ORIGINAL ARTICLE

Boiling heat transfer coefficient of R-22 refrigerant and its alternatives in horizontal tube : small refrigerator scale

Uthen Kuntha [1] and Tanongkiat Kiatsiriroat [2]

Abstract

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Boiling heat transfer coefficients and pressure drop of R-22 and its alternatives, which are R32/R125a/R134 (23%/25%/52%) and R32/R125a (50%/50%), flowing inside smooth and grooved tubes have been determined 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 -35 to -4 ºC and at the condenser temperatures of 40 to 50 ºC. The boiling heat transfer coefficients of the refrigerants in the grooved tubes are higher than those in the smooth tubes and R-22 shows the best performance for both tubes. The heat transfer correlations have also been developed. In case of the pressure drop, the two-phase friction multiplier ϕ_G^2 increases with Martinelli parameter and there is no effect of the tube roughness and the types of the refrigerants.

Key words : boiling, pressure drop, R-22 alternatives

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บทคัดย่อ

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การศึกษาค่าสัมประสิทธิ์การถ่ายเทความร้อนของการเดือดของสารทำความเย็น R-22 และสารทำความเย็นทดแทนในท่อแนวนอน : ระดับตู้เย็นขนาดเล็ก

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งานวิจัยนี้ได้ทำการศึกษาค่าสัมประสิทธิ์การถ่ายเทความร้อนของการเดือดของสารทำความเย็นที่ใช้ทดแทน R-22 ซึ่งได้แก่ R32/R125a/R134 (23%/25%/52%) และ R32/R125a (50%/50%) ในท่อเรียบและท่อมีร่องในแนวนอนภายใต้สภาวะตู้เย็นขนาดเล็ก โดยปรับอัตราการไหลของสารทำความเย็นให้อยู่ในช่วง 0.0025 - 0.0125 kg/s ที่อุณหภูมิอีวาปอเรเตอร์ -35 ถึง -4 °C และอุณหภูมิคอนเด็นเซอร์ 40-50 °C ผลการทดลองพบว่าค่าสัมประสิทธิ์การถ่ายเทความร้อนของการเดือดของสารทำความเย็นในท่อมีร่องจะมีค่าสูงกว่าท่อเรียบและพบว่า R-22 มีค่าสัมประสิทธิ์การถ่ายเทความร้อนของการเดือดสูงสุดในท่อทั้งสองชนิดและได้ทำการหาสมการความสัมพันธ์เพื่อทำนายค่าสัมประสิทธิ์การถ่ายเทความร้อนของการเดือดของสารทำความเย็นดังกล่าวด้วย สำหรับกรณีของค่าความดันลดพบว่าค่าตัวคูณความเสียดทานของของไหลเมื่ออยู่ในสถานะแก๊ส (ϕ_G^2) มีค่าเพิ่มขึ้นตามพารามิเตอร์มาร์ติเนลลี และพบว่าค่าความขรุขระของท่อกับชนิดของสารทำความเย็นไม่มีผลต่อค่าความดันลด

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At present, refrigeration and air-conditioning industries are still largely based on chlorofluorocarbon (CFCs) and hydrochlorofluorocarbon (HCFC) refrigerants due to their advantageous thermodynamic 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 protocol calls for accelerated development of new alternative refrigerants. In recent years, NARM blends with the commercial names, SUVA-9000 (R-32, R-125a and R-134 at 23%, 25% and 52%) and SUVA-9100 (R-32, R125a at 50% and 50%) have been proposed. These particular refrigerants are quite beneficial because these mimic the properties of R-22 and do not require a major overhaul.

Research of the boiling heat transfer characteristics and pressure drop of NARMs is still in its infancy. Recently, some researchers investigated boiling heat transfer coefficients of R-22 and SUVA-9000 in smooth and microfin tubes. However, little informations on these refrigerants has been published, particularity the flow in grooved tube, especially for small refrigerator scale.

In this study, the aim of the project is to test the new and environmentally friendly refrigerants. The refrigerants of interest are HCFC-22, SUVA-9000 and SUVA-9100. This paper concerns horizontal flow boiling inside smooth and grooved tubes for the scale of small refrigerator with the emphasis on the heat transfer and the 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 a 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.

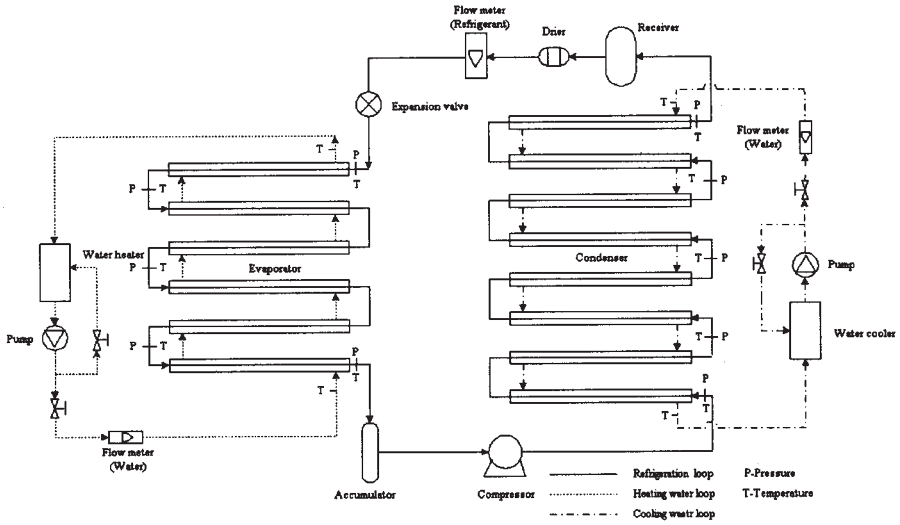

Figure 1. Experimental setup

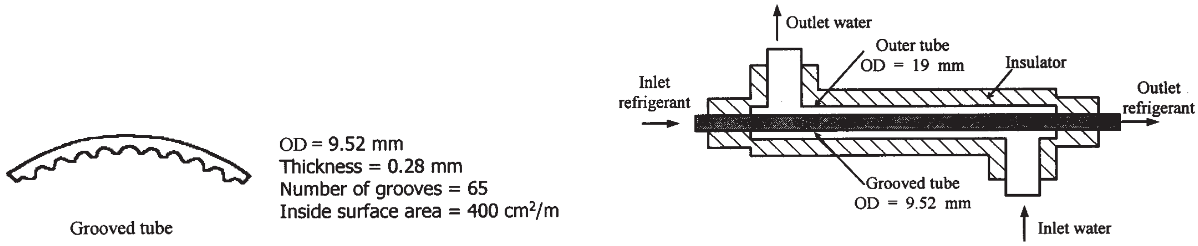

Figure 2. Detail of the test section (double pipes heat exchanger)

Refrigeration loop

The refrigeration loop consists 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 and 8.52-mm, respectively. The outer tube is also made of copper with the outside diameter of 19-mm and the 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-22, SUVA-9000 and SUVA-9100 at the mass flow rate of 0.0025 - 0.0125 kg/s, which is applicable in small cooler units.

Heating water loop

The evaporator water loop consists of a centrifugal pump (0.5 hp), a water storage tank (60 litres) and a cooling system (1/3 TR). The water flow could be maintained at 0.05 kg/s and the temperatures at 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 -35 to -4°C. These temperatures approximate the conditions of a small refrigerator.

Cooling water loop

Analogous condenser water loop with similar components to 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.

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

Where h_r represents the boiling heat transfer coefficient (W/m^2 °C), U_o is the overall heat transfer coefficient (W/m^2 °C), $ln(r_o/r_i)/2\pi kL$ denotes wall resistance and h_w is the water heat transfer coefficient (W/m^2 °C), that is modified from Sieder and Tate correlation (Incropera, 1990) by assuming that the fluid flow is fully developed and closed to that in the straight line.

$$h_w = 0.027\ Re^{0.8} Pr^{1/3} k\, D^2_{o,\,in} / (D^2_{i,\,out} - D^2_{o,\,in}), \quad (2)$$

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

$$\dot{Q} = \dot{m} C_p (T_{w,\,in} - T_{w,\,out}), \quad (3)$$

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

The heat transfer rate could also be estimated by

$$\dot{Q} = U_o A_o \Delta T_{LMTD} , \quad (4)$$

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

Then the overall thermal resistance $1/U_oA_o$ could be

$$\frac{1}{U_o A_o} = \frac{\Delta T_{LMTD}}{\dot{Q}} , \quad (5)$$

ΔT_{LMTD} is the log-mean temperature difference between the two fluid streams and is defined as

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

The calculation is carried out when the refrigerant is in a 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 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 terms of multipliers ϕ_G and ϕ_L, which could be defined as:

$$\phi^2_G = \frac{dP_F / dz}{(dP_F / dz)_G}, \quad (7)$$

and

$$\phi^2_L = \frac{dP_F / dz}{(dP_F / dz)_L} , \quad (8)$$

where dP_F/dz is the frictional two-phase pressure drop (N/m^2), and $(dP_F/dz)_G$ and $(dP_F/dz)_L$ are the pressure gradients for gas and liquid phases flowing alone in the tube (N/m^2), respectively. The pressure drops are defined as:

$$\left(\frac{dP_F}{dz}\right)_L = \frac{2f_L G^2(1-X)^2}{D_i \rho_L}, \tag{9}$$

$$\left(\frac{dP_F}{dz}\right)_G = \frac{2f_G G^2 X^2}{D_i \rho_G}, \tag{10}$$

G is the mass flux (kg/m^2 s), X is the quality, D_i is the inside diameter of the tube (m), and ρ_G and ρ_L are the gas and liquid densities based on inlet refrigerant temperature (kg/m^3). The friction factors f_G and f_L are related to the respective Reynolds numbers, 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 D_i is the inside diameter of the tube (m), μ_G is the gas-phase viscosity, μ_L is liquid-phase viscosity (Pa-s), and X is the quality. 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 a term of two-phase pressure drop calls Martinelli parameter, X_{tt}, where

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

Results and discussion

Temperature profiles

Figures 3-5 show the temperature profiles of R-22, SUVA-9000 and SUVA-9100 in the

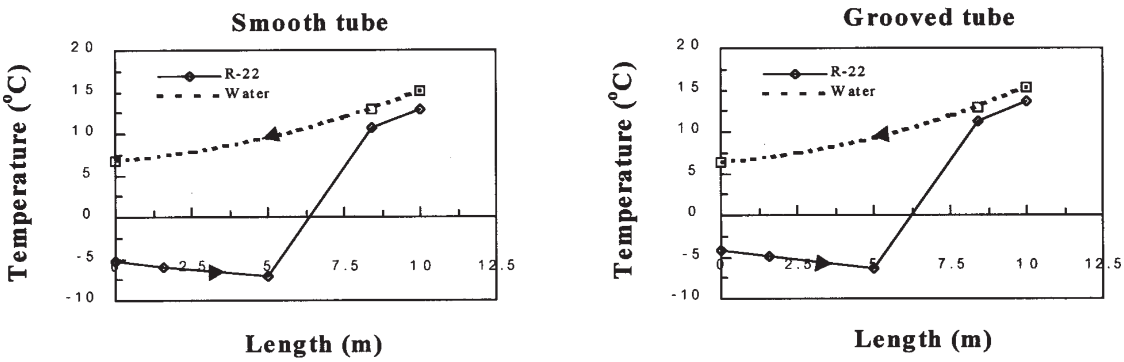

Figure 3. Evaporator temperature profiles of R-22 in smooth and grooved tubes

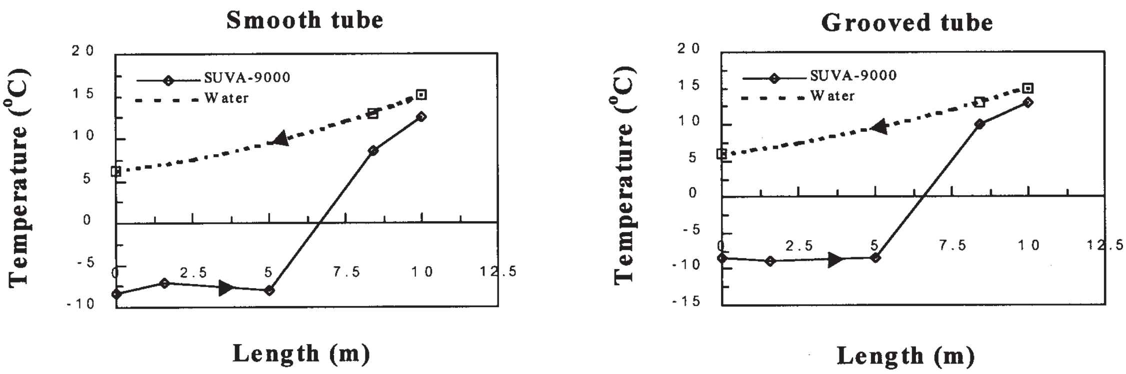

Figure 4. Evaporator temperature profiles of SUVA-9000 in smooth and grooved tubes

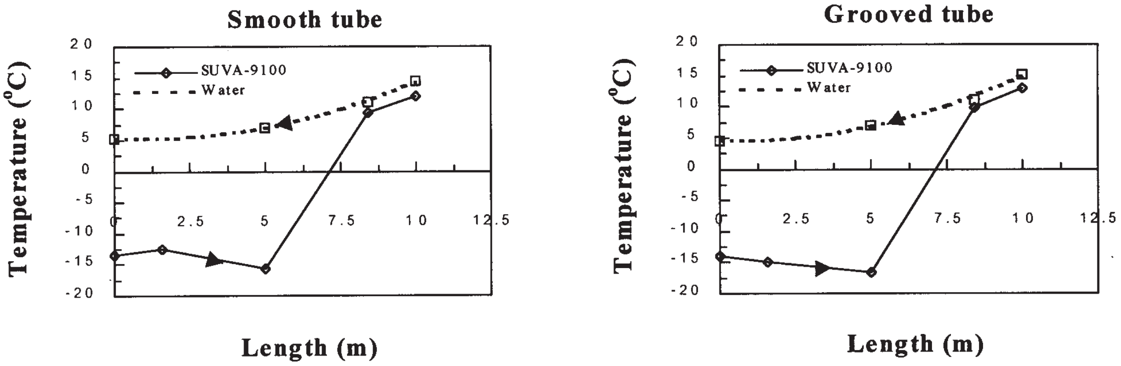

Figure 5. Evaporator temperature profiles of SUVA-9100 in smooth and grooved tubes

evaporator. It is found that a pressure drop occurs during boiling because of friction in the tube. The refrigerant temperature still decreases from the inlet until the saturated vapor point and there is a temperature increase (superheat vapor phase) along the remained tube length. R-22, SUVA-9000 and SUVA-9100 have decreased the heating water temperature about 10 °C. In each refrigerant, the water temperature in the grooved tube decreases more than that in the smooth tube because of its higher turbulance.

Boiling heat transfer coefficient of horizontal tube

Boiling heat transfer coefficient results for R-22, SUVA-9000 and SUVA-9100 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 number. The Reynolds number of all correlations is in the range of 1000 - 16000 and boiling heat transfer coefficients are in the range of 400 - 1100 W/m^2 °C. In each Reynolds number, boiling heat transfer coefficients of the grooved tube are higher than those of the smooth tube because of the higher turbulence obtained. In the smooth tube, the boiling heat transfer coefficients of R-22 are the highest, following by SUVA-9000 and SUVA-9100, respectively. In the grooved tube, the results of boiling heat transfer coefficients are similar to those in the smooth tube.

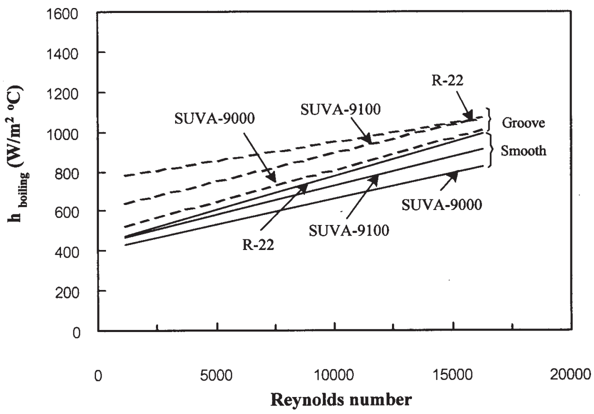

Figure 6. Boiling heat transfer coefficient of R-22 and its alternatives in smooth and grooved tubes

The boiling heat transfer coefficient and pressure drop data have been taken. The present study, correlation equations of boiling heat transfer coefficients for R-22, SUVA-9000 and SUVA-9100 in smooth and grooved tubes are

$$h_{TP} = h_{lo}\, a\, (1/Xtt)^b,$$

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 of R-22 compared to that evaluated from the correlations of Lavin and Young (1965), Chaddock-Brunemann (1967), Shah

Table 1. Constant values of R-22 alternatives boiling heat transfer coefficient correlations

Refrigerants	Tube type	a	b	% range of data
R-22	smooth	0.020	2.00	±20
R-22	grooved	0.023	1.99	±30
SUVA-9000	smooth	0.050	1.54	±30
SUVA-9000	grooved	0.025	1.70	±30
SUVA-9100	smooth	0.027	1.62	±20
SUVA-9100	grooved	0.030	1.74	±25

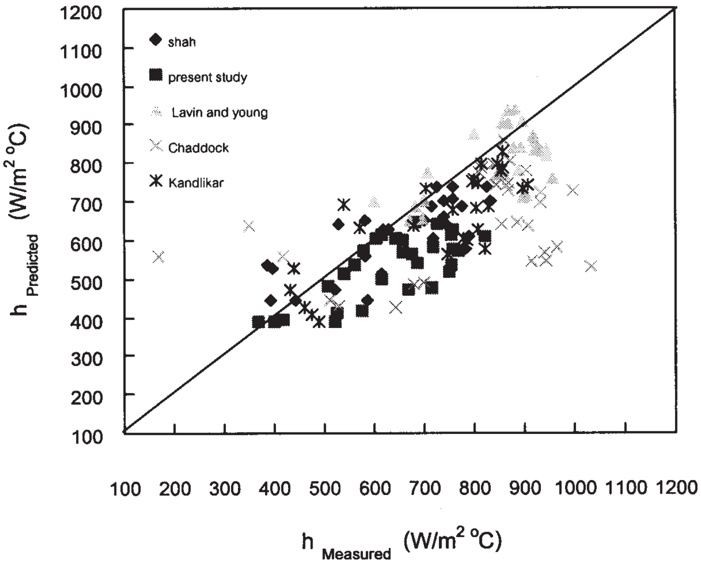

Figure 7. Comparison of measured boiling heat transfer coefficients of R-22 in smooth tube with other correlations

(1982) and Kandlikar (1987) as shown in Table 2. It could be seen that the present study boiling heat transfer coefficient is nearly the same as these correlations.

Figures 8-10 illustrate comparison of the correlations and the measured data for R-22, SUVA-9000 and SUVA-9100. The results agree well both smooth and grooved tubes with those of the experiments within 20 - 30% deviations.

Pressure drop

Pressure drops of R-22, SUVA-9000 and SUVA-9100 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 (ϕ_G^2) and Martinelli parameter (X_{tt}) . Chisholm (1967) proposed the following relation for a smooth tube: $\phi_G^2 = 1+CX_{tt}+X_{tt}^2$. For smooth tubes, the constant C ranges from 5 to 20, depending on whether the liquid and vapor phases are laminar or turbulent. Wang *et al.* (1998) proposed the correlation of R-22 as a function of two-phase friction multiplier for vapor flowing alone (ϕ_G^2) and Martinelli parameter (X_{tt}) . The correlation is $\phi_G^2 = 1+9.73X_{tt}^{0.65} + 0.487X_{tt}^{2.5}$. In smooth and grooved tubes for all refrigerants, the parameter X_{tt} increases from 0.1 -

Table 2. Boiling heat transfer correlations as applicable to experimental data of R-22 in horizontal smooth tube

Source	Correlation
Lavin and Young (1965)	$h = 6.59 h_l \left(\frac{1+X}{1-X}\right)^{1.16} \left(\frac{q}{G\,h_{fg}}\right)^{0.1}$
Chaddock and Brunemann (1967)	$h_{TP} = 1.91\, h_{lo}\left[Bo \times 10^4 + 1.5\left(\frac{1}{X_{tt}}\right)^{0.67}\right]^{0.6}$
Shah (1982)	$\frac{h_{TP}}{hl} = \psi$

ψ can be determined from the largest value of ψ_{nb}, ψ_{cb}, or ψ_{bs}, For horizontal tubes: $N = Co$

For horizontal tubes with $Fr_L \leq 0.04$ then, $N = 0.38 Fr_l^{-0.3} Co$

For N>1.0	$\psi_{nb} = 1 + 46\, Bo^{0.5}$ for $Bo < 0.3 \times 10^4$	
	$\psi_{nb} = 230\, Bo^{0.5}$ for $Bo < 0.3 \times 10^4$	
	$\psi_{cb} = \frac{1.8}{N^{0.8}}$	
For 0.1<N<1.0	$\psi_{bs} = F\, Bo^{0.5} e^{2.74\, N^{-0.1}}$	
For $N \leq 0.1$	$\psi_{bs} = F\, Bo^{0.5} e^{2.74\, N^{-0.15}}$	for $Bo \geq 11 \times 10^{-4}$ then F = 14.7 for $Bo \geq 11 \times 10^{-4}$ then F = 15.43

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 then $C1$ = 1.1360, $C2$ = -0.9, $C3$ = 667.2, $C4$ = 0.7 and $C5$ = 0.3
For Co > 0.5 then $C1$ = 0.6683, $C2$ = -0.02, $C3$ = 1058, $C4$ = 0.7 and $C5$ = 0.3
If the tube is horizontal, then $C5$ = 0 for all cases.

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

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

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

1 and ϕ_G^2 also increases from 2 - 40, which is closed to those of other correlations. The correlation of the parameters ϕ_G^2 and X_{tt} is

$$\phi_G^2 = 1 + aX_{tt}^b,$$

and the correlations for the studied refrigerants in the evaporator are illustrated in Table 3.

Table 3. Constant values of R-22 alternatives pressure drop correlations

Refrigerants	Tube type	a	b	Percentage of data within ±20%
R-22	smooth	40.030	1.2973	80.00
R-22	grooved	25.550	1.0443	80.00
SUVA-9000	smooth	78.461	2.2971	83.33
SUVA-9000	grooved	50.350	1.5864	95.00
SUVA-9100	smooth	28.432	0.9990	83.33
SUVA-9100	grooved	17.842	0.5330	89.29

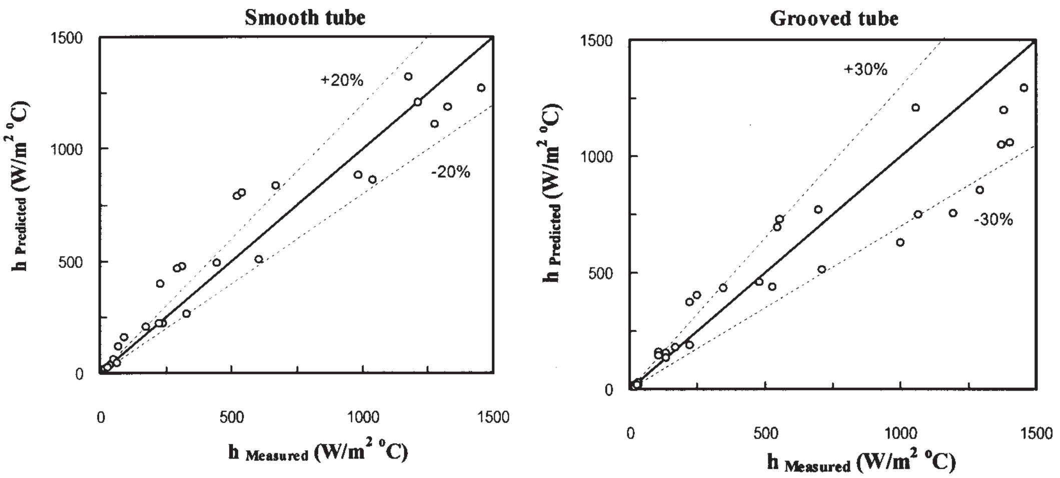

Figure 8. Comparison of measured and predicted boiling heat transfer coefficients of R-22 in smooth and grooved tubes in the present study

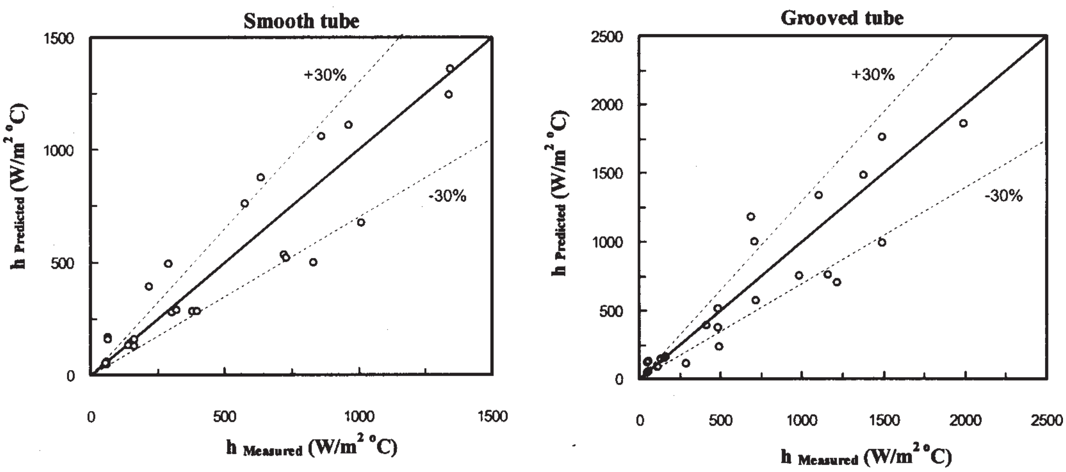

Figure 9. Comparison of measured and predicted boiling heat transfer coefficients of SUVA-9000 in smooth and grooved tubes in the present study

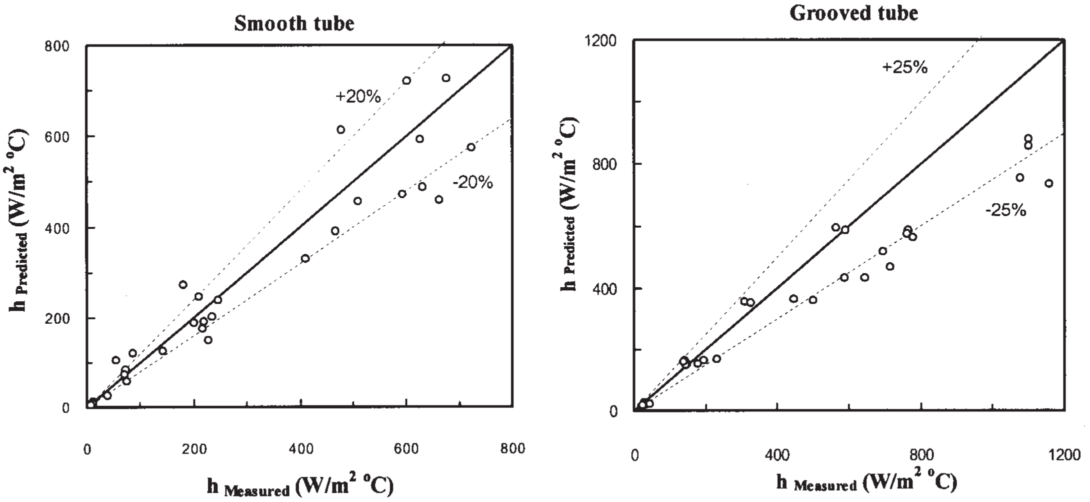

Figure 10. Comparison of measured and predicted boiling heat transfer coefficients of SUVA-9100 in smooth and grooved tubes in the present study

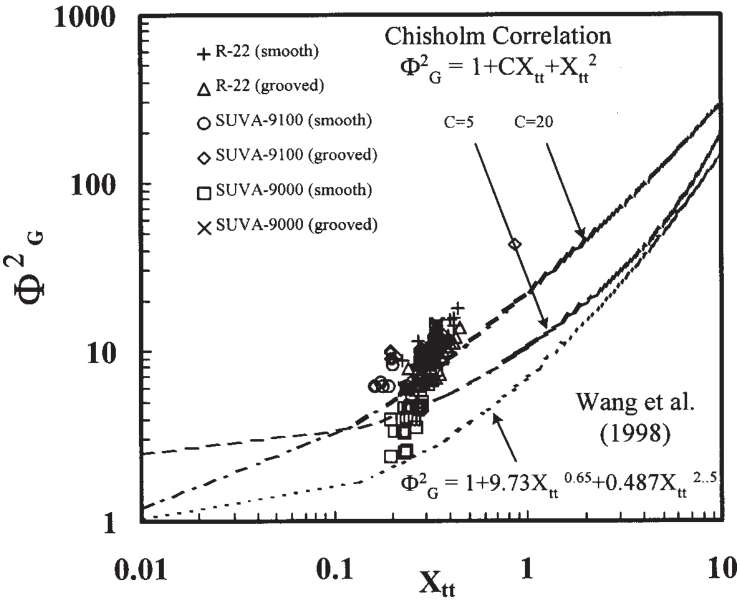

Figure 11. Pressure drops of R-22 alternative refrigerants in smooth and grooved tubes

Conclusion

This study shows the boiling heat transfer coefficient of R-22, SUVA-9000 and SUVA-9100 in horizontal tube for small refrigerator scale. In the evaporator, boiling heat transfer coefficients of the grooved tube are higher than those of the smooth tube because of the greater surface area and higher turbulence. In smooth and grooved tubes, R-22 shows the highest heat transfer coefficient, following by SUVA-9100 and SUVA-9000, respectively. Pressure drops of the refrigerants in smooth and grooved tubes of the evaporator are not different. The heat transfer and the pressure correlations developed could estimate the heat transfer coefficients and the pressure drop quite well.

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References

Chaddock, J.B. and Brunemann, H. 1967. Forced convection boiling of refrigerants in horizontal tubes-phase 3. Report HL-113. Duke University, Durham.

Hewitt, G.F., Shires, G.L. and Bott, T.R. 1994. Process heat transfer. CRC Press, Inc: USA.

Incropera, F.P. 1990. Fundamental of heat and mass transfer. John Wiley and Sons, Inc: Singapore.

Kandlikar, S.S. 1987. A general correlation for saturated two phase flow and boiling heat transfer inside horizontal and vertical tubes. ASME Winter Annual Meeting. Boston. MA. December 13-18. **In**: Boiling and Condensation in Heat Transfer Equipment. 85: 9-19.

Lavin, J.G. and Young, E.H. 1965. Heat transfer to evaporating refrigerants in two-phase flow. Intl. J. AlChE . 11:1124-1132.

Shah, M.M. 1982. Chart correlation for saturated boiling heat transfer : equations and further study, ASHRAE Transactions. 88: 66-86.

Wang, C.C., Chiang, C. and Yu, J. 1998. An experimental study of in-tube evaporation of R-22 inside a 6.5-mm smooth tube, Int. J. Heat and Fluid Flow: 259-269.0