SIMULATION ANALYSIS AND OPTIMIZATION RESEARCH ON COOLING AND DEHUMIDIFYING EFFECTS FOR EVAPORATOR

by

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The improvement performance of refrigerating dehumidification system was theoretically discusses based on a dehumidification model. The influence of evaporator inlet wind speed, dry bulb temperature and relative humidity on dehumidification were analysed by the model. The results show that, when inlet air temperature and humidity were kept constant, the dehumidification capacity increased first and then decreased with increase of the wind speed. When the moisture content and the wind speed of the inlet air were kept constant, the dehumidification capacity gradually decreased with increase of the inlet air dry bulb temperature. The inlet air dry bulb temperature was between 21-36 °C and the relative humidity was between 40% and 85%, the difference between the inlet air wet bulb temperature and the evaporation temperature at the optimum COP was about 10 °C. There was a nearly linear relationship between the corresponding evaporation temperature at the optimal COP and the evaporation temperature with the maximum dehumidification capacity, compared with the test value, the error was less than 10%.

Key words: *dehumidification capacity, simulation, evaporator, linear relationship*

Introduction

One of the major concerns of our days is to reduce energy consumption and the environmental impact of cities and buildings. [1]. In developed countries the HVAC systems consume around a third of the total energy consumption of the whole society [2]. Energy consumption of air conditioning systems is mostly associated with the summer season. In summer, it is necessary to dehumidification for air indoor air relative humidity is one important factor effecting on human thermal comfort. For controlling the air relative humidity, comparted with dehumidification, humidification is relatively simple. Air dehumidification system is complex, and high energy consummation. In the Yangtze River area of China, its climate is characterized by high temperatures in summer and high humidity throughout the year. The annual average relative humidity of major cities in the region is between 75% and 80%, sometimes up to 95-100%. High air humidity not only affects people's thermal comfort, but also have adverse effect on interior hygiene, human health, furniture and industrial equipment life [3]. In the southern rainy area of China, the temperature is low, but the humidity is high. In this case, the demand for dehumidification is often greater than the demand for refrigeration. For a long time,

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the indoor environment in the area is the worst in the country [4]. Therefore, dehumidification has become an important issue in the Yangtze river basin.

The methods usually used in air dehumidification include cooling-dehumidifying, adsorption dehumidification and absorption dehumidification. In addition, there are several new dehumidification methods such as membrane dehumidification, heat pump dehumidification, electrochemical dehumidification, HVAC dehumidification, *etc*. [5-8]. The refrigeration dehumidifier uses a refrigerating machine as the cold source, direct evaporative cooler as cooling equipment. The mechanism of refrigeration-dehumidifying is to cool the air below its dew point temperature, and thus make the water out from the air and reduce the absolute moisture content in the air. Due to its low energy consumption, simple operation and easy control, it has been widely used [9]. But there is one shortage in the actual use of practical engineering: high energy consumption. So, how to reduce the energy consumption of the refrigerating dehumidifier making it more efficient to operate is the key to research. Yuan *et al.* [10] and Geng *et al*. [11] added a more efficient plate-fin heat exchanger between the evaporator and the condenser, the wet air entering the dehumidifier was pre-cooled through the plate heat exchanger then entered into the evaporator for further cooling and dehumidification. From this the evaporator load was reduced and the dehumidification efficiency was effectively improved. Wan *et al*. [12] analyzed the influence of inlet air volume on cooling capacity, dehumidification capacity and unit input power dehumidification capacity, found that there was an optimal air volume to maximize the unit input power dehumidification capacity. Wang and Wang [13] analyzed the direction of frozen dehumidification energy recovery and proposed a total heat recovery refrigerating dehumidifier with a two-phase closed thermosiphon and an air water surface heat exchanger as a component, results showed that the system had greater energy saving potential than ordinary dehumidifiers. Zhao [14] established a system simulation program, and explored the effects of evaporator temperature and condenser inlet temperature and circulating air volume on the system impact on performance. The conclusion showed that with the improvement of the heat exchange conditions of the heat exchanger (the ambient temperature was close to the phase transition temperature of the refrigerant and the circulating air volume increases), the overall performance of the air conditioning system (refrigeration capacity, COP) wound be improved. There was a critical air volume for the performance improvement. The system performance improvement effect of increasing the circulating air volume after the critical air volume was not obvious. In Guo *et al*. [15], the 1-D simulation model of an automotive air conditioning system was built based on the structure parameters and respective test performance of the compressor, condenser, evaporator and thermal expansion valve. The model was verified by test data. The influences of temperature, humidity, air-flow rate and compressor speed were studied using design of experiment method. The key influencing factors of cooling capacity, COP and discharge pressure were determined. The effect of three controllable factors including compressor speed, evaporator air-flow rate and condenser air-flow rate were analyzed.

In this paper, through simulation and experimental verification methods, arrived the air-flow at the maximum dehumidification volume and the temperature difference range between the wet bulb temperature and the evaporation temperature at the maximum cooling capacity COP. The relationship between the difference between the evaporation temperature at the maximum COP and the evaporation temperature at the maximum dehumidification amount under different dry bulb temperatures is also given.

To summarize, not enough has been done on the research of the relationship between evaporation temperature and dehumidification capacity. This paper established a mathematical model of the refrigerating dehumidification system. The influence of various inlet parameters, such as inlet temperature, humidity, air volume on the dehumidification capacity of the system were analyzed, the relationship between the evaporation temperature at the optimal COP and the evaporation temperature at the maximum dehumidification capacity were researched. This can provide theoretical guidance on design of the refrigerating dehumidification system.

System working principle and theoretical model

System working principle

The refrigerating dehumidification system mainly consists of an evaporator, a gas-liquid separator, a compressor, a condenser and a throttle valve. Figure 1 shows the system principle diagram using R134a as the working fluid. When the system works, the working substance in the evaporator absorbs the heat of the air. The low temperature and low pressure two-phase working fluid from the evaporator is separated by the gas-liquid separator and then enters into the compressor. The high temperature and high pressure gas coming out from the compressor enters the condenser to condense and heat is released, the working substance discharges the heat into the outdoor environment and becomes a low temperature and high pressure liquid. After the working substance is throttled and depressurized in the throttle valve. It becomes a low temperature low pressure working substance and then enters the evaporator for the next cycle.

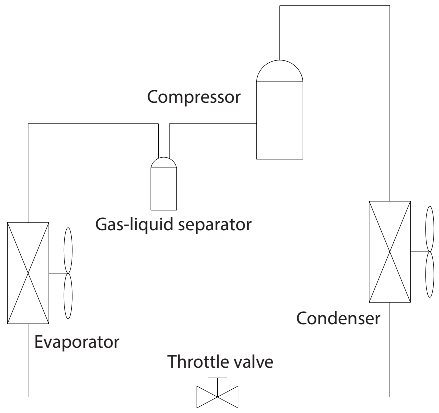

Figure 1. Principles of refrigerating dehumidification system

Evaporator model

The following assumptions are made in the evaporator model:
– Adopt uniform phase model.
– The physical properties of the air side and refrigerant side along the tube length are the same, regardless of the tube wall thermal resistance.
– Refrigerant pressure drop was ignored.

Refrigerant side model

– Single-phase region:

The convective heat transfer coefficient of the superheated region in the evaporator is calculated by the Dittus-Boelter [16]:

$$\mathrm{Nu} = 0.23\mathrm{Re}^{0.8}\mathrm{Pr}^{0.4} \tag{1}$$

where Re is refrigerant gas Reynolds number, and Pr – refrigerant gas Prandtl number.

– Two-phase region:

The convective heat transfer coefficient of the two-phase region in the evaporator is calculated by the Kandlikar proposed universal correlation of boiling in refrigerant tubes:

$$\frac{\alpha_{\mathrm{refr}}}{\alpha_{\mathrm{refr.l}}} = C_1\left(C_0\right)^{C_2}\left(25F_{\mathrm{rl}}\right)^{C_5} + C_3\left(B_0\right)^{C_4} F_{\mathrm{fl}} \tag{2}$$

$$\alpha_{\mathrm{ref.l}} = 0.023\,\mathrm{Re}_{\mathrm{l}}^{0.8}\,\mathrm{Pr}^{0.4}\,\frac{\lambda_1}{d_1} \tag{3}$$

$$\mathrm{Re}_\mathrm{l} = \frac{G(1-X)d_\mathrm{l}}{\mu_\mathrm{l}} \tag{4}$$

$$C_0 = \left(\frac{1-X}{X}\right)^{0.8}\left(\frac{\rho_\mathrm{g}}{\rho_\mathrm{l}}\right)^{0.5} \tag{5}$$

$$F_\mathrm{fl} = \frac{\left[G(1-X)\right]^2}{\mathrm{g}\rho_\mathrm{l}^2 d_\mathrm{l}} \tag{6}$$

$$B_0 = \frac{\psi}{r_\mathrm{refl}G} \tag{7}$$

Air side model

The calculation of heat transfer coefficient on the air side [16]:

$$\mathrm{Nu} = 0.982\,\mathrm{Re}^{0.424}\left(\frac{s}{e}\right)^{-0.0887}\left(\frac{Ns_1}{e}\right)^{-0.159} \tag{8}$$

$$h_2 = \xi h_1 \tag{9}$$

The calculation of dehumidification coefficient on wet surface [17]:

$$\xi = 1 + \frac{\gamma_0 + c_{p,\mathrm{v}}t_{1,2} - c_\mathrm{w}t_\mathrm{w}}{c_{p,\mathrm{m}}}\frac{d_{1,2} - d_\mathrm{w}}{t_{1,2} - t_\mathrm{w}} \tag{10}$$

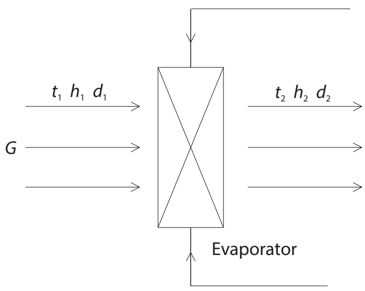

Figure 2. Evaporator air cooling diagram

Dehumidification model

When the evaporator wall temperature is lower than the dew point temperature of the air, condensation will occur. Figure 2 shows the evaporator air cooling process.

System cooling capacity is calculated [18]:

$$Q = G(h_1 - h_2) \tag{11}$$

where Q [kW] is cooling capacity, G [kgs^{-1}] – the supply air rate, h_1 [kJkg^{-1}] – the enthalpy of evaporator inlet, and h_2 [kJkg^{-1}] – the enthalpy of evaporator outlet.

The dehumidifying coefficient is equal to the ratio of the total heat exchange to the sensible heat exchange [19]:

$$\xi = \frac{h - h_b}{c_p(t - t_b)} \tag{12}$$

$$Q = q_s + q_r \tag{13}$$

The latent heat exchange capacity can be calculated by eqs. (12) and (13):

$$q_r = \frac{(\xi - 1)Q}{\xi} \tag{14}$$

where q_r [kW] is the latent heat, Q [kW] – the cooling capacity, and ξ – the dehumidification coefficient.

The dehumidification capacity is calculated [20]:

$$W = G\left(d_1 - d_2\right)\frac{1}{1000} \tag{15}$$

where W [kgs^{-1}] is the dehumidification amount, G [kgs^{-1}] – the air mass-flow rate, d_l [gkg^{-1}dry-air] – the moisture content of air inlet, and d_2 [gkg^{-1}dry-air] – the moisture content of air outlet.

The system unit energy consumption dehumidification capacity is calculated [21]:

$$m_e = \frac{3600\,m}{P} \tag{16}$$

where m_e [kgWh^{-1}] is the energy consumption dehumidification capacity, m [kgs^{-1}] – the dehumidification capacity, and P [kW] – the compressor power.

Other parts model

The following assumptions are made in the compressor model:
– Compressor suction and discharge pressure drop were ignored.
– The variable index and electrical efficiency of the compressor do not change with the working conditions.
– Heat transfer between compressor and environment was ignored.
– Assume that the speed of the compressor does not change with the operating conditions.

The condenser model is more or less the same as the evaporator model, the condensation temperature was kept at 45 °C, compressor suction superheat was kept at 7 °C.

Data comparison and analysis

Liu [22] according to the national standard *GB/T 199411-2003 Dehumidifier*, the selected test conditions are shown in tab. 1.

Table 1. Test condition parameters

Condition	Dry bulb temperature [°C]	Wet bulb temperature [°C]	Dew point temperature [°C]	Relative humidity [°C]	Moisture content [gkg^{-1}]	Enthalpy [kJkg^{-1}]
Nominal operating conditions	27.0	21.2	18.6	60%	13.424	61.50
Maximum load condition	32.0	23.0	19.1	46%	13.84	67.75
Condensing condition	27.0	24.0	22.8	78	17.57	72.08

Liu [22] fitted the experimental data of the compressor to obtain the relationship between COP and evaporation temperature, and solved the relationship between DCPP (unit energy consumption dehumidification capacity) and evaporation temperature of the dehumidifier under test conditions through MATLAB programming, and obtained different test conditions. The optimal evaporation temperature was compared with the simulation calculation results in this article. As shown in the tab. 2, the error between the two is controlled within 10%.

Table 2. Comparison of reference data and model fitting data

Condition	Experimental optimal evaporation temperature [°C]	Model optimal evaporation temperature [°C]	Model maximum dehumidification volume corresponding to evaporation temperature [°C]	Relative error [%]
Nominal operating conditions	10.2	9.24	8.28	–9.4
Maximum load condition	9.4	10.1	9.61	7.4
Condensing condition	16.2	14.7	13.74	–9.2

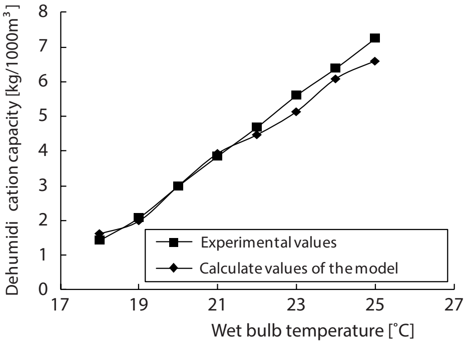

Figure 3. Comparison with simulation and experimental results

Li *et al.* [23] carried out an experimental study on frequency conversion air conditioner. The relationship between the dehumidification capacity and the inlet air parameters in the dehumidification mode of the variable-frequency air conditioner was researched. This paper compared the simulated data with that of experimental data. To ensure results data validation and accuracy, the test data was introduced into the relationship between the dehumidification capacity and the wet bulb temperature, the condition was dry bulb temperature of 27 °C, air volume 1000 m^3/h, as shown in fig. 3.

From fig. 3 we can see that, the calculate values of the model are basically consistent with the experimental values when wet bulb temperature was in the range of 18-21 °C. In the range of wet bulb temperature 22-25 °C, the deviates between calculates value of the model and the experimental value was less than 10%, this indicated that the model was consistent with the actual.

System simulation results analysis

In this paper, the programming iterative calculation method was used to simulate dehumidification process of the system. The simulation air conditions were: temperature 21-36 °C, relative humidity 40-85%. The result can show the relationship between the parameters such as system dehumidification capacity and evaporation temperature with the dry bulb temperature, relative humidity and inlet wind volume.

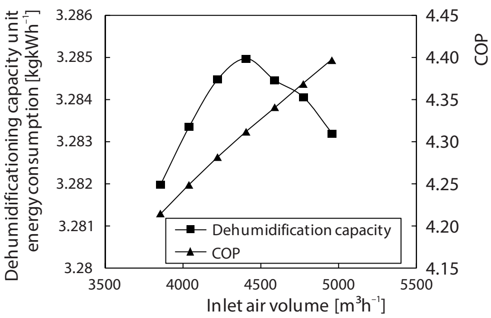

Figure 4. Influence of inlet air volume on working performance

Figure 4 shows the influence of inlet air volume on unit energy consumption dehumidification capacity and COP when the inlet air dry bulb temperature was 27 °C and the relative humidity was 75%. It can be seen from fig. 4 that, with the increase of the inlet air volume, the system energy consumption dehumidification capacity increased first and then decreased. When the inlet air volume was 4400 m^3/h, there was a maximum value of 3.285 kgkW/h. The simulation results were the same as [17]. It can be seen from fig. 4 that the system COP gradually increased. That was because with the increase of inlet air volume, heat exchange between air and evaporator was enhanced making evaporation temperature rose and cooling capacity increased. Since the system condensing temperature was assumed to be 45 °C, when the evaporation temperature and pressure increased, the compressor power consumption decreased, so the COP of the system increased gradually.

Figure 5 shows the effect of relative humidity of inlet air on the dehumidification capacity and cooling capacity of the system, under the condition of the inlet air temperature is 27 °C and the inlet air velocity is 2 m/s. With the increase of the relative humidity of the air, the dehumidification capacity of the system gradually increased from 1-23 kg/h. This was because with the increase of the relative humidity of the air, the difference of water vapor pressure and the air and the fin surface gradually increased, which made more water dehumidifying on the fin surface, and so the dehumidification capacity of the system increased. It can be seen from fig. 5 that, with the increase of air relative humidity, the cooling capacity of the system gradually increased from 20-28 kW. This was because when the relative humidity was large, the convective heat transfer and mass transfer coefficient of the heat exchanger increased, and this made the evaporation temperature increase.

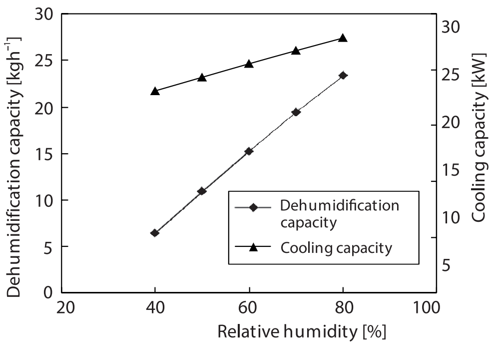

Figure 5. Effect of relative humidity of inlet air on dehumidification and cooling capacity

Figure 6 shows the effect of inlet air dry bulb temperature on dehumidification capacity and cooling capacity when the inlet air moisture content was 16 (g/kg dry-air) and the inlet air velocity was 2 m/s. With the increase of inlet air dry bulb temperature, the dehumidification capacity of the system gradually decreased from 21-15 kg/h, the cooling capacity gradually increased from 25-30 kW. This was because as the inlet air temperature increases, the surface temperature of the heat exchanger fins increased and the amount of the unit mass air reaching the dew point temperature decreased, so the dehumidification capacity gradually decreased.

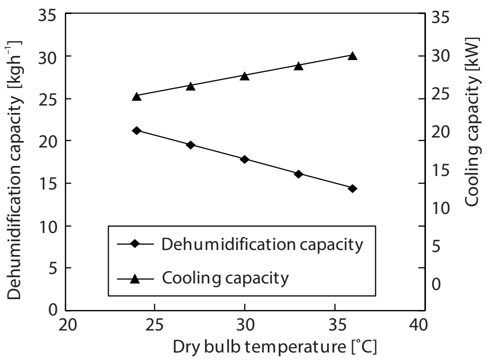

Figure 6. Effect of inlet air dry bulb temperature on dehumidification and cooling capacity

As can be seen from the fig. 7, Δt showed the difference between the inlet air wet bulb temperature and the evaporation temperature at the optimum COP. The relative humidity was between 40-85%, the difference between the inlet air wet bulb temperature and the evaporation temperature at the optimum COP was about 10 °C. Due to the assumption that the condensation temperature of the condenser model was kept at 45 °C in this system, the COP simulation results of the system showed a gradual increase. According to the engineering experience, this paper determined the evaporation temperature at the optimal COP of the system when the attenuation range of the cooling capacity growth rate was within 5-10%.

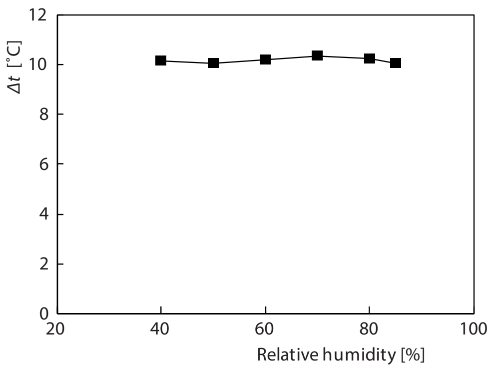

Figure 7. Effect of relative humidity on Δt

As can be seen from fig. 8, the evaporation temperature under the maximum dehumidification and the evaporation temperature under the optimal COP increased with the increase of relative humidity, the difference between the evaporation temperature under the optimum COP and the evaporation temperature under the maximum dehumidification rate decreased gradually.

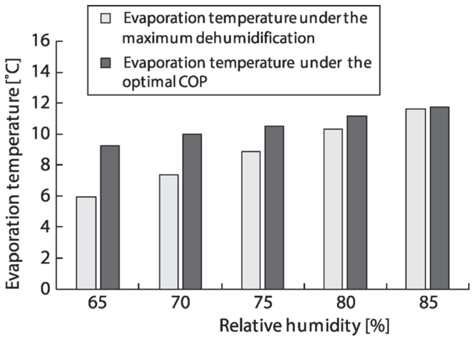

Figure 8. Effect of relative humidity on evaporation temperature

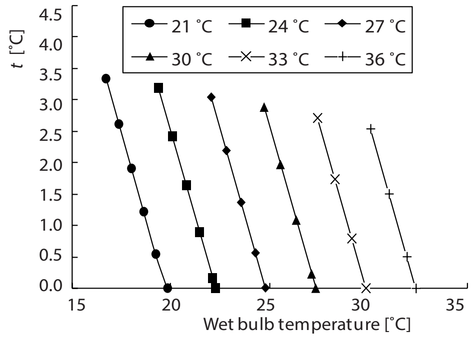

Figure 9. Effect of inlet air wet bulb temperature on Δt_1

Figure 9 shows the effect of inlet air wet bulb temperature on Δt_1 when the dry bulb temperature was 21-36 °C and relative humidity was 65-85%. The ordinate Δt_1 was the difference between the evaporation temperature at the optimal COP and the evaporation temperature at the maximum dehumidification capacity of the system. The fitting function:

$$y = -1.108t_s + 0.933t + 2.281 \tag{17}$$

where y [°C] is the difference between the evaporation temperature at the optimal COP and the evaporation temperature at the maximum dehumidification capacity of the system, t_s [°C] – the wet bulb temperature (°C), and t [°C] – the dry bulb temperature.

In the fig. 9, the left-most dry bulb temperature was 21 °C and the temperature interval from left to right was 3 °C. On the same dry bulb temperature line, the top relative humidity was 65%, and the relative interval humidity from top to bottom was 5%.

It can be seen from fig. 9 that, when the relative humidity was the same and the dry bulb temperature rose from 21-36 °C, Δt_1 gradually decreased. For example, within this dry bulb temperature range, Δt_1 gradually decreased from 3.4-2.6 °C when the relative humidity was 65%. This was because under the condition that the relative humidity was the same and the dry bulb temperature gradually rose, the increase amount of evaporation temperature at the maximum dehumidification capacity was greater than that of the evaporation temperature at the optimal COP of the system. Therefore, Δt_1 gradually decreased. In addition, the relative humidity at the intersection point with the X-axis from left to right in the figure was 90%, 86%, 84%, 82%, 80%, and 78%. when the relative humidity was exceeded the intersection of the X-axis, the value of Δt_1 was 0 °C. This was because the evaporation temperature at the maximum dehumidification capacity was basically equal to the evaporation temperature value at the optimal COP, and the optimal dehumidification condition was also the optimal refrigeration condition.

When the system simulated the working performance with dry bulb temperature at 21-36 °C and the relative humidity at 65-85%, the inlet air volume of the evaporator was reduced and the evaporation temperature decreased, and the dehumidification capacity of the system would be increased. When the relative humidity was below 65%, the air outlet temperature would be lower than the wall temperature of the evaporator, this did not meet the actual project. When the maximum dehumidification effect of the system could not be achieved only by reducing the inlet air volume, the dehumidification capacity of the system should be increased by adjusting the throttle valve to reduce the evaporation temperature.

Figure 10 shows the effect of inlet air wet bulb temperature on Δt_1 when the dry bulb temperature was 21-36 °C and relative humidity was 40-65%. It can be seen from the fig. 10

that, the difference between the evaporation temperature at the optimal COP of the system and the evaporation temperature at the maximum dehumidification of the system was linearly correlated. The fitting function:

$$y = -0.205t_s + 0.148t + 3.368 \tag{18}$$

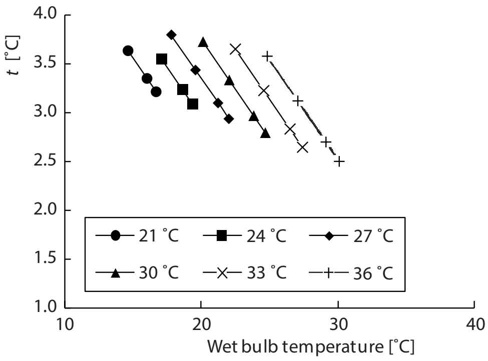

Figure 10. Effect of inlet air wet bulb temperature on Δt_1

In the figure, the leftmost dry bulb temperature was 21 °C, the temperature interval from left to right was 3 °C, on the same dry bulb temperature line the relative humidity from bottom to top, respectively were 65%, 60%, 50%, and 40%.

Compared figs. 9 and 10, it can be found that the variation of Δt_1 was smaller when the relative humidity was 40-65%. This was because the relative humidity in the range of 40-65% was through adjusting the throttle valve to reduce the evaporation temperature of the system, the evaporation temperature cannot be greatly reduced, otherwise it would lead to the cooling capacity and dehumidification capacity reduced of the system.

Dehumidification was not required when the relative humidity was below 40% in the comfort air conditioning range. So, the calculation of the humidity range in this paper did not include the relative humidity below 40%. In addition, in the case of low relative humidity of the evaporator inlet air, it was necessary to ensure that the evaporator wall temperature was greater than 0 °C to avoid frosting on the evaporator surface. Moreover, within the range of low humidity, there was no linear relationship between the evaporation temperature at the optimum COP and the evaporation temperature at the maximum dehumidification capacity.

Conclusions

This paper establishes the dehumidification model of the refrigeration system and theoretically study the dehumidification characteristics of the system. The dehumidification process was theoretically researched at a temperature of 21-36 °C and a relative humidity of 40-85%, the model can calculate the dehumidification characteristic parameters of the system. This research obtains the following conclusions.

- Under the condition of constant temperature and humidity, with the increase of the inlet air volume, the unit energy consumption dehumidification capacity first increased and then decreased, there was an optimal dehumidification air volume.
- The relative humidity was between 40% and 85%, the difference between the inlet air wet bulb temperature and the evaporation temperature at the optimum COP was about 10 °C.
- When the dry bulb temperature was 21-36 °C, the difference between the evaporation temperature at the optimum COP of the system and the evaporation temperature of the system's maximum dehumidification was linear.

  Relative humidity was 65-85%: $y = -1.108t_s + 0.933t + 2.281$.

  Relative humidity was 40-65%: $y = -0.205t_s + 0.148t + 3.368$
- Compared with the test value, the error of the simulation was less than 10 %.

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Nomenclature

B_0 – boiling feature number
C_0 – convection feature number
C_1-C_5 – R134a feature number $C_1 = 1.136$, $C_2 = -0.9$, $C_3 = 667.2$, $C_4 = 0.7$, $C_5 = 0.5$
c_w – specific heat of water (= 4.19), [kJkg^{-1}°C^{-1}]
$c_{p,v}$ – constant pressure specific heat of water vapor, [kJ kg^{-1}°C^{-1}]
$c_{p,m}$ – constant pressure specific heat of wet air, [kJ kg^{-1}°C^{-1}]
d_l – tube inside diameter
$d_{1,2}$ – average moister content of humid air as entering and exiting the evaporator, [gkg^{-1}dry-air]
d_w – moisture content of saturated air at wall temperature
e – wing root diameter
F_{fl} – liquid Froude number
G – mass flux, [kgm^{-2}s^{-1}]
g – gravity acceleration, [ms^{-2}]
h_1 – heat transfer coefficient of dry surface
h_2 – heat transfer coefficient of wet surface
h_b – enthalpy of saturated air at the surface of a water film, [kJkg^{-1}]
N – number of tube rows
Pr_l – liquid phase Prandtl number
q_s – sensible heat exchange capacity, [kW]
Re_l – refrigerant liquid Reynolds number
r_{refl} – latent heat of vaporization, [Jkg^{-1}]
s – fin spacing
s_1 – tube spacing along the air-flow direction
t – dry bulb temperature, [°C]
t_b – temperature of saturated air at the surface of a water film, [°C]
t_s – wet bulb temperature, [°C]
$t_{1,2}$ – average temperature of humid air as entering and exiting the evaporator, [°C]
t_w – wall temperature, [°C]
X – mass gas content rate (Dryness fraction)

Greek symbols

α_{refr} – two-phase surface heat transfer coefficient of boiling in tube, [Wm^{-2}K^{-1}]
$\alpha_{refr.l}$ – surface heat transfer coefficient of liquid phase in tube, [Wm^{-2}K^{-1}]
γ_0 – latent heat of water
μ_l – liquid phase dynamic viscosity, [Pa·s]
λ_l – liquid thermal conductivity, [Wm^{-2}K^{-1}]
ξ – dehumidification coefficient
ρ_g – gas phase density, [kgm^{-3}]
ρ_l – liquid phase density, [kgm^{-3}]
ψ – heat flux density, [Wm^{-2}]

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