INVESTIGATION OF THE PERFORMANCE OF HEAT PUMP WASTE HEAT REGENERATION RUNNER DEHUMIDIFICATION OF AIR CONDITIONING SYSTEMS

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Abstract - This study investigated the performance of a compound unit combining an air source heat pump (ASHP) and a small solid dehumidification runner. In this system, the exhaust gas and the condensation heat of the compressor were used as the regeneration energy of the runner dehumidifier. The detail of the exhaust temperature and the utilization energy characteristics of the heat pump were analyzed; the dehumidification runner under the waste energy utilization of the heat pump was also analyzed. The simulation result showed that the heating performance (COP) of the compound unit was 3.68 at the same evaporation temperature; but when the ASHP supplies heat, the condensing temperature becomes 55.0°C. Also, the primary energy utilization rate of the compound unit was 25 % higher than that of the ASHP. Also, the waste heat of the heat pump system was sufficient to realize the regeneration of the dehumidification runner under the dehumidification and cooling conditions in the hot season and met the prerequisites of the indoor wet load.

Keywords: Air Source Heat Pump, Dehumidification Runner, Waste Heat, Residential Buildings, Primary Energy Utilization.

1. Introduction

For residential buildings, air-source heat pumps have a wide range of applications because they can flexibly control the load, meet the needs of cooling, heating, and dehumidification, and meet the needs of individual differences in the human body [1-5]. However, there are still some problems in the application of air-source heat pumps, including low COP of cooling and heating conditions, large air supply temperature difference, and obvious blowing sensation, which affect the thermal comfort of the human body, and the inability to meet indoor requirements [2, 6-10], and the independent control requirements for temperature and humidity [11-16]. The study by [17] proposed an air conditioning system coupled with a dehumidification runner and a medium and high-temperature heat pump. The runner processes return fresh air and the heat pump is utilized to completely meet the recovery temperature and load requirements. In the study by [18], the use of condensation heat to preheat a hybrid dehumidification air-conditioning system combining a dehumidification runner with electric heating regeneration air and vapor compression refrigeration was proposed; the runner handles the wet load, and the cooling coil handles the sensible heat load which can be used to replace the traditional vapor compression refrigeration system. In [19], a hybrid dehumidification air-conditioning system integrating solid rotor desiccant dehumidification, indirect evaporative cooling, and vapor compression refrigeration was proposed. Furthermore, a multi-stage dehumidification system that combines a heat pump and a dehumidification board with a switchable position was proposed by [20]. The heat pump cools the processed air, and the waste heat of the heat pump can achieve a regeneration temperature of 40-50℃. The aforementioned study combined the heat pump condensation heat with the solid dehumidifier. The main issue is that the regeneration air temperature under normal operating conditions is low, resulting in insufficient dehumidification capacity; if the condensation temperature is increased, the unit performance will decrease [21-23].

This paper presents a compound unit of air source heat pump and solid runner using the high-temperature exhaust and condensing heat of the compressor as the regeneration energy of the runner dehumidifier. The runner dehumidification unit is only used to process fresh air, combined with inside building air from a radiant panel water system. In the humidity environment and indoor heat control, the framework is applied to private structures with little fresh air volume and low humidity load. The energy utilization characteristics and exhaust temperature characteristics of the heat pump unit were analyzed, and simulation and experimental studies were performed on the dehumidification runner in the case of heat pump waste heat utilization.

2. System Description and Compound

The process flow of the compound unit is shown in Figure 1. The operating conditions of the unit include cooling and dehumidification conditions, dehumidification condition, and heating condition. Figure 1 showed the rotary dehumidifier that consists of the pre-cooling heat exchanger, regenerative heat exchanger, fresh air blower, exhaust fan, heat exchange runner, and dehumidification runner.

The pre-cooling heat exchanger and the regenerative heat exchanger use finned coil heat exchangers and centrifugal fans; the heat exchange wheel has a diameter of 200 mm, a thickness of 200 mm, and a sensible heat exchange efficiency of 81 %. The dehumidification wheel uses a silica gel wheel of 200 mm diameter, 200 mm thickness, and 1:1 dehumidification to regeneration area ratio.

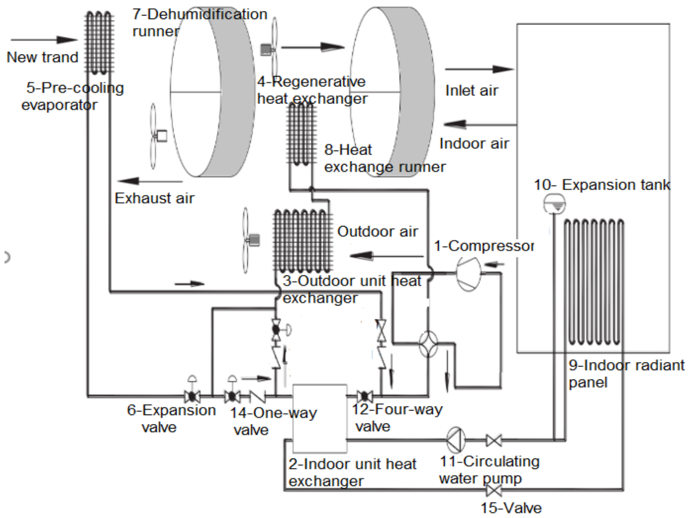

Figure 1. Process flow chart of the compound unit

In this working condition, the heat pump unit and the dehumidification unit operate together to bear latent heat load and the indoor sensible heat; the rotor dehumidification system bears the humidity load, and the indoor radiant panel bears the sensible heat load.

The refrigeration cycle process is as follows: the compressor at point 1 compressed the refrigerant and passed it to the reversing valve 12 (four-way type). After the high-temperature exhaust gas passes through the regenerative heat exchanger 4, it enters evaporator 5 and evaporator 2 at the same time. The evaporator 5 pre-cools the processed air and evaporates it. Device 2 is used to prepare indoor high-temperature cold water; the indoor return and supply water temperature is 20℃/24℃, and the evaporation temperature is 17.0℃. The process of the dehumidification unit is as follows: the fresh air enters the dehumidification rotor 7 to dehumidify after being pre-cooled by the evaporator; the fresh air is dehumidified and heated, and then enters the heat exchange rotor and then enters the room after being wet and cooled.

The exhaust air of the heat exchanges with the fresh air through the heat exchange rotor. After being heated, it is heated by the high-temperature exhaust gas of the compressor and then enters the dehumidification runner, analyze the water of the runner, and discharge to the outdoors.

Sometimes, it is only residual humidity with no residual heat indoors; this climatic feature usually appears in May and June. The outdoor air temperature is 18.0-25.0°C and the relative humidity is about 85%. In this working condition, only dehumidification is required, and no cooling is required. Due to the low load, the compressor exhaust heat is insufficient for regeneration, and the runner system and the indoor evaporator 2 loop are closed. The refrigeration cycle runs at a low load, and the system is at low evaporation. The operation temperature is t_0=5.0℃; the refrigerant is compressed by compressor 1 and passed through the reversing valve. After that, the exhaust gas passes through condenser 3, enters the loop of evaporator 5, and then returns to the compression through the four-way valve 12.

Machine 1 uses the refrigeration cycle to dehumidify the fresh air in this working condition.

In this working condition, the hot water at low temperature is used indoors for heating; the return and supply water temperatures are 30 and 35℃, respectively. Also, the condensing temperature of the system is t_k=41.0℃. The output of compressor 1 enters the indoor condenser 2 through the four-way reversing valve. After entering the outdoor unit evaporators 3 and 4 through the throttle valve 6, it returns to compressor 1 through the four-way reversing valve 12. The indoor heating cycle consists of a condenser 2, a radiant panel 9, an expansion tank 10, and a circulating pump 11. At this time, the dehumidification runner unit stops running, and the fresh air system adopts natural ventilation.

3. The Proposed Model

3.1. Mathematical Model of the Runner

The mathematical model of the dehumidification runner assumes 1) One-dimensional flows; 2) Mass transfer and conduction heat in the axial direction of the fluid are ignored; 3) There is no leakage between the dehumidification zone and the regeneration zone; 4) There is no permeation between each channel and Adiabatic; 5) Thermodynamic properties are constant; 6) The mass and heat transfers are constant along the channel. The mass and energy conservation equations of the one-dimensional dehumidification runner can be obtained based on the above assumptions [6~8]; the processing airside is equation (1) and equation (2), and the hygroscopic agent side is equation (3) and equation (4):

$$\frac{\partial Y_a}{\partial z} = \frac{h_m P_p}{u_a \rho_a A_P}(Y_w - Y_a) \quad (1)$$

$$\left(c_{pa} + Y_a c_{pv}\right)\frac{\partial T_a}{\partial z} = \frac{h_m P_p}{u_a \rho_a A_P}(Y_w - Y_a) \quad (2)$$

$$\frac{\partial W}{\partial t} = \frac{h_m P_w}{f_m \rho_w A_w}(Y_a - Y_w) \quad (3)$$

$$\left(c_{pw} + f_m c_{pl}\right)\frac{\partial T_w}{\partial t} = \frac{h P_w}{\rho_w A_w}(T_a - T_w) + \frac{h_m H_{sor} P_w}{\rho_w A_w}(Y_w - Y_a) \quad (4)$$

Where, Y_a is the air content of moisture, kg (water vapor)/kg (dry air); h_m is the mass transfer coefficient with the difference in moisture content as the driving force, kg/(m2•s); P_p is the airflow channel circumference, m; u_a is air velocity, m/sec.; c_{pa} is the air density, kg/m^3; A_p is the processing airflow channel area, m^2; Y_w is the air layer adsorbent surface moisture; c_{pa} is the specific heat of air at constant pressure, J/(kg.K); c_{pv} is the specific heat of water vapor at constant pressure specific, J/(kg.K); T_a is the air temperature, K; h is the heat exchange between air and adsorbent Number, W/(m^2.K); T_w is the dehumidifier temperature, K; W is the adsorbent moisture adsorption rate; t is the response time, sec.; P_w is the channel direction and air contact moisture absorption along the length of the agent, m; f_m is the mass ratio of the hygroscopic agent in the runner; ρ_w is the density of the hygroscopic agent, kg/m3; A_w is the end area of the hygroscopic agent in the direction of the channel, m^2; c_{pw} is the specific heat at a constant pressure of the hygroscopic agent, J/(kg.K); c_{pl} is the specific heat at a constant pressure of water, J/(kg.K); H_{sor} is the adsorption heat, J/kg.

The rate of primary energy utilization is used to analyze the energy utilization capacity of the built unit and the customary ASHP. The rate of primary energy utilization is the energy ratio obtained to the primary energy consumed to obtain the energy:

$$PER = \frac{Q_{gain}}{Q} \quad (5)$$

Where, Q_{gain} is the amount of cold or heat obtained by, kW; Q is the primary energy consumption required to obtain the energy, kW.

For air source heat pumps, there are:

$$PER = \frac{Q_{gain}}{Q} = \frac{Q_{gain}}{W} \cdot \frac{W}{Q} = COP.\eta_f(1 - \eta_s) \quad (6)$$

Where, W is the present heat pump input electric power, kW; COP is the cooling or heating performance coefficient of the air source heat pump; η_f is the secondary energy generation efficiency; η_s is the power transmission loss rate.

3.2. Boundary and Initial Conditions

For the processing part:

$$\begin{aligned} &A = \pi r^2 T_{wp}(o,z) = T_{wr}(t_r, L - z) \\ &Y_{wp}(o,z) = Y_{wr}(t_r, L - z) \\ &T_{ap}(t,0) = T_{ap,in} \ \text{,} Y_{ap}(t,0) = Y_{ap,in} \end{aligned} \quad (7)$$

For the regeneration part:

$$\begin{aligned} &T_{wr}(o,z) = T_{wp}(t_p, L - z) \\ &Y_{wr}(o,z) = Y_{wp}(t_p, L - z) \\ &T_{ar}(t,0) = T_{ar,in} \ \text{,} Y_{ar}(t,0) = Y_{ar,in} \end{aligned} \quad (8)$$

Where t is the time (sec); z and L are the lateral coordinates and runner thickness, respectively (m);

a is the air; w is the moisture absorbent; p is the treated air; r is the regeneration air.

4. Results and Discussion

The mathematical model was implemented using Matlab programming, combined with the given initial conditions and boundary conditions. The solution was calculated, and finally, the state parameters (temperature, moisture content) of the air after passing through the runner were obtained. Table 1 showed the simulation calculation parameters of the runner using the values. The number of grids in the circumferential direction was 400, and the number of grids in the thickness direction of the runner was 100. The time advance method was used to solve the problem, and space derivative term was written in the format of forwarding difference. In the process of solving, the unknown heat of adsorption and the moisture content of the desiccant wall air was solved by supplementary equations.

Table 1. The dehumidification parameters running at constant pressure

Parameters	Value
c_{pw} Silica gel specific heat	921 J/kg.K
ρ_w Silica gel density	720 kg/m^3
ρ_a Air density	1.1614 kg/m^{-3}
c_{pv} Specific heat of water vapor	1872 J/kg.K
c_{pa} Specific heat of the air	1005 J/kg.K
n Rotating speed	10 r/hour
U_a air velocity	2.5 m/sec
h_m Mass transfer coefficient is driven by difference in moisture content	0.03801 kg/m.sec
P_p the airflow channel of the circumference	8.38625×10^{-3} m
A_p area of air flow channel	3.0625×10^{-6} m^2
f_m Mass ratio of hygroscopic agent in runner	0.7
c_{pl} Specific heat of water	4186 J/kg.K

Figure 2 presented the relationship between the condensation temperature and the temperature at the exhaust temperature. The system evaporation temperature is running at 17.0°C in summer under the condition. The following calculation conditions were considered: the degree of sub-cooling = 4.0°C, the isentropic efficiency = 0.75, and the volumetric efficiency = 0.9. Figure 2 showed the linear change of the exhaust temperature with the condensation temperature; when the refrigerant is R22, the compressor discharge temperature becomes the highest, followed by R410a, and R134a is the lowest. It is more appropriate to choose refrigerant R22 as the circulating working fluid. When the condensation temperature of the R22 refrigerant is 55.0°C, the exhaust temperature is 85.7°C.

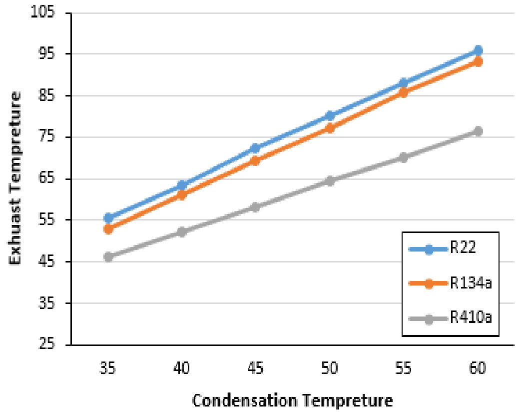

Figure 2. Exhaust temperature and condensation temperature

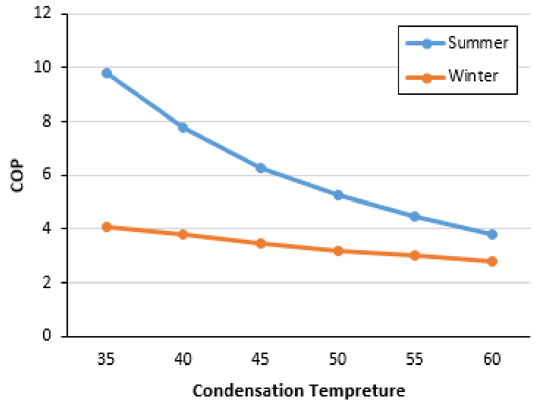

Figure 3. The relation between COP and condensation temperature

Figure 3 showed variations in the COP with condensation temperature under the evaporation temperature of 17.0°C in summer and -12.0°C in winter. Radiant cooling and heating devices are used at the end of the system to reduce the difference in heat exchange temperature. In summer conditions, due to the high evaporating temperature, high COP can still be achieved at the high condensation temperature. When the condensation temperature is 55.0°C, the COP is 4.278; in winter conditions, the system COP increases with the condensation temperature. The COP, at a condensing temperature of 41.0°C, increased by 25.9% compared to the COP of a condensing temperature of 55.0°C (the heating and condensing temperature of the indoor unit traditional convection air supply). The heat pump compound unit operates under the working conditions of the evaporation temperature of 17.0°C and the condensing temperature of 55.0°C; the COP is 4.278 based on simulation studies (Figure 3).

Figure 4 is the regeneration temperature that can be obtained at different compressor discharge temperatures in the experiment. It can be seen from Figure 4 that using the condensation heat and high-temperature exhaust heat, a higher regeneration temperature can be obtained. When the exhaust temperature is 85.6°C, the regeneration temperature can reach 74.9°C. It can be observed that the sensible heat of the high-temperature exhaust is small; the experimental scheme is to use the exhaust of the unit that bears all the cooling load for the regeneration of a small amount of fresh air dehumidification rotor.

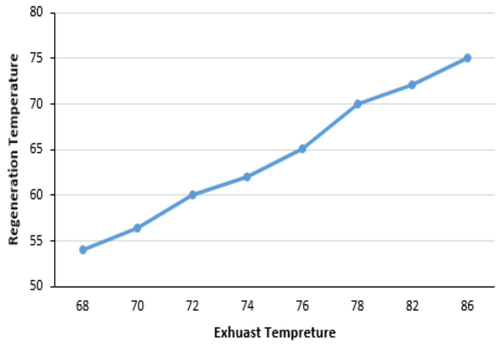

Figure 4. The regeneration temperature at different exhaust temperatures

In Figure 5, the simulated conditions were carried out under ideal conditions, and during field testing, there will be some errors due to many factors (such as measurement error, the efficiency of the runner, unstable surrounding environment, etc.). It can be seen that the variation between the experimental and simulated results was small, indicating that the developed mathematical model is valid.

Figure 5 showed the changes in the dehumidification capacity with the processing air temperature, processing air humidity, regeneration air humidity, and regeneration air inlet temperature. Figure 5a showed that as the air inlet temperature decreases, the dehumidification capacity increases significantly. In actual operation, the processing air temperature is the outdoor temperature. Pre-cooling the processing air through the pre-cooler 5 can increase the dehumidification capacity. Figure 5b showed that the amount of dehumidification will increase significantly as the air moisture content increases.

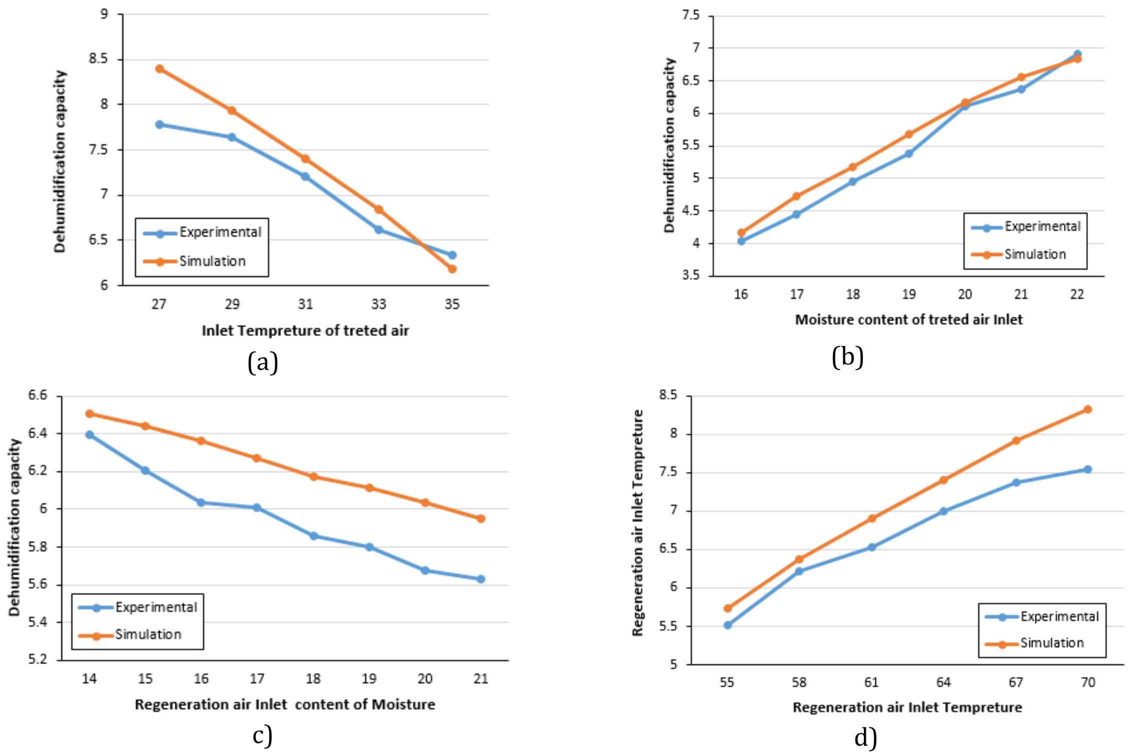

Figure 5. Changes in the dehumidification capacity with a) Inlet air temperature, b) Treated air content moisture, c) Regeneration air inlet moisture, and d) regeneration air inlet temperature

The adjustment in the air moisture content in the test was ensured by the enthalpy difference laboratory. In actual operation, the moisture content represents the outdoor air humidity. Figure 5c showed that the amount of dehumidification decreases with increases in the moisture content of the regeneration air. In actual operation, the moisture content of the regeneration air is the indoor moisture content. Figure 5d showed that when the temperature of the regeneration increases, the dehumidification capability also increases. According to the building model given, it can be calculated that when the indoor design temperature is 27℃ and the relative humidity is 70%, the total humidity load is 787.3g/h. It can be calculated that when the dehumidification capacity is 7.5g/kg, the dehumidification requirement can be met. From the simulated values, it can be concluded that when the regeneration temperature reaches 70℃, it can meet the requirements for indoor dehumidification.

Table 2 showed the energy consumption analysis of different source utilization units under cooling and heating conditions. When the power transmission loss rate is 5%, the primary energy generation efficiency is 33%. The ASHP air-conditioning unit temperature of the evaporator is 5℃ and the temperature of the condenser is 49℃. The results of the calculation showed that the rate of primary energy utilization of the compound unit was 36% higher than the ASHP values. While calculating the rate of energy utilization of the heating condition, the compound unit provides low-temperature water warming under the heating condition. The heat exchange between water and refrigerant requires a lower heat exchange temperature difference, so, the condensing temperature is lower, and the condensing temperature is 41℃.

Table 2. The essential energy utilization rate of different units

Unit	Compound unit	Air source heat pump
COP(Cooling)	4.75	3.46
Primary energy utilization rate (cooling)	1.34	0.98
COP(Heating)	3.68	2.93
Primary energy utilization rate (heating)	1.04	0.83

5. Conclusion

From this study, we concluded as follows:

1. A compound unit combining an ASHP and a small solid dehumidification runner has been modeled. The high-temperature exhaust of the compressor and the condensation heat released by the condenser was used as the regeneration energy of the runner dehumidifier. It can independently control the room temperature and humidity under the condition of meeting residential buildings' thermal comfort.
2. The simulation results of the heat pump showed that using R22 as the heat pump working fluid can offer higher exhaust temperatures. The high-temperature exhaust of the compressor and the condensation heat released by the condenser can be used as a renewable energy source for the dehumidifier.
3. Through the simulation studies, the regeneration air temperature was found to greatly affect the dehumidification performance of the runner. The lowering of the processing air temperature increases the humidity content and reduces the humidity content of the regeneration air, thereby increasing the dehumidification performance.
4. Under the cooling and dehumidification conditions, the compound unit utilization rate of primary energy is 36% higher than the ASHP; under the heating condition, the compound unit showed a better energy-saving capability.

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