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A study on the determination method of defrosting start time by calculation of frost volume

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Abstract

In cold winter season, most air heat pumps suffer from the frost formed on the surface of the outdoor heat exchanger. This frost must be removed, otherwise the heat pump performance will keep degrading and deviate from the design conditions, causing severe damage to the components. For this reason, it is necessary to determine the appropriate defrost start time. Several methods have been suggested, however, most of them are either based on empirical studies that require large number of experiments or less practical because they use expensive sensors. In this study, a new method is proposed to determine the defrost start time and verified with experiment. Three steps are required. First, calculate the frost mass that is generated per unit time using the heat and mass transfer analogy. Second, obtain the volume by dividing the frost mass with the density of frost. Third, the frost volume generated per unit time is accumulated and compared with the heat exchanger volume to determine the defrost start timing. With this method, the defrosting operation can be appropriately initiated.

Keywords: heat pump, heat exchanger, frost, defrost start time, frost volume, blocking ratio;

1. Introduction

In cold winter season, most air heat pumps in humid region suffer from the frost. It forms naturally due to the sublimation from air to the surface of the outdoor heat exchanger when the surface temperature becomes lower than the freezing temperature of water and the dew point temperature of the ambient air. This frost disturbs the air flow in the outdoor heat exchanger and lowers the heat exchange rate, thus reduce the overall performance of the heat pump system. In the experiment from Yan et al. [1], heat transfer rate and overall heat transfer coefficient drop while pressure drop of the heat exchanger rises under frosting conditions. In addition, Shao et al. [2] showed the decline of both heating capacity and COP in fin-tube microchannel heat exchangers during frost formation process. The frost on the surface must be removed, otherwise the heat pump performance will keep degrading and cause significant energy loss. Moreover, heat pump operation might deviate from the design conditions. Thereby determining the proper defrosting start time to prevent severe damage to the components is one of the important issue in heat pumps

Up to now, several methods have been suggested to judge defrosting start time. For direct frost detection methods, photoelectric sensors with emitter and receiver that measure the intensity of infrared ray between the inlet and outlet of air flow at the outdoor heat exchanger is proposed by Byun et al. [3]. Similarly, from the study of Chung et al. [4], the differential pressure can be measured to monitor the frost formation. The direct detection method measures a certain spot of the heat exchanger and let us know whether there is frost in that spot or not. This approach generally raises the issue of representativeness as to whether the point can represent the whole. For indirect frost detection methods, Kim and Lee [5] proposed the effective mass flow fraction (EMF) that follows the trend heating capacity. Zhu et al. [6] suggested temperature–humidity–time (T-H-T) method that utilizes a frosting map which is evolved from simple temperature-time (T-T) method. In this

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method, a concept of accumulated time is introduced to determine the defrost start time. However, it only considered outdoor air condition and not the conditions of heat pump itself. The indirect detection methods generally describe overall system performance change but proposed methods are either hard to implement or limited on certain cases.

In this study, a new indirect frost detection method is proposed to determine the defrost start time and verified with experiment. The accumulated frost volume is calculated and compared with the volume of heat exchanger. The concept of accumulation is important because the state of the heat pump continues to change during its operation. This method also considers both factors of outdoor air (temperature and humidity) and heat pump itself (speed of compressor, the temperature of heat exchanger).

2. The calculation of frost volume and blocking ratio

Yoo et el. [7] explains the frost formation depends on the air temperature, humidity, and surface temperature of heat exchanger and heat exchange rate at evaporator. To form frosts two conditions must be met. The evaporator temperature should be lower than 0°C and the humidity ratio of outdoor air needs to be higher than that of air at the saturated condition at surface temperature of heat exchanger. The point 1 in Fig. 1 explains this condition. Otherwise, frost will not form. (point 2)

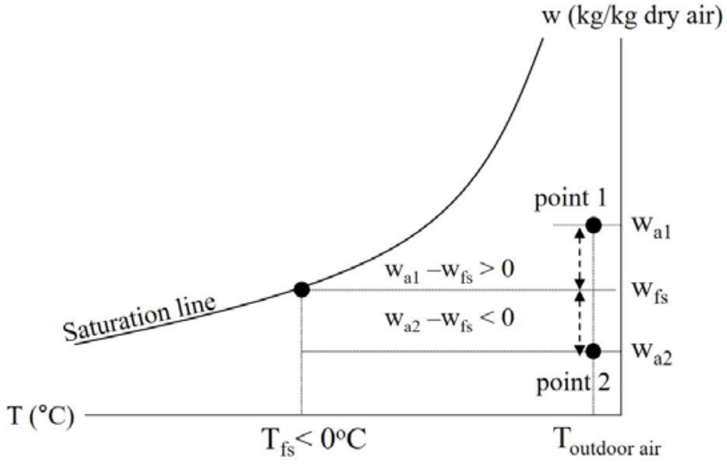

Fig. 1. Humidity ratio in psychrometric chart [7]

For the calculation of accumulated frost volume, three steps are required. First, we calculated the frost mass that is generated per unit time (Eq. (2)) by using the heat and mass transfer analogy in Eq. (1). Detailed derivation is represented in the study of Yoo et al. [7]

$$h_m = \frac{h_a}{C_{p,a}\rho_a Le^{2/3}} \tag{1}$$

$$\Delta m_f(t) = \frac{\dot{m}_{ref}\left(h_{comp,in} - h_{cond,out}\right) \times (w_a - w_{fs})\Delta t}{C_{p,a}Le^{2/3}\left(T_a - T_{fs}\right) + \left(w_a - w_{fs}\right)L_{ig}} \tag{2}$$

Second, we obtained the frost volume per unit time by dividing the frost mass with the density which is Eq. (3).

$$\Delta V_f(t) = \frac{\Delta m_f(t)}{\rho_f(t)} \tag{3}$$

The equations of frost density were developed over time. We used the equation (Eq. (4)) that has the same form from Hayashi *et al*. [8]

$$\rho_f = f(T_{fs}) = C_1 \exp\,(C_2 T_{fs}) \tag{4}$$

Third, the frost volume generated per unit time is accumulated (Eq. (5)) and compared with the heat exchanger volume to determine the defrost start timing. In Eq. (6), blocking ratio (BR) is defined to evaluate the ratio between two volumes.

$$V_{f,accum}(t) = \sum_{\tau=0}^{t} \Delta V_f(\tau) \tag{5}$$

$$BR = V_{f,accum}/V_{heat\ exchanger} \tag{6}$$

With the BR, which is an index of the ratio of frost volume and heat exchanger size, the time of defrost can be determined. In this work, the defrost process starts at the timing when BR becomes 1.

3. Experiment setup and test condition

In order to calculate the frost volume and blocking ratio that were presented in section 2, several sensor data are needed which are dry bulb temperature and relative humidity of outdoor air inlet, the evaporator inlet temperature, the pressure and temperature at the inlet of compressor and the outlet of condenser. Heat pump data are also required which are outdoor heat exchanger size, compressor displacement, current compressor rotational speed. Fig. 2 provides the measurement information of heat pump experiment. The properties of air such as density, enthalpy, specific heat and humidity ratio, were calculated by the equations from ASHRAE Handbook [9]. The air flow rate of the indoor units was obtained according to the ANSI/AMCA 210 [10].

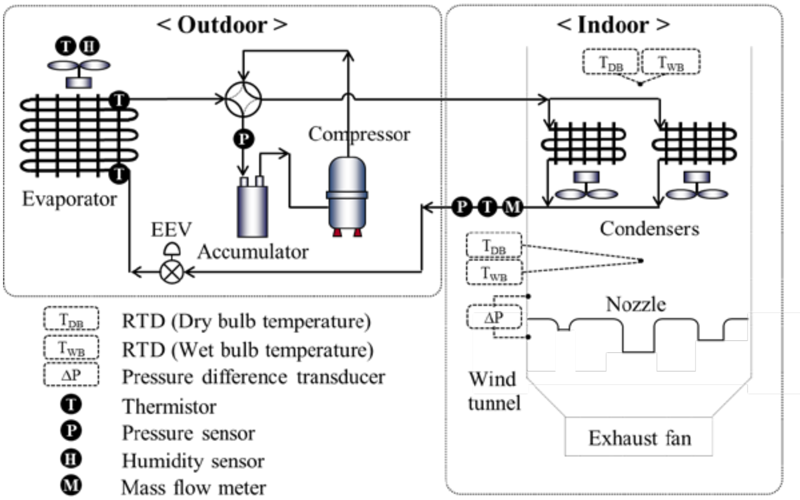

Fig. 2. The schematic of the heat pump system for experiment [7]

Table 1 gives the specification of the heat pump. It has variable speed compressor and fin-tube type heat exchangers, electronic expansion valve and uses refrigerant R410A.

Table 1. Specifications of heat pump system

Component	Specifications	
Compressor	Inverter-driven scroll compressor / Stroke volume : 62.1 cm^3	
Condenser side (2 Indoor units)	Fin-tube heat exchangers Tube inner / outer diameter (mm): 6.3 / 7.7 Width / length / height (mm): 3000 / 40 / 200	 Row / Step: 2 / 10 Fin pitch (mm): 1.5
Evaporator side (1 Outdoor unit)	Fin-tube heat exchanger Tube inner / outer diameter (mm): 6.3 / 7.7 Width / length / height (mm): 1650 / 60 / 1220	 Row / Step: 3 / 45 Fin pitch (mm): 1.7
Expansion device	Electronic expansion valve	
Refrigerant	R410A	

In Table 2, experiment conditions and results are presented. To verify the proposed frost volume calculation method, total 7 experiments were conducted. In case 1, 2 and 3, outdoor dry bulb temperature is set at 2, -4, -10˚C and all the other variables were fixed. Likewise, in case 1, 4 and 5, outdoor relative humidity changed from 68 to 83.8 %, in case 1, 6 and 7, compressor rotational speed was varied from 60 to 100 Hz.

Table 2. Experiment conditions and results

Experimental case	1	2	3	4	5	6	7
Indoor inlet T_{DB} (˚C)				20			
Indoor inlet T_{WB} (˚C)				15			
Outdoor inlet T_{DB} (˚C)	2	-4	-10	2	2	2	2
Outdoor inlet RH (%)	83.8	83.8	83.8	75	68	83.8	83.8
Compressor speed (Hz)	100	100	100	100	100	80	60
Q drop at the defrosting start time at BR=1 (%)	12.8	17.1	16.6	17.3	13.3	12.8	15.4
COP drop at the defrosting start time at BR=1 (%)	8.3	12.3	12.2	10.7	10.0	9.6	14.2
Time from start of operation to BR=1 (minutes)	72.0	70.0	65.2	96.5	135.1	82	108.6

The uncertainty analysis was done with respect to each cases, according to ASHRAE Guideline 2 [11]. At 95% confidential level, the maximum uncertainty of heating capacity and COP were 5.43% and 5.49% respectively.

4. Results

4.1. Determination of the defrost start time with the calculated blocking ratio

With the calculated BR derived from section 2, the defrost starts at the timing of BR equals 1 in all 7 experiment cases. The change of heating capacity, COP and BR for every cases is shown in Fig. 3. In Case 1 (Fig.3 (a)), when heat pump starts to operate the heating capacity and COP increases until 15 minutes. After that, heating capacity and COP does not show as drastic change between 15 and 60 minutes. However, after 60 minutes, the performance drop becomes noticeable. 72 minutes later, BR reaches 1 and at that time, the heating capacity and COP are dropped 12.8 % and 8.3% respectively. On the other hand, BR continues to grow from the beginning of operation because the value of humidity ratio difference in Eq. (2) is positive during operation and increases the value of frost volume in Eq. (5). All the other cases show similar trend which is divided into three stages. After heat pump is on, the performance value increases at first, shows only minor change at certain amount of time and begins to drop prominently. At the condition of BR =1, the defrost start timing for all cases lies at the 3rd stage where performance drop occurs and the specific percentage at that time is shown in Table. 2. Heating capacity drop is between 12.8 and 17.3 % while COP drop is between 8.3 and 14.2 %. With the proposed volume calculation method, it properly judges the defrost start time.

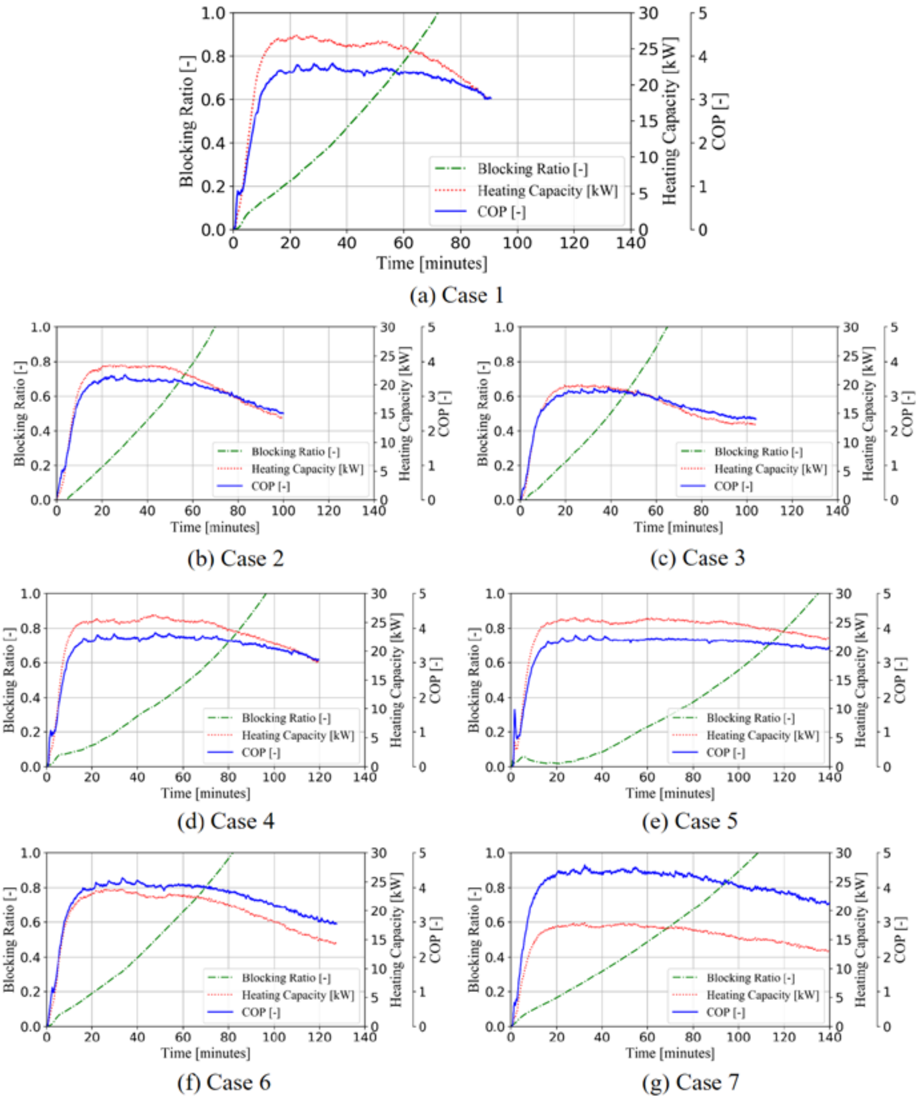

Fig. 3. Time vs. locking ratio, Heating capacity and COP at each cases

To support usefulness of the volume calculation method, the pictures of both front and right side of the outdoor heat exchanger were taken for every cases in Fig.4. At the defrosting start time, the majority of the area is covered with frost. No experimental cases show any mal-defrost.

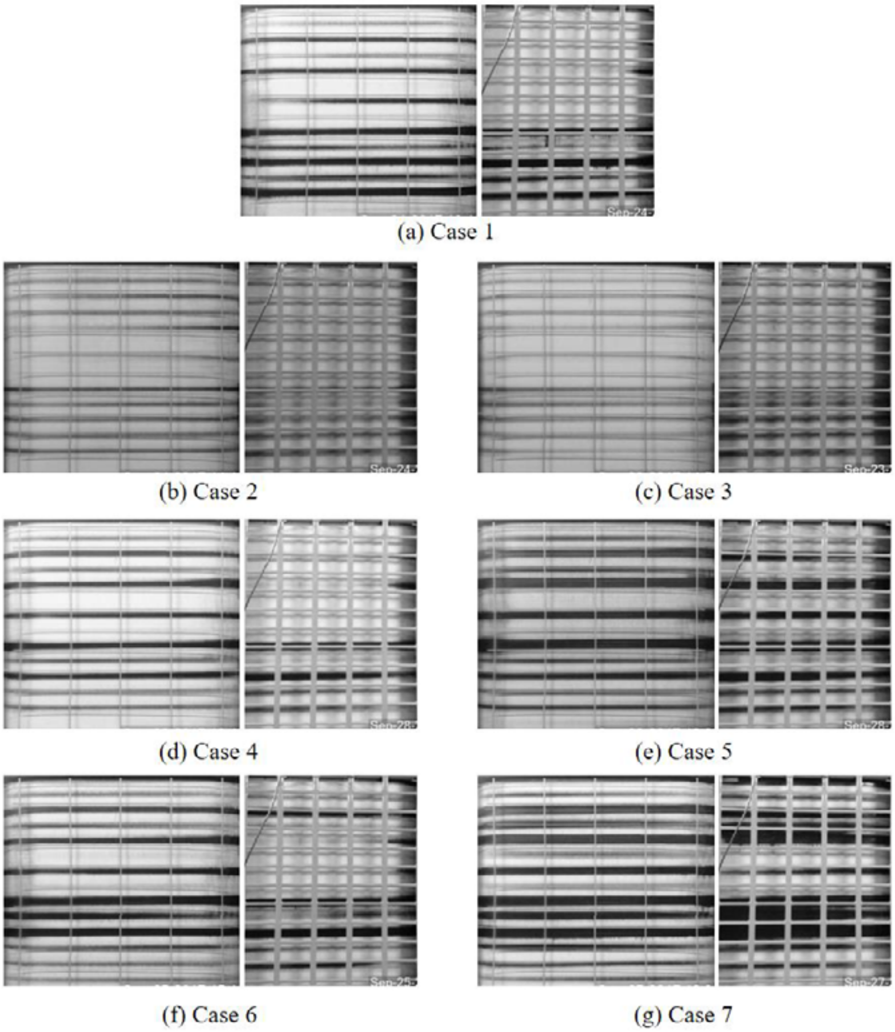

Fig. 4. The picture of outdoor heat exchanger at the defrost start time (front and right side)

4.2. The effect of outdoor dry bulb temperature, relative humidity and compressor rotational speed

With the 7 experiments in Table 2, the effect of outdoor dry bulb temperature, relative humidity and compressor rotational speed can be found. In case 1-3, all the other conditions are the same except outdoor dry bulb temperature. As the dry bulb temperature decreases, the start timing becomes shorter because the frost covers heat exchanger faster due to the lower frost density. (larger specific volume) When comparing case 1, 4 and 5, only the outdoor relative humidity is different. The relative humidity drop causes longer time from start of operation to BR=1. This is because the difference in humidity ratio ((w_a-w_{fs}) in Fig.1) decreases and forms less frost mass from Eq.(2).The effect of compressor rotational speed can be compared from case 1, 6 and 7. As the speed gets slower the mass flow rate of the refrigerant gets lower thereby creates less frost from Eq.(2).

5. Conclusion

In this study, a frost volume and blocking ratio calculation method is suggested in order to determine the defrosting start time. Detail calculation of BR and experiment setup is presented in section 2. To verify the method at various operating conditions, total 7 experiment were done at different outdoor dry bulb temperature, relative humidity and compressor rotational speed. In all cases, the defrosting start time (when BR becomes 1) were properly predicted at the performance decrease stage. The performance drop of heating capacity and COP is between 12.8 and 17.3 %, 8.3 and 14.2 % respectively at the defrost start time.

Acknowledgements

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