TWO-PHASE CONDENSATION HEAT TRANSFER COEFFICIENTS AND PRESSURE DROPS OF R-404A FOR DIFFERENT CONDENSING TEMPERATURES IN A SMOOTH AND MICRO-FIN TUBE

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Abstract

Two phase heat transfer coefficients and pressure drops of R-404A in a smooth (8.56 mm ID) and micro-fin tube (8.96 mm ID) are experimentally investigated. Different from previous studies, the present experiments are performed for different condensing temperatures, with superheating and sub cooling and using hermetically sealed compressor. The test runs are done at average saturated condensing temperatures ranging from 35°C to 60°C. The mass fluxes are between 90 and 800 kg $m^{-2}s^{-1}$. The experimental results from both smooth and micro-fin tubes show that the average heat transfer coefficient and pressure drop increases with mass flux but decreases with increasing condensing temperature. The average heat transfer coefficient is 30-210% higher for micro-fin tube than that of smooth tube, with moderate increase in pressure drop ranging from 10-55%. New correlations based on the data gathered during the experimentation for predicting condensation heat transfer coefficients are proposed for wide range of practical applications.

Keywords: *Condensation; Enhancement; Micro-fin tube; R404a*

NOMENCLATURE

A_i	inner surface area of tube (m^2) (=$\pi d_i L$)
A_o	outer surface area of tube (m^2) (= $\pi d_o L$)
Ac	cross sectional area (m^2) (=$\frac{\pi}{4} d_i^2$)
Bo	Bond number, $\left(Bo = \frac{\rho_l g \pi d_i e_f}{8 \sigma N} \right)$
C_{pl}	specific heat of liquid refrigerant (kJ $kg^{-1}.K^{-1}$)
C_{pv}	specific heat of vapour refrigerant (kJ $kg^{-1}.K^{-1}$)
C_{pw}	specific heat of water (kJ $kg^{-1}.K^{-1}$)
d_i	maximum inner diameter of tube (m)
d_o	outer diameter of tube (m)
d_t	fin tip diameter (m)

e_f fin height (m)

Fr Froude number $\left(Fr_l = \dfrac{G^2}{\rho_l{}^2 g d_i} \right)$

G mass flux of refrigerant (kg $s^{-1}.m^{-2}$) (= m_r/A_c)

h_i film coefficient inner side (refrigerant) (W m^{-2} K^{-1})

h_o outside heat transfer coefficient (water side) (W m^{-2} K^{-1})

h_{ti} specific enthalpy at test condenser inlet (kJ kg^{-1})

h_{to} specific enthalpy at test condenser outlet (kJ kg^{-1})

Ja modified Jacob number, $\left\{ Ja = \dfrac{C_{p_v}\,(T_{R_i} - T_{h_i}) + C_{p_l}\,(T_{h_o} - T_{R_o})}{h_{t_i} - h_{t_o}} \right\}$

k_l thermal conductivity of liquid refrigerant (W $m^{-1}.K^{-1}$)

k_t thermal conductivity of tube material ($Wm^{-1}.K^{-1}$)

k_w thermal conductivity of water (W $m^{-1}K^{-1}$)

L length of U-tube (m)

$LMTD$ average weighted logarithmic mean temperature difference (^oC)

$LMTD_c$ logarithmic mean temperature difference (^oC) for condensation process

$LMTD_d$ logarithmic mean temperature difference (^oC) for desuperheating process

$LMTD_s$ logarithmic mean temperature difference (^oC) for sub cooling process

m_r mass flow rate of refrigerant (kg s^{-1})

m_w mass flow rate of water (kg s^{-1})

N number of fins

Nu Nusselt number $\left(= \dfrac{h_i d_i}{k_l}\right)$

P saturation pressure (bar)

Pr_l Prandtl number for liquid refrigerant $\left(= \dfrac{\mu_l C_{p_l}}{k_l}\right)$

P_{cr} critical pressure, (bar)

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_{Eq} equivalent Reynolds number, (=$Re_l + (\frac{\mu_g}{\mu_l})(\frac{\rho_f}{\rho_g})^{0.5} Re_g$)

Re_l Reynolds number for liquid refrigerant

Re_g Reynolds number for vapour refrigerant

R_X heat transfer area enhancement factor for micro-fin tube, (=$\{\frac{2e_f N(1-\sin\beta/2)}{\pi d_i \cos\beta/2}+1\}/\cos\gamma$)

T_h average refrigerant saturation temperature of test condenser (^{o} C), (= ($T_{hi} + T_{ho}$) / 2)

T_{hi} refrigerant saturation temperature at the inlet of condenser (^{o} C)

T_{ho} refrigerant saturation temperature at the outlet of condenser (^{o}C)

T_{ri} refrigerant temperature at the inlet of condenser (^{o}C)

T_{ro} refrigerant temperature at the outlet of condenser (^{o} C)

T_{wc} estimated water temperature at the end of only condensation of refrigerant (^{o}C)

T_{wd} estimated water temperature at the end of desuperheating of refrigerant (^{o}C)

T_{wi} cooling water temperature at the inlet of shell (^{o}C)

T_{wo} cooling water temperature at the outlet of shell (^{o}C)

U_o overall heat transfer coefficient based on outer surface area (W m^{-2}.K^{-1})

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})

β fin tip angle

γ helix angle

σ surface tension of refrigerant at liquid-vapour interface

1. Introduction

Refrigerant vapor discharged from a compressor in refrigeration equipment is generally cooled and condensed in a condenser via heat transfer to a secondary heat transfer fluid such as air or water. If the condenser does not dissipate the heat at the required rate, the discharge pressure would build up resulting in an increase in compressor power. One way of increasing the condenser effectiveness is to enlarge the size of condensers. This, however, may not necessarily be practical from the viewpoint of system maintenance and initial cost since more refrigerant is to be charged. Therefore, the best way to design effective condensers would be to keep the size as small as possible with special heat transfer enhancement mechanisms like use of micro-fin tubing [see Fig.17 for actual photograph]. For this it is important to know the flow condensation heat transfer coefficients (HTCs) of a working fluid in a horizontal tube for air-cooled condensers.

Through much research efforts, flow condensation phenomenon in horizontal tubes is well understood and few correlations have been proposed. An excellent state of the review on condensation phenomenon associated using micro-fin tube with various refrigerants is made by Wang H.S. and Honda H. in 2003 .Dongsoo Jung, et al, (2004) suggested a correlation for refrigerants HFC-134a, R-407C, R-410A and R-22 for both smooth and micro-fin tube for 40^{o}C condensing temperature and 100-300 kg m^{-2} s^{-1}. In 1974 Cavallini and Zecchin suggested a semi empirical correlation based upon a heat and momentum similarity in annular flow regime for smooth tube. Louay M. Chamra et al.,(2004) has done a comparative study of three correlations developed for micro-fin tube and found the Yu and Koyama model displayed poor prediction on most of the R22 pure refrigerant data used for its validation. In 1995 Tandon et al.suggested a modified correlation from that of Akers and Rosson. In 1994 and 1998 Dobson and Chato measured flow condensation HTCs of R12 and R134a and suggested an empirical correlation in annular flow regime which is similar to that of the convective evaporation contribution in additive type flow boiling correlations. Minh. Luu.and Bergles (1980) developed correlation for horizontal micro-fin tube using condensation of refrigerant-113. Cavallini etal. (1995) suggested correlations for condensation of new refrigerants inside smooth and enhanced tubes. Kedzierski and Goncalves (1999) proposed correlation for horizontal convective condensation of alternative refrigerants within a micro-fin tube. Thipjak et al., (2003) modified the Cavallini et al. (1995) correlation for high mass flux of HFC-134a for both smooth and micro-fin tubes to improve the accuracy.

Even though many researchers suggested various correlations, they carried out flow condensation heat transfer research with working fluids mainly ozone depleting substances such as CFCs and HCFCs. Most of the experimental work was carried out using refrigerant pump, in very few studies compressor was used. Also most of the correlations have been developed for limited range of operating conditions such as condensing temperatures, mass flux etc [Agrawal et al. (2004), Dongsoo Jung et al. (2003, 2004), Thipjak et al. (2003), Yiong and Srinivas (2003) Smit Meyer (2002), Keumnam et al. (2000), Eckels and Tesene (1999), Dobson and Chato (1998), Eckels and Pate (1991), Schlager et al. (1990), Khanpara et al. (1986), Stoecker and Kornota (1985), Tandon et al. (1985),Tichy et al. (1985), Wang et al.(1985), Said and Azer (1983), Minh and Bergles (1980),]. No correlation was found which was developed for high condensing temperature such as 55-65^{o}C.

The correlations proposed in the present study are developed for wide range of condensing temperatures and mass flux. The range of experimental conditions tested in this study is listed in Table 1. As the hermetically sealed compressor is used instead of pump in the test facility, the test conditions are close to practical condensation process. Moreover in the refrigerant condenser, the condensation process is always associated desuperheating and sub cooling which is considered in this study during experimentation. Even though the present study has not thrown any light on effect of vapour quality on heat transfer coefficient, its effects has been incorporated in the average heat transfer coefficient.

Table1 Experimental conditions

Operating Parameters	**Range**
Saturation temperature, T_h, ^{o}C	35-60^{0}C
Mass flux, G, kg m^{-2}s^{-1}	100-800
Cooling water temperature, T_w, ^{o}C	5-30^{0}C

2. Description of the test apparatus

The test apparatus, as shown schematically in Fig.1 [see Fig.18 for actual photograph] consists of four circuits namely, refrigerant main, refrigerant auxiliary, cooling water and chilled water circuit. Details of these circuits are given below.

The refrigerant main circuit joins 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 tube. Thermostatic expansion valve is used as an expansion device. The evaporator is of tank and coil type with refrigerant flows through coil surrounded by water in the tank, heaters are immersed in the tank to provide heat source for evaporator as well as to 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, chilled water flowing through the shell of diameter 50.8 mm. In order to induce turbulence and direct the water flow outside the tubes, segmental type baffles are employed. The dimensions of the smooth and micro-fin tubes are listed in Table 2

The chilled water is used in test rig which flows in closed cycle between evaporator and test condenser. The circuit mainly joins components such as, evaporator to pump to rotameter to test condenser and back to evaporator. 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 maintain the desired temperature an arrangement of heaters are provided in the evaporator.

The cooling water circuit as shown in Fig. 2 is used to cool the water circulating from the main condenser, the heat absorbed in the main condenser by cooling water is rejected in the forced drought cooling

Table 2 Smooth and micro-fin tube dimensions

Parameter	**Smooth Tube**	**Micro-fin Tube**
Outside diameter, d_o, mm	9.42	9.52
Bottom thickness, t, mm	0.64	0.28
Number of fins, N	--------	60
Spiral angle, γ, degree	--------	18
Apex angle, β, degree	--------	45
Fin height, e_f, mm	--------	0.2
Fin tip diameter, d_t, mm	--------	8.56
Max. inside diameter, d_i, mm	8.14	8.96
Length of tube, L, m	4.5	4.5
Cross sectional area, A_c, mm^2	52.04	63.053

tower and circulating back from main condenser with the help of pump. Plate type valves are used in lines to regulate the flow of refrigerant and water. The desired condensing temperature is achieved with the help of this circuit by adjusting the flow rate of cooling water.

3. Instrumentation

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

3.1 Temperature measurements

- Inlet and outlet of the test condenser tube (refrigerant circuit)
- Inlet and outlet of the test condenser shell (chilled water circuit)
- In the evaporator tank, to measure the temperature of chilled water in order to monitor the steady state.

3.2 Pressure measurements

- At inlet and outlet of the test condenser, to measure the refrigerant pressures, required to calculate the pressure drop across the test condenser.
- Mounted on main condenser, to measure condenser pressure, to monitor the condensing temperature and to ensure the system balancing when the refrigerant flow rate is changed

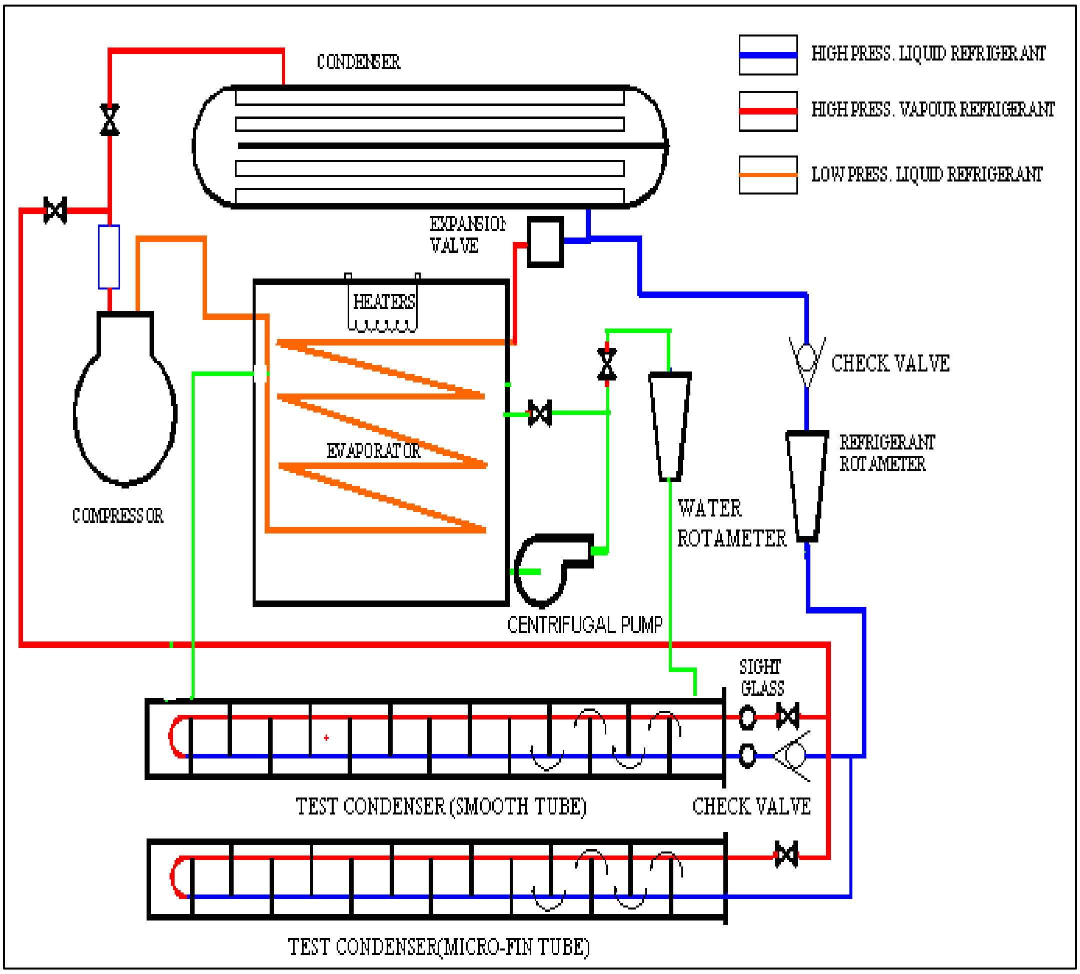

Fig.1 Experimental test facility

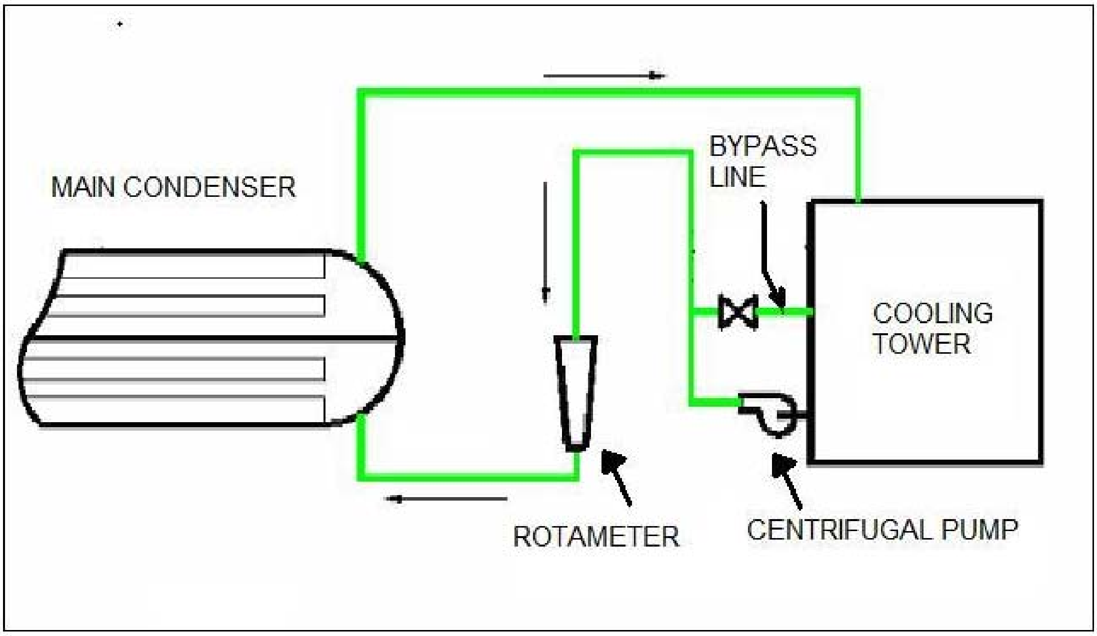

Fig.2 Cooling water circuit for main refrigerant circuit

3.3. Flow measurements

- In the auxiliary refrigerant circuit, to measure the refrigerant flow rate in the test condenser, required to calculate mass flux and heat rejected by refrigerant.
- 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.

PT100 (Resistance Temperature Detector made of platinum and has a base 100 Ω at 0^{o}C) 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 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.

4. Data reduction

The data-analysis procedure determines the average convective heat transfer coefficient of refrigerant side, which also takes into account oil present in the refrigerant (because refrigerant carries the compressor oil in the test condenser). In addition, the data analysis determines the correlation constants required for average convective heat transfer coefficient of water using modified Wilson plot technique (Jose et al., 2005). 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 %.

$$O_r = Q_{sv} + Q_c + Q_{sl} \quad (1)$$

Where

$$Q_{sv} = m_r C_{pv} (T_{ri} - T_{hi}) \quad (2)$$

$$Q_{sl} = m_r C_{pl} (T_{ho} - T_{ro}) \quad (3)$$

$$Q_c = m_r (h_{ti} - h_{to}) \quad (4)$$

$$Q_w = m_w C_{pw} (T_{wo} - T_{wi}) \quad (5)$$

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

$$T_{wd} = T_{wi} + \frac{Q_{sv}}{m_w C_{pw}} \quad (6)$$

$$T_{wc} = T_{wd} + \frac{Q_c}{m_w C_{pw}} \quad (7)$$

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

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

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

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

The overall heat transfer coefficient is determined by using;

$$U_o = \frac{Q_r}{A_o LMTD} \qquad (12)$$

Assuming that the resistance of the copper tube is negligible, the following formula is obtained for the average heat transfer coefficient on the inner surface of the test tube:

$$h_i = \frac{1}{\left(\frac{1}{U_o} - \frac{1}{h_o}\right)\frac{A_o}{A_i}} \qquad (13)$$

The annulus–side heat transfer coefficient (h_o) is determined with a correlation developed using modified Wilson-plot technique (Jose et al., 2005) specifically for the annulus of test sections. 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 (Jose et al., 2005) with a coefficient of correlation above 0.95.

The maximum inner diameter of the micro-fin tube is defined as the outer diameter of the micro-fin tube minus twice the bottom wall thickness [see Fig.3]. Once the overall heat transfer coefficient and the annulus–side heat transfer coefficient are determined, the average in-tube heat transfer coefficient can be determined using Eq. (13).

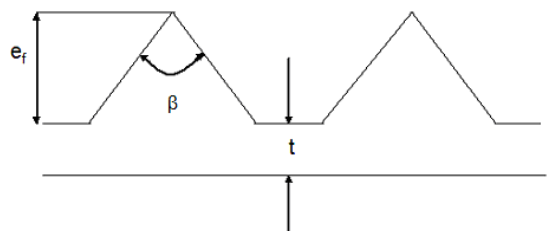

Fig. 3 Cross section of the micro-fin tube

5. Results and discussion

The heat transfer coefficients and pressure drops of R-404A are measured in a smooth and micro-fin tube at different condensing temperatures of 35, 40, 45, 50, 55 and 60^{o}C. Around 560 data points are taken during experimentation on smooth and micro-fin tubes.

5.1 Condensation heat transfer

Condensation heat transfer data for smooth tube and micro-fin tube with R-404A are shown in Fig.4. For both tubes the heat transfer coefficient increases with mass flux but decreases with increasing condensing temperature. The heat transfer coefficients obtained for micro-fin tube are greater than that of smooth tube for all condensing temperatures and mass fluxes.

5.2. Pressure drop

Average pressure drop data during condensation of R-404A for smooth tube and micro-fin tube are as shown in Fig.5. As with heat transfer coefficients, the pressure drop varies considerably with mass flux and condensing temperature. The values of pressure drop in the Fig.5 are measured for 4.5 m length of test section.

5.3. 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, and pressure level.

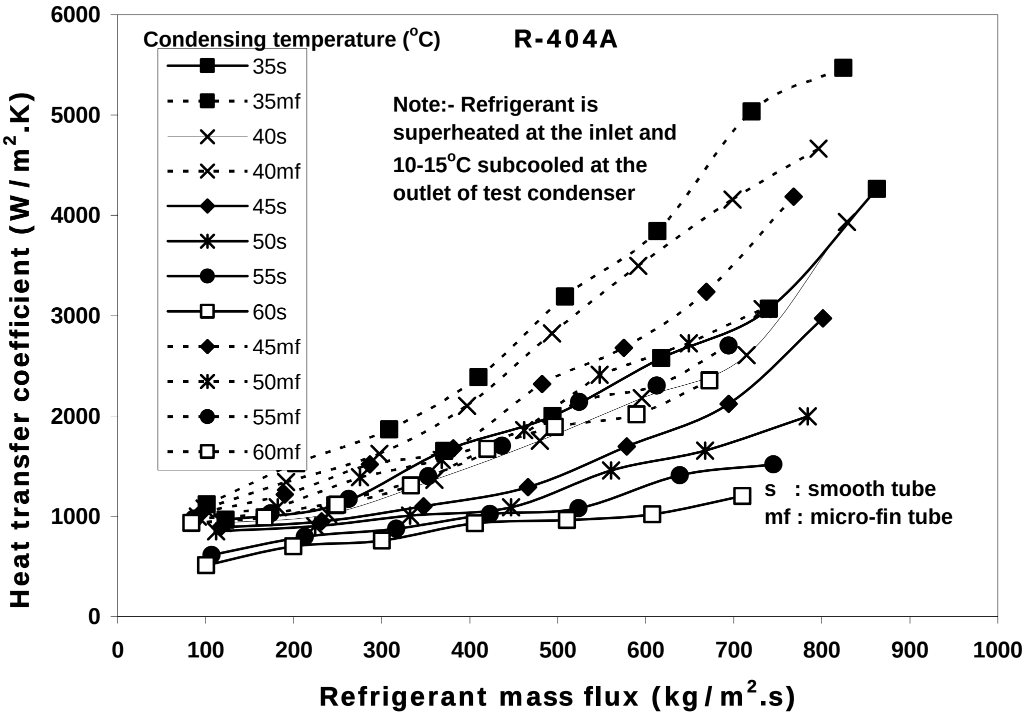

Fig.4 R-404A Condensation heat transfer coefficient in a smooth and micro-fin tube

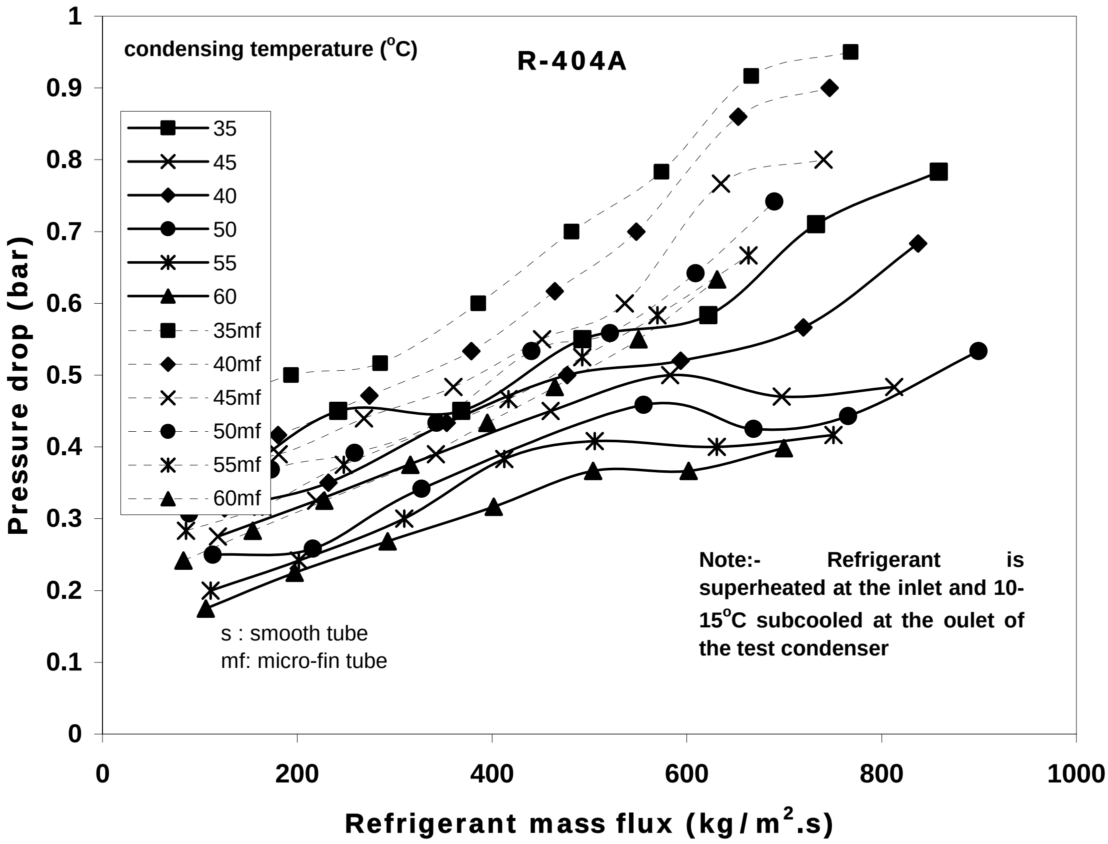

Fig.5 R-404A Condensation pressure drop in a smooth and micro-fin tube

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

Fig6. shows both heat transfer enhancement factors, EF, and pressure drop penalty factors, PF, for the micro-fin tube with R-404A. The EFs vary from minimum of 1.5 at the low mass flux to maximum of 2.1 at high mass flux. The PFs are also shown in Fig.6. and varies from minimum 1 at low mass flux to a maximum 1.55 at high mass flux. There appears to be an overall trend of increasing enhancement factors with increasing mass flux and increasing penalty factors with increasing mass flux. At any given mass flux and condensing temperature EFs are greater than PFs

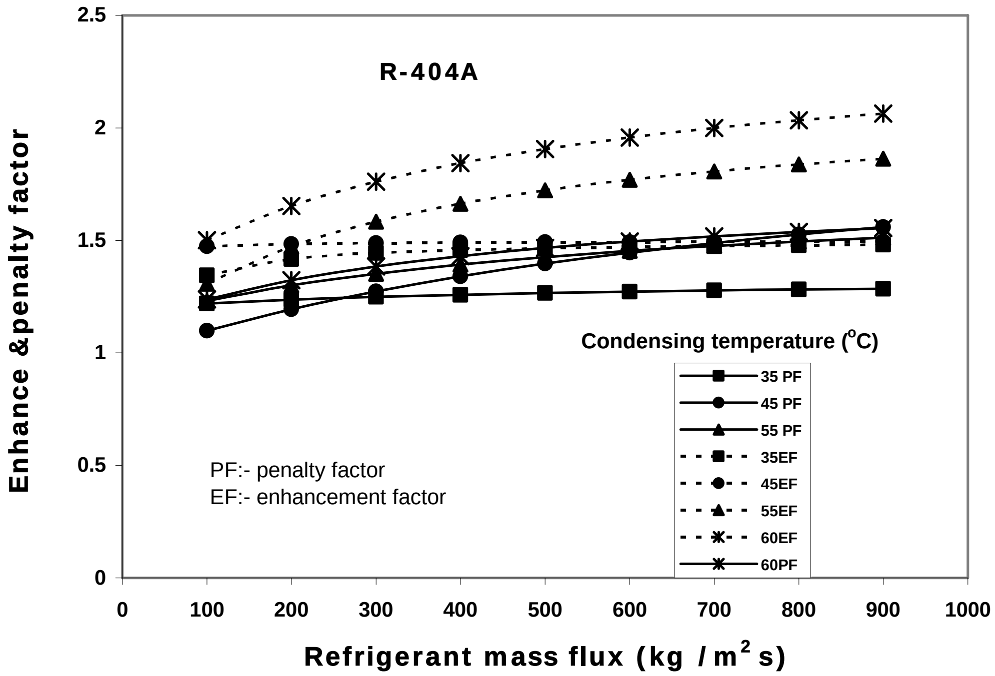

Fig.6 R-404A heat transfer enhancement and pressure drop penalty factor

5.4 Experimental uncertainty

A propagation of error analysis (Kline and McClintock 1953) is used to obtain the uncertainty listed in Table 3 with a confidence interval of 85-90%.The average uncertainties in smooth tube and micro-fin tube for various parameters are nearly equal except the uncertainty of refrigerant side heat transfer coefficient.

Table 3 Average Uncertainties of various parameters

Sr.No	Parameters	Smooth tube	Micro-fin tube
		%	%
1	LMTD	±13.2	±15.34
2	Refrigerant mass flow rate	±2	±2
3	Water flow rate	±2.81	±2.45
4	Heat rejected by refrigerant	±4.72	±4.09
5	Heat absorbed by cooling water	±9.22	±7.15
6	Overall heat transfer coefficient (U_o)	±13.3	±11.49
7	Refrigerant side heat transfer coefficient (h_i)	±18.2	±24.47
8	Water side heat transfer coefficient (h_o)	±5.03	±3.04
9	Pressure drop	±4.75	±5.61

5.5 Predicted correlations of the present work

The new correlations (see Fig.7 and Fig.8) are developed for determination of heat transfer coefficients in smooth and micro-fin tubes. The new correlations take into account the effect of condensing temperature, super heating and sub cooling of refrigerant. The Fig.7 and Fig.8 show the deviation of experimental values with the predicted line of heat transfer coefficient for smooth and micro-fin tube respectively. The maximum deviation in both cases is within ± 20%. The modified Jacob number (Ja) takes care of effect of superheating and sub cooling on condensation heat transfer coefficient. The inclusion of pressure ratio (P/P_{cr}) makes the correlation suitable for wide range of condensing temperatures. Multiple linear regression technique

(Chapra, 1990) is used to develop correlations. The coefficient of correlations obtained for smooth tube is 0.9341 and that for micro-fin tube is 0.9310.

In case of micro-fin tube, area enhancement factor (R_x) is also a governing parameter in addition to other dimensionless numbers. The product of Froude and Bond number takes in to account the effect of relative magnitude of inertia force and surface tension force.

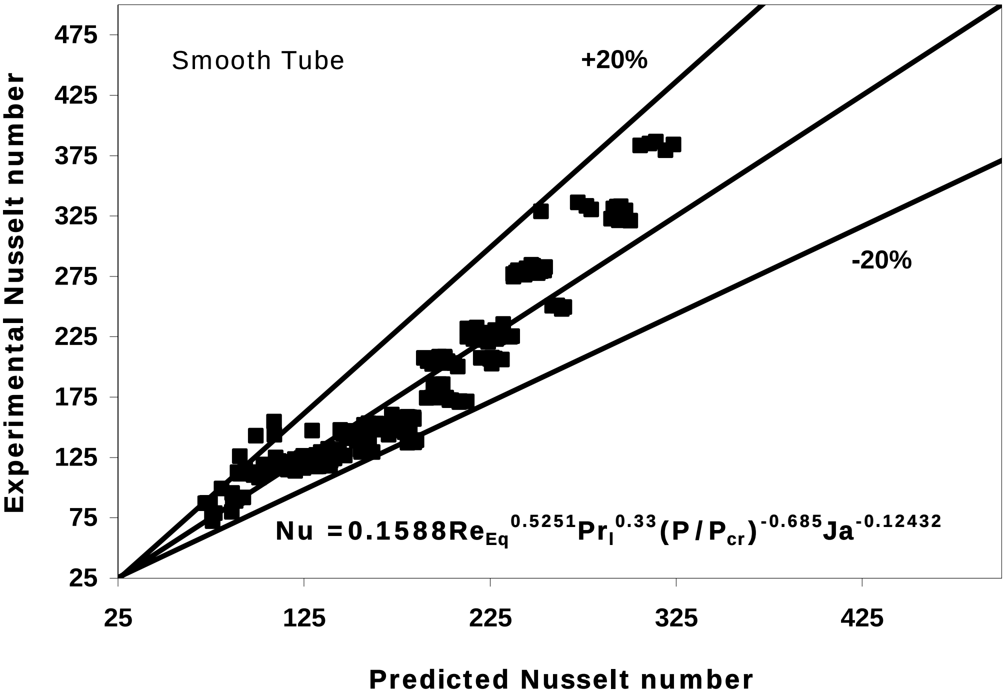

Fig.7 Predicted condensation heat transfer coefficient correlation for R-404A in a smooth tube

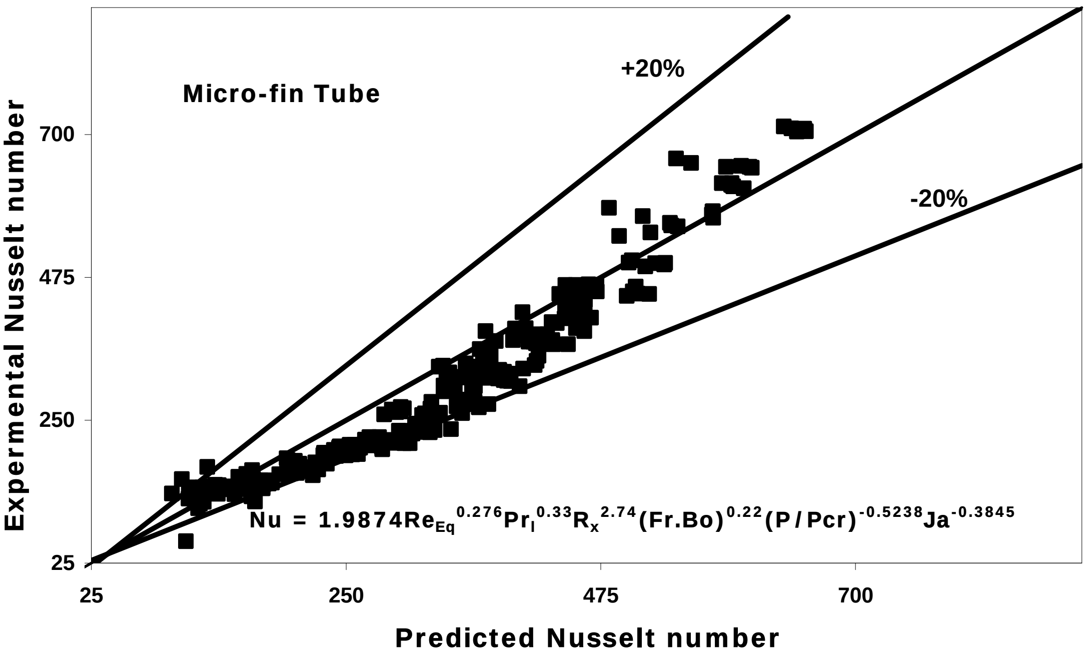

Fig.8 Predicted condensation heat transfer coefficient correlation for R-404A in a micro-fin tube

5.6. Correlation comparison

The Nusselt numbers of smooth tube and micro-fin tube obtained from predicted correlations are compared with the Nusselt numbers obtained from available correlations. Their deviations for different condensing temperatures are presented from Fig.9 to Fig.16. For smooth tube, values obtained from Tandan et al. (1985) and few values of Akers et al. (1959) fall within 20% of the predicted correlation, rest all points fall between 20% and 50%, values of Cavallini etal. (1974) fall just above 50 % of predicted correlation, for condensing temperature 35°C and 40°C. [Fig.10] The deviation increases with increasing condensing temperature. For condensing temperatures 55, 60, 65°C [Fig.12] all the values fall between 20% to 60% zone. This is because most of correlations are developed for condensing temperatures in the range between 30-50°C.

In case of micro-fin tube values obtained from Thipjak (2003) and Bergles (1980) fall within the ±20%, while values obtained from Kedzierski etal. (1999) under predict the proposed correlation and fall between -20% and -50% of predicted correlation line, for condensing temperature 35°C and 40°C [Fig.14]. However for condensing temperature 55 and 60°C [Fig.16] only values obtained from Bergles (1980) and few from Kedzierski et al. (1999) fall within ±20%. The deviation for Cavaillini et al. (1995) values is well above +60% for all condensing temperature.

6. 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 is 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 instrumentations used for measurements are calibrated from recognized calibration centres.

The condensation and pressure drop of R-404A in smooth and micro-fin tubes are measured and obtained the values of condensation heat transfer coefficients for different mass flux and condensing temperatures. 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 new correlations are developed for determination of condensation heat transfer coefficients for smooth and micro-fin tubes which are suitable for wide range of condensing temperatures. The correlations also take into account major practical aspects like, superheating, sub cooling of refrigerant and effect of change in condensing temperature etc. The correlations are especially important for designing R-404A air-cooled condensers suitable for wide range of ambient conditions. R-404A is a promising alternative refrigerant for low temperature commercial refrigeration applications such as deep freezer.

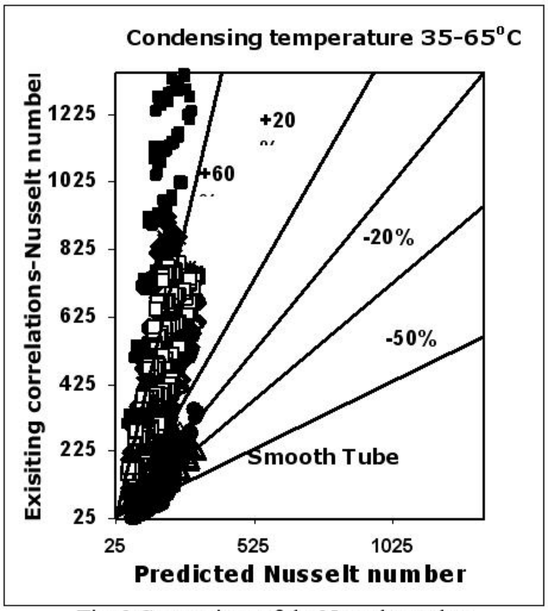

Fig. 9 Comparison of the Nusselt number

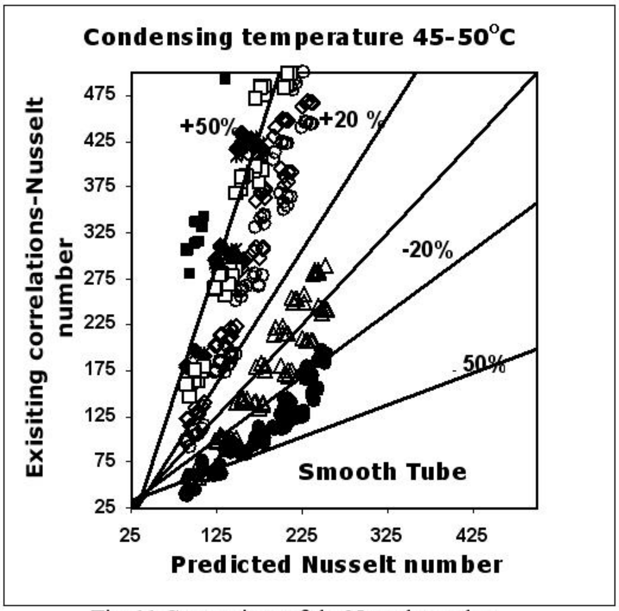

Fig. 11 Comparison of the Nusselt number

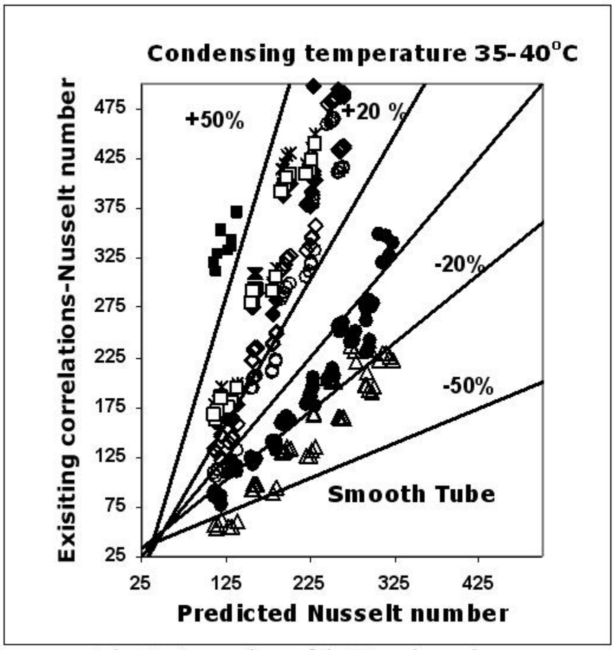

Fig. 10 Comparison of the Nusselt number

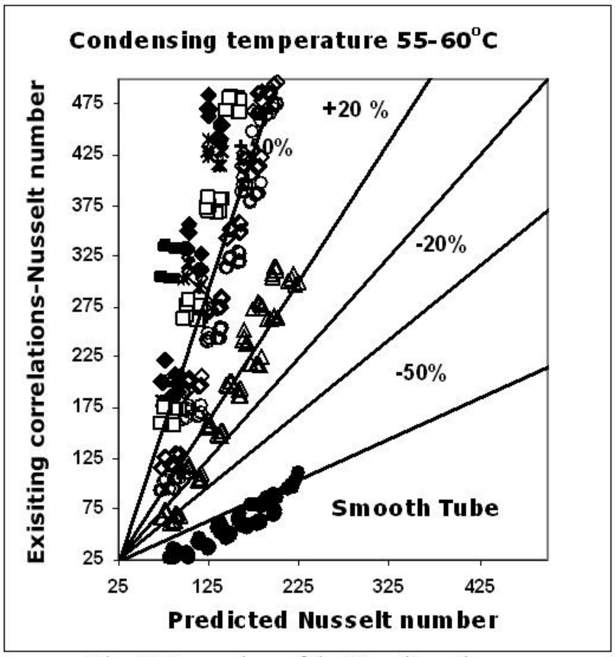

Fig. 12 Comparison of the Nusselt number

▪ Cavallini-Zecchin(1974)
⁎ Shah (1979)
• Tandon,Varma, and Gupta (1985)
◇ Akers etal (1959)
△ Boyko and Kruzhillin (1967)
○ Azer et al (1971, 1972)
♦ Dobson (1998)
□ Akers,Deans and Crossef (1958)

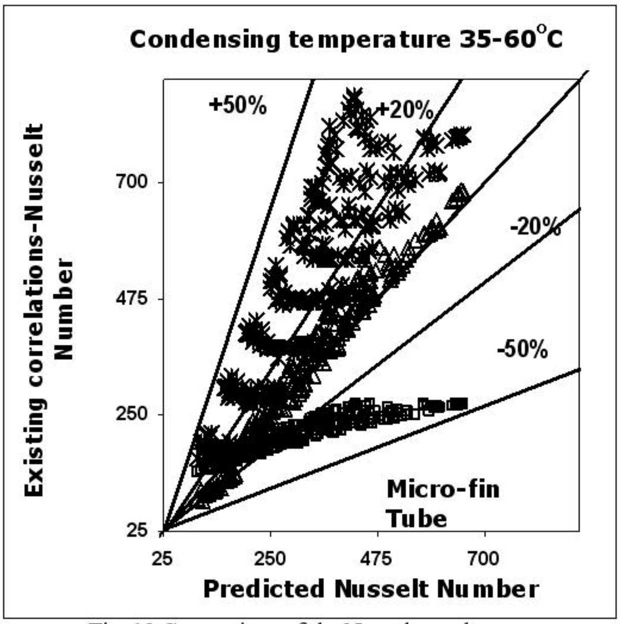

Fig. 13 Comparison of the Nusselt number

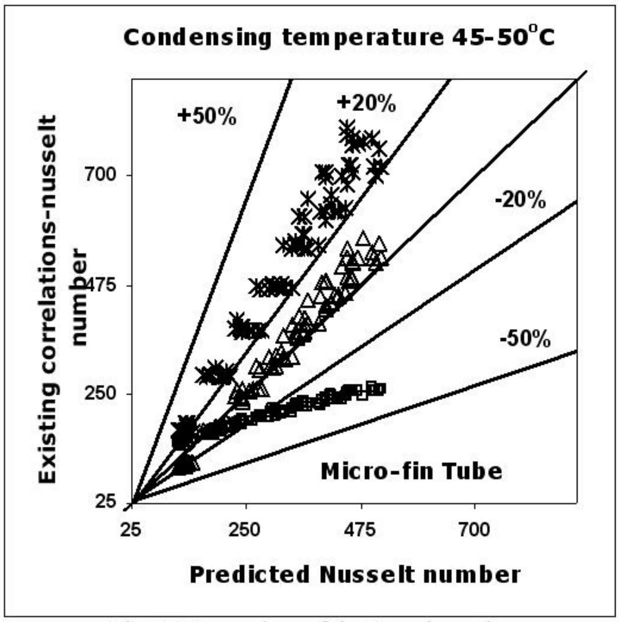

Fig. 15 Comparison of the Nusselt number

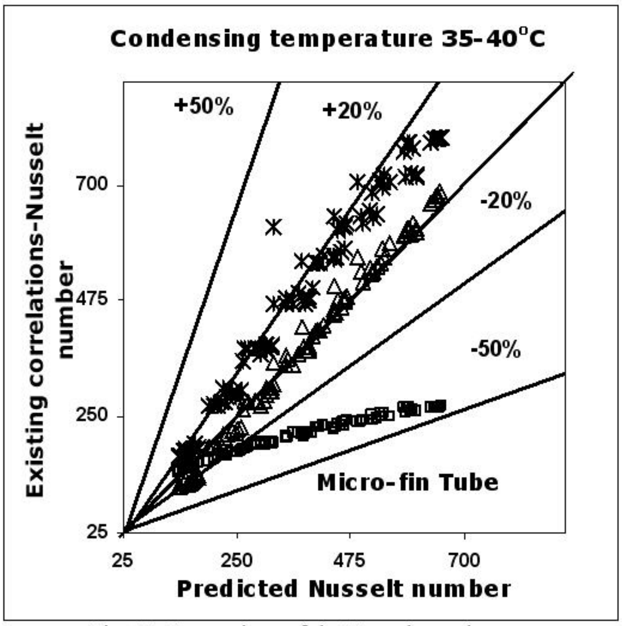

Fig. 14 Comparison of the Nusselt number

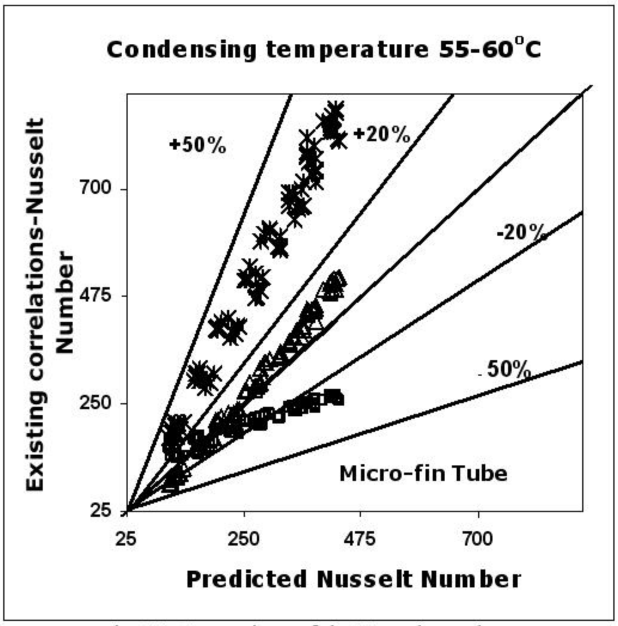

Fig. 16 Comparison of the Nusselt number

The smooth tube correlation is;

$$Nu = 0.1588.\mathrm{Re}_{Eq}{}^{0.5251}.\mathrm{Pr}_l{}^{0.33}.\left(\frac{P}{Pcr}\right)^{-0.685} Ja^{-0.1243}$$

The micro-fin tube correlation is;

$$Nu = 1.987.\mathrm{Re}_{Eq}{}^{0.276}.\mathrm{Pr}_l{}^{0.33}.Ja^{-0.3845}.\left(\frac{P}{Pcr}\right)^{-0.5238}.Rx^{2.74}.(Fr.Bo)^{0.22}$$

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References

[1] Agrawal, N.K., Ravi Kumar, and Varma, K.H., 2004. Heat transfer augmentation by segmented tape inserts during condensation of r-22 inside a horizontal tube. ASHRAE Transactions, 143-149.

[2] Akers, W.W., Deans H.A., Crosser O.K., 1958. Condensation heat transfer within horizontal tubes. Proce. 2nd Nat. Heat Transfer Conf., ASME/Ch E.

[3] Akers, W.W., Deans, H.A., Crosser, O.K., 1959. Condensation heat transfer within horizontal tubes. Chemical Engineering Progress Symposium Series, 55 (29), 171–176.

[4] Azer, N.Z., et al., 1972. Local heat transfer coefficients during annular-flow condensation. ASHRAE Transaction 78 (2): 135-313.
[5] Boyko, L.D., Kruzhilin, G.N., 1967. Heat transfer and hydraulic resistance during condensation of steam in a horizontal tube and in a bundle of tubes. Int J Heat Mass Transfer;10, 361–73.
[6] Cavallini, A., Doretti, L., Klammsteiner, N., Longo, L.G. , Rossetto, L., 1995. Condensation of new refrigerants inside smooth and enhanced tubes. In: Proc. 19th Int. Cong. Refrigeration, vol. IV, Hague, The Netherlands, 105–14.
[7] Cavallini, A., Zecchin, R.A., 1974. A dimensionless correlation for heat transfer in forced convection condensation. proceedings of the Sixth International Heat Transfer Conference, 3, 309–313.
[8] Chapra, Steven, C., and Canale Raymond P., 1990. Numerical Methods for Engineers. McGRAW-HILL INTERNATIONAL EDITIONS, pp 330-360.
[9] Dobson, M.K., and Chato, J.C., Feb.1998. Condensation in smooth horizontal tubes. ASME Journal of heat transfer, 193-210.
[10] Dongsoo, Jung etal., 2003. Flow condensation heat transfer coefficients of pure refrigerants. International Journal of Refrigeration, Volume 26, Issue 1, Pages 4-11
[11] Dongsoo, Jung, etal., 2004. Flow condensation heat transfer coefficients of R22, R134a, R407C, and R410A inside plain and micro-fin tubes. International Journal of Refrigeration, Volume 27, Issue 1 January 2004, Pages 25-32
[12] Eckels, S.J., and Pate, M. B., 1991. In-tube evaporation and condensation of refrigerant-lubricant mixtures of HFC-134a and CFC-12. ASHRAE Transactions, Vol. 97, Part 2.
[13] Eckels, Steve, J., Tesene, B., 1999. A Comparision of R-22, R-134a, R-410a, and R-407C Condensation Performance in Smooth and Enhanced tubes: Part I, Heat Transfer. ASHRAE Transactions, 428-451
[14] Jose, F.S., Francisco, J.U., Jaime, S., and Antonio, C., 2005. Experimental apparatus for measuring heat transfer coefficients by the Wilson plot method, European Journal of Physics 26, N1-N11.
[15] Kedzierski, M. A., Goncalves, J. M., 1999. Horizontal convective condensation of alternative refrigerants within a microfin tubes. Enhanced Heat Transfer, 6, 161–178.
[16] Kern, 2003. Process Heat Transfer Second edition. Tata Mc Graw- Hill Publishing Company Ltd., India
[17] Keumnam, Cho, Sang-Jin, Tae., 2000. Condensation heat transfer for R-22 and R-407C refrigerant-oil mixtures in a micro-fin tube with a U-bend. International Journal of Heat Transfer, 2043-2051.
[18] Khanpara, J.C, Bergles, A.E., Pate, M.B., 1986-2B. Augmentation of R-113 in-tube evaporation with micro-fin tubes. ASHRAE Transactions.
[19] Khanpara, J.C., Pate, M.B., and Bergles, A.E., 1986. Augmentation of R-113 In-tube Condensation with Micro-Fin tubes. Heat Transfer in Air Conditioning and Refrigeration Equipment, HTD-65, New York, American Society of Mechanical Engineers., 21-32.
[20] Kline, S.J, and McClintock, 1953. Describing Uncertainties in Single-Sample Experiments. Mech.Eng, 3.
[21] Louay, M., Chamra, etal. 2004. Evaluation of existing condensation heat transfer models in horizontal micro-fin tubes, Experimental Thermal and Fluid Science, 617–628.
[22] Minh, Luu, and Bergles, 1980. Enhancement of horizontal in-tube condensation of R-113. ASHRAE Transactions, 293-310.
[23] Said, S.A., and Azer, N.Z., 1983. Heat transfer and pressure drop during condensation inside horizontal finned tubes. ASHRAE Transactions, 114-135.
[24] Schlager, L.M, Pate, M.B, Bergles, A.E., Nov. 1990. evaporation and condensation heat transfer and pressure drop in horizontal, 12.7 mm Micro-Fin tubes with Refrigerant 22. ASME Journal of Heat Transfer.
[25] Shah, M. M., 1979. A general correlation for heat transfer during film condensation inside pipes. Int J Heat Mass Transf, 22 (4) 547–556.
[26] Smit, F.J, Meyer, J.P., Oct. 2002. R-22 and Zeotropic R-22/R-142b mixture condensation in micro-fin, high-fin, and twisted tape insert tubes, ASME Journal of Heat Transfer.
[27] Stoecker, W.F., and Kornota, E., 1985. Condensing coefficients when using refrigerant mixtures, ASHRAE Transactions, 1350-1367.
[28] Tandon, T.N., Varma, H.K., and Gupta, C.P., 1985. An experimental investigation of forced convection condensation during annular flow inside a horizontal tube. ASHRAE Transactions, 343-355.
[29] Thipjak, etal., 2003. Two-phase condensation heat transfer coefficient of HFC-134a at high mass flux in a smooth and micro-fin tubes. Int.Comm.Heat Mass Transfer, Vol 30, No.4, pp 577-590.

[30] Tichy, J.A., Macken, N.A., and Duval, W.M.B., 1985. An experimental investigation of heat transfer in forced convection condensation of oil-refrigerant mixtures. ASHRAE Transactions, 297-309.
[31] Wang, H. S., Honda, H., 2003. Condensation of refrigerants in horizontal micro-fin tubes: Comparison of prediction methods for heat transfer. International Journal of Refrigeration, 26 (2003), 452–460.
[32] Wang, J.C., Kalamchi, A., and Fazio, P., 1985. Experimental study on condensation of refrigerant-oil mixtures: part I- design of the test apparatus. ASHRAE Transactions, 216-237.
[33] Yiong, J., and Srinivas, G., 2003. Heat transfer and pressure drop for condensation of refrigerant R-404A at near critical pressures. ASHRAE Transactions, 677-686.

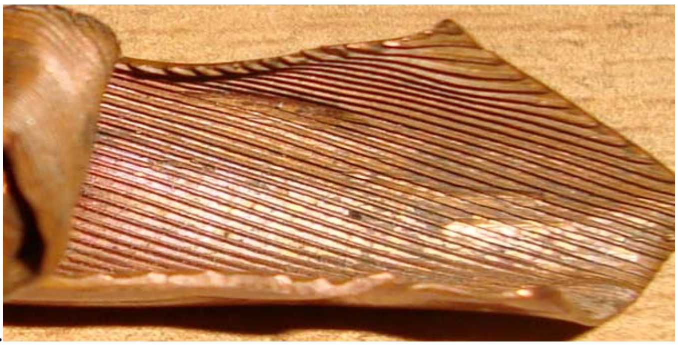

Fig.17 Photograph of micro-fin tube

Fig.18 Photograph of test facility