Numerical Analysis of the Channel Wheel Fresh Air Ventilator Under Frosting Conditions

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Abstract: As new equipment, the channel wheel fresh air ventilator has become increasingly popular in recent years. However, when such equipment is operated under low ambient temperature in the freezing area in winter, the formation of frost on the outdoor waste air surface becomes problematic, leading to the degradation of the channel wheel fresh air ventilator's performance or even the shutdown of equipment. Therefore, it is necessary to have a detailed investigation on the operational characteristics of the channel wheel fresh air ventilator under frosting in order to guide its application.

This paper first reports on the development of a detailed model for the channel wheel heat exchanger, which is the core part of the channel wheel fresh air ventilator under frosting conditions. The model developed, first seen in open literature, consists of a frosting sub-model and a channel wheel heat exchanger sub-model. This is followed by reporting an evaluation of the operational characteristics of a frosted channel wheel heat exchanger under different ambient conditions using the model developed. These include frost formation on the surface of the channel wheel heat exchanger, and impacts on the operational performance of the channel wheel fresh air ventilator. Furthermore, the interval of defrosting is obtained, which provides the basis for the adoption of effective defrosting measures, and thus increasing the channel wheel fresh air ventilator's energy efficiency and operating reliability.

Key words: channels wheel fresh air ventilator; frosting; numerical calculation

1. INTRODUCTION

Channels wheel fresh air ventilator, a new ventilating and energy recovery equipment for building heating, ventilation and air conditioning (HVAC) systems, becomes increasingly popular in commercial districts in major cities in China[1]. However, when such a equipment is operated under low ambient temperature in the freezing area in winter, the formation of frost on the exit of outdoorwaste air surface becomes problematic, leading to the degradation of the fresh air ventilator's performance or even the shutdown of a equipment. Therefore, it is necessary to have a detailed investigation on the operational characteristic of the channels wheel fresh air ventilator under frosting in order to guide its application.

Currently, a substantial amount of literature is available on frost properties and growth[2-4]. A large number of experimental and theoretical investigations on frost properties, the mechanism of frost growth, and the heat transfer involved in frost growth have been performed and reported for simple geometry heat exchangers, such as tubule[5], annuli[6], cylinders[7,8], flat plates and parallel plates[9-11]. However, for heat exchangers with a more complex geometry, such as channels wheel heat exchanger, unfortunately, there is little knowledge in open literature about this new-style equipment (China patent number: ZL 03 1 23754.1).

This paper firstly reports on the development of a detailed model for the channels wheel heatexchanger which is the core part of the channels wheel fresh air ventilator under frosting condition.The model developed, firstly seen in open literature, consists of a frosting sub-model and a channels wheel heat exchanger sub-model. This is followed byreporting an evaluation of the operational characteristics of a frosted channels wheel heat exchanger under different ambient conditions using the model developed. These include frost formationon the surface of channels wheel heat exchanger, andtheir impacts on the operational performance of channels wheel fresh air ventilator. Based on the evaluation, the obtaining the defrosting time interval which provide the basis for the adoption of effective defrosting measure, thus, increasing the channels wheel fresh air ventilator's energyefficiency and operating reliability.

2. MODEL DEVELOPMENT

The model can be divided into two sub-models, i.e., a frosting sub-model and a channels wheelheat exchanger sub-model.

In developing the model, the followingassumptions have been made:

(1) Both the cold air in channel of fresh air and the warm air in channel of waste air are not compressed and in thermal equilibrium.

(2) The mass flow in both channel of fresh air channel of waste air is equivalency.

(3) The mass flow in both channel of fresh air andchannel of waste air are constants.

(4) The mass flow in one dimension along the heat exchanger channel axis.

(5) The thermal conductivity of entire exchanger surface is constant.

(6) Both kinetic energy and potential energy are neglected in energy conservation equation.

(7) The heat transfer by radiation between the moist air and frost is negligible.

(8) The problem of frosting is assumed to be of the nature of quasi-steady-state.

(9) Frost is formed layer by layer. The density andthermal conductivity of each frost layer is entirely dependent on the interface temperaturebetween air and frost.

3. THE FROSTING SUB-MODEL

The frost accumulation rate is determined by a loss of water vapor in the air, as water vapor condenses on the exit of outdoor waste air surface of the channels wheel fresh air ventilator. It is expressed by:

$$\dot{m}_{fr} = \dot{m}\,(d_{1in} - d_{1out}) \tag{1}$$

Where:

$\dot{m}_{fr}$—frost accumulation rate (kg s^{-1});

$\dot{m}$—mass flow（kg s^{-1}）;

d_{1in}—moisture content of indoor waste air (kg water vapor kg^{-1} dry air);

d_{1out}—moisture content of outdoor waste air (kg water vapor kg^{-1} dry air).

Based on the physical change of frost in term of both density and thickness, Equation (1) can berewritten as:

$$\dot{m}_{\mathrm{fr}} = \dot{m}_{\delta} + \dot{m}_{\rho} \tag{2}$$

Where:

$\dot{m}_{\delta}$—frost thickness accumulation rate (kg s^{-1});

$\dot{m}_{\rho}$—frost density accumulation rate (kg s^{-1}).

By this arrangement, the total amount of watervapor approaching a frost layer for solidification is now separated into two terms. The first term $\dot{m}_\delta$ represents the amount of water vapor condensed contributing to the increase of frost thickness, andthesecond term $\dot{m}_\rho$ to the increase of frost density. Yang Yao[12]] used the following expression for themass of water vapor diffusing through a porous frost layer:

$$\dot{m}_\rho = \frac{\dot{Q}}{i_{sv} + \dfrac{\lambda_{fr} R T_s^2 (v_v - v_i)}{D_s [i_{sv} - P_V (v_v - v_i)](1 - \dfrac{\rho_{fr}}{\rho i})/[1 + (\dfrac{\rho_{fr}}{\rho_i})^{0.5}]}} \quad (3)$$

Where:

$D_s = 2.302\ (0.98\times10^5/P_a)\ (T_s/256)^{1.81} \times 10^{-5}$

$\dot{Q}$ —overall energy transfer (W);

i_{sv} —sublimation potential heat of water vapor (J kg^{-1});

λ_{fr} —thermal conductivity of frost (W m^{-1} K^{-1});

R—gas constant of water vapor (461.9J kg^{-1} K^{-1});

T_s —temperature of the frost surface (K);

P_V —fractional pressure of vapor (Pa);

v_v —water vapor specific volume of frost (m^3 kg^{-1});

v_i —ice specific volume of frost (m^3 kg^{-1});

D_s —diffusivity of water vapor in air (m^2 s^{-1});

ρ_{fr} —frost density (kg m^{-3});

ρ_i —ice density (kg m^{-3});

The frost density ρ_{fr}, in Equation (3) is affected by the surface temperature of the heat exchanger, ambient air temperature and relative humidity, and frosting time. The longer the frosting time is, the greater the frost density will be. When performing simulation, an initial density would have to be assumed, and for each time step the change offrost density and frost thickness may be determinedby:

$$\Delta\delta_{fr} = (\dot{m}_\delta / A_t\, \rho_{fr})\,\Delta t \quad (4)$$

$$\Delta\rho_{fr} = (\dot{m}_\rho / A_t\, \delta_{fr})\,\Delta t \quad (5)$$

Where:

Δ t—time step (s);

δ_{fr} —frost thickness (m);

$\Delta\delta_{fr}$ —the change of frost thickness in time step (m s^{-1});

$\Delta\rho_{fr}$ —the change of frost density in time step (m s^{-1});

A_t —total area (m^2).

These increments are added to the values of frost density and thickness, respectively, at the previoustime step in order to obtain the density and thickness at the present computing time step. The thermalconductivity of frost was evaluated by[13]:

$$\lambda_{fr} = 0.001202\,\rho_{fr}^{0.963} \quad (6)$$

4. THE CHANNELS WHEEL HEAT EXCHANGER SUB-MODEL

The channels wheel heat exchanger is the core part of a channels wheel fresh air ventilator, with its detailed specifications shown in Table 1 and Figs.1.The channel wheel heat exchanger is made up of 36 channels which are made of corrugated flexible aluminum foil as shown in Fig.1 and were embedded radially into the shaft which driven by motor every

10° around the azimuth. The channels wheel heat exchanger, with complex configuration, revolved in high speed when it in operating condition. Thus, this paper take the ε-NTU method to gained thermal conductivity between the two channelsinstead of the traditional Dittus-Boelter methodwhich could not fit in this condition.

Tab. 1.Geometry details of the heat exchangcoil used in simulation

Item	Name	Material/number/ description
1	Type	LY-600GM
2	Voltage	220V
3	Power	60W
4	Capacity of fresh air	270m3/h
5	Thermal efficiency	≥69%
6	Weight	20kg
7	Filtration level	G3
8	Number of channel	36
9	Channel material	Aluminum
10	Length of channels wheel exchanger	720mm
11	Diameter of channels wheel exchanger	200mm

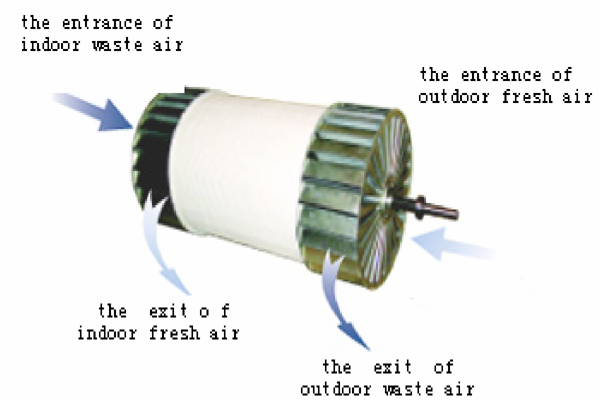

Fig.1. Sketch map of principle of channels wheel fresh air ventilator

In adverse current condition, ε-NTU is expressed by:

$$\varepsilon=\frac{1-\exp[(-NTU)(1-\frac{(Mc)_{\min}}{(Mc)_{\max}})]}{1-\frac{(Mc)_{\min}}{(Mc)_{\max}}\exp[(-NTU)(1-\frac{(Mc)_{\min}}{(Mc)_{\max}})]} \quad (7)$$

Where:

ε —heat exchanger efficiency (%);

NTU —number of transfer equipment;

$(Mc)_{\min}$ —the minimum of liquid specific heat (W^{-1});

$(Mc)_{\max}$ —the maximum of liquid specific heat (W □$^{-1}$).

According to the assumptions (2), we can deduce:

$$1-\frac{(Mc)_{\min}}{(Mc)_{\max}}\to 0 \quad (8)$$

Equation (7) can be simplified by principle ofL'Hospital:

$$\varepsilon=\frac{NTU}{1+NTU} \quad (9)$$

$$K=\frac{\varepsilon(Mc)_{\min}}{(1-\varepsilon)A_c} \quad (10)$$

Where:

K—thermal conductivity in frostless condition ($Wm^{-2}K^{-1}$);

A_c—the area between two adjacent channels' wall (m^2).

Neglecting the aluminum foil's thermal resistance, the thermal conductivity in frostless condition can be expressed by:

$$K=\frac{1}{\frac{1}{\alpha_1}+\frac{1}{\alpha_2}} \quad (11)$$

Where:

α_1—the heat transfer coefficient of indoor waste air side (Wm^{-2}K^{-1});

α_2—the heat transfer coefficient of outdoor fresh air side (Wm^{-2}K^{-1}).

The general heat transfer coefficient of indoor waste air side in frost condition can be expressed by:

$$\frac{1}{\alpha_1'} = \frac{1}{\alpha_{1fr}} + \frac{\delta_{fr}}{\lambda_{fr}} \tag{12}$$

Where:

α_1' —the general heat transfer coefficient of indoor waste air side in frost condition (Wm^{-2}K^{-1});

α_{1fr}—the heat transfer coefficient of frost surface (Wm^{-2}K^{-1}).

For the increasing of frost surface's roughness, the heat transfer coefficient between the moist air and frost is evaluated by:

$$\alpha_{1fr} = (1.2\sim1.3)\alpha_1 \xi \tag{13}$$

In this paper, we take $\alpha_{1fr} = 1.25\alpha_1 \xi$, so the general thermal conductivity in frost condition in the surface of exchanger's thermal conductivity can be expressed by:

$$K_{fr} = \frac{1}{\dfrac{1}{1.25\alpha_1\xi} + \dfrac{\delta_{fr}}{\lambda_{fr}} + \dfrac{1}{\alpha_2}} \tag{14}$$

Where:

K_{fr} —thermal conductivity in frost condition (Wm^{-2}K^{-1});

h_{fr}—solidification potential heat of water vapor in the indoor waste air side (J kg^{-1}).

ξ—liberation of water factor.

$$\xi = \frac{h_1' - h_1''}{C_p(t_1' - t_1'')} \tag{15}$$

Applying energy conservation in indoor waste air side yields:

$$Q_1 = \dot{m}(h_1' - h_1'' + h_{fr}) \tag{16}$$

Q_1—exothermic quantity of indoor waste air side (W);

h_1' —the enthalpy of indoor waste air (J kg^{-1});

h_1''—the enthalpy of outdoor waste air (J kg^{-1});

C_p—specific heat at constant pressure for dry air (J kg^{-1} □$^{-1}$).

Applying energy conservation in outdoor fresh air side yields:

$$Q_2 = \dot{m}(h_2'' - h_2') \tag{17}$$

Q_2—endothermic quantity of outdoor fresh air side (W);

h_2''—the enthalpy of indoor fresh air (J kg^{-1});

h_2'—the enthalpy of outdoor fresh air (J kg^{-1}).

Applying heat transfer equation between indoor waste air side and outdoor fresh air side yields:

$$Q_3 = K_{fr} A_c \Delta t_m \tag{18}$$

Q_3—quantity of heat passed between two adjacent channels' wall (W);

Δt_m—logarithmic mean temperature difference (□).

Applying humidity conservation and according to the assumptions (3) yields:

$$P_{q.b}(t_{w1})\,\varphi_{tw1} = P_{q.b}(t_1'')\varphi_{1out} \tag{19}$$

$t_1^{''}$—temperature of the exist of outdoor waste air (□);

t_{w1}—temperature of the exist of outdoor waste air wall surface (□);

$P_{q.b}(t_1^{''})$—saturated steam particle pressure of the $t_1^{''}$ (Pa);

$P_{q.b}(t_{w1})$—saturated steam particle pressure of the t_{w1} (Pa);

φ_{tw1}—relative humidity of air near the exist of outdoor waste air wall surface (%);

φ_{1out}—relative humidity of outdoor waste air (%).

In frost condition, φ_{tw1}=100%. Equation (19) can be rewritten as:

$$P_{q.b}(t_{w1}) = P_{q.b}(t_1^{''})\varphi_{1out} \qquad (20)$$

5. MODEL VALIDATION

Channels wheel fresh air ventilator, with its detailed specifications shown in Table 1 and Figs.1,be used for carrying out simulation using the governing equations discussed above.

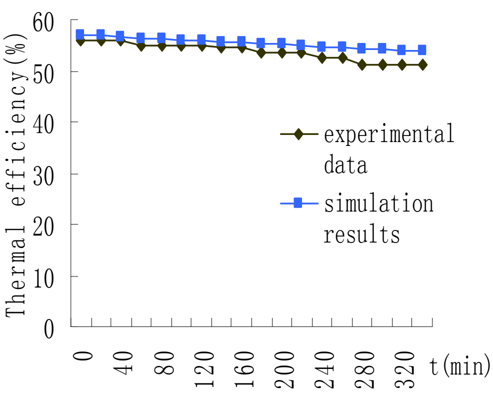

Fig.2. Comparison between the simulation and previous experimental result of thermal efficiency of channels wheel heat exchanger vs. time

Fig.2 presents an example of the comparisonsbetween the simulation results using the model developed in this paper and the previous published experimental results on thermal efficiency in condition K. The simulated operating indoor air temperature was 20□, indoor air relative humidity was 30%, outdoor air temperature was -15 □ and outdoor air relative humidity was 40%. It is seen that the simulated results using the current model agreed well with the experimental data[14], with a maximum error of 6.51% at 340 min.

6. OPERATIONAL CHARACTERRISTICS EVALUATION

The channels wheel fresh air ventilator underfrosting used in simulation for operational characteristics evaluation is the same as that used for model validation. The operating conditions used in simulations are given in Table 2.

Examples of simulation results for equipmentoperational characteristics under frosting such as frost accumulation, thickness under different operating conditions, and for the impacts of frosting on the operational performance of an equipment under frosting, such as the quantity of heat exchange are presented in Figs.3–11.

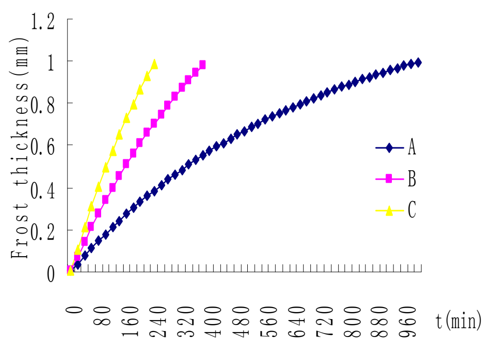

Fig.3. Frost thickness vs. time under three different three sets of indoor air relative humidity

Figs.3 presents the simulated results for frost thickness against time and Fig.4 simulated the frost mass accumulation against time under Conditions: A–C, respectively. As can be seen in Fig.3, the frost thick-ness increases with time elapsing; the higher the indoor air relative humidity, the faster the increase of frost thickness. As shown in Fig.4, more frost will be formed on cold surface with the increase of time; a higher indoor air relative humidity would result in more frost accumulated.

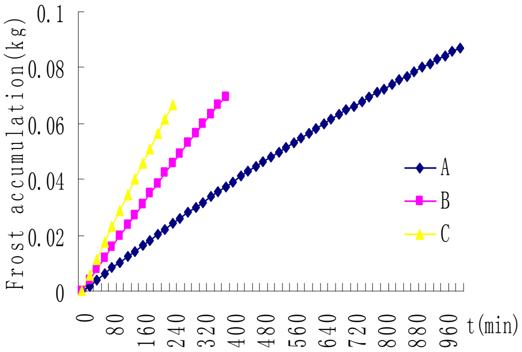

Fig.4. Frost mass accumulation vs. time under different sets of indoor air relative humidity

These are expected,since the driving potential of mass transfer between air and cold exchanger surface is greater at higher frosting, such as the quantity of heat exchanger are indoor air humidity. This helps explain why the fresh conditions, and for the impacts of frosting on the air ventilator frosts severely at high indoor relativeoperational performance of an equipment under humidity.

Fig.5 shows the simulated total heat transfer of the channels wheel heat exchanger against time underconditions A–C, respectively. It is found that the total heat transfer will be reduced with the increase offrost thickness; the higher the indoor air relativehumidity, the faster the reduction of the total heat transfer. This may be explained by the fact that athicker frost formation on heat exchanger surface leads to a greater reduction of airflow rate and a higher heat transfer resistance, thus resulting in a greater reduction of the heat exchange coefficient, leading to a reduced total heat transfer, at a higher indoor air relative humidity. As the time elapses, with more frost accumulated on the surface of a heat exchanger, the heat exchange coefficient may be significantly reduced, and heat transfer efficiency is severely deteriorated, so that the total heat transfer is decreased.

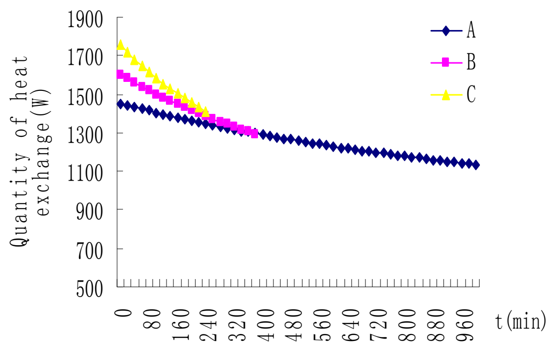

Fig.5. Quantity of heat exchange vs. time under three different sets of indoor air relative humidity

Tab..2 Calculation condition

condition	Indoor air temperature （□）	Indoor air relative humidity （%）	Outdoor air temperature （□）	Outdoor air relative humidity（%）
A	18	30%	-20	50%
B	18	50%	-20	50%
C	18	70%	-20	50%
D	18	40%	-10	50%
E	18	40%	-22	50%
F	18	40%	-35	50%
G	12	40%	-20	50%
H	16	40%	-20	50%
I	20	40%	-20	50%
J	24	40%	-20	50%

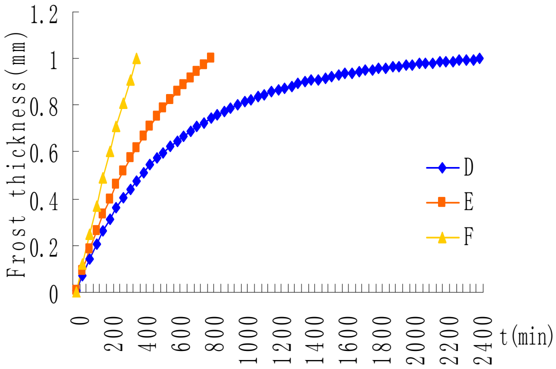

Fig.6. Frost thickness vs. time under threedifferent outdoor air temperature

Figs.6-8 illustrate the simulated frost thickness, frost mass accumulation and total heat transfer of thechannels wheel heat exchanger against time, respectively, under Conditions D-F. It is seen in Fig.6 that at lower outdoor air temperature, the faster the increase of frost thickness. Fig.6 shows that at -35□, it takes 360 min for frost thickness to reach to 1.0 mm on heat exchanger surface. However, at -22□ and -10□, it would take 800 min and 2400 min to reach the same thickness. Therefore, it may be concluded that at constant relative humidity and indoor air temperature, the lower the outdoor air temperature, the faster the increase of frost thickness. This is obvious because at the lower the outdoor air temperature, the lower the decrease heat exchanger surface's temperature, more water vapor is accumulated the colder heat exchanger surface.

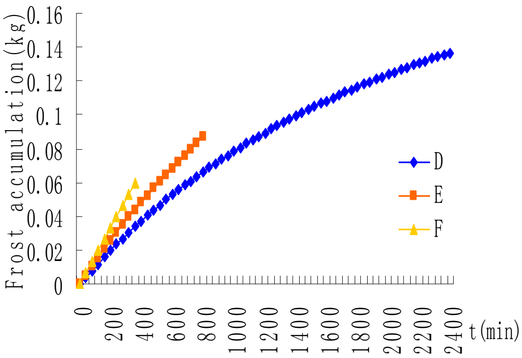

Fig.7. Frost mass accumulation vs. time underthree different outdoor air temperature

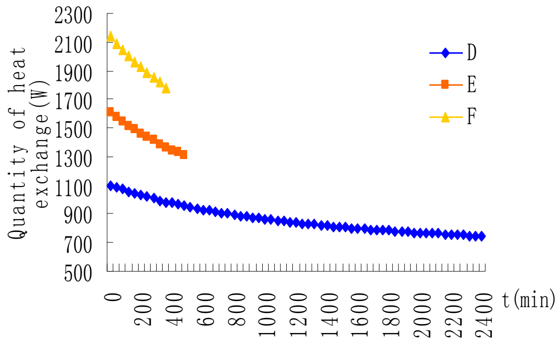

Fig.8. Quantity of heat exchange vs. time underthree different outdoor air temperature

By the same token, as can be seen in Fig.7, it can be deduced that at constant relative humidity andindoor air temperature, the lower the outdoor airtemperature, the more frost accumulate. As alsoshown in Fig.8, the total heat transfer of the channels wheel heat exchanger decrease with time going; the lower the outdoor air temperature, the faster the decrease of the total heat transfer of the channels wheel heat exchanger.

Fig.9-11 presents the simulated frost thickness, frost mass accumulation and total heat transfer of therespectively, under conditions G-J. As can be seen inFig.9, the frost thickness increases with time elapsing; channels wheel heat exchanger against time, the higher the indoor air temperature, the faster the

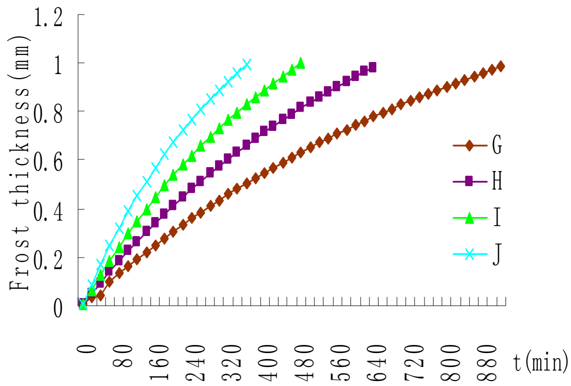

Fig.9. Frost thickness vs. time under different indoor air temperature

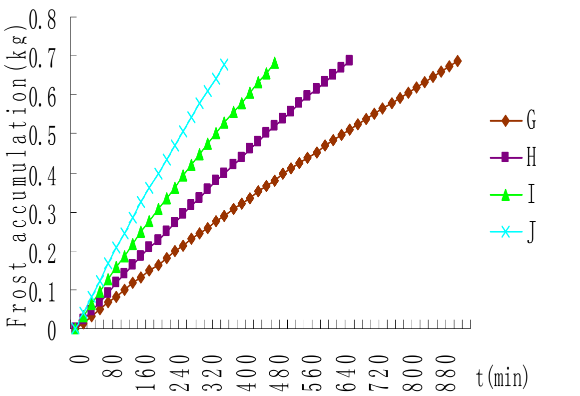

Fig.10. Frost mass accumulation vs. time under different indoor air temperature

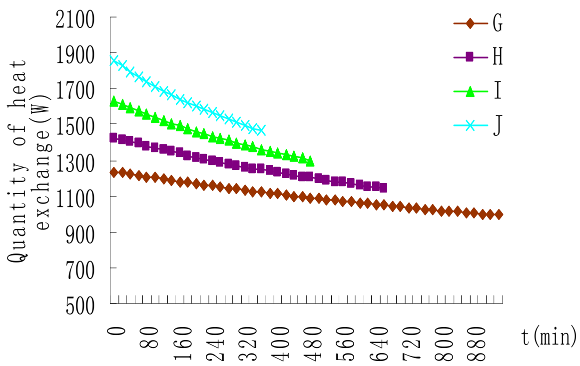

Fig.11. Quantity of heat exchange vs. time under increase of frost thickness.

As shown in Fig.10, morefrost will be formed on cold surface with the increaseof time; a higher indoor air temperature would resultin more frost accumulated. Furthermore, as can berevealed in Fig.11, the total heat transfer of thechannels wheel heat exchanger decrease with time going; the higher the indoor air temperature, thefaster the decrease of the total heat transfer of thechannels wheel heat exchanger. Therefore, it may be deduced that at constant indoor air relative humidityand outdoor air temperature, the higher the indoor

air temperature, the faster the increase of frost thickness and the more frost be accumulated. This is obvious because indoor air with a higher temperature at a constant relative humidity contains more water vapor.

As the time elapses, with more frost accumulated on the surface of a heat exchanger, if the frost thickness is increased to a certain value (e.g.1.0 mm), defrosting will be required. A by-pass pipe, which can merge the circulating air and heating circulating air, is set to defrost in our laboratory. Experiments prove the method is feasible and effective for the channels wheel fresh air ventilator under frosting condition[15].

7. CONCLUSIONS

A model of the channels wheel heat exchanger under frosting has been developed and validatedusing previous experimental data in our laboratory.The operational characteristics of the channels wheel fresh air ventilator under frosting and their effects onequipment's performance have been investigated byusing the model. The following conclusions may be drawn:

- The mathematical model developed for a channelswheel heat exchanger under frosting is derived from energy and humidity conservations, which is firstlyseen in open literature. Therefore, it can be applied to channels wheel types of heat exchangers havingdifferent parameters.
- The frost formation/growth varies with ambientconditions. At the indoor air temperature and the outdoor air temperature are constant, the higher the indoor air relative humidity, the more serious the frost formation, and the shorter the defrosting interval. At the indoor air relative humidity and the indoor air temperature are constants, the frost formation is more serious at a low outdoor air temperature than that at a high outdoor air temperature. Furthermore, at the indoor air relative humidity and the outdoor air temperature are constants, the higher the indoor air temperature, the more water vapor contains, therefore, the more serious the frost formation.
- With the increase of frost accumulation, the totalheat transfer of channels wheel heat exchanger will decrease, and the defrosting time interval is evaluated.

All these are expected to be useful in finding key points which influence the frost formation and obtaining the defrosting time interval which provide the basis for the adoption of effective defrosting measure, thus, increasing the channels wheel fresh air ventilator's energy efficiency and operating reliability.

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