DEFROST CONTROL METHOD FOR AIR HEAT PUMP BASED ON AVERAGE HEATING CAPACITY

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Abstract

The defrosting system should operate timely for an air heat pump at lower temperature. Investigations have indicated that the period for defrosting is one of the important factors that affects the operating characteristics of the air heat pump. To frame the control method, several aspects should be studied, including frost growth process, system operating point variation and the control method. In this paper, system-operating characteristics were analyzed and a defrosting control method is presented based on average heating capacity [1].

1 INTRODUCTION

In cold and watery climate, the evaporator of air-source heat pump operates under frosting conditions for part of the heating season. Frost accumulation on the evaporator can reduce the heating capacity of the heat pump, so that the evaporator needs to be defrosted periodically in order to maintain adequate performance under frosting conditions. The loss of capacity during the defrost operation is usually made up using auxiliary heat, often in the form of electric heating.

Among the several defrosting methods for the evaporator of air-source units, the most effective may be passing refrigerant in its superheated vapor state through the evaporator tubes. There are two methods for this [2]: (a) Hot gas by-pass, that means the superheated refrigerant vapor from the compressor passed into the evaporator, by-passing the condenser and the expansion device. (b) Reverse cycle, the normal heating operation is reversed by using a 4-way valve so that the outdoor coil becomes the condenser and the indoor coil the evaporator. The reverse cycle method is generally used in reversible air-to-air systems designed to provide both heating and cooling.

Recently there are several defrost control methods [3] used for air-cooling heat pump and other refrigeration facilities running in low temperature environment, including: (a) Fixed time defrost control method. This method would not be suitable to variable environment conditions. (b) Time-temperature or time-pressure control method. As frost layer grows, systems evaporating temperature will fall down correspondingly. So that the evaporating temperature or evaporating pressure could be used as a criterion of defrost. (c) Air pressure differential control method. In frost conditions frost will block air flow channel and make air pressure differential larger than which in normal conditions. (d) Voice-operated control, get frost layer thickness by noise agitator's resonant frequency installed in evaporator. (e) Average heating capacity control method [1], defined by:

$$Q_m = \frac{1}{t + T_{def}} \int_0^t Q(\tau) d\tau \tag{1}$$

Analysis indicates that Q_m could reach its maximum when t and t_{def} confirmed suitable. This conception is good as one idea but there are many troubles to go into practice.

No matter which defrosting project will be chosen, a good control method should be designed to determine both the startup and the time length of defrosting action. To plan this, several factors must be studied: (a) Frost formation and development as a function of time, environment conditions and evaporating temperature. (b) Evaporating and condensation temperature change caused by frost depositing. (c) Coefficient of heat transfer between outdoor air and refrigerant. (d) Quantity change of air flow through the evaporator.

2 CALCULATION OF FROST THERMAL CONDUCTIVITY AND FROST DEVELOPMENT

The thermal conductivity of frost layer plays an important part in its structure and rate of formation. The frost contains air and crystals of ice, the air-ice thermal conductivity of frost k_e should be somewhere between the thermal conductivity of air and ice. The thermal conductivity of air is given by [4]

$$k_a = 2.646 \times 10^{-3} [\frac{T^{1/2}}{1 + (245/T) \times 10^{-12/T}}] \tag{2}$$

and that of ice by [4]

$$k_i = 630/T \tag{3}$$

Researches denote that radiation effective conductivity and ventilation enthalpy rate term can be neglected. The thermal conductivity due to the water vapor latent heat flux k_v can be defined by [4]

$$k_v \equiv \frac{\dot{m}_d L_s}{(dT/dx)} = \frac{L_s DB}{(1-\chi)\tau_s} \left(\frac{P_v}{R_v T^2} \right) \left(\frac{L_s}{R_v T} - 1 \right) \tag{4}$$

$$k = k_e + k_v \tag{5}$$

Since the water vapor diffusion occurs only in the air portion of the frost, Biguria and Wenzel [5] suggest that if the effective air thermal conductivity instead of the true air thermal conductivity is used, better results can be expected. The effective air thermal conductivity can be obtained from the relation

$$k_{eff\_air} = k_a + \frac{DP_t}{P_t - P_v}\left(\frac{L_s P_v}{R_v T^2}\right)\left(\frac{L_s}{R_v T} - 1\right) = k_a + \frac{\tau_s k_v}{B} \tag{6}$$

The proposed model in reference [4] made the following assumptions about the frost structure, as shown in figure 1 and 2. At low frost density or at high porosity, two types of frost structure predominate. One is the ice cylinders created by the diffusion of water onto the ice, which result in a parallel conductive heat transfer. The other portion is the ice spheres created by nucleation of water vapor or water droplets, resulting in a much lower conductive heat transfer. Specific analysis and calculations please refer to reference [4]. Simplification expression of k is:

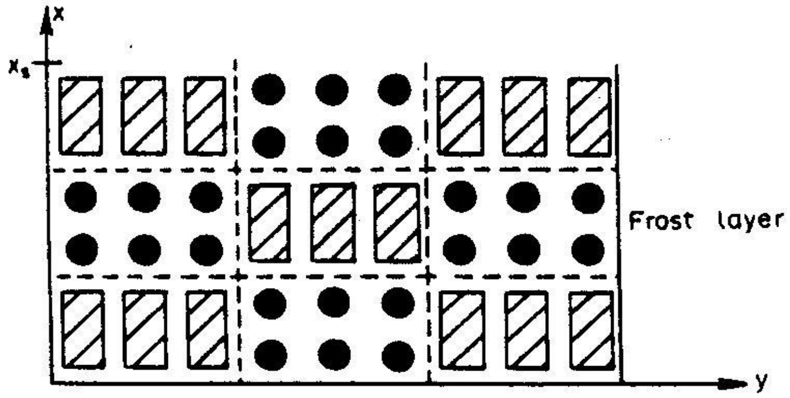


Figure 1. Frost structure model: random mixture of ice cylinders

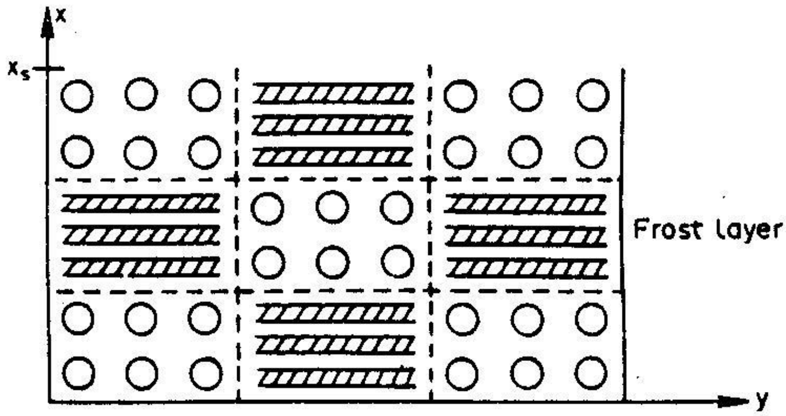


Figure 2. Frost structure model: random mixture of ice planes and bubbles at low porosities or high frost densities

$$k = (k_u + k_l)/2 \tag{7}$$

$$k_u = (1-B)k_b + Bk_c \tag{8}$$

$$k_b = k_i\left[1 - 2B\left(\frac{1-a}{2+a}\right)\right] \Big/ \left[1 + B\left(\frac{1-a}{2+a}\right)\right] \tag{9}$$

$$a = k_{eff\_air}/k_i \tag{10}$$

$$k_c = (1-B)k_i + Bk_{eff\_air} \tag{11}$$

$$k_l = (1-B)k_p + Bk_s \tag{12}$$

$$k_s = k_i\,[3 + 2B(a-1)] \Big/ \left[3 - B\left(\frac{a-1}{a}\right)\right] \tag{13}$$

$$k_p = \frac{k_i k_{eff\_air}}{(1-B)k_{eff\_air} + k_i B} \quad (14)$$

To describe variation of $Q(\tau)$ with time, curves of frost growth should be portrayed. In this paper an approximate way was used. Analogy convectional mass transfer to heat transfer, quantity of frost deposition is gained. Based on frost layer density usually is 0.5 g/cm^3, frost conductivity k can be work out to be about 1.2 $W/m \cdot K$, with equation (7)~(14). Ignore frost specific heat, temperature inside frost layer has a linear distribution. The total calculation processes are given by

$$\alpha = 15.5 \times V_{\max}^{0.578} \quad (15)$$

$$\frac{\alpha}{\alpha_D} = \rho C_p Le^{2/3} \quad (16)$$

$$\dot{m}_s = \alpha_D \left(\rho_{v,\infty} - \rho_{v,s}\right) \quad (17)$$

$$\dot{X} = \dot{m}_s / \rho_f \quad (18)$$

$$k \frac{T_s - T_w}{X} = \alpha(T_\infty - T_s) + \alpha_D \left(\rho_{v,\infty} - \rho_{v,s}\right)\Delta H \quad (19)$$

A calculation results is given as an example. Assume that ambient temperature is –5℃ and relative humidity is 50%, initial evaporating temperature is –15℃, figure 3 gives out curves of frost thickness, surface temperature, heat exchanged and wall temperature.

3 HEAT PUMP RUNNING STATES DURING FROSTING CONDITIONS

To portray variation of $Q(\tau)$, an example was given in this paper. A ZR84KCE-TF5/TFD compressor is chosen. Figure 4 gives out its performance curves.

Based on calculation results above, refer to specific compressor performance curves, we can portray variation curve of heating capacity $Q(\tau)$ with time in figure 5. Figure 5 shows the key factor to design defrosting control method, which is portrayed based on data from Figure 4.

Process from point O to A is initiative step, heating capacity begin to comeback. Process from A to B is steady step. System runs on no frost stage. From B to D is frost step. In this stage, heating capacity and COP deteriorate mainly for air flow decreasing. From D on, evaporator coils badly frosted, no air flow pass through the heat exchanger then. Evaporator transfers heat with free convection method.

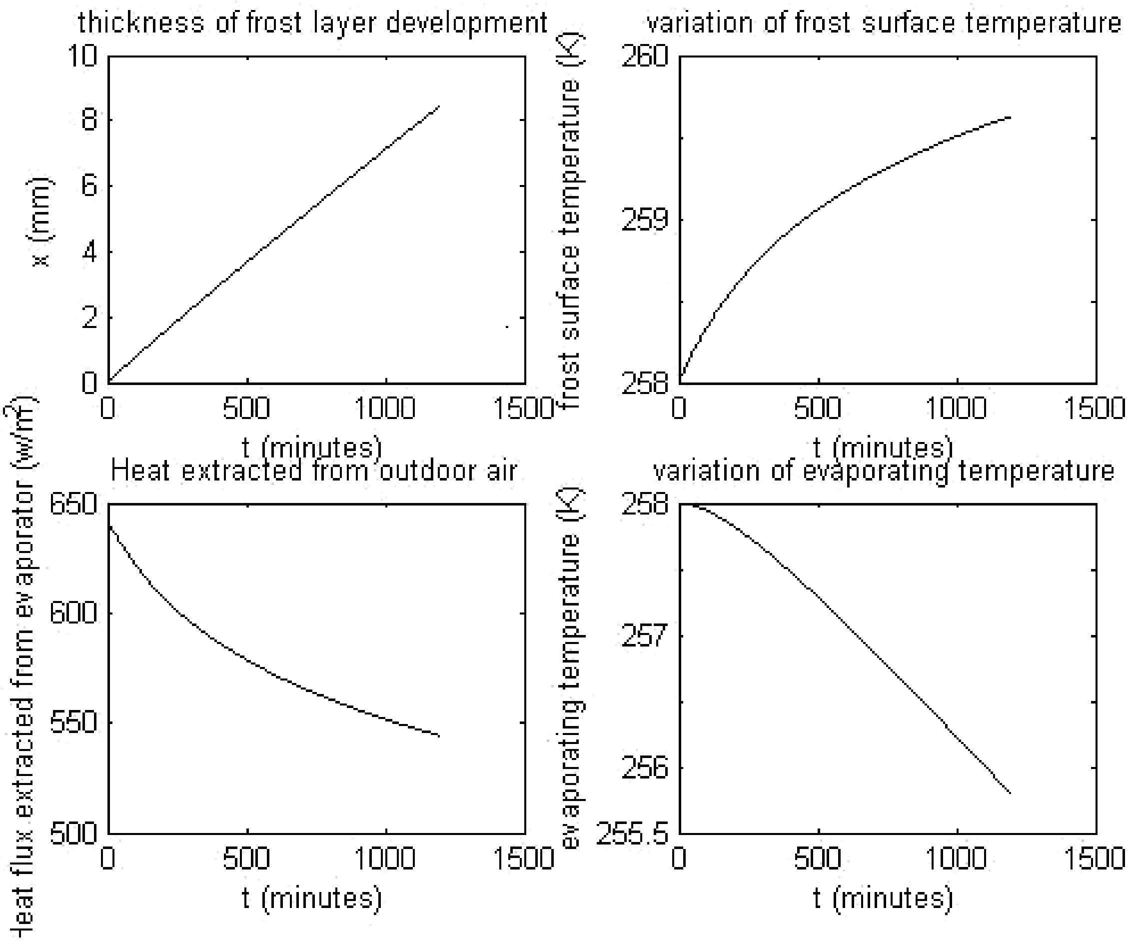
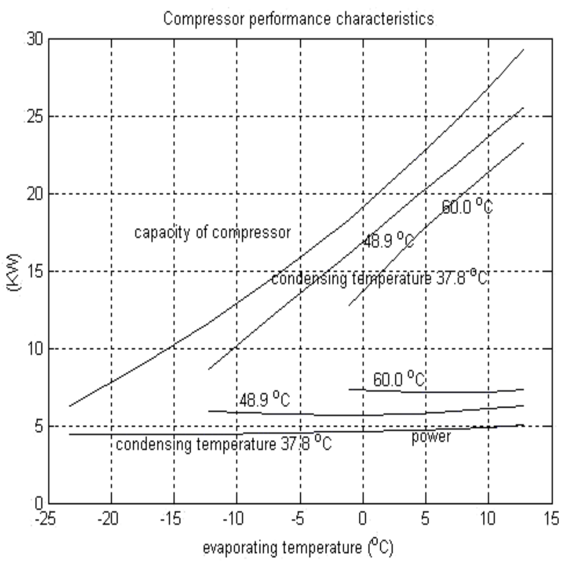


Figure3. Frost layer and Heat transfer variation by time



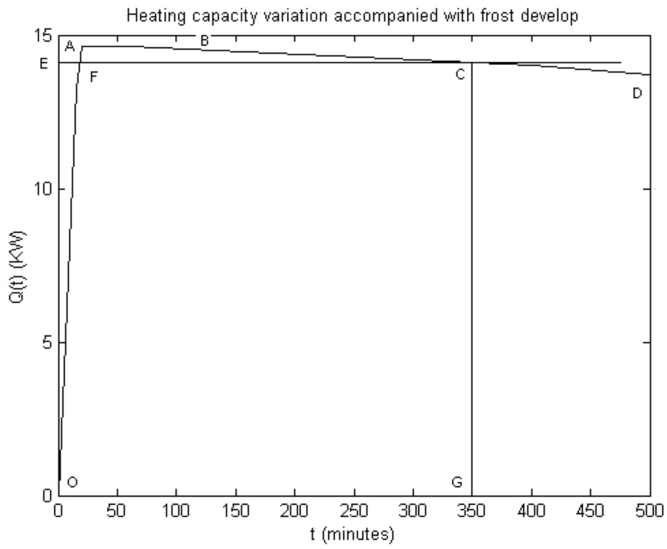


Figure 4. Performance curves of compressor

Figure 5. Heating capacity variation with time

Using this method, heating capacity change with frost develop can be work out under different environment conditions and specific facilities. Heat pump operating points are clearly painted by frost formation process, then it's easier to study defrost control method applied in specific surroundings.

4 DEFROST CONTROL METHOD USING AVERAGE HEATING CAPACITY

Average heating capacity could be a criterion to judge whether defrosting system should be started or not, when defrosting process ends, heating capacity need time to resume. If defroster acts very frequently, heating capacity can not come back to a good level, if defrost action starts too late, frost will develop unboundedly and heating capacity will fall down badly due to frost thermal resistance and flux of air. So heat pump average heating capacity could reach the best level if the defrosting period are appropriate. That means heating capacity will have enough time to resume after defrosting, while frosting-growth process will still be controlled.

Purpose of using heat pump is providing heat, one of the most important performance evaluating indicators is heating capacity besides COP (coefficient of performance). So using average heating capacity as a criterion of defrosts control have instructional significance. To work out the control method, two factors should be confirmed. One is t, time length of heating during one period, the other is t_{def}, defrosting time length.

When heat pump runs under frosting conditions, frost formation can increase conduction thermal resistance, reduce flow passage of air through evaporator and enhance flow resisting force. Finally quantity of air flow receding is induced. Total quantity of exchanged heat falls down, system runs in bad work condition.

Running in specific surroundings, frost development is a function of time τ. $X(\tau)$ is frost layer's thickness, assuming that frost has a uniform distribution on the evaporator surface. $Q(\tau)$ is a function of time, system working state and surrounding conditions, but all of these can be expressed using time. Figure will be used to express relations of $Q(\tau)$ and τ.

During t, heat pump runs in heating state, frost deposits on the evaporator, thickness is $X(t)$, total mass expressed as $m(t)$. Then defrosting actions start up. In t_{def} time, frost should be removed. That means :

$$t_{def} = {m(t) \times L}\big/{q \times \eta} \tag{20}$$

So

$$Q_m = \frac{1}{t + t_{def}} \int_0^t Q(\tau) d\tau = f(t) \tag{21}$$

$Q(\tau)$ is portrayed in figure 5.

Choose a point C, making area of OEFO and FACF is equal. When heating period time exceeds t_G, average heating capacity in heating period t will reduce. And defrosting time t_{def} will be longer too, so total heating capacity must be less.

If a short t is given, defrosting action starts up frequently. When defrosting process ends, heating capacity need time to resume, heating capacity could not come back to a good level. During defrosting work state, house needs auxiliary heat, often in the form of electric resistance heating. If defrost action starts too late, frost will develop unboundedly and heating capacity will fall down badly due to frost thermal resistance and decreased air flux. So heat pump average heating capacity could be bad either t is short or long. An appropriate t can make Q_m to be its maximum. Define t and t_{def} correctly means heating capacity will have enough time to resume after defrosting, while frosting-growth process will still be controlled.

5 DISCUSSION

In engineering application, when defrosting action ends, if dehydration do not complete, water will cling to evaporator surface. Then heat pumps go into heating operating mode. Evaporator surface temperature will be low enough to make the water ice up. After several periods, things go worse and worse. This is dangerous for system’s performance. So defrost method must consider how to contain evaporator surface at no-ice state. A long dehydration periodic can be embodied in control project to ensure after a few defrosting periods, the last defrosting cycle can dehumidify the evaporator surface absolutely.

6 CONCLUSION

Based on the above analysis, conclusions can be drawn as below:

(1) COP decline caused by frost develop do not only due to conduction thermal resistance or rate of air flow change, but also due to decreasing air flow induced by increasing flow resisting force. Compared the two factors, air flow variable seems the main trigger.

(2) Average heating capacity control method can be applied. Aim to specific whether record in specific area, numerical simulation can give out rough automation factors. Application of programmable timer and programmable controller can ensure heat pump’s COP keep in a good level.

ACKNOWLEDGEMENTS

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NOMERCLATURE

Q	heat pump heating capacity
Q_m	maximum heating capacity
τ	time
t	time length of heating during one period
t_{def}	time length of defrosting during one period
k_a	thermal conductivity of air
k_i	thermal conductivity of ice
k_v	water vapor thermal conductivity
k	thermal conductivity of frost
k_e	effective thermal conductivity of the combined heat conductivity proportions of air and ice
D	ordinary diffusion coefficient, $D = \frac{2.256}{P_t}\left(\frac{T}{256}\right)^{1.81}$ [6]
B	frost porosity, $(\rho_i - \rho_f)/(\rho_i - \rho_a)$
P_t	ambient pressure
P_v	water vapor pressure
R_v	water vapor gas constant
T	frost temperature
L_s	latent heat of ice sublimation
α	convection heat transfer coefficient
α_D	convection mass transfer coefficient
ρ_a	air density
C_p	effective specific heat of forced air flow in frost layer
Le	Lewis Number
m_s	mass velocity of frost formation

ρ_f frost density

$\rho_{v,s}$ vapor density on frost surface

$\rho_{v,\infty}$ ambient vapor density

X thickness of frost layer

T_∞ ambient air temperature

T_s frost surface temperature

T_w evaporator surface temperature

L frost melt latent heat

q defrosting heat flux

η defrosting efficiency

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