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Determination method of defrosting start-time based on temperature measurements

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HIGHLIGHTS

- A novel determination method of defrosting start-time is investigated.
- Effective mass-flow fraction (EMF) is calculated only by temperature measurements.
- The EMF detected variations in heat transfer rate of heat exchanger.
- The EMF provided accurate defrosting start-times under varied frosting conditions.

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ABSTRACT

We report a novel defrosting control method for refrigeration systems including refrigerators and heat pumps that commonly use time control or time–temperature control to determine the defrosting start-time. However, these methods do not provide accurate defrosting start-times due to variations in the operating conditions. Here, we describe the use of the effective mass-flow fraction (EMF) to determine the defrosting start-time; this method qualitatively detects variation trend of heat transfer rate based on only temperature measurements. The performance of the EMF control was compared with that of time control under varied frosting conditions of an experimental system. The time control method determined the defrosting start-time with an error of ±50%, whereas the EMF control precisely determined the defrosting start-time with an error of ±10%.

1. Introduction

Frosting occurs in the evaporators of refrigerators and in the outdoor units of air-source heat pumps. When the surface temperature of the heat exchanger is cooled below the dew point, water vapor in humid air condenses onto the surface of the heat exchanger. If the temperature of this surface is also below the freezing point, the condensed water droplets form a frost layer. If frosting occurs, the thermal performance of the heat exchanger degrades as a result of an increase in both the flow resistance and the thermal resistance [1–4]. To recover the thermal performance of the heat exchanger, defrosting cycles are required to remove the frost layer from the surface. If defrosting cycles are not conducted properly, the energy efficiency of the system suffers due to either a decline in the thermal performance of the heat exchanger (caused by excessive frost) or to excessive defrosting cycles [5–7].

There have been a number of researches on defrosting control. The most common control techniques are time control (TC) and time–temperature control (TTC), which are commonly applied to commercially available products. With TC, the operating time of the compressor is measured, and defrosting starts when the operating time of the compressor reaches a pre-set value. This technique is commonly used in commercial products due to its low cost [8]. However, TC may result in inaccurate defrosting start-times as variations in the operating conditions are not taken into account; TC control neglects information affecting frost formation such as the air temperature, humidity, air velocity, and refrigerant temperature. TTC was developed as an improvement to TC, TTC measures not only the operating time of the compressor but also the evaporation temperature or degree of superheating to determine the defrosting time [9,10]. However, Wang et al. [5] reported that TTC also results in inaccurate defrosting start-times under variable operating conditions; they termed this problem "mal-defrost". For example, TTC may perform a defrost cycle in the absence of a frost layer under low humidity conditions. In addition, a defrost cycle may not be carried out even

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Nomenclature

A	area, m^2
BR	blocking ratio, %
c_p	specific heat, J kg^{-1} K^{-1}
EMF	effective mass-flow fraction
FPI	fins per inch
L_h	latent heat of sublimation, J kg^{-1}
$\dot{m}$	mass flow rate, kg s^{-1}
Q	heat transfer rate, W
T	temperature, °C
t	time, s
w	absolute humidity, kg kg_{DA}^{-1}

Subscripts

a	air
ds	defrosting start
eff	effective
f	frosting
in	inlet
ineff	ineffective
lat	latent heat
max	maximum
out	outlet
pro	proper
r	refrigerant
sen	sensible heat
TC	time control
th	threshold
tot	total
tube	tube of heat exchanger

though the air path is completely blocked by frost layer under high humidity conditions.

Other defrosting control methods have been developed, including differential pressure control and optical control [11–13]. Differential pressure control measures the flow resistance between the inlet and outlet of air-side to determine whether a frost layer is present. Although this technique can effectively determine defrosting start-times, it is expensive and utilizes a transducer with a short lifespan; therefore, it is not suitable for commercial products. Optical control uses photo couplers to determine the appropriate defrosting start-time by measuring the thickness of the frost layer; however, this technique measures the thickness of the frost layer in only one location. For this reason, this technique may also result in inappropriate defrosting start-times. Furthermore, optical sensors are expensive and have a short lifespan, rendering this method is unsuitable for commercial products.

Two main problems exist in current defrosting control methods. First, TC and TTC exhibit the problems associated with mal-defrost; second, differential pressure control and optical control require expensive sensors. Here, we introduce effective mass-flow fraction (EMF) as a defrosting control method. The EMF can be calculated by measuring only the temperature; therefore, this method has the advantage of utilizing only low-cost sensing devices. Furthermore, this technique can precisely determine appropriate defrosting start-times, even under variable operating conditions. To validate the performance of EMF control, the accuracy of the defrosting start-times obtained with both TC and EMF methods are compared under various frosting conditions.

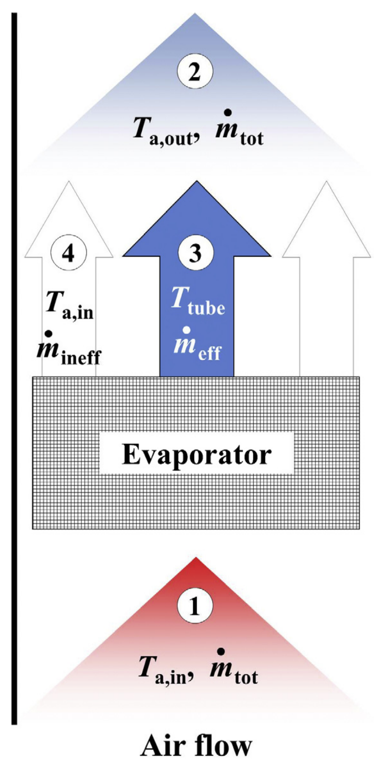

Fig. 1. A schematic diagram of the principle of EMF.

2. Methodology and experiments

2.1. Methodology

Fig. 1 describes a schematic diagram of the principles of EMF. The air flows into the duct with temperature $T_{a,in}$ and mass-flow rate $\dot{m}_{tot}$ (state ①). The air is cooled so that it exits the evaporator with temperature $T_{a,out}$ (state ②). When air passes through the evaporator, we assume that some fraction of the mass flow $\dot{m}_{eff}$ is cooled to temperature T_{tube} (state ③), and that the remaining mass flow $\dot{m}_{ineff}$ is at temperature $T_{a,in}$; i.e., this temperature is identical to that of the inlet air (state ④). States ③ and ④ are mixed and are equivalent to state ②. The terms $\dot{m}_{eff}$ and $\dot{m}_{ineff}$ are imaginary air mass-flow rates used to explain the concept of EMF, and the mass-flow rate $\dot{m}_{tot}$ is the measured total air mass-flow rate. The temperatures $T_{a,in}$, $T_{a,out}$, and T_{tube} can all be measured.

The energy balance based on the enthalpy at the outlet of the evaporator can be expressed by setting the sum of the masses of air in states ③ and ④ equal to the mass of air in state ②:

$$\dot{m}_{eff}c_{p,a}T_{tube} + \dot{m}_{ineff}c_{p,a}T_{a,in} = \dot{m}_{tot}c_{p,a}T_{a,out}, \quad (1)$$

where c_p is the specific heat capacity of the air under constant pressure. For practical applications in refrigerators and air-source heat pumps, we assume that the temperature will be in the range of −50 to 50 °C; therefore, $c_{p,a}$ can be assumed to be constant. (We