บทความวิจัย

Boiling heat transfer coefficient of R-12 alternatives in horizontal tube : small refrigerator scale

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Abstract

Boiling heat transfer coefficient and pressure drop of R-12 and its alternatives which are R-134a and R-22/R-152a/R-124 (33/15/52% by mass) flowing inside smooth and grooved horizontal tubes have been carried out with the conditions similar to those in small refrigerators. The range of mass flow rates examined is between 0.0025 and 0.0125 kg/s. The data have been taken at the evaporator temperatures of -20 to $3°C$ and at the condenser temperatures of $40-50°C$. The boiling heat transfer coefficients of the refrigerants in the grooved tubes are found to be higher than those in the smooth tubes. The heat exchange correlations have also been developed. In case of the pressure drop, the two-phase friction multiplier ϕ_G^2 increases linearly with Martinelli parameter and there is no effect of the tube roughness and the types of the refrigerants.

Introduction

Nowadays, refrigeration and air-conditioning industries are still largely based on chlorofluorocarbon (CFCs) and hydrochloro-fluorocarbon (HCFC) refrigerants due to their advantageous thermo-dynamic and chemical characteristics. Both these groups of refrigerants are now classified as “Ozone Depleting Substances” and “Controlled Substances” under “The Montreal Protocol on Substances that Deplete the Ozone Layer”.

Since the Montreal Protocol will ban CFCs and HCFC by the year 1996 and 2020, respectively. This protocal calls for accelerated development of new alternative refrigerants. In recent years, numbers of new refrigerants have been proposed to replace CFCs, either as pure or as constituents of mixtures, for example R-134a and non-azeotopic refrigerant mixtures (NARMs) whose ozone potential depletion is very small.

At present, R-134a has been used as an alternative to R-12 in automobile air-conditioning and domestic refrigerators. However, many components potentially require including the compressor and lubricant replacement , therefore, the cost of replacement is rather expensive (Kiatsiriroat *et al.*, 1997 and Jung *et al.*, 1999). As a solution to this problem, a NARM blend with the commercial name, MP-52 (R-22, R-152a and R-124 with 33%, 15% and 52% mass fractions) has been proposed. This particular refrigerant is quite beneficial because it mimics the properties of R-12 and does not require a major overhaul.

Research of the boiling heat transfer characteristics and pressure drop of NARMs is still in its infancy. Recently, Sami (1994) and Changyam (1997) investigated boiling and condensation heat transfer coefficients of R-22/R-152a/R-124 in smooth tube compared with R-12. However, very few informations of these refrigerants have been published especially the flow in grooved tube, especially for small refrigerator scale.

In this study, the aim of the project is to test new and environmentally friendly refrigerants. The refrigerants of interest are HFC-134a and MP-52. This paper concerns horizontal flow boiling inside smooth and grooved tubes with the emphasis on heat transfer and pressure drop.

Experimental set-up

The experimental setup for studying of boiling heat transfer coefficients of refrigerants in horizontal tube is illustrated in Figure 1.The refrigeration test rig consists of 3 loops, namely, the refrigeration loop, the heating water loop and the cooling water loop. The test section is double pipe in which the inner tube contains the refrigerant and the annulus space has water flow for heating (evaporator section). The condenser of the refrigeration cycle is similar to the evaporator and is water-cooled.

Refrigeration loop

The refrigeration loop is consisted of an evaporator, a condenser, a compressor, an accumulator, a receiver, a drier and an expansion valve. The evaporator is a double pipe of which the inner tubes are smooth and grooved tubes each is made of copper with outside diameter of 9.52-mm (3/8 inch.) and inside diameter of 7.92-mm. The outer tube is also made of copper with outside diameter of 19-mm (6/8 inch.) and inside diameter of 17-mm as shown in Figure 2. The test section is 1.20 m long from the evaporator length of 9 m. Temperatures are measured by a set of thermocouples and recorded by a data logger at steady state conditions. The refrigerants tested are R-12, R-134a and MP-52 at the mass flow rates of 0.0025 - 0.0125 kg/s, which is applicable in small cooler unit.

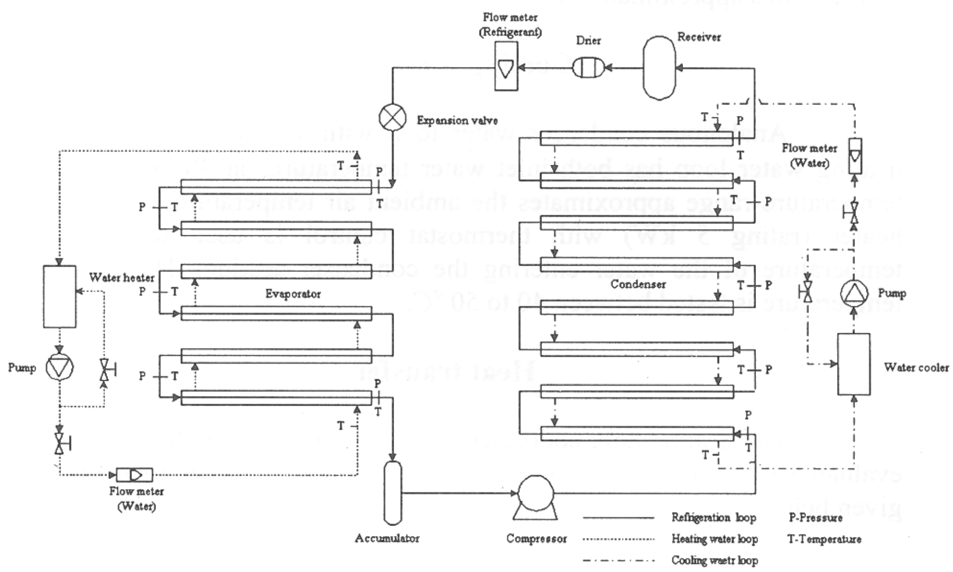

Figure 1 : Experimental setup

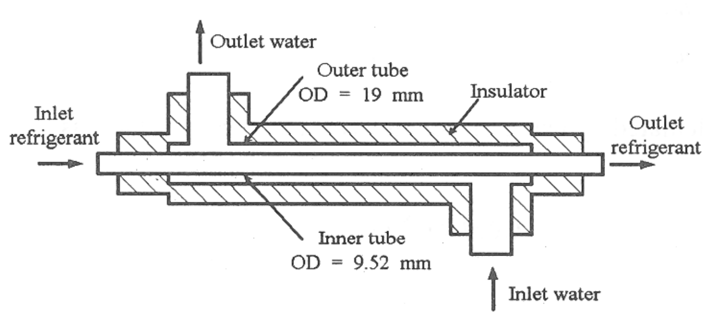

Figure 2 : Detail of the test section (double pipes heat exchanger)

Heating water loop

The evaporator water loop is consisted of a centrifugal pump (0.5 hp), a water storage tank (60 litres) and cooling system (1/3 TR). The water flow could be maintained at 0.05 kg/s and temperatures of 5, 10 and 15°C. For the condition at 5°C, the water contains 28% ethylene glycol to prevent freezing. The refrigerant temperature is controlled between –20 to 3°C. These temperatures approximate the conditions as those in small refrigerator.

Cooling water loop

Analogous condenser water loop with similar components as the heating water loop has both inlet water temperatures at 30 and 40°C. This temperature range approximates the ambient air temperature in Thailand. A heater (rating 5 kW) with thermostat control is used to control the temperature of the water entering the condenser section. The condensing temperature is tested between 40 to 50 °C.

Heat transfer

The heat transfer coefficients of the evaporator, in this study, can be evaluated from conduction and forced convection in heat transfer theory as given below.

$$\frac{1}{h_r A_i} = \frac{1}{U_o A_o} - \frac{1}{h_w A_o} - \frac{ln(r_o / r_i)}{2\pi kL} \tag{1}$$

where h_r = boiling heat transfer coefficient (W/m^2 °C)

U_o = overall heat transfer coefficient (W/m^2 °C)

$ln(r_o / r_i) / 2\pi kL$ = wall resistance

h_w = water heat transfer coefficient (W/m^2 °C)

h_w is calculated by using Sieder and Tate (Incropera, 1990) as

$$h_w = 0.027\, Re^{0.8}\, Pr^{1/3}\, k \frac{D_{o,in}^2}{D_{i,out}^2 - D_{o,in}^2} \tag{2}$$

The heat transfer rate at the test section can be determined from the heat balance of the water flow in the annulus and could be estimated by

$$\dot{Q} = \dot{m}_w C_p \left(T_{w,in} - T_{w,out}\right) \tag{3}$$

where $\dot{Q}$ = heat transfer rate (W)
$\dot{m}_w$ = mass flow rate of water (kg/s)
C_p = specific heat of water (kJ/kg °C)
$T_{w,in}$ and $T_{w,out}$ = inlet and outlet water temperature (°C)

The heat transfer rate could also be estimated by

$$\dot{Q} = U_o A_o \Delta T_{LMTD} \tag{4}$$

where A_o = outside surface area of the tube (m^2)

Then the overall thermal resistance $1/U_o A_o$ could be

$$\frac{1}{U_o A_o} = \frac{\Delta T_{LMTD}}{\dot{Q}} \tag{5}$$

where ΔT_{LMTD} = log-mean temperature difference between the two fluids streams (°C)

and ΔT_{LMTD} is defined as

$$\Delta T_{LMTD} = \frac{\left(T_{h,in} - T_{c,out}\right) - \left(T_{h,out} - T_{c,in}\right)}{ln\left(\dfrac{T_{h,in} - T_{c,out}}{T_{h,out} - T_{c,in}}\right)} \tag{6}$$

The calculaion is carried out when the refrigerant is in two-phase condition. T_h and T_c mean the hot stream and the cold stream temperatures, respectively.

With the data of the inlet and outlet of the fluids exchanging heat, the water mass flow rate, the wall resistance then the value of h_i could be evaluated.

Pressure drop

The pressure drop is analyzed by using the concept of two-phase pressure drop in separated-flow model developed by Hewitt *et al.* (1994). In the horizontal tube, the accelerational and gravitational pressure drop can be neglected, therefore, only the frictional two-phase pressure drop is considered.

Two-phase pressure drop is convenient to relate the frictional pressure gradient for the gas phase or liquid phase flowing alone in the channel, in term of multiplier ϕ_G and ϕ_L, which could be defined as:

$$\phi_G^2 = \frac{dP_F / dz}{(dP_F / dz)_G} \tag{7}$$

and

$$\phi_L^2 = \frac{dP_F / dz}{(dP_F / dz)_L} \tag{8}$$

where dP_F / dz = frictional two-phase pressure drop (N/m^2)
$(dP_F / dz)_G$ and $(dP_F / dz)_L$ = pressure gradients for gas and liquid phases flowing alone in the tube (N/m^2)

The pressure drops are defined as:

$$\left(\frac{dP_F}{dz}\right)_G = \frac{2 f_G G^2 X^2}{D_i \rho_G} \tag{9}$$

and

$$\left(\frac{dP_F}{dz}\right)_L = \frac{2 f_L G^2 (1-X)^2}{D_i \rho_L} \tag{10}$$

where f_G and f_L = gas and liquid friction factor
G = mass flux (kg/m^2 s)
X = quality
D_i = inside diameter of the tube (m)
ρ_G and ρ_L = gas and liquid densities (kg/m^3)

f_G and f_L are related to the respectively Reynolds numbers that are defined as follows:

$$Re_G = \frac{GXD_i}{\mu_G} \tag{11}$$

and

$$Re_L = \frac{G(1-X)D_i}{\mu_L} \tag{12}$$

where Re_G and Re_L = Reynolds numbers in gas and liquid phase
G = mass flux (kg/m^2 s)
X = quality
D_i = inside diameter of the tube (m)
μ_G and μ_L = gas and liquid phase viscosity (Pa-s)

For laminar flow $(Re < 2000)$, $f = 16/Re$; for turbulent flow $(Re > 2000)$, the Blasius equation, $f = 0.079(GD_i/\mu)^{-0.25}$, is often used. The parameter often uses in the correlation of two-phase pressure drop is Martinelli parameter, X_{tt}, where

$$X_{tt}^2 = \frac{(dP_F/dz)_L}{(dP_F/dz)_G} \tag{13}$$

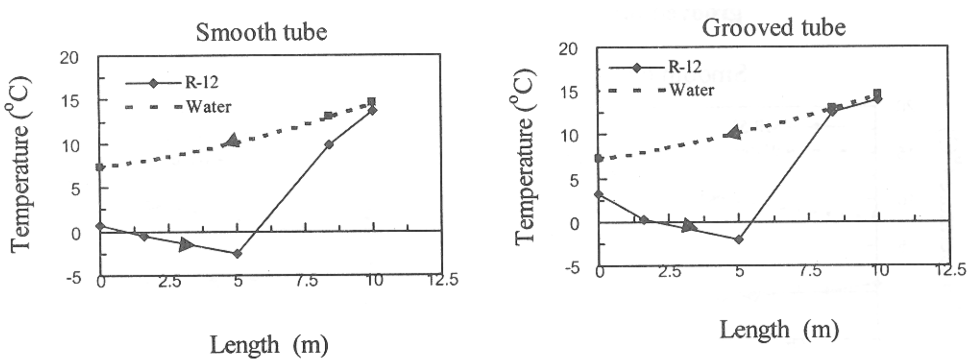

Figure 3 : Evaporator temperature profiles of R-12 in smooth and grooved tubes

Results and discussions

Figures 3-5 show the temperature profiles of R-12, R-134a and MP-52 in the evaporator at 15°C of inlet water temperature. It is found that there is a pressure drop during boiling because of friction in the tube. The refrigerant temperature decreases from the inlet until saturation condition is achieved and after that the temperatures slightly increase because of the pressure change along the rest of the tube length in each refrigerant. For the grooved tube, the temperature decreases more than that in the smooth tube.

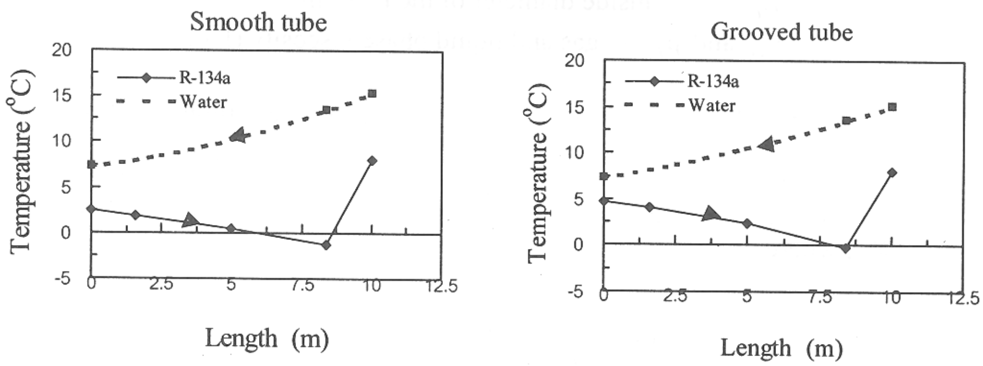

Figure 4 : Evaporator temperature profiles of R-134a in smooth and grooved tubes

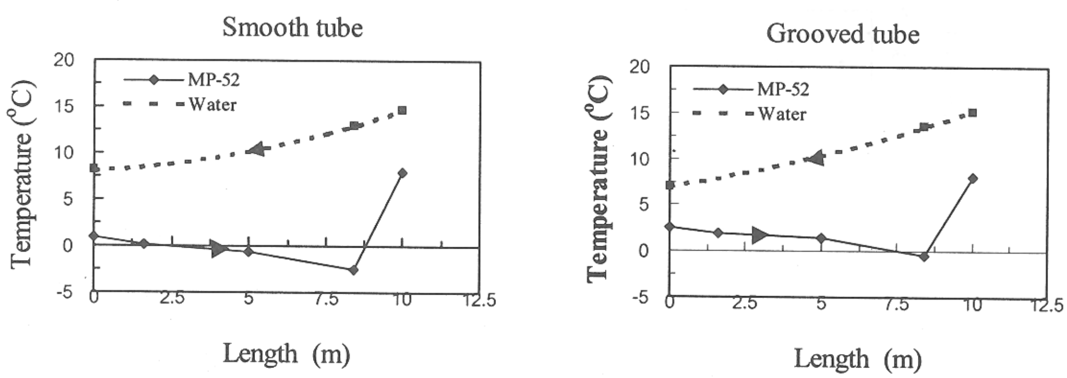

Figure 5 : Evaporator temperature profiles of MP-52 in smooth and grooved tubes

Boiling heat transfer coefficient results for R-12, R-134a and MP-52 in smooth and grooved tubes are plotted in Figure 6 as a function of Reynolds number. The Figure shows that the heat transfer coefficient increases with Reynolds numbers. The Reynolds number is in the range of 3500 – 13000. In each refrigerant, boiling heat transfer coefficients of the grooved tube is higher than the smooth tube because it has more inside surface area and higher turbulence is obtained. In smooth tube, the evaporative heat transfer coefficients of MP-52 is the lowest because the component of high boiling point remains in the evaporator. For R-12 and R-134a, the results are nearly the same value. For the grooved tube, boiling heat transfer coefficient of R-134a is the highest, following by R-12 and MP-52, respectively. Therefore, R-134a is suitable to replace R-12 in both smooth and grooved tubes because of its performance is similar or better than R-12.

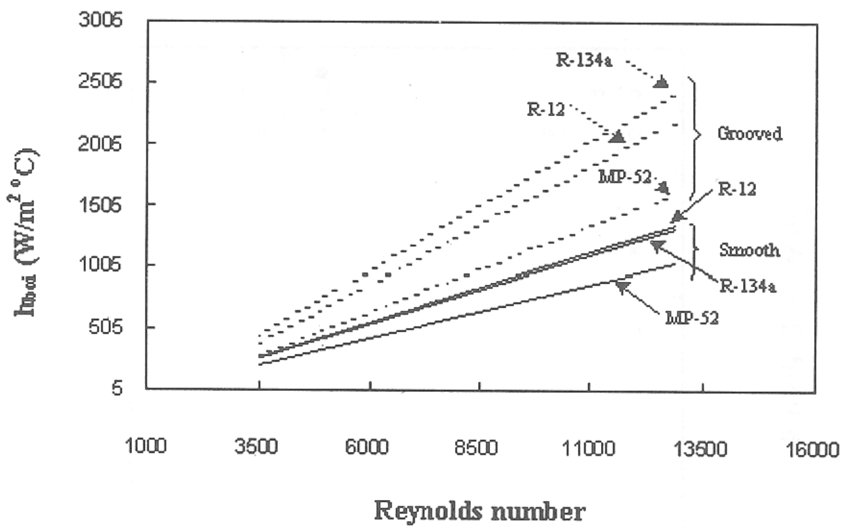

Figure 6 : Boiling heat transfer coefficient of R-12 alternative refrigerants in smooth and grooved tubes

The boiling heat transfer coefficient and pressure drop data have been taken. The correlation equations of boiling heat transfer coefficients for R-12, R-134a and MP-52 in smooth and grooved tubes are modified from Chaddock and Brunemann (1967) equation. The equation is

$$h_{TP} = ah_{lo}\left[\left(Bo\times 10^{4}\right)+1.5\left(\frac{1}{X_{tt}}\right)^{0.67}\right]^{b}$$

where $h_{lo} = 0.023(k_l / D_i)(GD_i / \mu_l)^{0.8} Pr^{0.4}$. The empirical constants a and b are shown in Table 1. Figure 7 shows the present study heat transfer coefficient for smooth tube for R-12 compared to those evaluated from the correlations of Pierre (1955), Lavin and Young (1965), Chaddock-Brunemann (1967), Kandlikar (1987), Shah (1982) and Changyam (1997) as shown in Table 3 in the appendix . It could be seen that for small-scale refrigerator, the previous correlations give a high deviation in evaluating the heat transfer coefficient.

Figures 8-10 illustrate comparison of the results from the correlation and the measured data for R-12, R-134a and MP-52. The results agree well with the experiments within ±15% deviation.

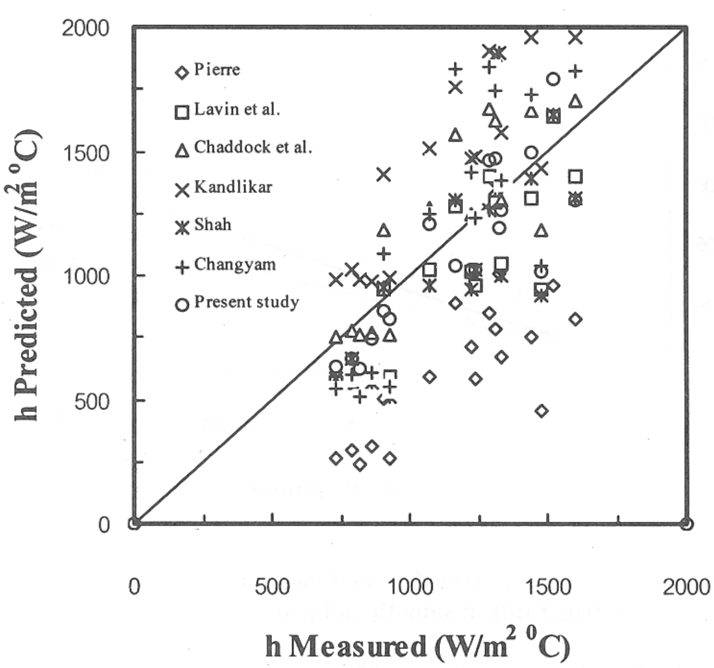

Figure 7 : Comparison of measured boiling heat transfer coefficients of R-12 in smooth tube with other correlations

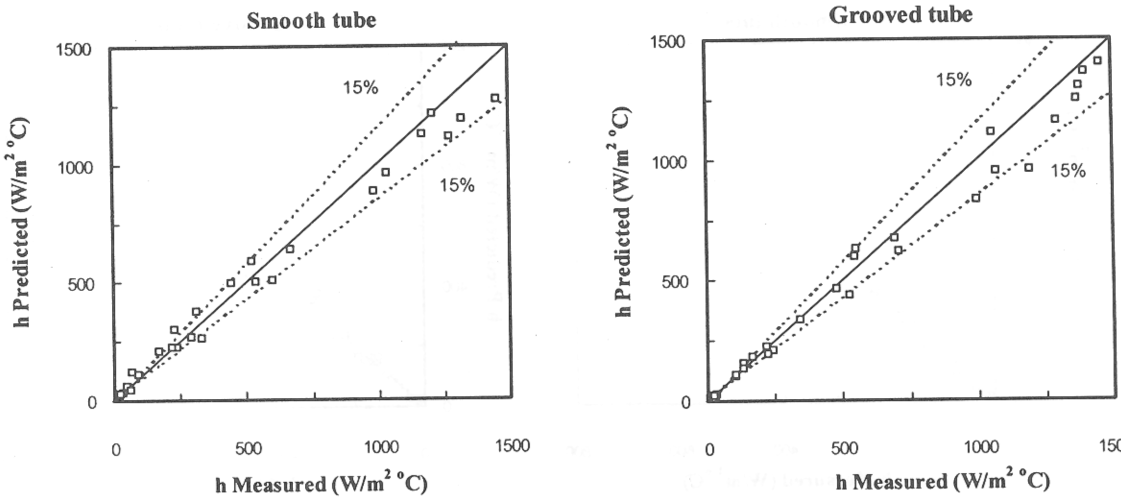

Figure 8 : Comparison of measured boiling heat transfer coefficients of R-12 in smooth and grooved tubes with the present study

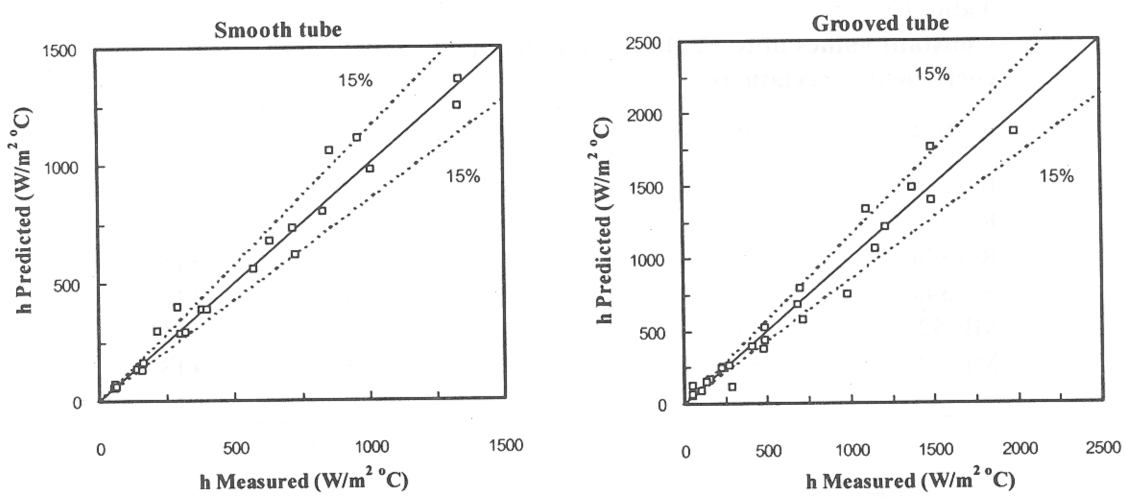

Figure 9 : Comparison of measured boiling heat transfer coefficients of R-134a in smooth and grooved tubes with the present study

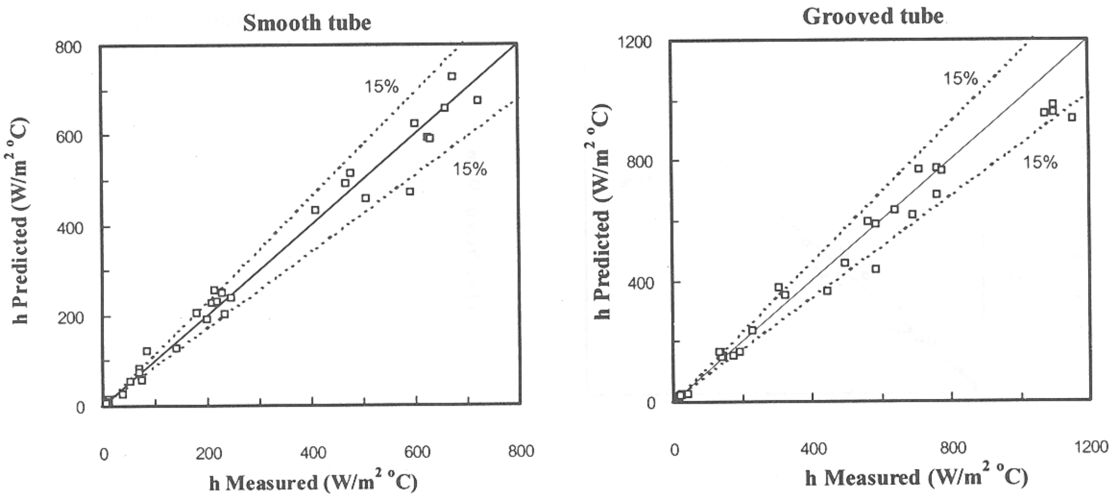

Figure 10 : Comparison of measured boiling heat transfer coefficients of MP-52 in smooth and grooved tubes with the present study

Table 1 :
Constant values of R-12 alternatives boiling heat transfer coefficient correlations

Refrigerants	Tube type	a	b	% range of data
R-12	smooth	2.20	0.51	±15
R-12	grooved	1.99	0.57	±15
R-134a	smooth	1.40	0.82	±15
R-134a	grooved	1.70	0.74	±15
MP-52	smooth	2.10	0.55	±15
MP-52	grooved	1.50	0.75	±15

Pressure drop of R-12, R-134a and MP-52 in smooth and grooved tubes of the evaporator are presented in Figure 11 as a function of two-phase friction multiplier for vapor flowing alone $\left(\phi_G^2\right)$ and Martinelli parameter (X_{tt}). Chisholm proposed the following relation for a smooth tube: $\phi_G^2 = 1+CX_{tt}+X_{tt}^2$, (Hewitt, 1994). For smooth tubes, the constant C ranges from 5 to 20, depending on whether the liquid and vapor phase are laminar or turbulent. For smooth and grooved tubes of all refrigerants in the experiment, the parameter X_{tt} is formed to be in a range of 0.1 – 0.45. ϕ_G^2 increases with the parameter X_{tt} and could be simplified by $\phi_G^2 = a + bX_{tt}$ within the range of 3 - 10. The constants a and b are illustrated in Table 2.

Table 2 : Constant values of R-12 alternatives pressure drop correlations

Refrigerants type	Tube	a	b	Mean Deviation
R-12	smooth	20.539	0.9336	±0.13%
R-12	grooved	20.518	0.9373	±0.14%
R-134a	smooth	20.291	1.0045	±0.48%
R-134a	grooved	20.291	1.0045	±0.48%
MP-52	smooth	21.255	0.7390	±1.92%
MP-52	grooved	20.419	0.9602	±0.09%

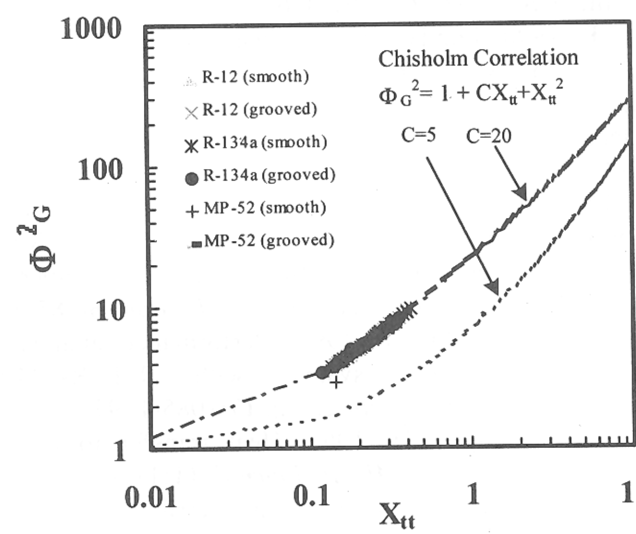

Figure 11 : Pressure drops of R-12 alternative refrigerants in smooth and grooved tubes

Conclusion

This study shows the boiling heat transfer coefficient of R-12, R-134a and MP-52 in smooth and grooved tubes. The boiling heat transfer coefficients of the grooved tube is higher than those of the smooth tube because of its bigger surface area and higher turbulence. For the grooved tube, R-134a shows the highest heat transfer coefficient following by R-12 and MP-52, respectively. For the smooth tube, boiling heat transfer coefficients of R-12 and R-134a are nearly the same and MP-52 is the lowest because the component of higher boiling point remains in the evaporator. Pressure drops at the evaporator in smooth and grooved tubes are nearly the same for all types of the refrigerants.

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Appendix 3

Table 3 : Boiling heat transfer correlations as applicable to experimental data of R-12 in horizontal smooth tube

Source	Correlation
Pierre (1955, 1957)	$h = C_1 \left(\frac{k_l}{D_i}\right)\left[\left(\frac{GD_i}{\mu_l}\right)^2 \left(\frac{J\Delta X h_{fg}}{L}\right)\right]^n$ where $C_1 = 0.0009$ and $n = 0.5$ for exit equalities $\leq 90\%$ and $C_1 = 0.0082$ and $n = 0.4$ for 6 K superheat at exit
Lavin and Young (1964)	$h = 6.59 h_l \left(\frac{1+X}{1-X}\right)^{1.16} \left(\frac{q}{Gh_{fg}}\right)^{0.1}$
Chaddock and Brunemann (1967)	$h_{TP} = 1.91 h_{l0} \left[Bo \times 10^4 + 1.5\left(\frac{1}{X_{tt}}\right)^{0.67}\right]^{0.6}$
Kandlikar (1987)	$\frac{h_{TP}}{h_l} = C1 Co^{C2} (25 Fr_l)^{C5} + C3 Bo^{C4} F_{fl}$ $Co = \left(\frac{1-X}{X}\right)^{0.8} \left(\frac{\rho_g}{\rho_l}\right)^{0.5}$ For $Co < 0.5$ and $C1 = 1.1360$, $C2 = -0.9$, $C3 = 1058$, $C4 = 0.7$ and $C5 = 0.3$. If the tube is horizontal, then $C5 = 0$ for all cases.

Table 3 : (continue)

Source	Correlation
Shah (1982)	$\frac{h_{TP}}{h_l} = \psi$ ψ can be determined from the largest value of ψ_{nb}, ψ_{cb} or ψ_{bs}, for horizontal tube: $N = Co$ For horizontal tubes with $Fr_l \le 0.04$ then, $N = 0.38 Fr_l^{-0.3} Co$ $Fr_l = \frac{G^2}{\rho_l^2 g D_i}$ $\psi_{nb} = 1 + 46 Bo^{0.5}$ for $Bo < 0.3\times10^{-4}$ $\psi_{nb} = 230 Bo^{0.5}$ for $Bo > 0.3\times10^{-4}$ $\psi_{cb} = \frac{1.8}{N^{0.8}}$ for $N > 1.0$ $\psi_{bs} = F Bo^{0.5} e^{2.74 N^{-0.1}}$ for $0.1 < N < 1.0$ $\psi_{bs} = F Bo^{0.5} e^{2.74 N^{-0.15}}$ for $N \le 0.1$ for $Bo \ge 11\times10^{-4}$ then $F = 14.7$ for $Bo \le 11\times10^{-4}$ then $F = 15.43$
Changyam (1997)	$Nu = 0.000843\, Re^{1.046203}\, K_f^{0.3}$ $K_f = \frac{(Xo - Xi) h_{fg}}{Lg}$

In the above corrrelations the following definitions are used : Dittus-Boelter correlation for single-phase transfer:

$$h_l = \frac{0.023 k_l}{D_i}\left[\frac{D_i G(1-x)}{\mu_l}\right]^{0.8} (Pr)_l^{0.4} \qquad h_{lo} = \frac{0.023 k_l}{D_i}\left[\frac{D_i G}{\mu_l}\right]^{0.8} (Pr)_l^{0.4}$$