A Study on the Influence of External Conditions on Refrigeration Capacity of Libr-H2o Double-Effect Absorption Chillers

Hai Liu [1,2], YongWang[2], Yong Song [2], AqiangZhang [2], and JitianHan [*1]

[1.] *School of Energy and Power Engineering,* Shandong University, Jinan, 250061, China
[2.] *School of Mechanical, Electrical and Information Engineering,* Shandong University at Weihai, Weihai, 264209, China

***Abstract* — We introduce a LiBr-H_2O double-effect absorption chiller system based on a distributed co-supply system of cooling, heat, and power. By analyzing the physical process, a dynamic mathematical model is established, and a simulation system is built using Simulink. The principal factors that affect the coefficient of performance (COP) and cooling capacity of the refrigeration machine are analyzed by changing external parameters of the chiller. In addition, we examine the relationships between the refrigerating machine's external parameters and the refrigeration effect, which can be used to help optimize the design and debug the chiller in future work.**

Keywords - LiBr-H_2O double-effect absorption chiller; dynamic mathematical model; refrigeration effect;simulation analysis

I. INTRODUCTION

With the rapid development of the global economy and expansion of the population, energy demands are exponentially increasing. This is expected to continue in the future. Especially in China, the contradiction between population and development is very critical. Thus, the challenge is to continue supplying energy to society in an economically viable way while minimizing the impact on the environment. The world is facing pressures from an insufficient energy supply and the need for environmental protection. To address these issues, countries are seeking effective coping strategies [1-6].Combined cooling, heating, and power (CCHP) systems have been developed to address these needs. We propose a CCHP system that represents an improvement over a traditional integrated energy system. And it can accomplish step utilization of the energy by meeting the demands of the user. In the proposed system, the energy utilization efficiency is greatly improved [7].

ACCHP system consists of a power generation unit (PGU), an absorption chiller (AC), and other equipment such as heat exchangers and an electricity-driven chiller (EC)[8].The absorption chiller is a key component. The LiBr-H_2O absorption chiller system uses water as the refrigerant; LiBr solution as the absorbent; and gas, steam, or hot water as the heat source. This system is functional, has high efficiency, and is environmentally sound [9-11]. There are two main types of LiBr-H_2O absorption chillers in production: single-effect and double-effect. The chiller's performance determines the energy utilization rate and practicality. The COP and cooling capacity are the two key parameters of an absorption refrigeration machine system. Therefore, improving the COP and cooling capacity of the system becomes particularly important. The external conditions are the main factors affecting performance, which include the parameters of the heat source cooling water and chilled water. Chen Yaping et al. proposed a 1.x stage LiBr-H_2O absorption refrigeration cycle utilizing solar energy [12]. They analyzed the effects of the temperature of hot water, chilled water, and cooling water on a LiBr-H_2O absorption chiller's performance. Chen Ying et al. presented a refrigeration cycle driven by a low temperature heat resource with relatively high cooling capacity [13]. In this system, the COP and cooling capacity for different heat source temperatures were analyzed in detail. Bahador Bakhtiari etal performed an experimental and simulation analysis of a laboratory single-stage H_2O-LiBr absorption heat pump with a cooling capacity of 14 kW[14]. The machine performance was measured at different flow rates and temperatures of the external cool and hot water loops, and for different temperatures of produced chilled water. Currently, few studies of the effects of external conditions on the performance of LiBr-H_2O steam-driven double-effect absorption chillers are available in the literature. Based on previous studies, we examined LiBr-H_2O steam-driven double-effect absorption chillers and built a dynamic mathematical model of this system. Steady state simulation experiments were conducted for various external conditions. Finally, connections between the parameters and the system's performance were found based on our theoretical considerations, simulations, and experimentation results.

II. THE PRINCIPLE OF LIBR-H_2O DOUBLE-EFFECT CHILLERS

The LiBr-H_2Odouble-effect chiller is a typical absorption refrigerating machine that is mainly comprised of two generators (a high pressure generator (HPG) and a low pressure generator (LPG)), a condenser, an evaporator, an absorber, two heat exchangers (a high temperature heat exchanger (HTHE),and a low temperature heat exchanger (LTHE)). Compared with single-effect chillers, double-effect chillers have a greater COP and higher temperature heat sources. In comparison with cascading chillers, paralleling chillers exhibit an even larger COP [9,10]. Therefore, we adopted the scheme of double-effect and paralleling chillers.

A schematic of aLiBr-H_2O steam-driven double-effect chiller is shown in Fig. 1.

LiBr-H_2O absorption chillers use a LiBr solution (absorbent) and water (refrigerant) as working pairs. When an absorption chiller operates, the LiBr solution in the HPG is gradually heated by external steam. The concentration of the LiBr solution increases because of water evaporation. Then, generated high pressure steam enters into the condenser pipe of the HPG and heats the LiBr solution in the LPG by heat transfer. The water evaporates and enters the condenser, and the temperature of the solution increases. Simultaneously, the vapor in the high pressure condenser pipe turns into liquid water and flows into the condenser. After entering the condenser, the vapor is liquefied. The vapor is cooled by cooling water and then mixes with the liquid water that comes from the high pressure condenser pipe. After entering the evaporator, the condensed water absorbs heat from the cooled water and vaporizes rapidly due to the sharply declining pressure. This process achieves the cooling effect. Meanwhile, in order to increase the evaporation capacity, the unevaporated liquid water is sprayed onto the external surface of the cooled water pipe through the evaporation pump. As the vaporized steam enters the absorber, the concentrated solution from the two heat exchangers is diluted with the steam by contact. The heat generated in the absorbing process is brought outside by cooling water. Drained out of the absorber, the dilute solution exchanges heat with concentrated solution generated in the two generators, and then flows into the generators. The recycling continues in this way.

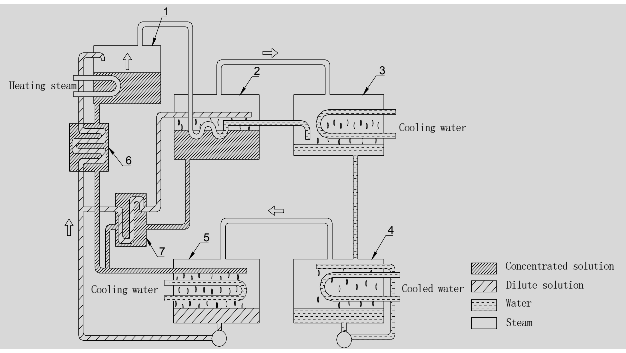

Figure 1. Schematic of the LiBr-H_2O steam-driven double-effect chiller. 1-HPG, 2-LPG, 3-Condenser, 4-Evporator, 5-Absorber, 6-HTHE, 7-LTHE.

III. MODELING AND SIMULATION OF LIBR-H_2O STEAM-DRIVEN DOUBLE-EFFECT ABSORPTION CHILLERS

AnLiBr-H_2O absorption chiller is a complex system. Based on the conservation laws of energy, matter, etc, a lumped parameter mathematical model is established by simplifying. Then, the principle of LiBr-H_2O absorption chillers was simulated and verified using Simulink. In the process of modeling, several assumptions for this system were made as follows [15]:

(1) Heat exchange between the shell and the environment is ignored.
(2) The pressure loss of all linkers is not considered.
(3) The fluids are incompressible and their velocity is the same in all linkers.
(4) The heat transfer coefficients between the shells of all parts remain constant and are chosen according to technical requirements.

A. *Models of the LiBr Absorption Chiller's Major Modules*

The models of the LiBr-H_2O steam-driven double-effect absorption chiller are comprised of an HPG model, an LPG model, a condenser model, an evaporator model, an absorber model, and a heat exchanger model [10,15]. To solve the problem, the nonlinear models are based on the following 42 algebraic equations.

1) HPG model

The continuity equation (Eq. (1)) of the solution and mass conservation equation (Eq.(2)) of LiBr for the HPG:

$$\frac{dG_s}{d\tau} = m_i - m_o - m_v \quad (1)$$

$$\frac{d(G_s x_o)}{d\tau} = m_i x_i - m_o x_o \quad (2)$$

The energy conservation equations (Eqs.(3)-(5))of the solution for the HPG:

$$\frac{d(G_s u_o)}{d\tau} = m_i h_i - m_o h_o - m_v h_v + Q_h - Q_{sh} \quad (3)$$

$$Q_h = K_h A_h[(t_h - t_i) - 0.65(t_o - t_i)] \quad (4)$$

$$Q_{sh} = G_{sh} C_{sh} \frac{dt_o}{d\tau} \quad (5)$$

The mass conservation equation (Eq. (6)) of steam, the energy conservation equation(Eq. (7)) of steam, and the total volume conservation equation (Eq. (8)) of steam and the solution for the HPG:

$$\frac{dG_v}{d\tau} = m_v - m_{vo} \quad (6)$$

$$\frac{d(G_v u_{vo})}{d\tau} = m_v h_v - m_{vo} h_{vo} \quad (7)$$

$$\frac{G_v}{\rho_{vo}} + \frac{G_s}{\rho_s} = V_h \quad (8)$$

2) LPG model

The pressure in the LTHE is higher than the pressure in the LPG, so flashing always takes place when dilute solution enters into the LPG. The flow rate, solution concentration, and specific enthalpy are as follows after flashing:

$$m_i^{'} = m_i(1 - \chi_l) \quad (9)$$

$$x_i^{'} = x_i/(1 - \chi_l) \quad (10)$$

$$h_i^{'} = (h_i - \chi_l h_i^{'})/(1 - \chi_l) \quad (11)$$

The definition of χ_l used here is the dryness after flashing.

The continuity equation (Eq. (12)) and energy conservation equation(Eq. (13)) of the solution for the LPG:

$$\frac{dG_S}{d\tau} = m_i - m_i^{'} - m_v \quad (12)$$

$$\frac{d(G_s u_o)}{d\tau} = m_i h_i - m_o h_o - m_v h_v + Q_l - Q_{sh} \quad (13)$$

The mass conservation equation (Eq. (14)) and energy conservation equation (Eq. (15))of the steam for the LPG:

$$\frac{dG_v}{d\tau} = m_v + m_i - m_i^{'} - m_{vo} \quad (14)$$

$$\frac{d(G_v u_{vo})}{d\tau} = \left(m_i - m_i^{'}\right) h_i + m_v h_v - m_{vo} h_{vo} \quad (15)$$

The mass conservation equation(Eq. (16)) of LiBr and total volume conservation equation(Eq. (17)) of steam and solution for the LPG:

$$\frac{d(G_s x_o)}{d\tau} = m_i x_i - m_o x_o \quad (16)$$

$$\frac{G_v}{\rho_{vo}} + \frac{G_s}{\rho_s} = V_l \quad (17)$$

3) Condenser model

After entering into the condenser, the saturated water from the high pressure steam pipe of the LPG begins to flash. The dryness after flashing is given by

$$\chi_c = (h_{ho} - h_l)/r_l \quad (18)$$

where h_{ho} is the specific enthalpy of condensate water in the high-pressure steam pipe outlet, h_l is the specific enthalpy of saturated water under condensing pressure, andr_l is the latent heat of the saturated water under condensing pressure.

The continuity equation (Eq. (19)) and energy conservation equation (Eq. (20)) of the saturated water for the condenser:

$$\frac{dG_s}{d\tau} = m_c + m_{ho}(1 - \chi_c) - m_{co} \quad (19)$$

wherem_c is the condensing capacity of the condenser.

$$\frac{d(G_s u_{co})}{d\tau} = m_{hi}(1 - \chi_c) h_{co} + m_c h_{co} - m_{co} h_{co} \quad (20)$$

The continuity equation of steam for the condenser:

$$m_c = m_{ho}\chi_c + m_{lo} \quad (21)$$

The energy conservation equations (Eqs. (22)and(23)) of the cooling water:

$$\rho_w V_c c_w \frac{dt_{wo}}{d\tau} = Q_c - m_w c_w (t_{wo} - t_{wi}) \quad (22)$$

$$Q_c = K_c A_c [(t_{wi} - t_c) - 0.65(t_{wo} - t_{wi})] \quad (23)$$

4) Evaporator model

The pressure in the evaporator is much lower than that in the condenser, so flashing occurs after condenser water in the condenser enters the evaporator. The steam ratio after flashing is given by

$$\chi_e = (h_{ei} - h_{ew})/r_l \quad (24)$$

where h_{ei} is the specific enthalpy of the cryogen water in the evaporator inlet, h_{ew} is the specific enthalpy of saturated water under the evaporating pressure, andr_l is the latent heat of the saturated water under the evaporating pressure.

The continuity equation of steam for the evaporator:

$$m_{ev} = m_{vo} - m_{ei}\chi_e \quad (25)$$

where m_{ev} is the evaporating capacity of the evaporator.

The energy conservation equations (Eqs. (26)and(27)) of cooled water:

$$\rho_w V_e c_w \frac{dt_{wo}}{d\tau} = m_w c_w (t_{wi} - t_{wo}) - Q_e \quad (26)$$

$$Q_e = K_e A_e [(t_{wi} - t_{va}) - 0.65(t_{wi} - t_{wo})] \quad (27)$$

Where t_{eva} is the evaporating temperature.

The energy conservation equation (Eq. (28)) of the cryogen in the condenser and continuity equation (Eq. (29))of the cryogen in the evaporator:

$$\frac{d(G_s u_{ew})}{d\tau} = m_{ei}(1 - \chi_e) h_{ew} - m_{ev} h_{ew} + Q_e - Q_{sh} \quad (28)$$

where $Q_{sh} = G_{sh} C_{sh} \frac{dt_{sh}}{d\tau}$, and $t_{sh} \approx t_{va}$.

$$\frac{dG_s}{d\tau} = m_{ei}(1 - x_e) - m_{ev} \quad (29)$$

5) Absorber model

The continuity equation (Eq. (30)) and energy conservation equation(Eq. (31)) of solution in the absorber:

$$\frac{dG_s}{d\tau} = m_{va} + m_i - m_o \quad (30)$$

$$\frac{d(G_s u_o)}{d\tau} = m_{va} h_{va} + m_i h_i - m_o h_o - Q_{sh} - Q_a \quad (31)$$

The mass conservation equation of LiBr for the absorber:

$$\frac{d(G_s x_o)}{d\tau} = m_i x_i - m_o x_o \quad (32)$$

The energy conservation equations (Eqs. (33)and(34))of the cooling water in the absorber:

$$\rho_w V_a c_w \frac{dt_{wo}}{d\tau} = Q_a - m_w c_w (t_{wo} - t_{wi}) \quad (33)$$

$$Q_a = K_a A_a [(t_i - t_{wi}) - 0.5(t_{wo} - t_{wi}) - 0.65(t_i - t_o)] \quad (34)$$

6) Heat exchanger model

The mass conservation equation (Eq. (35)) and energy conservation equation(Eq. (36)) of concentrated solution for the shell side:

$$\frac{dG_q}{d\tau} = m_{qi} - m_{qo} \quad (35)$$

$$\frac{d(G_q u_{qo})}{d\tau} = m_{qi} h_{qi} - m_{qo} h_{qo} - Q_t \quad (36)$$

The equation of concentration for the shell side:

$$x_{qi} = x_{qo} \quad (37)$$

The mass conservation equation (Eq. (38)) and energy conservation equation(Eq. (39)) of concentrated solution for the tube side:

$$\frac{dG_g}{d\tau} = m_{gi} - m_{go} \quad (38)$$

$$\frac{d(G_g u_{go})}{d\tau} = m_{gi}h_{gi} - m_{go}h_{go} + Q_t \quad (39)$$

The equation of concentration for the tube side:

$$x_{gi} = x_{go} \quad (40)$$

The heat transfer equations (Eqs. (41) and (42)) between the tube side and shell side:

$$m_{qi}c_{qi}t_{qi} - m_{gi}c_{gi}t_{gi} = Q_t \quad (41)$$

$$Q_t = K_t A_t[(t_{qi} - t_{gi}) - 0.35(t_{go} - t_{gi}) - 0.65(t_{qi} - t_{qo})] \quad (42)$$

B. Dynamic Modeling of the LiBr Absorption Chiller

Based on the mathematical model described above, and taking thermodynamic properties of the LiBr solution and water into consideration, the LiBr-H_2O absorption chiller system was modeled in Simulink as shown in Figs. 2-6.

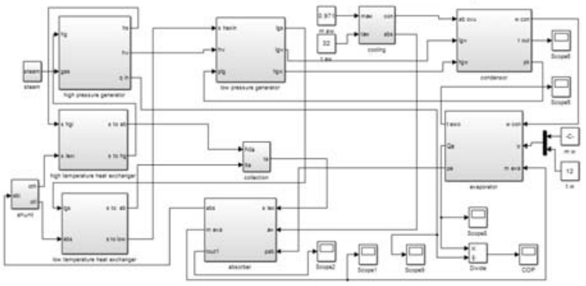

Figure 2. Simulink model of the LiBr absorption chiller.

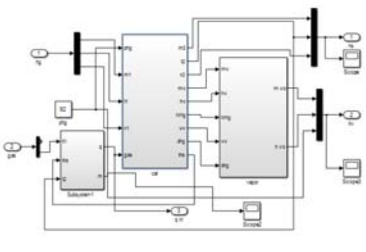

Figure 3. Simulation diagram of the HPG.

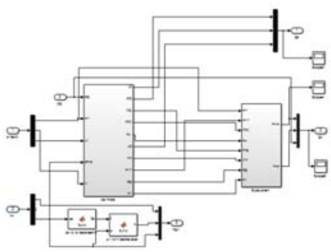

Figure 4. Simulation diagram of the LPG.

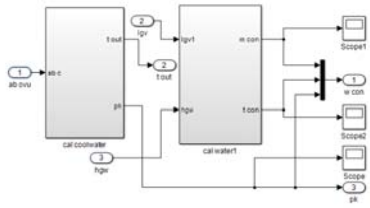

Figure 5. Simulation diagram of the condenser.

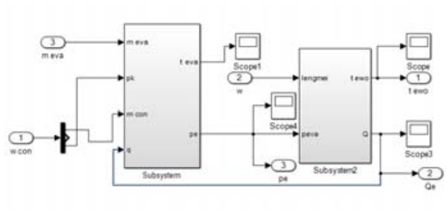

Figure 6. Simulation diagram of the evaporator.

According to the design principles of the basic parameters, various parameters of a small-sized LiBr absorption chiller were determined [9,10,16]. Operation parameters of the chiller are given in Table I.

TABLE I. OPERATION PARAMETERS OF THE LIBR-H_2OABSORPTION CHILLER

Pressure of heating steam (kPa)	800	Input heat inlet temperature (□)	170.4
Cooling capacity (kW)	11.81	Input heat capacity (kW)	9.24
Cooling water inlet temperature(□)	32	Cooling water outlet temperature(□)	37
Cooled water inlet temperature(□)	12	Cooled water outlet temperature(□)	7
Flow rate of cooling water (kg/s)	0.9710	Flow rate of cooled water (kg/s)	0.5706
Pressure of HPG(kPa)	92	Condensation pressure(kPa)	8.211
Evaporation pressure(kPa)	0.869	Absorption pressure(kPa)	0.869
Solution inlet temperature of HPG(□)	142.88	Solution inlet temperature of LPG(□)	83.52
Solution outlet temperature of HPG(□)	164.05	Solution outlet temperature of LPG(□)	91.76

As shown in Figs. 7-9, stabilization was gradually reached more than ten minutes after the system started, and ideal simulation results were obtained as follows.

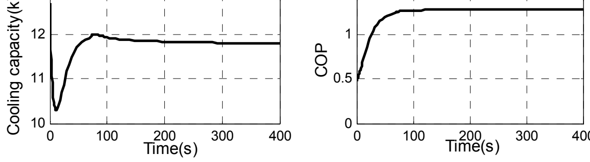

Figure 7. Dynamic response of cooling capacity and COP.

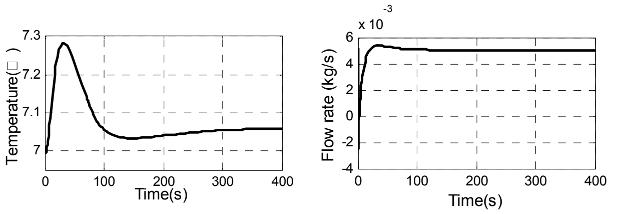

Figure 8. Dynamic response of cooled water outlet temperature and steam outlet flow in the evaporator.

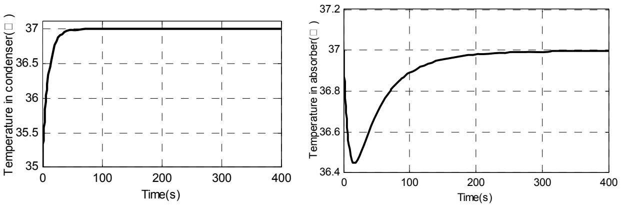

Figure 9. Dynamic response of cooling water outlet temperature in condenserand cooling water outlet temperaturein the absorber.

IV. ANALYSIS AND SIMULATION VERIFICATION OF THE EFFECTS OF EXTERNAL CONDITIONS ON THE PERFORMANCE OF LIBR ABSORPTION CHILLERS

The heat source, cooling water, and cooled water represent three important external conditions of LiBr-H_2O absorption chillers. Based on our simulation model, the operating data we retested under different external conditions by using the control variable method. By fitting curves, the relationships between external conditions and the refrigeration effect were found. By analyzing the effects of the external conditions on the system's operating parameters, the cooling output trends were obtained. Finally, conclusions were drawn by comparing the simulation results and theoretical results.

A. Effects of Heat Source

When the heating steam temperature rises (i.e., the heating steam pressure increases), the heat transfer performance of the generators is enhanced. Accordingly, the steam generating capacity in the generators is increased. Then, the HPG pressure and the condensing pressure are enhanced, the concentration of inlet solution from the absorber increases, and thus the absorptive capacity is strengthened. Consequently, the steam outlet flow rate and the cooling capacity are increased at last. Simultaneously, as the HPG pressure is enhanced, the heating capacity is decreased. Taking these two aspects into account, the COP of the chiller system increases. However, as the heating steam temperature rises, the reduction rate of the heating capacity slows, and the cooling capacity increases almost linearly. Therefore, the growth rate of the COP decreases.

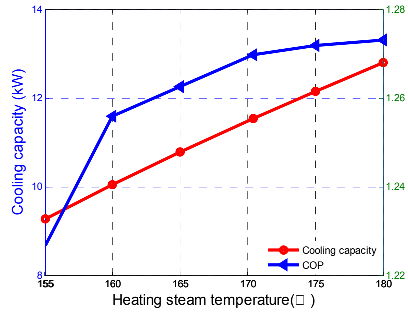

Figure 10. Variation of technical parameters with heating steam temperature

The simulation curves describing the chiller system's cooling capacity and COP versus the heating steam temperature are shown in Fig. 10. The figure shows that the COP and cooling capacity of the chiller system increase with an increase in the temperature of the heating steam. The cooling capacity versus the heating steam temperature is nearly linear, and the increased rate of the COP tends to decrease as the heating steam temperature increases. Thus, the simulation curves are in good agreement with the theory. In a practical application, the increase in the heating steam temperature can improve the COP and cooling capacity of the chiller system. However, if the heating steam temperature goes too high, it might lead to crystallization of the concentrated solution. In addition, a high level of LiBr solution will cause corrosion of components. Therefore, it is necessary to keep the heating steam pressure within a reasonable range.

B. Effects of Cooling Water

With the increment of cooling water inlet temperature, the dilute solution outlet temperature of the absorber goes up and the absorption effect becomes weak. Then, the outlet solution concentration of the absorber grows. Accordingly, the outlet steam content and the cooling capacity decrease. At the same time, the condensing water outlet temperature and condensing pressure of the condenser are elevated with the increasing cooling water inlet temperature. Hence, the steam content in the two generators declines, and the cooling capacity declines. The effects of these two aspects lead to a reduction of cooling capacity, and the COP of the chiller system decreases correspondingly.

With other parameters held constant and altering the inlet temperature of cooling water, the variation of the COP and cooling capacity with heating steam temperature is shown in Fig. 11based on simulation. We note that the COP and cooling capacity of the system decrease significantly as the cooling water temperature increases. This is in keeping with our theory analysis. Within a certain range, to increase the COP and cooling capacity, it is helpful to reduce the cooling water inlet temperature. However, in a practical application, the cooling water temperature is determined by the external environment. It is seldom to decrease the cooling water temperature.

When the cooling water flow rate increases, the solution outlet temperature of the absorber decreases and the absorption effect is strengthened. Then, the inlet solution concentration of the two generators is decreased while the steam content rises. Simultaneously, the condensing pressure and condensing water outlet temperature are reduced. This also raises the steam content. Thus, the COP and cooling capacity of the system increase. In addition, the increase in the cooling water flow rate can make the heat transfer coefficients *Kc* and *Ka* increase. Accordingly, the condensing pressure and the solution outlet temperature of the absorber are further reduced, and the COP and cooling capacity of the system increase naturally. But with the increased cooling water flow rate, the increasing rate of the condensing pressure and the solution outlet temperature of the absorber slow down, and the heat transfer coefficients *Kc* and *Ka* seem to be stable. Thus, the increasing rates of the COP and cooling capacity are slowed.

Variations of the COP and cooling capacity for different cooling water flow rates with other parameters constant, based on simulation results, are shown in Fig. 11. The figure shows that the two indices are enhanced with an increase in the cooling water flow rate, and the growth rate slows as the flow rate increases. These simulation results nearly fit with our theory analysis results. In an actual application system, it is worthwhile to improve chiller performance by increasing the cooling water flow rate. However, cooling water pipes will be seriously corroded if the flow rate is too large, and thus their service life will be affected.

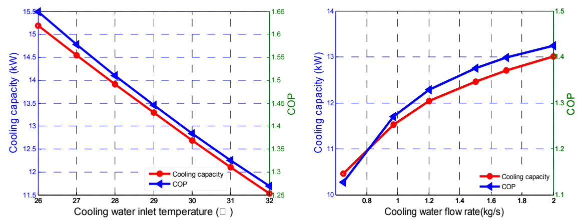

Figure 11. Variation of technical parameters withcooling water temperature and cooling water flow rate.

C. Effects of Cooled Water

With an increase in the cooled water outlet temperature, the evaporating pressure increases and the absorption capacity of the absorber is strengthened. Therefore, the solution outlet temperature of the absorber decreases. The steam content of the two generators rises, and the thermal load of the condenser and the condensing pressure increase. Eventually, the evaporating capacity of the evaporator and the cooling capacity are further increased. Simultaneously, the heating steam consumption is also elevated. Thus, the input heat increases, but the growth rate is slower than the cooling capacity. Hence, the COP of the system increases.

For the proposed simulation system, Fig. 12wasobtained by varying the cooled water outlet temperature and keeping the others constant. We note that the COP and cooling capacity increase nearly linearly with the increased cooled water outlet temperature, as shown in Fig.12. This coincides exactly with the theory results. It is evident that raising the outlet temperature of the cooled water can improve the performance of the chiller. But if the cooled water outlet temperature istoo high, the cryogen water's level in the solution capsule of the absorber will drop, or even starve the cryogen pump. Meanwhile, the cooling capacity will no longer increase after the temperature exceeds a certain value.

The cooled water inlet temperature and evaporating pressure decrease with the increased cooled water flow rate, which lowers the absorbing capacity of the absorber. Then, the solution outlet concentration of the absorber goes up, and the steam content in the generators and the cryogen of the refrigeration cycle decrease. Thus, the COP and cooling capacity of the system decline. On the other hand, the heat transfer coefficient *Ke* increases with the increased flow rate. Subsequently, the heat transfer capacity of the evaporator is improved. This makes the COP and cooling capacity of the system increase. Eventually, the effects of both are offset, so the change in the COP and cooling capacity are almost negligible for a different cooled water flow rate.

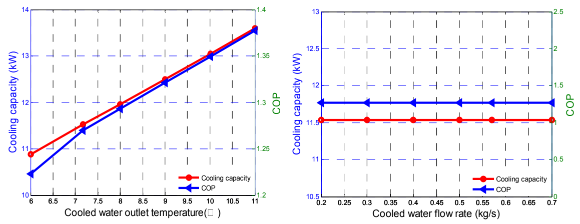

Figure 12. Variation of technical parameters with cooled water outlet temperature and cooled water flow rate.

Similarly, in the simulation system, by changing the cooled water flow rate and holding the other parameters constant, the performance curves are obtained. As shown in Fig. 12, the COP and cooling capacity of the system almost stay the same when the cooled water flow rate varies. This is in good agreement with the theory analysis. In an actual application, a cooled water flow rate that is either too high or too low is unsuitable. A high flow rate leads to corrosion of the water pipes, which will affect their service life. A low flow rate results in a sharp drop in the heat transfer coefficient, which will significantly reduce the COP and cooling capacity the system.

V. CONCLUSION

In this study, a simulation system of LiBr-H_2O double-effect absorption chillers was developed by building dynamic mathematical models of the components based on Simulink. Relationships between the refrigeration cycle and external conditions were found by using the controlling variables method. Changes in the heating steam pressure, cooling water temperature, cooling water flow rate, cooled water temperature, or cooled water flow rate can affect the state of the chiller system. Therefore, the proposed system could be helpful in guiding the design of a practical application, and in the optimization of control in a field test.

As described in Section 3, the effects of the main external conditions were obtained through simulation and analysis. The relationship between external parameters and refrigeration effects are summarized in Table II. Within a certain range, increasing the heating steam pressure, cooling water flow rate, and cooled water outlet temperature, and decreasing the cooling water inlet temperature, will improve the performance of the chiller system. However, changes in the cooled water flow rate have no effect on performance.

When the load of the chiller system varies, it is necessary to match the load by adjusting the parameters of the external conditions. This satisfies the needs of users, and increases the energy use. For example, suppose the cooled water outlet temperature is the regulatory signal of the LiBr-H_2O absorption chiller. When the load increases, the cooled water inlet temperature increases and the cooled water outlet temperature increases. At this point, increasing the heating steam temperature (which increases the heating steam pressure), decreasing the cooling water inlet temperature, or increasing the cooling water flow rate can make the cooled water outlet temperature return to the set value. The three regulation methods can also improve the COP of the system. When the load decreases, the inlet temperature and outlet temperature of the cooled water increase. At this moment, decreasing the heating steam temperature (i.e., decreasing the heating steam pressure), increasing the cooling water inlet temperature, or decreasing the cooling water flow rate can keep the cooled water outlet temperature constant. However, in this situation, the COP of the system will be reduced by the three methods. During practical applications, it is difficult to alter the temperature of cooling water. Therefore, altering the heating steam temperature and cooling water flow rate are the main ways to match the load of the chiller system.

Table II. ARIATION OF THE COP AND COOLING CAPACITY WITH EXTERNAL CONDITIONS

	Variation trend of cooling capacity	Variation trend of COP
Increasing heating steam temperature	↑	↑
Increasing cooling water inlet temperature	↓	↓
Increasing cooling water flow rate	↑	↑
Increasing cooled water outlet temperature	↑	↑
Increasing cooled water flow rate	—	—

ACKNOWLEDGMENT

This work is supported by the National Natural Science Foundation of China (nos. 61473174 and51376110), the Specialized Research Fund for the Doctoral Program of Higher Education (no.20130131130006), a project funded by the China Postdoctoral Science Foundation (no.2014M551907),and the Independent Innovation Foundation of Shandong University (no. 2013ZRQP002).

Nomenclature

A	effective heat transfer area (m^2)	*c*	specific heat capacity ($kJkg^{-1}$□$^{-1}$)	*G*	mass (kg)
h	specific enthalpy ($kJ\ kg^{-1}$)	*K*	heat-transfer coefficient (kWm^{-2}□$^{-1}$)	*m*	mass flow rate (kg/s)
Q	thermal power (kW)	*t*	temperature (□)	*u*	internal energy power (kW)
V	volume (m^3)	*x*	concentration(%)	*ρ*	density (kg/m³)
τ	time (s)				

Subscripts

a	absorber	*c*	condenser	*co*	outlet cryogen water of condenser
e	evaporator	*ei*	inlet of evaporator	*va*	vapor
g	tube side of heat exchanger	*gi*	tube side inlet of heat exchanger	*go*	tube side outlet of heat exchanger
h	HPG	*ho*	outlet cryogen water of condensing pipes in LPG	*i*	inlet
l	LPG	*lo*	outlet vapor of LPG	*o*	outlet
q	shell side of heat exchanger	*qi*	shell side inlet of heat exchanger	*qo*	shell side outlet of heat exchanger
s	solution	*sh*	barrel	*t*	heat exchanger
v	generated steam	*vo*	outlet vapor	*w*	cooling water or cooled water
wi	inlet of cooling water or cooled water	*wo*	outlet of cooling water or cooled water		

REFERENCES

[1] S. Li,Z. Xu,"The research trends overview of solar heat utilization," J. Anhui Agricultural Sci, vol. 41, pp. 3103-3104, 2013.

[2] G. Chicco,P.Mancarefla,"Distributed multi-generation: A comprehensive view," Renewable and Sustainable Energy Rev,vol. 13, pp. 535-551,2009.

[3] National Bureau of Statistics of China (NBS),China Statistical Yearbook,Chinese Statistics Press, Beijing 2012.

[4] World Bank,China 2030: Building a Modern, Harmonious, and Creative HighIncome Society, World Bank,Washington, DC, 2013.

[5] F.J. Sáez-Martínez, J.Mondéjar-Jiménez, J.A. Mondéjar-Jiménez,"The energy challenge: energy and environment knowledge week E2KW 2013," J. Cleaner Production, vol. 86, pp.471-473, 2015.

[6] M.Shan, P.Wang, J.Li, G.Yue, X.Yang,"Energy and environment in Chinese rural buildings: Situations, challenges, and intervention strategies,"Building and Environment, vol. 91,pp. 271-282, 2015.

[7] L.Li, H.Mu, W.Gao, M.Li,"Optimization and analysis of CCHP system based on energy loads coupling of residential and office buildings,"Appl.Energy, vol. 136, pp. 206-216, 2014.

[8] C.Qin, J.Tang, Y.Zhang,"An efficient algorithm for CCHP system sizing and an operational optimization model based on LP,"J. Natural Gas Sci.and Eng, vol. 25, pp.189-196, 2015.

[9] C.P. Xin,The Practical Tutorial of LiBr-waterAbsorption Refrigerating Machine, Electronics Industry Press, Beijing, pp. 93-113, 2004.

[10] L. Wang,Principle and application of small-sized absorption refrigerator,China Building Industry Press,Beijing, pp. 21-110, 2011.

[11] C.B. Wei, S.X.Lu,Z.Y. Zhou, Practical Handbook of Refrigeration and Air Conditioning Engineering(second edition),Machinery Industry Press,Beijing, pp. 230-321, 2010.

[12] Y.P.Chen, K.Y. Wang,M.H.Shi, "Analysis on performance of 1.x lift LiBr absorption refrigeration cycle,"J.Eng.Thermophysics, vol. 26, pp. 193-195,2005.

[13] Y.Chen, Y.Q.Zhu, W.Geng,"SE/DL absorption refrigeration cycle driven by low temperature heat resources," Acta Energiae Solaris Sinica, vol. 23, pp.1-6, 2002.

[14] B. Bakhtiari, L. Fradette, R. Legros,J. Paris,"Model for analysis and design of H_2O-LiBr absorption heat pumps,"Energy Conversion and

Management, vol.52, pp.1439-1448, 2011.

[15] L. Xie,H.S.Zhang,"Research of dynamic state simulation of double effect LiBr absorption refrigeration system,"Fluid Machinery. Vol.37, pp. 75-79, 2009.

[16] M.M. Maghanki, B. Ghobadian, G. Najafi et al.,"Micro combined heat and power (MGHP) technologies and applications,"Renewable and Sustainable Energy Rev.vol. 28, pp. 510-524, 2013.