Experimental Study on Evaporation Heat Transfer Characteristics of R32 in A Plate Heat Exchanger

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Abstract - This study experimentally investigates the evaporation heat transfer and frictional pressure drop characteristics of R32 in a plate heat exchanger with a chevron angle of 60°. The evaporation heat transfer coefficient and friction factor are measured at mass fluxes of 50, 70, and 90 kg/m²s, a heat flux of 6.45 kW/m², and saturation temperatures of 5 and 15 °C. The heat transfer coefficient increases with the mass flux, saturation temperature, and vapor quality because the forced convective boiling is dominant in this study. The frictional pressure drop also increases gradually with the vapor quality owing to the high velocity difference between the two phases. R32 exhibits a higher heat transfer coefficient and frictional pressure drop than R410A due to the high thermal conductivity and low vapor density. Furthermore, the friction factor of R32 decreases up to a certain quality (0.4) and then increases slightly afterwards. The frictional pressure drop of R32 is greater than R410A owing to the influence of density despite the lower viscosity. The friction factor of R32 is also higher than R410A due to the lower liquid and vapor density.

Keywords: R32, Heat transfer coefficient, Friction factor, Frictional pressure drop, Plate heat exchanger

Nomenclature

a = amplitude of sinusoidal surface corrugation, m
 b = channel spacing, m
 C_p = specific heat, J/kg°C
 D_h = hydraulic diameter, m
 f = friction factor, dimensionless
 g = gravitational constant, m/s²
 G = mass flux, kg/m²s
 h = heat transfer coefficient, kW/m²°C
 H = enthalpy, kJ/kg
 i_{fg} = enthalpy of vaporization, kJ/kg
 k = thermal conductivity, W/m°C
 L = channel length from center of inlet port to center of exit port, m
LMTD = log mean temperature difference, °C
 $\dot{m}$ = mass flow rate, kg/s
 N_{pass} = number of passes of an exchanger
 P = pressure, kPa
 q = heat flux, kW/m²
 Q = heat transfer rate, kW
 R_f = fouling factor
 R_{wall} = thermal resistance of the wall
 Re = Reynolds number, dimensionless
 T = temperature, °C
 U = overall heat transfer coefficient, kW/m²°C
 u = flow velocity, m/s
 v = specific volume, m³/kg

w = width of the plate, m
 X = vapor quality
 ΔP = pressure drop, kPa
 ΔT = temperature difference, °C
 ΔX = quality change in the exchanger
 β = chevron angle of plate heat exchanger, degrees
 Λ = corrugation wavelength, m
 ρ = density, kg/m³
 μ = viscosity, Ns/m²
 Subscripts
 a = acceleration
 ave = average
 ele = elevation
 exp = experiment
 fg = difference between liquid and vapor phase
 i, o = at inlet and exit of test section
 l, g = liquid and vapor phase
 m = average value for the two-phase mixture or between the inlet and outlet
 R = refrigerant
 sat = saturated state
 w = water
 wall = wall of the plate heat exchanger
 pre = pre-heater
ov = overall

1. Introduction

EU F-gas regulation [1] makes the selection of refrigerants more difficult by regulating high GWP refrigerants to phase out 79% of F-gas by 2030. Among the conventional refrigerants, R410A, which is mainly used in chillers or heat pumps, should be replaced due to its high GWP (2088). There are many candidates for replacement of R410A. Properties of the alternative refrigerants of R410A are shown in Table 1. Among the given candidates, R32 can be considered as the most promising alternative refrigerant because of its low GWP and density, high thermal conductivity and latent heat.

Table 1: Properties of R410A and its alternatives.

	R410A	R32	R446A	R452B
Critical pressure [kPa]	4901	5780	5725.2	5220.1
Critical temperature [K]	344.5	351.1	359.1	350.2
Boiling temperature [K]	221.7	221.3	223.4	222.5
Latent heat [kJ/kg]	272.9	368.9	327.2	331.8
Liquid Thermal conductivity [mW/m·K]	94.73	133.54	113.79	103.57
Vapor Thermal conductivity [mW/m·K]	14.09	13.43	14.7	15.16
Liquid viscosity [μ Pa/s]	134.11	127.31	125.2	114.7
Vapor viscosity [μ Pa/s]	13.02	12.26	12.73	12.93
Liquid density [kg/m ³]	1106.3	1000.9	1001.4	993.5
Vapor density [kg/m ³]	48.86	35.19	42.51	52.39
GWP	2088	650	470	677
Safety class	A1	A2L	A2L	A2L

2. Experimental setup and test conditions

2.1. Experimental setup

Fig. 1 shows a schematic of the experimental setup. The experimental setup consisted of a refrigerant, a water, and a chiller loop. A magnetic pump circulated the refrigerant through the test section, and the vapor quality of the refrigerant at the inlet of the test section was controlled by a pre-heater. The refrigerant evaporated in the test section by exchanging heat with water and condensed to the initial condition through exchanging heat with ethylene-glycol. All the temperatures and pressures were recorded by sensors and used to calculate the evaporation heat transfer coefficient and frictional pressure drop. The operating conditions of this experiment are shown in Table 2.

Fig. 1: Schematic diagram of the experimental setup.

Table 2: Operating conditions for experiments.

Data	Operating condition
Evaporating temperature (°C)	5, 15
Mass flux ($kg/s \cdot m^2$)	50, 70, 90
Quality	0.01–0.9
Heat flux (kW/m^2)	6.45

2.2. Plate heat exchanger

A plate heat exchanger is advantageous in terms of maintenance by separating each single plate. Also, it is widely used because of its high heat transfer performance and low fouling effect. The plate heat exchanger is characterized by two factors: corrugation aspect factor (γ) and enlargement factor (ϕ), as given in Eqs. (1) and (2).

$$\gamma = \frac{4a}{\Lambda} = \frac{2b}{\Lambda} \quad (1)$$

where a denotes the corrugation amplitude and Λ denotes the corrugation wavelength. The enlargement factor (ϕ) can be approximated by using a non-dimensional parameter A , as shown in Eqs. (2) and (3) [2].

$$A = \frac{2\pi a}{\Lambda} \quad (2)$$

$$\phi(A) \cong \frac{1}{6} \left(1 + \sqrt{1 + A^2} + 4\sqrt{1 + A^2 / 2} \right) \quad (3)$$

As shown in Fig. 2, the plate heat exchanger with a chevron angle of 60° was used as an evaporator.

Fig. 2: Geometry of the PHE.

3. Data reduction

3.1. Heat transfer coefficient

As shown in Eq. (4), the water-side heat transfer coefficient was calculated through the modified Wilson plot method technique [3] using the water-to-water preliminary experiment data. Validation graph of R_{ov} actual to R_{ov} calculated by Nu_w correlation is shown in Fig. 3.

$$Nu_w = 0.02628 Re_w^{1.21} Pr_w^{1/3} \quad (4)$$

The overall heat transfer coefficient was calculated by the LMTD method, which is shown in Eqs. (5) – (7).

Fig. 3 Validation graph of R_{ov} actual to R_{ov} calculated

$$\Delta T_{lm} = \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1 / \Delta T_2)}, \quad \Delta T_1 = T_{w,in} - T_{r,out}, \quad \Delta T_2 = T_{w,out} - T_{r,in} \quad (5)$$

$$U = \frac{Q_w}{A \Delta T_{lm}} \quad (6)$$

$$Q_w = \dot{m}_w C_{p,w} \Delta T_w \quad (7)$$

where Q_w denotes the heat gain from the evaporator, A denotes the heat transfer area of the PHE, and ΔT_{lm} denotes the LMTD. The properties of the refrigerant was obtained using REFPROP 10.0 database [4]. The total thermal resistance of the PHE, as shown in Eq. (8), can be obtained by the sum of the heat transfer coefficients of the water-side and refrigerant-side, and the conduction resistance.

$$\frac{1}{U} = \frac{1}{h_r} + \frac{1}{h_w} + R_{cond} \quad (8)$$

3.2. Two-phase frictional pressure drop

As shown in Eq. (9), the total pressure drop in the plate heat exchanger is composed of pressure drops by gravitation, acceleration, friction, and inlet/outlet ports.

$$\Delta P_{total} = \Delta P_{grav} + \Delta P_{acc} + \Delta P_{fric} + \Delta P_{port} \quad (9)$$

Pressure drops by the gravitation and acceleration are shown in Eqs. (10) and (11), respectively [5].

$$\Delta P_{grav} = g L \rho_m \quad (10)$$

$$\Delta P_{acc} = v_{fg} G^2 \Delta x \quad (11)$$

Also, the port pressure drop can be calculated by Eq. (12) [6].

$$\Delta P_{port} \cong 0.75 u_m^2 \rho_m \quad (12)$$

Consequently, the two-phase frictional pressure drop was calculated by Eqs. (9) – (12), and the two-phase friction factor f_{tp} was calculated by Eq. (13).

$$f_{tp} = -\frac{\rho_m D_h}{2 G^2 L} \Delta P_{fric} \quad (13)$$

4. Results and discussion

4.1. Heat transfer coefficient

Fig. 4 shows the evaporation heat transfer coefficient of R32 in the plate heat exchanger at a heat flux of 6.45 kW/m² and saturation temperatures of 5 °C and 15 °C. In the vapor qualities of 0.05 to 0.9, the heat transfer coefficient increased with the vapor quality. Since the forced convective boiling was the dominant heat transfer mechanism, the evaporation of the liquid to vapor phase accelerated with an increase in the vapor quality, promoting heat transfer. The heat transfer coefficient also increased with an increase in the mass flux.

Fig. 4 Heat transfer coefficient of R32 with respect to mass flux at (a) $T=5\text{ }^{\circ}\text{C}$ and (b) $T=15\text{ }^{\circ}\text{C}$.

Fig. 3 shows the heat transfer coefficient of R32 compared to R410A at various mass fluxes and vapor qualities with a heat flux of 6.45 kW/m^2 and a saturation temperature of $5\text{ }^{\circ}\text{C}$. At the same conditions, the heat transfer coefficient of R32 was higher than R410A because of its higher liquid thermal conductivity and lower vapor density.

Fig. 5 Heat transfer coefficients of R32 and R410A with respect to mass flux at $T=5^{\circ}\text{C}$.

4.2. Frictional pressure drop

Fig. 6 shows the frictional pressure drop and friction factor of R32 at various mass fluxes and vapor qualities with a heat flux of 6.45 kW/m^2 and saturation temperature of $5\text{ }^{\circ}\text{C}$. The frictional pressure drop increased with the vapor quality and the increasing trend became dramatic as the vapor quality increased. This was because the flow velocity increased as the vapor quality and mass flux increased, resulting in the higher friction. In relatively low vapor quality region ($0.05\text{--}0.4$), the friction factor decreased as the vapor quality increased because the decreasing rate of ρ_m in Eq. (13) was larger than the decreasing rate of ΔP_{fric} . On the other hand, at vapor qualities higher than 0.4 , the friction factor tended to increase slightly, indicating large enough ΔP_{fric} . This was because the friction increased as the vapor quality increased.

(a) (b)

Fig. 6 (a) Frictional pressure drop and (b) friction factor of R32 with respect to mass flux and vapor quality.

Fig. 7 shows the frictional pressure drop and friction factor of R32 compared to R410A. The frictional pressure drop of R32 was higher than R410A because the effect of the lower density of R32 was more dominant even with the lower viscosity compared to R410A. However, the friction factor of R32 was lower than R410A due to the effect of the lower density, as shown in Eq. (14).

Fig. 7 (a) Frictional pressure drop and (b) friction factor of R32 compared to R410A.

5. Conclusion

The evaporation heat transfer and frictional pressure drop characteristics of R32 were measured in a plate heat exchanger with a chevron angle of 60°. The evaporation heat transfer coefficient of R32 in the plate heat exchanger increased as the mass flux and vapor quality increased. Furthermore, the heat transfer coefficient of R32 was higher than R410A in all conditions owing to the higher thermal conductivity and lower vapor density of R32, promoting the forced convective boiling. Additionally, the frictional pressure drop of R32 increased with increases in the mass flux and vapor quality. The friction factor decreased significantly to a certain vapor quality and then slightly increased depending on which factor had the greater impact at each condition.

References

- [1] Schulz, M., and D. Kourkoulas. "Regulation (EU) No 517/2014 of the European Parliament and of the council of 16 April 2014 on fluorinated greenhouse gases and repealing Regulation (EC) No 842/2006." *Off. J. Eur. Union* 2014.517 (2014): L150.
- [2] Martin, Holger. "A theoretical approach to predict the performance of chevron-type plate heat exchangers." *Chemical Engineering and Processing: Process Intensification* 35.4 (1996): 301-310.
- [3] Wilson, E.E. "A basis for rational design of heat transfer apparatus." *The J. Am. Soc. Mech. Engrs.* 37 (1915): 546-551.
- [4] Lemmon, E. W., Bell, I. H., Huber, M. L., & McLinden, M. O. "NIST standard reference database 23: Reference fluid thermodynamic and transport Properties-REFPROP, Version 10.0, National Institute of Standards and Technology." *Standard Reference Data Program, Gaithersburg* (2018).
- [5] Collier, J.G., and J.R. Thome. *Convective boiling and condensation*. Clarendon Press, 1994.
- [6] Shah, R.K., and W.W. Focke. "Plate heat exchangers and their design theory." *Heat Transfer Equipment Design* 227 (1988): 254.