 35, 40, 45, 50, 55 and 60°C. About 280 data points each are taken during experimentation on smooth and micro-fin tubes. Condensation of refrigerant at specific conditions (mass flow of refrigerant and condensing temperature) is achieved for different flow rates and temperatures of cooling water for obtaining constants of co-relations using modified Wilson plot technique as shown in Figures 4 and 5.

Modified Wilson plot method

The modified Wilson plot method is applied to experimental data according to iteration procedure indicated in experimental procedure. The constants C1 and C2 are obtained as indicated in Figure 4. The Wilson plot is implemented for estimating heat transfer coefficient for every mass flow rate. The experimental data with particular refrigerant flow rate and condensing temperature are considered for each plot.

Figure 5 shows the values of the term ln [1/(Rov-C1)] plotted as a function of ln (Re), taking into account the values of the overall thermal resistance and the constant C1 obtained from least square technique as indicated in Figure 4. If the obtained value of 'm' from regression technique as indicated in Figure 5 is equal to assumed

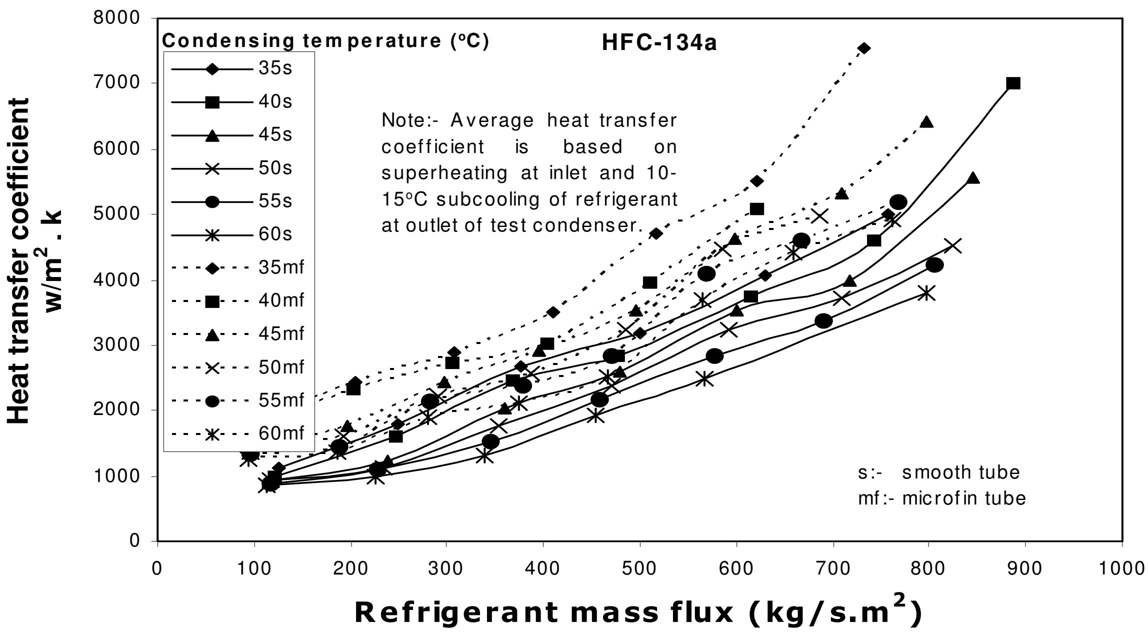

Figure 6. HFC-134a Condensation heat transfer coefficient in a smooth and micro-fin tube.

value of m from Figure 4, the iteration procedure is completed, otherwise repeat the procedure as indicated in Figure 4. This technique is applied for each condensing temperature and for all mass flow rate of refrigerant. Total 42 Wilson plots each are developed with correlation coefficient of above 0.9.

Condensation heat transfer

Condensation heat transfer data for smooth tube and micro-fin tube with HFC-134a are shown in Figure 6. For both tubes, the heat transfer coefficient increases with mass flux but decreases with increasing condensing temperature. The value of heat transfer coefficients is obtained using Eq. (18) and Eq. (15). The heat transfer coefficients obtained for micro-fin tube are greater than that of smooth tube for all condensing temperatures and mass fluxes.

Pressure drop

Frictional pressure drop data obtained using equation (26) during condensation of HFC-134a for smooth tube and micro-fin tube are as shown in Figure 7.

As with heat transfer coefficients, the pressure drop varies considerably with mass flux and condensing temperature.

Enhancement and penalty factors

Another approach for comparing the micro-fin tube heat transfer performance with that of the smooth tube is to form heat transfer enhancement factors, EF, defined as the ratio of micro-fin tube heat transfer coefficient to that of comparable smooth tube at a similar mass flux, heat flux, pressure level, and inlet and oulet quality. Pressure drop performance comparisons between the micro-fin tube and smooth tube can be made by forming ratios of pressures drop in a manner similar to that used to form heat transfer enhancement factors. These ratios are hereafter referred to as pressure drop penalty factors (PF). Figure 8 shows both heat transfer enhancement factors, EF, and pressure drop penalty factors, PF, for the micro-fin tube with HFC-134a. The EFs vary from maximum of 2.42 at low mass flux to a minimum of 1.24 for highest mass flux. The PFs are also shown in Figure 8 and vary from minimum 1 at low mass flux to maximum 1.77 at high mass flux. The penalty factors appear to be nearly constant above 400 kg/s.m^2 mass flux.

Experimental uncertainty

The maximum uncertainties are ±13.2% for the LMTD, ±1.8% for the mass flow rate of water, ±2.81% for the mass flow rate of refrigerant, ±4.72% for the heat dissipation by refrigerant in the test section, ±9.22% for the heat absorbed by the water in the test section, ±13.3 for overall heat transfer coefficient, ±18.2% for refrigerant side heat transfer coefficient and ±13.3% for the pressure drop. A propagation of error analysis (Kline and McClintock, 1953) is used to obtain the uncertainty listed above with a confidence interval of 85 - 90% with a coefficient of correlation above 0.9.

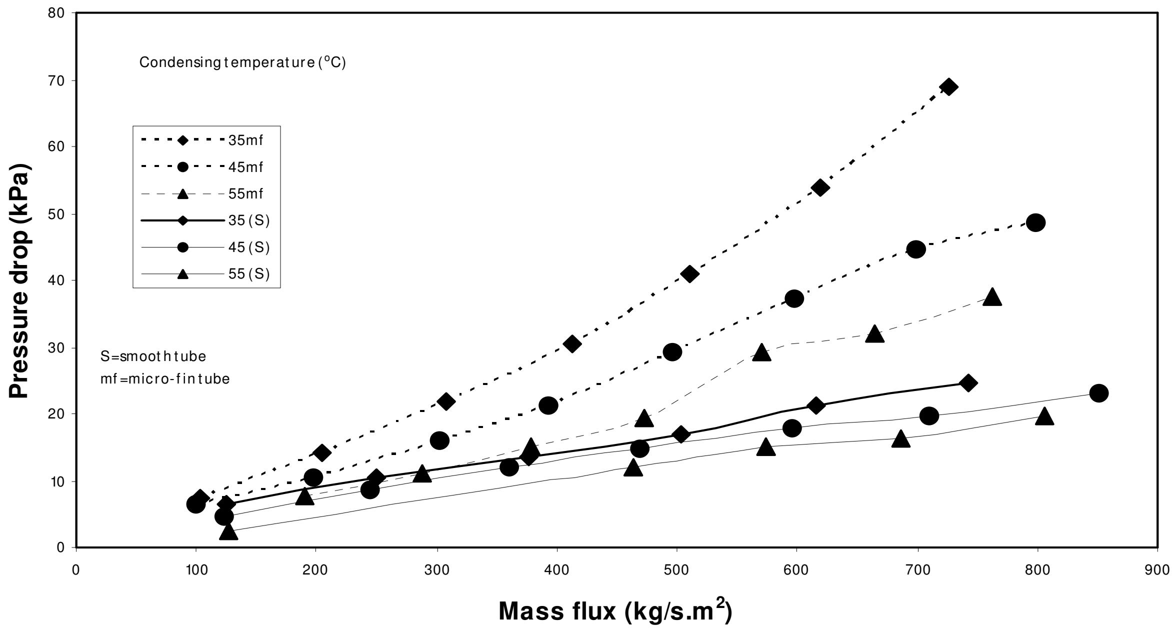

Figure 7. HFC-134a Condensation pressure drop in a smooth and micro-fin tube.

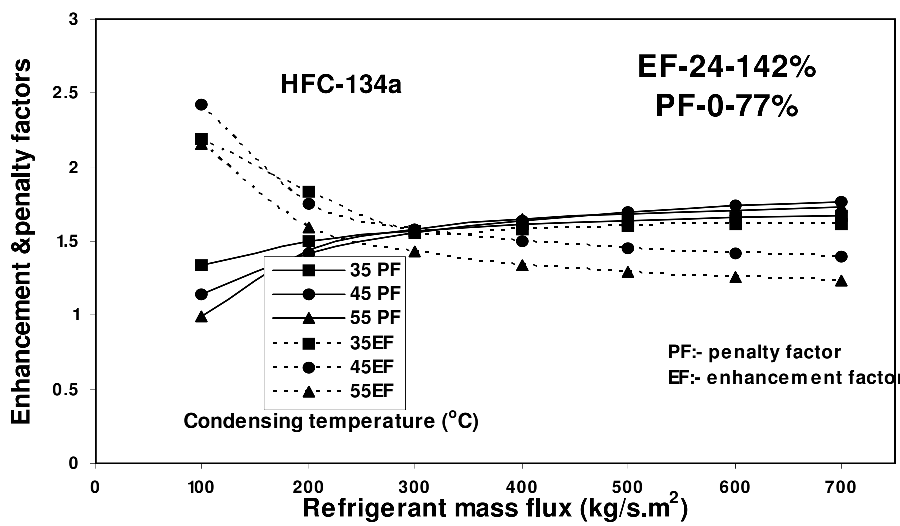

Figure 8. HFC-134a heat transfer enhancement and pressure drop penalty factor.

Correlation comparison

The experimental heat transfer and pressure drop data of smooth and micro-fin tubes are also compared with some available correlations and only the best two correlations for each case is discussed as follows:

Heat transfer

Boyko and Kruzhilin (1967) correlation captures 83.91% HFC-134a data within ±20%. Akers et al. (1959) correlation captures 78.32% HFC-134a data for smooth tube as shown in Figure 9.

For micro-fin tube, Luu and Bergles (1980) correlation captures maximum data points amongst all, capturing 74.64% of HFC-134a data within ±20. Most of the data points corresponding to low mass flux are under predicted, however almost all values corresponding to 60℃ condensing temperatures are over predicted by this correlation. Hiroshi Honda, Huasheng Wang and Shigeru Nozu's correlation captures 47.84% of HFC-134a data

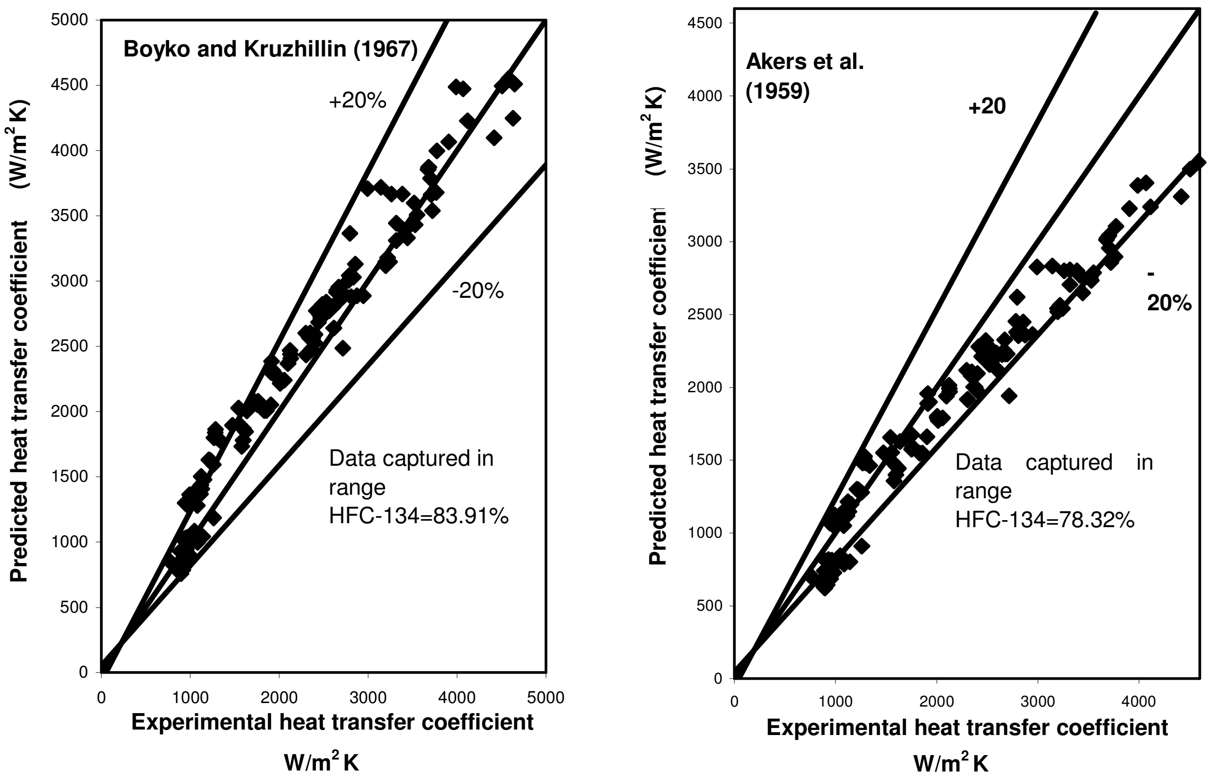

Figure 9. Comparison of smooth tube heat transfer data with existing correlations.

within ±20 (Hiroshi et al., 2002). Most experimental data between 50 and 60°C condensing temperatures are over predicted and low mass flux data between 35 and 55°C is under predicted by this correlation as shown in Figure 10.

Pressure drop

In case of smooth tube, (Friedel, 1979) correlation captures maximum data points amongst all, capturing 75% data of HFC-134a data within ±30%. The experimental data of mass fluxes below 200 kg/s.m^2 are under predicted by this correlation. M¨uller-Steinhagen and Heck (1986) correlation captures 57.57% of HFC-134a data within ±30%. Most of the experimental data from low mass flux area and high condensing temperature are under predicted by this correlation as shown in Figure 11. Choi et al. (2001) correlation captures maximum data points of micro-fin tube amongst all, capturing 69.88% data of HFC-134a within ±30%. The experimental data of mass fluxes below 200 kg/s.m^2 and some of data corresponding to 35 and 40°C condensing temperatures are under predicted by this correlation. Kedzierski and Goncalves (1999) correlation captures 64.2% of HFC-134a data within ±30%. Most of the experimental data from low mass flux area are under predicted, and few data points corresponding to high mass flux are over predicted by this correlation as shown in Figure 12.

Conclusion

The experimental test facility has been designed and developed, which is used to determine the condensation heat transfer coefficient and pressure drop in smooth and micro-fin tubes for various HFC refrigerants namely HFC-134a, R-404A, R-407C, R-507A. As the hermetically sealed compressor used for circulating refrigerant, effect of oil present in the refrigerant during condensation is also taken into account. The experimentation covers wide range of operating parameters such as mass flux and condensing temperatures. The instruments used for measurements are calibrated from recognized calibration centers.

The condensation and pressure drop of HFC-134a in smooth and micro-fin tubes are measured and the values of condensation heat transfer coefficients for different mass flux and condensing temperatures are obtained using modified Wilson plot technique with correlation coefficient above 0.9. The condensation heat transfer coefficient and pressure drop increases with increasing mass flux and decreases with increasing condensing temperature for both smooth and micro-fin tubes. The heat transfer coefficients and pressure drops obtained for micro-fin tube are greater than that of smooth tube for all condensing temperatures and mass fluxes. The EFs obtained varies from 1.24 to 2.42, while PFs varies from 1 to 1.77.

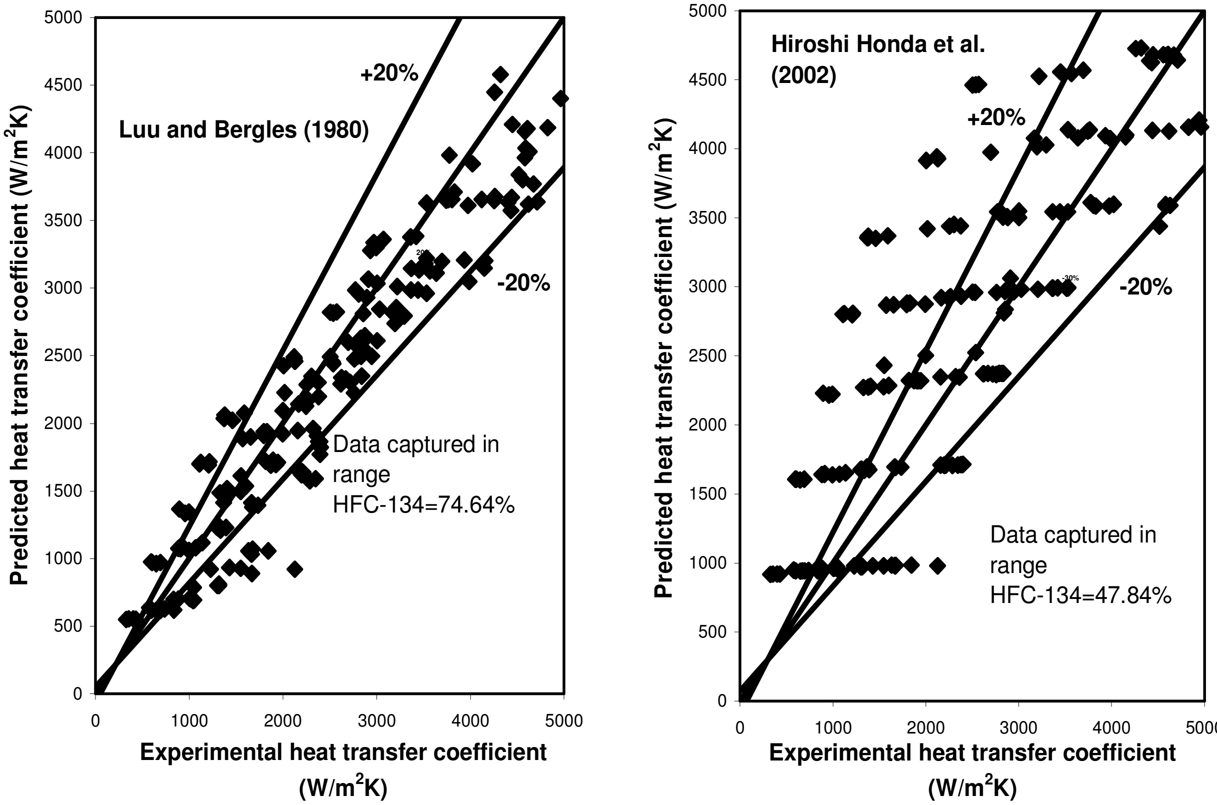

Figure 10. Comparison of micro-fin tube heat transfer data with existing correlations.

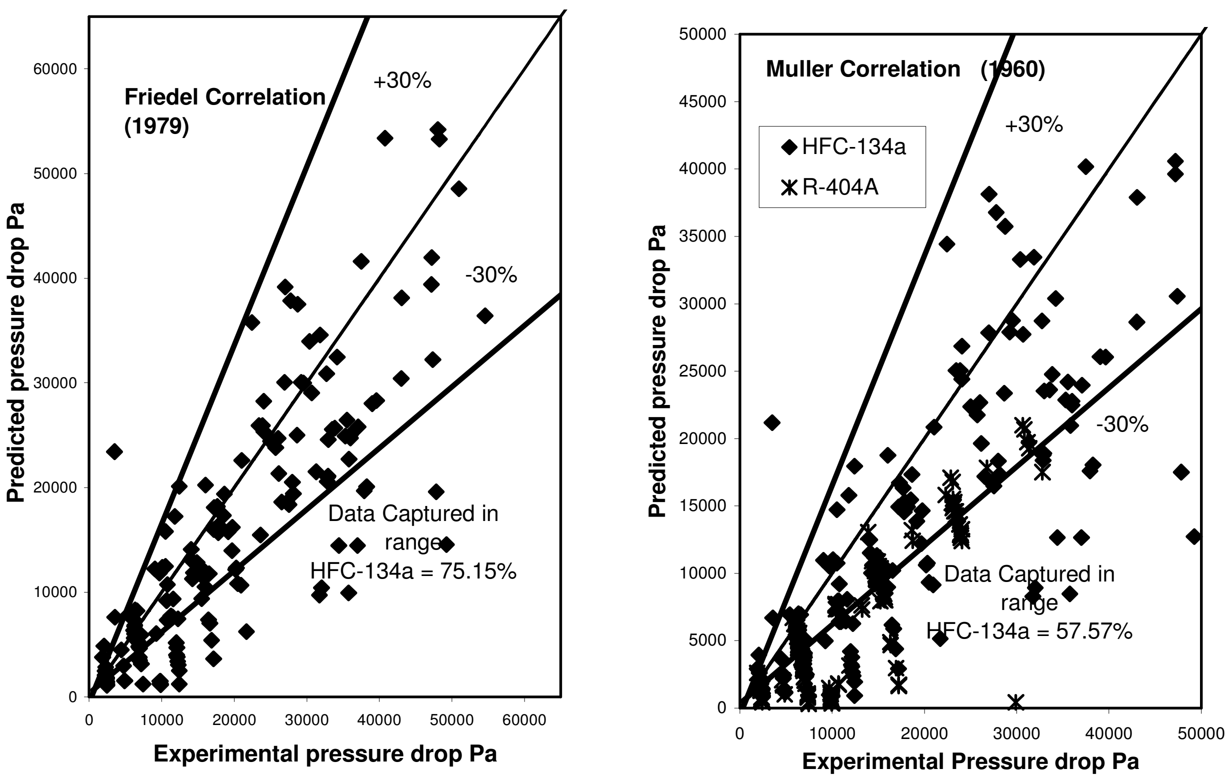

Figure 11. Comparison of smooth tube pressure drop data with existing correlations.

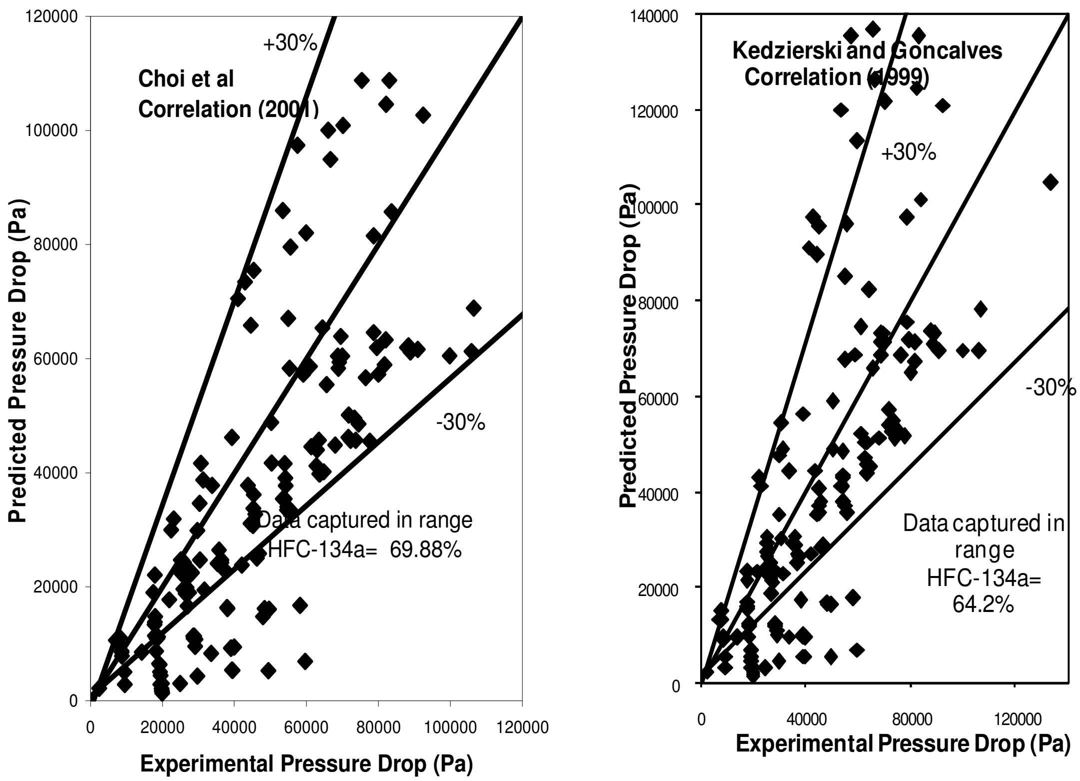

Figure 12. Comparison of micro-fin tube pressure drop data with existing correlations.

The results are compared with existing available correlations for validation of test facility. The experimental data points have good association with few available correlations except some data points from low and high mass flux and data points from higher condensing temperatures, which did not fall within ±20%.

ACKNOWLEDGEMENTS

The authors express their deep appreciation for the financial support provided for this setup by ASHRAE, USA. We would also like to thank our undergraduate students; Mr. Pushkar Natu, Mr. Sagar Shirolkar, Mr. Rahul Piese and Mr. Sachin Pole for their efforts during the fabrication of this test facility, and Mr. Yeole Madhusudan, Mr. Nehete Yatin, Mr. Chaudhari Ashish and Mr. Waykos Yogesh for their efforts during experimentation on smooth tube condensation. We would also like to thank our College authorities Dr. J. D. Bapat and Prof. S. V. Chaitanya for giving support at administration level.

NOMENCLATURE

A_i inner surface area of tube (m^2) $=\pi d_i L$
A_o outer surface area of tube (m^2) $= \pi d_o L$
a_f cross flow area (m2) $=IDxCxB/P_T$
B baffle space (m)
C clearance in U-tube (m)
C_{pl} specific heat of liquid refrigerant (kJ/kg.K)
C_{pv} specific heat of vapour refrigerant (kJ/kg.K)
C_{pw} specific heat of water (kJ/kg.K)
D characteristic diameter of tube (m)
D_e equivalent diameter of shell (m) $=4x\ (P_T^2 - \pi d_o^2/4)/(\pi do)$
d_i inner diameter of tube (m)
d_o outer diameter of tube (m)
G mass velocity of water ($kg/m^2.s$) $= m_w/a_f$
h_i film coefficient inner side (refrigerant) (W/m^2K)
h_o outside heat transfer coefficient (water side) (W/m^2K)
h_{ti} enthalpy at test condenser inlet (kJ/kg)
h_{to} enthalpy at test condenser outlet (kJ/kg)
ID inner diameter of shell (m)
k_t thermal conductivity of liquid refrigerant (W/m.K)
k_t thermal conductivity of tube material (W/m.K)
k_w thermal conductivity of water (W/m^2K)
L length of U-tube (m)
$LMTD$ average weighted logarithmic mean temperature difference (°C)
$LMTD_c$ logarithmic mean temperature difference (°C) for condensation process
$LMTD_d$ logarithmic mean temperature difference (°C) for

desuperheating process
$LMTD_s$ logarithmic mean temperature difference (℃) for sub cooling process
m_r mass flow rate of refrigerant (kg/s)
m_w mass flow rate of water (kg/s)
Nu Nusselt number
P saturation pressure (bar)
Pr_l Prandtl number for liquid refrigerant
Pr_w Prandtl nuber for water
P_{rc} reduced pressure=(P/Pcr)
P_T pitch of U-tube
Q_c rate of heat rejected by refrigerant during only condensation (kW)
Q_r total rate of heat rejected by refrigerant (kW)
Q_{sl} rate of heat rejected by refrigerant during sub cooling of refrigerant (kW)
Q_{sv} rate of heat rejected by refrigerant during desuperheating of refrigerant (kW)
Q_w rate of heat absorbed by cooling water (kW)
Re_l Reynolds number for liquid refrigerant
Re_g Reynolds number for vapour refrigerant
Re_w Reynolds number for water
R_i thermal resistance due to inner film coefficient (K/W)
R_o thermal resistance due to outer heat transfer coefficient (K/W)
R_{ov} overall thermal resistance (K/W)
R_t thermal resistance due to tube wall (K/W)
T_{hi} = refrigerant saturation temperature at the inlet of condenser (℃)
T_{ho} refrigerant saturation temperature at the outlet of condenser (℃)
T_{ri} refrigerant temperature at the inlet of condenser (℃)
T_{ro} refrigerant temperature at the outlet of condenser (℃)
T_{wc} estimated water temperature at the end of only condensation of refrigerant (℃)
T_{wd} estimated water temperature at the end of desuperheating of refrigerant (℃)
T_{wi} cooling water temperature at the inlet of shell (℃)
T_{wo} cooling water temperature at the outlet of shell (℃)
U_o overall heat transfer coefficient based on outer surface area (W/m^2.K)
X vapour quality of refrigerant
μ_w dynamic viscosity of water (N.s/m^2)
μ_g dynamic viscosity of liquid refrigerant (N.s/m^2)
μ_l dynamic viscosity of vapour refrigerant (N.s/m^2)
ρ_f density of liquid refrigerant (kg/m^3)
ρ_g density of vapour refrigerant (kg/m^3)
ΔP_{total} measured pressure drop during experimentation

$$\Delta P_{mom} \quad G^2\left\{\left[\frac{1}{\rho_l}\right]_{out} - \left[\frac{1}{\rho_g}\right]_{in}\right\}$$

ΔP_l pressure drop occurred during sub cooling process $= \left(\dfrac{4f_l(L_l/d_i)G^2}{2\rho_l}\right)$

ΔP_g pressure drop occurred during desuperheating process $= \left(\dfrac{4f_g(L_g/d_i)G^2}{2\rho_g}\right)$

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