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Performance analysis on a novel water chiller using liquid desiccant

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Abstract: A new type of liquid desiccant water chiller for applications on air-conditioning and refrigeration is introduced. The system can be driven by low-grade heat sources with temperatures of 60 to 80 ℃, which can be easily obtained by a flat plat solar collector, waste heat, etc. A numerical model is developed to study the system performance. The effects of different parameters on performance are discussed, including evaporating temperature, regenerating temperature, ambient condition, and mass flow rates of closed moist air and regenerating air. The results show that an acceptable performance of a cooling capacity of 2. 5 kW and a coefficient of performance of 0. 37 can be achieved in a reference case. The regenerating temperature and the humidity ratios of ambient air are two main factors affecting system performance, while the temperature of ambient air functions less. In addition, the mass flow rate of regenerating air and closed moist air should be carefully determined for economical operation.

Key words: liquid desiccant; water chiller; coefficient of performance; low-grade energy utilization

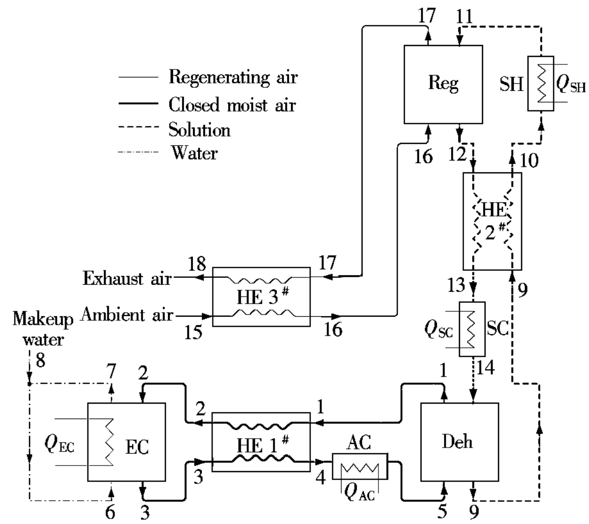
Due to the excessive energy electrical consumption problems in traditional air-conditioning systems, more and more attention has been paid to liquid desiccant cooling systems, which have advantages of environmentally friendly working fluid, low maintenance cost and good use of low-grade energy.

For years, liquid desiccant systems have become a popular research topic. Al-Farayedhi et al. [1] conducted an experimental study on a hybrid liquid desiccant system, and the coefficient of performance was used to evaluate the system performance. Gommed and Grossman[2] presented a solar liquid desiccant system to dehumidify the ambient air directly supplied to offices, and some data and calculated performance parameters were obtained based on practical operation. A solar powered liquid desiccant pre-cooling system was investigated by Katejanekarn and Kumar[3], and some influential parameters were suggested. Tu et al. [4] performed a simulation study on a liquid desiccant system, where the cooling capacity of exhaust air was recovered by an indirect evaporative cooler.

This paper presents a novel liquid desiccant system, which produces chilled water by a closed moist air loop running between a dehumidifier and an evaporative cooler. The system prevents cooling objectives from direct contact with the liquid desiccant, avoiding the possibility of pollution and corrosion. The configuration is modified based on the system presented by Yin et al. [5]. This work aims to numerically study the performance of the new system, and to preliminarily optimize the design parameters.

1 System Description

The schematic of the novel water chiller using liquid desiccant is shown in Fig. 1. The system mainly comprises three loops: the closed moist air loop, the liquid desiccant solution loop, and the regenerating air loop.



Deh—Dehumidifier; Reg—Regenerator; EC—Evaporative cooler; HE—Heat exchanger; SH—Solution heater; SC—Solution cooler; AC—Air cooler;

Fig. 1 Schematic of the liquid desiccant water chiller (Arabic numerals represent state points)

The closed moist air loop is composed of a dehumidifier (Deh), an air-to-air heat exchanger (HE 1#), an evaporative cooler (EC), and an air cooler (AC). As shown in Fig. 1, moist air (state 5) enters into the dehumidifier, where it directly contacts with the concentrated liquid desiccant entering the unit at state 14 and leaving at state 9. During this process, water vapor is removed from the air stream by the concentrated solution. Then the dehumidified air stream goes through HE 1# into EC (state 2), where the dry air is cooled by an evaporative cooling process, and the cooling capacity Q_{EC} can be obtained for the chilled medium by water evaporation. The humid air leaving the EC at state 3, with lower temperature than that at its inlet, is inducted through HE 1# again, providing pre-cooling for the air stream from state 1 to state 2. After being cooled in the AC (states 4 and 5) where water from a cooling tower can be adopted as the coolant, the air stream is finally drawn back to the dehumidifier, ending the circulation.

In the solution loop, the diluted solution leaving the dehumidifier (state 9) is first preheated to state 10 in the solution-to-solution heat exchanger (HE 2#), then heated in the solu-

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tion heater(SH), and finally re-concentrated in the regenerator(Reg). The heat Q_{SH} needed for solution re-concentration can be supplied by low temperature heat sources, such as flat plate solar collectors. The concentrated solution(state 12) is pre-cooled to state 13 in HE $2^{\#}$, then cooled in the solution cooler(SC) to state 14, and finally sprayed into the dehumidifier. The solution-to-solution heat exchanger here facilitates preheating the weak solution(states 9 and 10) and pre-cooling the concentrated solution(states 12 and 13). The same as for the air cooler, the coolant in the solution cooler can also be supplied by a cooling tower.

While in the regenerating air loop, the air-to-air heat exchanger(HE $3^{\#}$) facilitates recovering heat from the air stream leaving the regenerator(state 17), so that the regenerating air(state 16) entering the regenerator can reach high temperature. Thus, the air stream absorbs less heat from the solution stream in the regenerator and the temperature decrease of the solution stream can be depressed. Therefore, the regenerating heat Q_{SH} supplied by a low temperature heat source can be less, and a greater portion of it can be employed for solution re-concentration.

2 Mathematical Model of the System

The detailed performance analysis of the system is performed using a computer code. Each basic component is simulated with a mathematical description. Then, the main program links the components together corresponding to the system configuration; each state point is calculated and the performance of the whole system can be evaluated.

The following assumptions are made to make the numerical study possible: 1) The analysis is based on the steady-state of each component and the overall system; 2) The power needed for air and liquid circulation is omitted; 3) HE $1^{\#}$ has enough heat transfer area. The mathematical models of the various components are presented as follows.

2.1 Dehumidifier and regenerator

The dehumidifier and the regenerator play important roles in the liquid desiccant system. In both units, counter-current flow operation is used. And the NTU-Le_f(number of transfer units and the Lewis factor) model based on the finite difference method is adopted in this paper. The equations are expressed as follows:

$$m_a \mathrm{d}\omega_a = \mathrm{d}m_s \tag{1}$$

$$m_a \mathrm{d}h_a = c_{p,s}\mathrm{d}(m_s t_s) \tag{2}$$

$$\mathrm{d}h_a = Le_f\left[(h_{s,sat} - h_a) - \left(1 - \frac{1}{Le_f}\right)(\omega_{s,sat} - \omega_a) r\right]\mathrm{dNTU} \tag{3}$$

$$\mathrm{d}\omega_a = (\omega_{s,sat} - \omega_a)\,\mathrm{dNTU} \tag{4}$$

where m is the mass flow rate; ω is the humidity ratio; h is the enthalpy; t is the temperature; c_p is the specific heat; r is the latent heat of vaporization; subscripts "a" and "s" stand for moist air and solution, while "sat" symbolizes the status of moist air in equilibrium with the desiccant solution.

Eqs. (1) and (2) govern the conservation of mass and energy, while Eqs. (3) and (4) are heat transfer and mass transfer equations. Detailed deductions can be referred to Refs. [6–7]. For simplification, the Lewis factors of both the dehumidifier and the regenerator are set at a constant value 1.0. However, the NTUs are assumed to be equal to 3.5 and 2.0, respectively. The NTU of the regenerator should be less than that of the dehumidifier, because the vapor pressure difference between the solution and the air in the regenerator is greater than that in the dehumidifier.

2.2 Solution heater, the solution cooler and air cooler

In the solution heater, the solution is heated to meet the requirements of the regeneration temperature. While, in the solution cooler, as mentioned above, the coolant can be supplied by a cooling tower, so the solution is cooled to a temperature of 3 ℃ above the ambient wet bulb temperature. Thus, the heat gain from the solution heater, Q_{SH}, and the heat release from the solution cooler, Q_{SC}, can be defined as

$$Q_{SH} \text{ or } Q_{SC} = \int_{t_{in}}^{t_{out}} m_s c_{p,s}(t, X)\,\mathrm{d}t \tag{5}$$

where X is the mass concentration of the solution in salt, and scripts "in" and "out" stand for the inlet and the outlet of corresponding components, respectively.

Similar to the solution cooler, the function of the air cooler is to cool the moist air to the same temperature before entering the dehumidifier. The heat emission from the moist air Q_{AC} in the air cooler can be expressed as

$$Q_{AC} = m_5(h_5 - h_4) = m_5[h(t_5, d_3) - h(t_4, d_3)] \tag{6}$$

where Arabic numerals represent state points.

2.3 Heat exchangers

The heat exchangers adopted in this paper are two air-to-air heat exchangers and a solution-to-solution heat exchanger, whose performance can be presented in terms of heat exchange effectiveness ε and actual heat transfer rate Q, as defined by

$$\varepsilon = \frac{t_{hot,in} - t_{hot,out}}{t_{hot,in} - t_{cold,in}} \tag{7}$$

$$Q = \varepsilon\left\{\int_{t_{cold,in}}^{t_{hot,in}} m_{hot} C_{p,hot}\mathrm{d}t, \int_{t_{cold,in}}^{t_{hot,in}} m_{cold} C_{p,cold}\mathrm{d}t\right\}_{min} \tag{8}$$

where the scripts "cold" and "hot" stand for cold and hot fluid, respectively.

The effectivenesses of 0.75 and 0.85 are assumed for HE $3^{\#}$ and $2^{\#}$, respectively. However, the effectiveness of HE $1^{\#}$ is idealized to be 1.0, since the area of heat exchange is assumed to be large enough. Then, the process in EC can be treated as isothermal humidification.

2.4 System performance

Coefficient of performance(COP) is employed to evaluate the system performance, which measures the fraction of energy consumption that is converted to useful energy,

$$\mathrm{COP} = \frac{Q_{EC}}{Q_{SH}} = \frac{m_5(h_3 - h_2)}{\int_{t_{10}}^{t_{11}} m_9 c_{p,s}(t, X)\,\mathrm{d}t} \tag{9}$$

where Q_{EC} is the cooling capacity produced by the system.

3 Results and Discussion

The aqueous solution of lithium chloride is chosen as the desiccant due to its high performance and stability, and the property formulations derived from the work of Conde[8] are employed. Based on the models described above, the simulation yields the cooling capacity and the COP of the novel system. In order to study the performance of the cycle, a sensitivity analysis is conducted by varying one parameter and keeping others constant. The conditions of the reference case are selected as listed in Tab. 1, which are marked out by dots on each figure in the following graphs. A cooling capacity of about 2. 5 kW and a COP of 0. 37 can be obtained in the reference case. And it should be mentioned that the mass flow rate of the solute is maintained at 0. 115 kg/s as a constant.

Tab. 1 System parameters for reference case

Parameters	Value
Evaporating temperature t_3/℃	7
Regenerating temperature t_{11}/℃	70
Dry bulb of ambient air t_{15}/℃	35
Relative humidity of ambient air φ_{15}/%	60
Cooling temperature t_5, t_{14}/℃	$t_{w,am}+3.0$
Mass flow rate of closed moist air m_5/(kg·s^{-1})	0. 4
Mass flow rate of regenerating air m_{15}/(kg·s^{-1})	0. 16

Note: $t_{w,am}$ is the ambient wet bulb temperature.

3. 1 Evaporating temperature and regenerating temperature

Fig. 2 describes the effects of evaporating temperature on cooling capacity and COP. Obviously, the higher the evaporating temperature, the easier cooling requirements and, thus, the higher cooling capacity and the higher COP can be obtained. As evidence, the cooling capacity and the COP increase rapidly as the evaporating temperature rises. The increases in cooling capacity by 126% and the COP by 64% compared with the reference case can be achieved for an evaporating temperature of 16 ℃. The system seems to be more suitable for applications on high-temperature air-conditioning equipment, such as supplying cooling water for chilled panels or fan coils for operation under dry conditions.

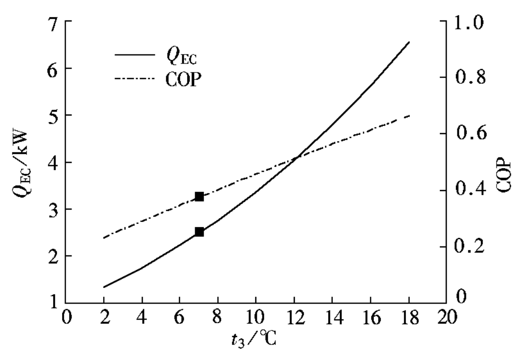

Fig. 2 Effects of evaporating temperature on cooling capacity and COP

As shown in Fig. 3, cooling capacity increases as the inlet solution temperature to the regenerator(state 11, regenerating temperature) increases. And the COP increases rapidly at low temperature, but changes quite mildly at high temperature. It can be explained that higher regenerating temperature leads to greater vapor pressure difference in the regenerator and more concentrated circulating desiccant solution. Thus, the outlet air of the dehumidifier has lower humidity, bringing higher cooling capacity; while stronger solution leads to more difficult re-concentration. As a result, the curve of the COP climbs slowly at higher regenerating temperature. Furthermore, there is a minimum value for solution re-concentration(about 54 ℃ in this figure) where the cooling capacity and the COP trend to be zero.

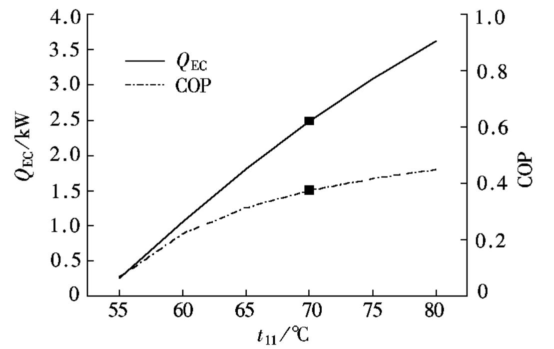

Fig. 3 Effects of regenerating temperature on cooling capacity and COP

3. 2 Ambient air temperature and humidity

The impacts of ambient temperature on system cooling capacity and the COP are shown in Fig. 4. As shown in Fig. 1, the ambient air(state 15) is used for desiccant re-concentration. Higher temperature of ambient air at a constant relative humidity means a higher ambient humidity ratio and a higher cooling temperature at states 14 and 5 due to the increase in the ambient wet bulb temperature. Therefore, the concentration of the solution circulating in the system becomes weaker, and the solution temperature in the dehumidifier becomes higher. Thus, cooling capacity declines rapidly due to a much smaller vapor pressure difference in the dehumidifier. Although the higher temperature of the ambient air helps to re-concentrate the solution, the humidity ratio seems to play a more important role, since the cycle COP decreases as the ambient temperature increases at constant relative humidity. As expected, the increase of relative humidity at a constant temperature pulls down cooling capacity and the COP, as shown in Fig. 5.

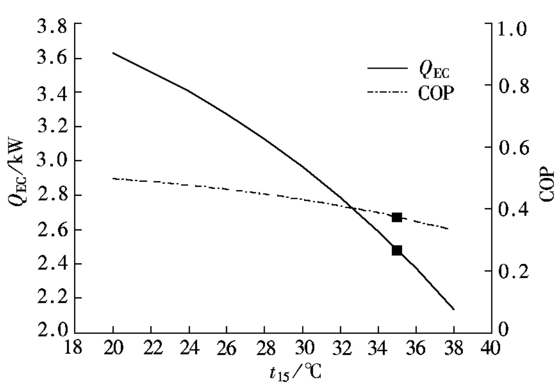

Fig. 4 Cooling capacity and COP as functions of dry bulb temperature of ambient air

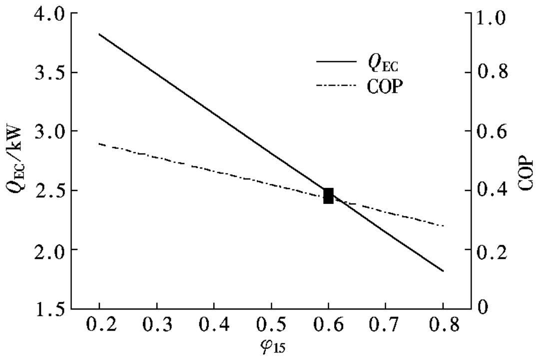

Fig. 5 Cooling capacity and COP as functions of relative humidity of ambient air

3.3 Mass flow rates of regenerating air and closed moist air

Fig. 6 describes the impacts of the mass flow rate of regenerating air on cooling capacity and the COP. Clearly, both curves climb and then level off as the mass flow rate of regenerated air increases. Though the increase in the regenerating air mass flow rate leads to more vapor being removed from the diluted solution, corresponding to the vapor humidified into air which produces a cooling capacity in the evaporative cooler, the limited concentration of the solution at state 12 is dependent on the temperature and humidity of the regenerating air at state 16. Hence, both curves behave as exhibited. However, as the mass flow rate increases, the sensible heat carried from the regenerator by regenerating air increases, that is why the COP even drops a little at high mass flow rates.

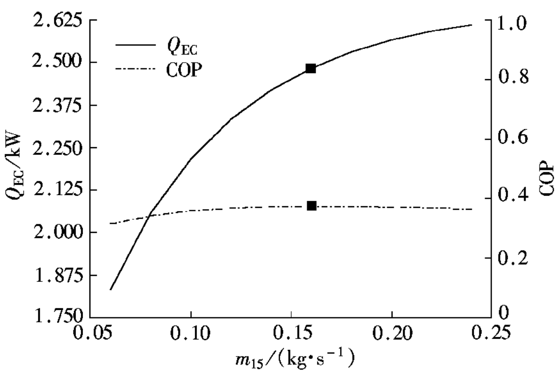

Fig. 6 Effects of regenerating air mass flow rate on cooling capacity and COP

The effects of the mass flow rate of closed moist air, m_3, on cooling capacity and the COP are shown in Fig. 7. The cooling capacity and the COP increase 25.5% and 15.6%, respectively, at a 0.6 kg/s closed moist air mass flow rate compared to the reference point. In spite of this, a mass flow rate of 0.4 kg/s is chosen to be the design parameter for closed moist air. The less the mass flow rate of closed moist air is, the dryer the moist air is at the outlet of the dehumidifier. Thus, a higher cooling capacity per unit mass flow rate can be achieved and a greater mass flow rate requires much more power to circulate the moist air.

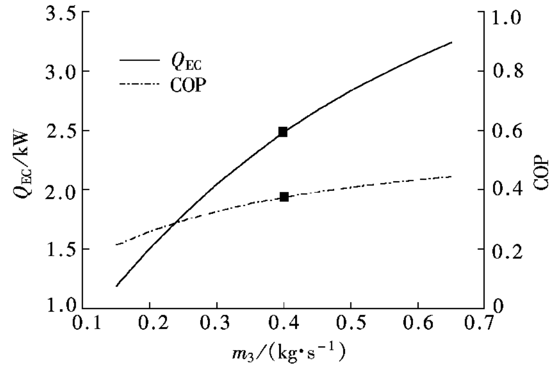

Fig. 7 Effects of closed moist air mass flow rate on cooling capacity and COP

4 Conclusion

Compared with conventional vapor-compression refrigeration systems driven by electricity, the proposed system makes it feasible to use low-grade heat sources, such as flat plate solar collectors and waste heat, to obtain chilled water. With a closed moist air loop, the solution does not contact directly with cooled working fluid in the evaporative cooler, avoiding the possibility of pollution and corrosion. The system seems to provide acceptable performance with a cooling capacity of about 2.5 kW and a COP of 0.37 in the reference case.

With numerical simulation, parametric study is conducted to investigate the effects of different parameters on the performance of the system. There exists a minimum regeneration temperature for each operation condition. In addition, it is found that the humidity ratio of ambient air significantly impacts the system performance, while the effect of ambient air temperature is rather negligible. The excessive increase in the mass flow rate of regeneration air is unnecessary and useless, since the diluted solution can be regenerated adequately. And the mass flow rate of closed moist air should be carefully determined for economical operation.

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一种新型溶液除湿冷水机组性能分析

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摘要:介绍了一种应用于制冷空调领域的新型溶液除湿冷水机组,该系统可以由60~80 ℃的低品位热能驱动,如太阳能平板集热、余热废热等.对该系统建立了数学模型,理论分析了蒸发温度、再生温度、环境空气温湿度、封闭制冷循环和再生循环空气流量等参数对系统性能的影响.结果表明:系统在参考工况下制冷量为2.5 kW,性能系数达0.37,制冷性能良好;再生温度和环境空气含湿量对系统制冷量和性能系数的影响较大,而环境空气温度的影响较小;同时,为了使得系统能够经济运行,再生空气流量不宜过大,而封闭制冷循环空气的流量也需要合理选择.

关键词:溶液除湿;冷水机组;性能系数;低品位能源利用

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