Effect of Saturation Temperature Change Due to Pressure Loss of Refrigerants on Heat Transfer Capacity of Heat Exchanger

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ABSTRACT

Heat exchanger is an essential component in the HVAC&R applications. The heat exchanger development methods are divided into simulation and experiment. Simulation is often used by researchers because it can save a lot of time and money over experimental method. However, it is difficult to model all physical phenomena in simulation. Some phenomena are simulated with ideal assumptions based on experience. To analyze an ideal heat exchanger without pressure loss of the refrigerant, the saturation temperature of the heat exchanger is assumed to be constant. Whereas, saturated temperature drop occurs due to the pressure loss of the refrigerant in a real case scenario which affects the heat transfer capacity of the heat exchanger. In this study, a theoretical method to evaluate the effect of saturated pressure loss of refrigerant on the heat transfer capacity in heat exchanger was proposed and analyzed. The proposed method was verified by simulation. R134a, R410A, R600a, R32, and R1234yf were selected as refrigerants for analysis which are used in air conditioners and refrigerators. As a result, the heat transfer capacity ratio of the cycle using R134a showed 96.11% under the condensing condition of 15 kPa pressure loss, and 108.97% and 123.17% for air conditioner and refrigerator evaporating conditions, respectively. Moreover, R600a showed the greatest performance change, and R32 showed the smallest performance change.

1. INTRODUCTION

A heat exchanger is an essential component in the HVAC&R applications. The heat exchanger development method is separated into simulation and experiment. Experimental approaches are often used to improve the performance of heat exchangers, but they are costly and time-consuming. Simulation is often used because it has the advantage of saving time and money. However, it is necessary to model complicated phenomena related to the thermal-hydraulic characteristics of the heat exchanger. Sometimes, complex phenomena are ideally assumed and simulated, which can cause errors between real and theoretical results.

Pressure loss occurs when refrigerant passes through a heat exchanger. The pressure loss of the two-phase flow in the heat exchanger not only increases the power consumption of the system but also affects the heat transfer capacity of the heat exchanger by reducing the saturation temperature of the refrigerants. Figure 1 schematically shows an example of temperature change of refrigerants in a condenser. In the ideal situations, the pressure of the refrigerants in the two-phase flow region remains the same, and the saturation temperature is also kept the same. However, in real situations, the refrigerants pass through the heat exchanger tubes and the saturation temperature decreases. Therefore, it is important to clarify the error between the ideal situation and the real situation.

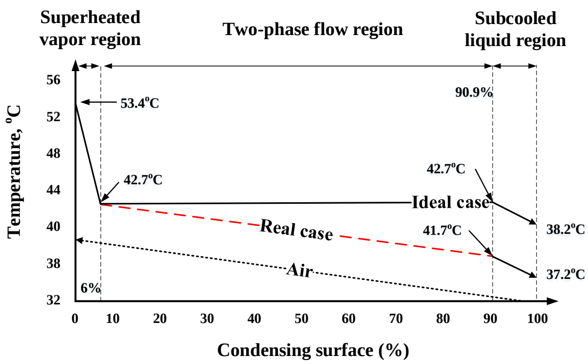


Figure 1. Variation of temperature through a condensing heat exchanger (Handbook of ASHRAE (2008)).

Recently, some researchers studied the effects of saturation temperature changes due to pressure loss on heat transfer capacity. Zhang and Webb (2001) observed that the lower the saturation temperature of the refrigerants, the greater the pressure loss of the refrigerants. This means that the lower the saturation temperature of the refrigerants, the larger the saturation temperature drop of the refrigerants. For this reason, in order to analyze the effect of pressure loss on heat transfer capacity, it is necessary to individually evaluate the effect of saturation pressure changes on condensation and evaporation conditions. ASHRAE (2008) recommends a saturation temperature drop below 2.2 ℃ due to refrigerants pressure loss in the condensing heat exchanger. Under the evaporation conditions, Didi et al. (2002) mentioned the saturation temperature drop due to refrigerant pressure loss to be below 1.4 ℃. However, ASHRAE and Didi et al.'s work do not explain why saturation temperature drop thresholds are proposed, and the relationship between saturation temperature drop and pressure loss. Larminat and Wang (2017) applied the COP to explain the saturation temperature drop including the temperature glide of mixed refrigerants. However, saturation temperature drop due to pressure loss was assumed. Since the relationship between pressure loss and saturation temperature drop is not described, it is difficult to analyze the effect of pressure loss on the heat transfer capacity of the heat exchanger. The irreversibility caused by the pressure loss of the refrigerants in a heat exchanger was considered by Liang et al. (2000) and Lee and Jeong (2016) based on the analysis of exergy and entropy, respectively. However, these studies only consider the irreversibility caused by the pressure loss in the heat exchanger and it is hard to explain the effect of the saturation temperature change on the heat transfer capacity because the potential for the heat transfer cannot be considered.

The authors did not come across any previous studies that investigated the effect of saturation temperature changes on heat transfer capacity due to head loss in heat exchangers. In this study, the effect of pressure loss on heat transfer capacity is presented and calculated as a theoretical model.

2. Analysis of heat transfer capacity through a theoretical model

The heat balance equation for the heat exchanger is represented by equation (1).

$$Q = UA\Delta T \tag{1}$$

In the ideal situation of a heat exchanger that does not take into account the pressure loss of the refrigerant, the refrigerant uses latent heat during the heat exchange. Therefore, the temperature of the refrigerant at the inlet and the outlet of the heat exchanger is the same as the ideal situation shown in Figure 1 and is expressed by equation (2).

$$\Delta T_{ideal} = (T_i - T_{air}) = (T_o - T_{air}) \tag{2}$$

However, in the real situation, even if the temperature of the refrigerant is saturated due to the pressure loss in the heat exchanger, the saturation temperature is decreased by the pressure loss of the refrigerant. Therefore, the temperature difference of the refrigerant occurs at the inlet and outlet of the heat exchanger in the real situation as shown in Figure 1. Thus, to obtain the temperature difference of the heat exchanger, the log mean temperature difference (ΔT_{LM}), is used and expressed in equation (3).

$$\Delta T_{LM,real} = \frac{(T_i - T_{air}) - (T_o - T_{air})}{ln\left(\dfrac{T_i - T_{air}}{T_o - T_{air}}\right)} \tag{3}$$

The total thermal resistance is expressed as 1/UA as shown in equation (4).

$$\frac{1}{UA}=R_{air}+R_w+R_{ref}=\frac{1}{(\eta hA)_{air}}+R_w+\frac{1}{(hA)_{ref}} \tag{4}$$

Where R_{ref} is the refrigerant side convection heat resistance, R_{air} is the air side convection heat resistance, R_w is the conduction heat resistance of the wall surface, and η is the overall surface efficiency.

From equations (1) to (4), the ratio of heat transfer rate in an ideal heat exchanger and heat transfer rate in a real heat exchanger can be expressed by equation (5). By using equations (5), heat transfer capacity of a heat exchanger can be analyzed.

$$\frac{Q_{real}}{Q_{ideal}}=\frac{(UA)_{real}\,\Delta T_{LM,real}}{(UA)_{ideal}\,\Delta T_{ideal}}\times 100\ (\%) \tag{5}$$

In the equation (5), T_o is used to calculate $\Delta T_{LM,real}$ using the Clapeyron-Clausius equation. The Clapeyron-Clausius equation is a relation that can calculate the saturation pressure change with temperature and is rearranged as Equation (6).

$$T_o=\left[\frac{1}{T_i}-\frac{R_c}{i_{fg}}ln\left(\frac{P_i-\Delta P}{P_i}\right)_{sat}\right]^{-1},\qquad \Delta P=P_i-P_o \tag{6}$$

If the inlet condition of the heat exchanger and the type of refrigerant are determined in Equation (6), the outlet temperature of the heat exchanger according to the pressure loss of the refrigerant can be obtained. By calculating T_o, $\Delta T_{LM,\,real}\,/\,\Delta T_{ideal}$ of equation (5) can be calculated and can be used for heat transfer capacity analysis. However, the heat transfer capacity analysis should analyze not only $\Delta T_{LM,\,real}\,/\,\Delta T_{ideal}$ but also $UA_{real}\,/\,UA_{ideal}$. Therefore, Equation (4) is expressed as Equation (7) as a ratio representing the change in *UA* due to pressure loss.

$$\frac{UA_{real}}{UA_{ideal}}=\left(\frac{1}{C_1h_{ref}}+C_2\right)^{-1}_{real}\Bigg/\left(\frac{1}{C_1h_{ref}}+C_2\right)^{-1}_{ideal}\quad where\ \ UA=\left(\frac{1}{C_1h_{ref}}+C_2\right)^{-1},C_1=A_{ref},\ C_2=(R_{air}+R_w) \tag{7}$$

Subsequently, equation (7) can be taken as the case where the thermal resistance of the refrigerant is too small or too large as shown in equations (8), respectively.

$$if\quad \frac{1}{C_1h_{ref}}<<C_2,\ \frac{UA_{real}}{UA_{ideal}}=\frac{(C_2)^{-1}_{real}}{(C_2)^{-1}_{ideal}}=1\qquad\qquad if\quad \frac{1}{C_1h_{ref}}>>C_2,\ \frac{UA_{real}}{UA_{ideal}}=\frac{(C_1h_{ref})_{real}}{(C_1h_{ref})_{ideal}}=\frac{h_{ref,real}}{h_{ref,ideal}} \tag{8}$$

Since condensation and evaporation have different physical phenomena, the change of convective heat transfer coefficient for pressure loss is evaluated by separating condensing condition and evaporating condition. For condensation conditions, the correlation of Shah (1979) is used and is given by equation (9).

$$h_{ref,\,cond}=0.023\,\mathrm{Re}_l^{0.8}\,\mathrm{Pr}_l^{0.4}\left(\frac{k_l}{D_h}\right)\left[(1-x)^{0.8}+\frac{3.8x^{0.76}(1-x)^{0.04}}{P_R^{\ 0.38}}\right] \tag{9}$$

In order to analyze the change of *UA* due to the refrigerant pressure loss, equation (8) can be expressed as equation (10) by summarizing the variables irrespective of the state of the refrigerant.

$$\left(\frac{UA_{real}}{UA_{ideal}}\right)_{cond}=\left(\frac{h_{ref,\,real}}{h_{ref,\,ideal}}\right)_{cond}=\frac{\left(\mathrm{Re}_l^{0.8}\,\mathrm{Pr}_l^{0.4}\left(\frac{k_l}{D_h}\right)\left[(1-x)^{0.8}+\frac{3.8x^{0.76}(1-x)^{0.04}}{P_R^{\ 0.38}}\right]\right)_{real}}{\left(\mathrm{Re}_l^{0.8}\,\mathrm{Pr}_l^{0.4}\left(\frac{k_l}{D_h}\right)\left[(1-x)^{0.8}+\frac{3.8x^{0.76}(1-x)^{0.04}}{P_R^{\ 0.38}}\right]\right)_{ideal}} \tag{10}$$

Table 1. Calculating conditions.

List	Refrigerants saturation temperature		Air temperature		List	Value	
Condensing condition for air-conditioner and refrigerator	51.7	℃	35	℃	Hydraulic diameter	12	mm
Evaporating condition for air-conditioner	1.7	℃	10	℃	Mass flux	200	kg/m^{2}s
Evaporating condition of refrigerator	-25	℃	-18	℃	Quality	0.7	
					Pressure loss	15	kPa

For evaporation condition, Gungor and Winterton (1987) correlation is used and is expressed as equation (11).

$$h_{ref,eva} = 0.023\mathrm{Re}_l^{0.8}\,\mathrm{Pr}_l^{0.4}\left(\frac{k_l}{D_h}\right)\left(EE_2 + SS_2\right)$$

$$\text{where } E = 1 + 3000Bo^{0.86}, \; E_2 = Fr_l^{(0.1-2Fr_l)}, \; S = 1.12\left[\frac{x}{(1-x)}\right]^{0.75}\left(\frac{\rho_l}{\rho_v}\right)^{0.41}, \; S_2 = Fr_l^{1/2} \tag{11}$$

Similar to the condensing conditions, by summarizing the variables irrespective of the state of the refrigerant, equation (8) is expressed as

$$\left(\frac{UA_{real}}{UA_{ideal}}\right)_{eva} = \left(\frac{h_{ref,real}}{h_{ref,ideal}}\right)_{eva} = \frac{\left(\mathrm{Re}_l^{0.8}\,\mathrm{Pr}_l^{0.4}(k_l)(EE_2+SS_2)\right)_{real}}{\left(\mathrm{Re}_l^{0.8}\,\mathrm{Pr}_l^{0.4}(k_l)(EE_2+SS_2)\right)_{ideal}} \tag{12}$$

To calculate the total thermal resistance, the temperature conditions were selected according to the AHRI and ISO standards. For the condensation condition, AHRI 460 (2005) was used. Moreover, for the evaporation condition, AHRI 420 (2008) and ISO 15502 (2005) were used. The hydraulic diameter, mass flux, and quality were selected as 12 mm, 200 kg/m2-s, and 0.7, respectively as stated in ASHRAE (2009). The range of pressure loss was determined to be 15 kPa by the saturated temperature drop reported by ASHRAE (2008) and Didi et al. (2002). It should be noted that the refrigerant properties were obtained from the REFPROP 9.1. The conditions used for the heat transfer capacity analysis are summarized in Table 1.

Figure 2 shows the absolute values of the UA ratio and temperature ratio according to the pressure loss of R134a under different conditions. For 15 kPa of pressure loss, the UA ratio is less than 0.1 times the change in temperature ratio. Therefore, condensation and evaporation condition can be considered as $UA_{ideal} \cong UA_{real}$. Hence, equation (5) can be expressed by equation (13).

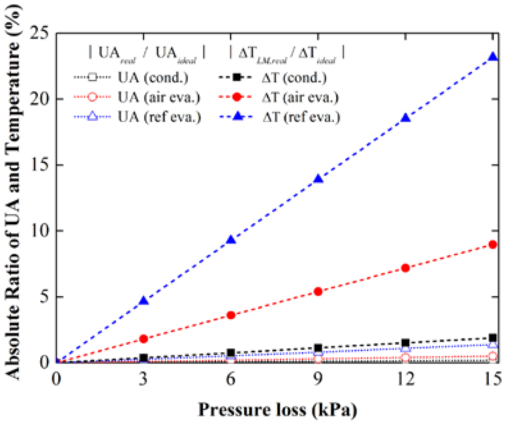
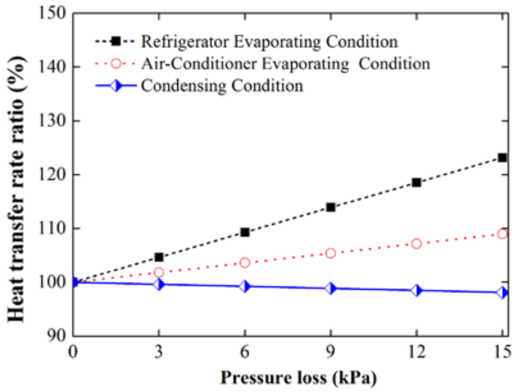


Figure 2. Comparison of UA with temperature in variable conditions.

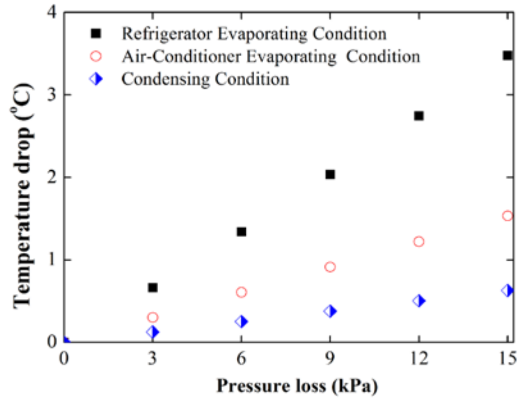
$$\frac{Q_{real}}{Q_{ideal}} = \frac{(UA)_{real}\,\Delta T_{LM,real}}{(UA)_{ideal}\,\Delta T_{ideal}} \cong \frac{\Delta T_{LM,real}}{\Delta T_{ideal}} \times 100(\%) \tag{13}$$

Using equations (6) and (13), the heat transfer capacity due to the pressure loss in the heat exchanger can be analyzed as the ratio of heat transfer rate.

Figure 3 shows the heat transfer rate ratio and a temperature drop of R134a for pressure loss under various conditions. In Figure 3 (a), at the pressure loss of 15 kPa, the heat transfer rate ratio represents 98.11%, 108.97%, and 123.17% under conditions of condensation, air-conditioning evaporation, and refrigerator evaporation, respectively. This is due to the difference in temperature drop.

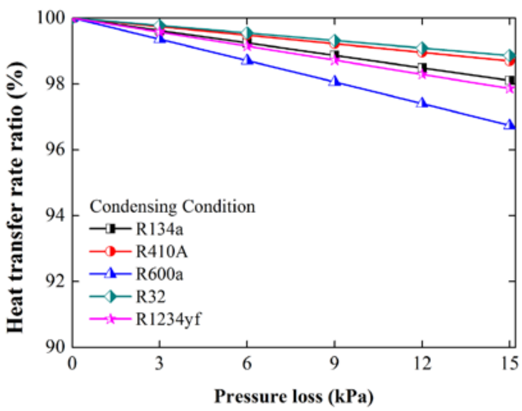


(a)

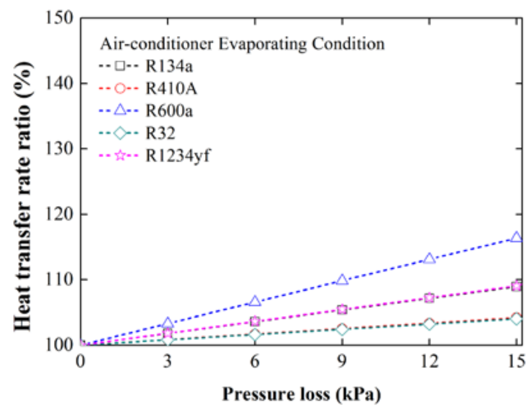


(b)

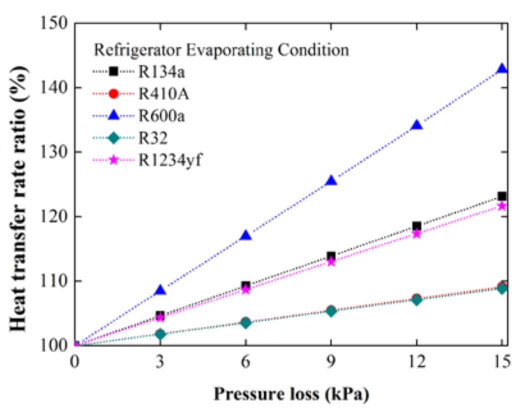
Figure 3. Effect of Pressure loss on the (a) heat transfer rate ratio, and (b) temperature drop of R134a.



(a)



(b)



(c)

Figure 4. Effect of pressure loss on heat transfer rate ratio in (a) condensing condition, (b) air-conditioner evaporating condition, (c) refrigerator evaporating condition, for various refrigerants.

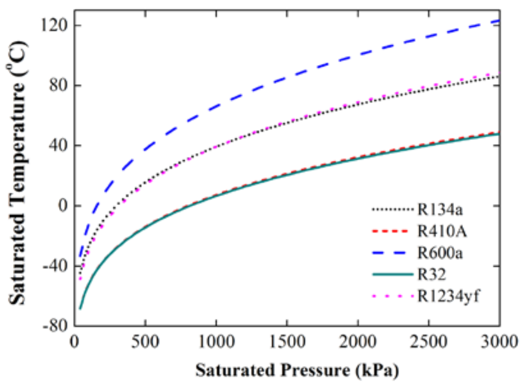


Figure 5. Saturated temperature versus saturated pressure for different refrigerants.

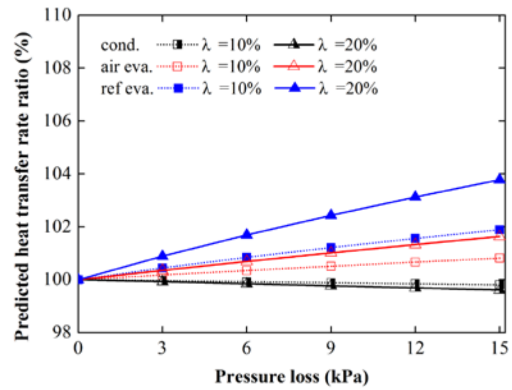


Figure 6. Effect of pressure loss by λ on the predicted heat transfer rate ratio for R134a.

In Figure 3 (b), the temperature drops to 0.63 ℃ in condensation condition and 3.48 ℃ in evaporation condition. This means that, the lower saturation temperature of the refrigerants showed the higher impact on the temperature drop concerning the pressure loss. Therefore, the low saturation temperature can have a significant effect on the heat transfer capacity of the heat exchanger.

Since the capacity of the heat exchanger depends on the refrigerant, the heat transfer capacity of the heat exchanger has been evaluated using R134a, R410A, R600a, R32 and R1234yf. Figure 4 compared the heat transfer rate ratio of various refrigerants with the pressure loss under different conditions. For condensation conditions, heat transfer rate ratio is arranged in the following order: R600a, R1234yf, R134a, R410A, R32. At a pressure loss of 15 kPa, R600a has the highest heat transfer rate ratio at 96.74%, and R32 is the smallest at 98.86%. In case of evaporating conditions for the air-conditioner, the heat transfer rate ratio of refrigerants shows the similar tendency as condensing conditions in order of R600a, R1234yf, R134a, R410A, and R32. However, for refrigerator evaporating condition, the heat transfer rate ratio changes in the order of R600a, R134a, R1234yf, R410A, and R32, with different order of R134a and R1234yf from other conditions. This is caused by the difference in temperature drop due to the pressure loss of each refrigerant. Figure 5 shows the saturation pressure and saturation temperature for each refrigerant. R32 has the lowest saturation temperature drop with saturation pressure loss, and R600a has the largest one. In case of R134a and R1234yf, the saturation temperature drop versus the saturation pressure loss is reversed in some sections. Also, the saturation temperature of the refrigerants decreases sharply as the saturation pressure of the refrigerants approaches the vacuum as shown in Figure 5. The lower the saturation temperature of the refrigerant, the greater the saturation temperature drop due to the refrigerant pressure loss. This is consistent with the observation by Zhang and Webb (2001).

3. Prediction of heat transfer capacity by correlation error

Pressure loss in two-phase flow, the acceleration terms due to the density and the gravity is calculated theoretically. However, since the phenomenon of pressure loss due to frictional term is complicated, it is difficult to calculate by the theoretical method, and is calculated using the experimental method instead. The error of the friction correlation equation provides not only the error of the predicted value of the pressure drop but also the error of the saturated temperature predicted value. That is, the error of the pressure drop correlation affects the predicted value of the heat transfer capacity for the heat exchanger. Therefore, to analyze the predicted value of the heat transfer capacity in the heat exchanger due to the correlation error of pressure loss, the correlation error is defined as equation (14).

$$\lambda = \frac{\Delta P_{pred}}{\Delta P_{real}} \times 100 \ (\%) \tag{14}$$

The effect of the correlation error on the predicted value of the heat transfer capacity in the heat exchanger can be analyzed using the ratio of the real heat transfer rate and the heat transfer rate due to the correlation error, as shown in equation (15).

$$\frac{Q_{pred}}{Q_{real}} = \frac{(UA)_{pred}\,\Delta T_{LM,\,pred}}{(UA)_{real}\,\Delta T_{LM,\,real}} \times 100\ (\%)$$

$$\text{Where,}\quad \Delta T_{LM,\,pred} = \frac{(T_i - T_{air}) - (T_{o,\lambda} - T_{air})}{ln\left(\dfrac{T_i - T_{air}}{T_{o,\lambda} - T_{air}}\right)},\quad T_{o,\lambda} = \left[\frac{1}{T_i} - \frac{R_c}{i_{lv}} ln\left(\frac{P_i - \Delta P_{pred}}{P_i}\right)_{sat}\right]^{-1} \tag{15}$$

The change of UA due to pressure loss was analyzed in equation (7-12). When the error of the correlation is less than 100%, $\frac{UA_{pred}}{UA_{real}} < \frac{UA_{real}}{UA_{ideal}}$. Therefore, equation (15) is summarized as equation (16).

$$\frac{Q_{pred}}{Q_{real}} = \frac{(UA)_{pred}\,\Delta T_{LM,\,pred}}{(UA)_{real}\,\Delta T_{LM,\,real}} \cong \frac{\Delta T_{LM,\,pred}}{\Delta T_{LM,\,real}} \times 100(\%) \tag{16}$$

Using the equations (16), the heat transfer capacity of the heat exchanger due to the correlation error can be analyzed by ratio of the predicted heat transfer rate.

Figure 6 demonstrates the predicted values of the heat transfer rate ratio due to the correlation error of R134a for the λ under various conditions. The predicted heat transfer rate ratio is lowest in condensation and highest in refrigerator condition. For condensation condition, the predicted heat transfer rate ratio is 99.61%, which is 0.39% overpredicted when λ=20% at a pressure loss of 15 kPa. The predicted heat transfer rate ratio is 99.61% at λ = 20%, and by decreasing λ to 10%, the error is improved by 0.19%. For refrigerator evaporation condition, when the pressure loss is 15 kPa and λ=20%, the predicted heat transfer rate ratio is 103.78%, which is 3.78% overpredicted. If λ is improved by an error of 10%, the predicted heat transfer rate ratio is 101.89%, 1.89% better than when λ=20%. It means that the predicted heat transfer rate ratio is overpredicted by 3.78% at 20% correlation error in refrigerator evaporating condition. It implies that the heat transfer capacity is 3.78% less than the design capacity of the heat exchanger.

4. CONCLUSIONS

In this study, the effect of refrigerant pressure loss on heat transfer capacity of the heat exchanger was analyzed through the theoretical model and simulation. Also, the influence of the pressure loss correlation error on the heat exchanger capacity was investigated. As a result, analysis of heat transfer capacity for R134a using theoretical model represents that the heat transfer rate ratio is 98.11% under condensing condition, 108.97% under the air-conditioning evaporating condition, and 123.17% under refrigerator evaporating condition. The lower saturation temperature of the refrigerants shows the higher impact on the temperature drop due to the pressure loss. Also, in case of various refrigerants comparisons, the heat transfer capacity of R134a, R410A, R600a, R32, and R1234yf is compared which indicates that R600a has the maximum and R32 has the minimum impact. In the study of the predicted heat transfer capacity by the correlation error, the change of heat transfer capacity due to the correlation error was greatest in the refrigerator evaporation condition. The heat exchange capacity error was improved by 1.89% when the correlation error was improved by 10% under refrigerator evaporation conditions.

NOMENCLATURE

A	Area	(m^2)
Bo	Boiling number	
C	Constant	
D_h	Hydraulic diameter, m	
Fr	Froude number	
G	Mass flux	(kg/m^2-s)
g	Gravity	(m/s^2)
h	Heat transfer coefficient	(W/m^2-K)
i_{lv}	Enthalpy	(kJ/kg)
k	Thermal conductivity	(W/m^2-K)
P	Pressure	(kPa)
P_R	Reduced pressure	
Pr	Prandtl Number	
Q	Heat transfer rate	(W)
q''_h	Heat flux	(W/m^2)
R	Thermal resistance	(K/W)
R_c	Gas constant	(kJ/kg-K)
Re	Reynolds Number	
T	Temperature	(°C)
U	Overall heat transfer coefficient	(W/m^2-K)
x	Quality	
η	Fin efficiency	
λ	Correlation error	
μ	Dynamic viscosity	(N-s/m^2)
ρ	Density	(kg/m^3)

Subscript

air	Airside
i	Inlet
l	Liquid
o	Outlet
pred	Predicted
ref	Refrigerants side
sat	Saturated
v	Vapor
w	Wall side

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