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A State-Space Model for Dynamic Simulation of a Single-Effect LiBr/H_2O Absorption Chiller

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ABSTRACT A simplified transient dynamic model for a single-effect absorption refrigeration chiller using LiBr/H_2O as the working pair is developed in this paper. The model contains a series of sub-models presenting the components composing the chiller. The development of the sub-models of the four main components (i.e. generator, condenser, evaporator, and absorber), where the solution or refrigerant goes through phase change with complex heat and mass transfer progress, is based on the conservation of mass and energy, by means of dynamic modeling approaches. The throttling devices, solution pump, and solution heat exchanger (SHE) adopt quasi-steady models because their thermal inertia is much smaller compared with the main components. The system model proposed in this study is finally simplified and expressed in the form of state space matrix, which enhances the model portability and computational efficiency. Two approaches are carried out to verify the model accuracy at the end of the paper, static comparison between the steady simulation results and the design parameters under the on-design condition, the dynamic comparison between the simulation results and experimental data under the same inputs. Both comparison results show the model accuracy.

INDEX TERMS Absorption chiller, dynamic model, heat transfer, simulation.

NOMENCLATURE

A	area (m^2)
COP	Coefficient of Performance (-)
$c_{p,w}$	specific heat at constant pressure (kJ/kg·K)
f	frequency (Hz)
g	gravitational acceleration (m/s^2)
h	specific enthalpy (kJ/kg)
$\dot{m}$	mass flow rate (kg/s)
M	mass (kg)
P	pressure (kPa)
T	temperature (°C)
t	time (s)
$\dot{Q}$	heat transfer rate (kW)
V	volume (m^3)
UA	global thermal surface conductivity (kW/K)
X	concentration of the LiBr/H_2O solution (-)
Z	level height (m)
z	height difference (m)

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Greek symbols

ρ	density (kg/m^3)
η	efficiency (-)

Subscripts

abs	absorber
con	condenser
clw	chilled water
cw	cooling water
d	on-desigin condition
eva	evaporator
gen	generator
hw	hot water
in, *out*	inlet, outlet
l	liquid phase
p	pump
r	refrigerant
strong	strong solution
she	solution heat exchanger
v	vapor phase
vl	valve
weak	weak solution

I. INTRODUCTION

In recent years, absorption refrigeration systems (ARSs) have been widely used in refrigeration and air-conditioning engineering because of their advantages of utilizing low-grade heat source, low running noise, a wide range of refrigerating capacity regulation, strong adaptability to the change in external conditions, simple manufacturing, convenient operation, environmental friendliness, and so on. However, the low efficiency of the absorption refrigeration system compared to the compressive refrigeration system limits its development and popularity. Many researchers devote to improve the coefficient of performance of the absorption refrigeration system.

While operating, the absorption chiller always works in the transition process from one part load condition to another due to the influence of environmental factors, as well as changes in user demands. It is quite an expensive and unrealistic work to study the dynamic behavior of the absorption refrigeration system in the field considering the huge size, high manufacturing costs and long adjustment time of the absorption chiller. Computer dynamic simulation can effectively reveal the interrelations of the internal parameters during the chiller running under off-design conditions, and thus provide favorable conditions for the design of an effective control scheme. Several theoretical and experimental studies have been carried out by various researchers to mathematically model an actual absorption refrigeration cycle.

Jeong *et al.* [1] proposed a dynamic lumped parameter model of a steam-driven LiBr/H_2O absorption heat pump to investigate the effects of various design parameters and operating conditions on the system performance. In this model, a set of partial differential equations are built for describing the transient behavior of the system. Solution mass storage and thermal capacity heat storage are assumed. The mass flow of vapor and solution is determined by the pressure difference of vessels. Fu *et al.* [2] developed a dynamic model library of absorption systems for a simulation software Dymola, in which lumped parameter method is used to describe each system component involving two-phase equilibrium. Matsushima *et al.* [3] proposed a dynamic simulation program for predicting the transient behavior of absorption chillers with arbitrary configurations. A new algorithm is put forward to calculate the solution flow rate based on the pressure difference and flow resistance between the generators and the absorber. Kohlenbach and Ziegler [4], [5] derived and experimentally verified a dynamic model for a single-effect LiBr/H_2O absorption refrigeration system based on external and internal steady-state enthalpy balances of each main component. Different from other works, the generator and absorber vessels are modeled as a serial connection of tube bundle heat exchanger and a solution sump in the model to maintain different temperatures and concentrations in one vessel. Moreover, all thermal capacities are divided into an external part and internal part influenced by external and internal temperature respectively. Simple constant physical parameters of water and solution are used instead of detailed calculation. Besides, mass storage terms, thermal heat storage terms and the delay time in the solution cycle are also discussed. Evola *et al.* [6] put forward a mathematical model of a single-effect LiBr/water absorption by applying mass and energy balances to the internal components of the system. This model also accounts for the non-steady behavior due to thermal and mass storage in the components. Ochoa *et al.* [7] implemented a dynamic model of a single-effect absorption chiller on the Matlab platform, and resolve the non-linear equations of the system through the finite difference method. This model can simulate and predict the behavior of internal and external parameters when the system suffers disturbances in the power supply or thermal load. Martinho *et al.* [8] built a dimensionless dynamic mathematical model to analyze the system response according to operating and geometric parameters. In this paper, mass accumulation in the components is neglected and the system is divided into the thermal compressor region and the pure refrigerant region.

Along with the widespread application of the computer, artificial intelligence technologies represented by the neural network have also been introduced to the modeling and simulation of the absorption refrigeration systems [9]–[11].

In this study, a simplified dynamic model of a single-effect absorption chiller driven by hot water is proposed to satisfy the demand for control system design and optimization. Firstly, the sub-models of each components composing the absorption system are developed based on a series of assumptions. Considering that the dynamic behavior of the main components influences the system performance much, dynamic modeling methods are applied. The rest components adopt quasi-steady modeling methods or empirical formulas. After that, an initial overall model of the chiller is established by integrating the sub-models. Then, the model is further simplified and rewritten in state space form with six state variables. Finally, the system model is verified through two approaches.

II. SYSTEM DESCRIPTION

Fig 1 represents the schematic diagram of a typical single-effect LiBr/H_2O absorption refrigeration chiller driven by hot water. The chiller basically consists of five heat exchangers (generator, condenser, evaporator, absorber and solution heat exchanger), a solution pump and two throttling devices, connected by tubes. The generator, condenser, evaporator and absorber are the main components of the chiller, where heat exchange takes place. The solution heat exchanger can recover part of the heat and improve the efficiency of the chiller. All the heat exchangers are shell-and-tube heat exchangers of the counter flow type. The solution pump supplies power for the circulation of the solution and is the unique moving part in the chiller. The solution pump, together with the two devices, divides the chiller into the high-pressure side (P_{con}) and the low-pressure side (P_{eva}). The concrete operational process of this absorption chiller has been described in the paper [6].

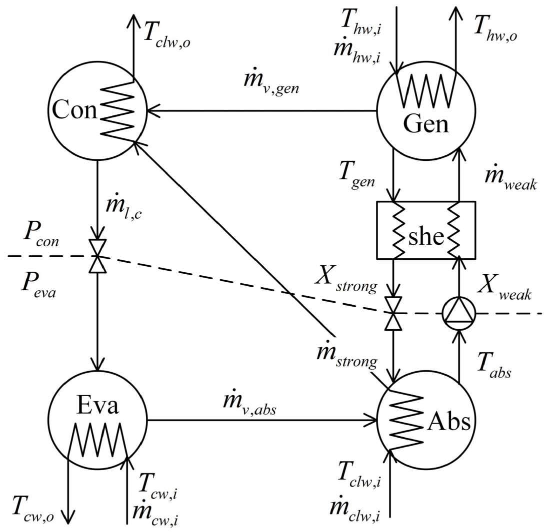

FIGURE 1. Schematic of a single-effect LiBr/H_2O absorption system.

III. FORMULATION OF THE DYNAMIC MODEL

The simulation procedure for the transient behavior of the absorption refrigeration system involves casting sub-models for each of the components making up the system, including the main components, the solution heat exchanger, the solution pump, the throttling devices and the external water circulation. Then, the overall system model can be obtained by combing the sub-models according to the input-output relationship between the components.

There are mainly two kinds of coupled dynamic behavior inside the chiller, the mass accumulation and the energy accumulation in the main components. The dynamic behavior influences the performance of the chiller much and is the key to build the system model. In each of the main components, the mass accumulation is respectively governed by the mass flow rate of the incoming and leaving fluids. The energy accumulation is a bit more complex, in relation with not only the energy of the inlet and outlet fluid but also the heat transfer rates between the external circulation and the internal circulation. As illustrated in Fig 1, the external circulation involves the hot water, cooling water and chilled water. The internal circulation involves the refrigerant, weak solution and strong solution. The overall system model refers to the mass flow and temperature of each circulation, as well as the concentrations of the solution.

The mathematical modeling of the main components is based on the laws of the conservation of mass and energy, and developed by considering that the variables vary in time. Lumped parameter method is adopted in the modeling process to describe each of the main components with a temperature, a solution concentration and a pressure. In order to facilitate the formulation and implementation of the absorption refrigeration system model, the following assumptions are made:

a. The temperature, pressure and concentration are homogeneous within each main components;
b. Only two pressure levels are considered: high (generator-condenser) and low (absorber-evaporator);
c. The fluid flows out each component at the same state with that inside the component;
d. The thermal storage of the shell and tube of the exchanger is neglected;
e. No pressure drop except through the throttling valves;
f. The throttling process is isenthalpic;
g. Heat loss to the surroundings is neglected;
h. The power of the solution pump is negligible;
i. Constant solution heat exchanger efficiency.

A. GENERATOR AND ABSORBER

The overall mass accumulation change rate over the time in the shell side of the generator depends on the mass flow of the three streams incoming and leaving the generator. The mass conservation equation can be written as:

$$\frac{dM_{gen}}{dt} = \dot{m}_{weak} - \dot{m}_{strong} - \dot{m}_{v,gen} \tag{1}$$

where M_{gen} is the cumulative mass in the generator at each moment, including the mass of the refrigerant vapor and the LiBr solution; $\dot{m}_{weak}$, $\dot{m}_{strong}$ and $\dot{m}_{v,gen}$ are the mass flow rates of the weak solution, strong solution and the refrigerant vapor, respectively.

Considering that the density of the liquid phase is tens of thousands of times larger than that of the vapor phase, the little mass and heat of the refrigerant vapor are negligible compared to that of the LiBr solution within the margin of error. Consequently, the cumulative mass in the generator M_{gen} can approximately represent the mass of the LiBr solution in the generator. This approximation can further simplify the modeling and will also be adopted in the following modeling process of other main components of the chiller.

The continuity equation for the absorbent (LiBr) in the generator is written as:

$$\frac{d}{dt}\left(M_{gen}X_{strong}\right) = \dot{m}_{weak}X_{weak} - \dot{m}_{strong}X_{strong} \tag{2}$$

where X_{weak} and X_{strong} respectively denotes the concentration of the weak solution and strong solution.

The energy balance equations for the LiBr solution in the generator is written as:

$$\frac{d}{dt}\left(M_{gen}h_{strong}\right) = \dot{m}_{weak}h_{weak,gen} - \dot{m}_{strong}h_{strong} - \dot{m}_{v,gen}h_{v,gen} + \dot{Q}_{gen} \tag{3}$$

where h_{strong} is the specific enthalpy of the solution at the outlet of the generator; $h_{weak,gen}$ is the specific enthalpy of the solution at the inlet of the generator; $h_{v,gen}$ is the specific enthalpy of the superheated refrigerant vapor in the generator; $\dot{Q}_{gen}$ is the heat transfer rate from the hot water to the generator.

Inserting Eq. (1) into Eq. (2) reads:

$$M_{gen}\frac{dX_{strong}}{dt} = \left(\dot{m}_{v,gen} - \dot{m}_{weak}\right)X_{strong} + \dot{m}_{weak}X_{weak} \tag{4}$$

Inserting Eq. (1) into Eq. (3) reads:

$$\left(M_{gen}c_{p,strong}\right)\frac{dT_{gen}}{dt}=\dot{m}_{weak}\left(h_{weak,gen}-h_{strong}\right)+\dot{m}_{v,gen}\left(h_{v,gen}-h_{strong}\right)+Q_{gen} \tag{5}$$

where T_{gen} and $c_{p,strong}$ are the average temperature and specific heat capacity of the solution in the shell side of the generator, respectively.

The sub-model of the generator is made up of the three differential equations, Eq.(1), (4) and (5). The model of the absorber can also be established by the same method.

$$\frac{dM_{abs}}{dt}=\dot{m}_{strong}-\dot{m}_{weak}+\dot{m}_{v,abs} \tag{6}$$

$$M_{abs}\frac{dX_{weak}}{dt}=\dot{m}_{strong}X_{strong}-\left(\dot{m}_{strong}+\dot{m}_{v,abs}\right)X_{weak} \tag{7}$$

$$\left(M_{abs}c_{p,weak}\right)\frac{dT_{abs}}{dt}=\dot{m}_{strong}\left(h_{strong,abs}-h_{weak}\right)+\dot{m}_{v,abs}\left(h_{v,abs}-h_{weak}\right)-Q_{abs} \tag{8}$$

where M_{abs} and T_{abs} are the accumulative mass and average temperature of the solution in the absorber; $\dot{m}_{v,abs}$ is the mass flow rate of the refrigerant vapor; $c_{p,weak}$ is the specific heat capacity of the weak solution; h_{weak} and $h_{strong,abs}$ are the specific enthalpy of the solution at the outlet and inlet of the absorber; $h_{v,abs}$ is the enthalpy of the vapor in the absorber.

B. CONDENSER AND EVAPORATOR

The mass accumulation change of the refrigerant water in the condenser over the time is govern by the mass flow rates of the inlet refrigerant water vapor and the outlet liquid refrigerant water.

$$\frac{dM_{con}}{dt}=\dot{m}_{v,gen}-\dot{m}_{l,c} \tag{9}$$

where M_{con} is the accumulative mass of the liquid refrigerant in the condenser; $\dot{m}_{l,c}$ is the mass flow rate of the outlet liquid refrigerant water.

The energy balance equation for the refrigerant in the condenser is:

$$\frac{d\left(M_{con}h_{l,c}\right)}{dt}=\dot{m}_{v,gen}h_{v,gen}-\dot{m}_{l,c}h_{l,c}-\dot{Q}_{con} \tag{10}$$

where T_{con} is the average temperature in the condenser; $\dot{Q}_{con}$ denotes the heat transfer rate from the condenser to the cooling water; $h_{l,c}$ is the specific enthalpy of the liquid water under condensing pressure.

Similarly, Eq. (9) and (10) can also be extended to the modeling of the evaporator:

$$\frac{dM_{eva}}{dt}=\dot{m}_{l,c}-\dot{m}_{v,abs} \tag{11}$$

$$\frac{d\left(M_{eva}h_{l,eva}\right)}{dt}=\dot{m}_{l,c}h_{l,c}-\dot{m}_{v,abs}h_{v,abs}+\dot{Q}_{eva} \tag{12}$$

where M_{eva} is the accumulative mass of the liquid refrigerant in the condenser; $h_{l,eva}$ is the specific enthalpy of the liquid refrigerant under evaporating pressure; $\dot{Q}_{eva}$ denotes the heat transfer rate from the chilled water to the evaporator.

C. THE SOLUTION HEAT EXCHANGER

The solution heat exchanger is considered as a double pipe counter-flow heat exchanger. The strong solution from the generator preheats the weak solution from the absorber in the solution heat exchanger. The heat transfer process can be simply expressed in terms of the effectiveness of the heat exchanger [12].

$$\eta_{she}=\frac{T_{strong}-T_{strong,abs}}{T_{strong}-T_{weak}} \tag{13}$$

Rearranging Eq. (13) reads,

$$T_{strong,abs}=T_{strong}-\eta_{she}\left(T_{strong}-T_{weak}\right) \tag{14}$$

The energy balance of the weak solution and strong solution in the solution heat exchanger can be expressed as follows:

$$\dot{m}_{strong}c_{p,strong}\left(T_{strong}-T_{strong,abs}\right)=\dot{m}_{weak}c_{p,weak}\times\left(T_{weak,gen}-T_{weak}\right) \tag{15}$$

Inserting Eq. (14) into Eq. (15) reads,

$$T_{weak,gen}=T_{weak}+\frac{\dot{m}_{strong}c_{p,strong}}{\dot{m}_{weak}c_{p,weak}}\eta_{she}\left(T_{strong}-T_{weak}\right) \tag{16}$$

D. THE SOLUTION PUMP

The mass flow rate of the weak solution from the absorber is determined by the solution pump. The volume flow rate of the solution pump is assumed to be in proportion to the pump rotary speed, adjusted by the frequency converter. The mass flow rate of the weak solution can be expressed as:

$$\dot{m}_{weak}=k\rho_{weak}f_pV_p \tag{17}$$

where k is a proportional coefficient determined by the properties of the pump; ρ_{weak} is the density of the weak solution; f_p is the solution pump frequency; V_p is the inner volume of the pump.

E. THE THROTTLING DEVICES

The liquid flow across the throttling devices is an adiabatic process, driven by the force of gravity and the pressure difference between connected components. The mass flow rate can be calculated from the following empirical formula of throttling valves.

$$\dot{m}_l=A_{vl}C_{vl}\sqrt{\rho_l\left(\Delta P+\rho_l g\left(Z+z\right)\right)} \tag{18}$$

where A_{vl} and C_{vl} are the minimum cross-section area and flow coefficient of the valve; ρ_l is the density of the liquid flowing through the valve; ΔP is the pressure difference between the connected components; g is the gravitational acceleration; z is height difference between the upper component outlet and the lower component inlet; Z is the level height of the liquid in the upper component, which is in proportion to the accumulative mass.

F. THE HEAT TRANSFER PROCESS

The heat transfer rates in the four main components ($\dot{Q}_{gen}$, $\dot{Q}_{con}$, $\dot{Q}_{eva}$, $\dot{Q}_{abs}$) between the internal circulation and the external circulation of the chiller are determined by the heat transfer equation of the heat exchanger $\dot{Q} = UA\Delta T$. The heat transfer equation is a function of the overall heat transfer coefficient (U), the effective area in contact with the two fluids in each main component (A) and the temperature difference between the two fluids (ΔT). Usually, there are two ways to determine the temperature difference, the logarithmic mean temperature difference (LMPD) approach [13] and the arithmetic mean temperature difference (AMPD) approach [4]. The LMPD approach is adopted in this paper. The heat transfer coefficient U is not a constant value. U can be calculated from the following formula:

$$U = U_d\,(\dot{m}/\dot{m}_d)^{0.8} \tag{19}$$

where U_d and $\dot{m}_d$ are the heat transfer coefficient and mass flow rate under designed condition. In this paper, the mass flow rate refers to the external water circulation. The heat transfer rate can also be calculated from the tube side of the heat exchanger:

$$\dot{Q} = \dot{m}_w c_{p,w}\left(T_{w,in} - T_{w,out}\right) \tag{20}$$

where $\dot{m}_w$, $c_{p,w}$, $T_{w,in}$ and $T_{w,out}$ are the mass flow rate, specific heat capacity, inlet temperature and the outlet temperature of the external water circulation.

Taking the generator as an example:

$$\dot{Q}_{gen} = \dot{m}_{hw} c_{p,w}\left(T_{hw,i} - T_{hw,o}\right) \tag{21}$$

$$\dot{Q}_{gen} = (UA)_{gen}\frac{\left(T_{hw,i} - T_{gen}\right) - \left(T_{hw,o} - T_{gen}\right)}{\ln\left(\left(T_{hw,i} - T_{gen}\right)/\left(T_{hw,o} - T_{gen}\right)\right)} \tag{22}$$

Combining Eq. (21) and (22), the heat transfer rate in the generator and the outlet temperature of the hot water can be calculated.

$$\dot{Q}_{gen} = \dot{m}_{hw} c_{p,w}\left(1 - e^{-(UA)_{gen}/\dot{m}_{hw}c_{p,w}}\right)\left(T_{hw,i} - T_{gen}\right) \tag{23}$$

$$T_{hw,o} = T_{hw,i} - \left(1 - e^{-(UA)_{gen}/\dot{m}_{hw}c_{p,w}}\right)\left(T_{hw,i} - T_{gen}\right) \tag{24}$$

Similar results can also be achieved for the other main components. Specially, owing to that the cooling water successively flows through the absorber and the condenser, the inlet cooling water temperature of the condenser equals to the outlet cooling water temperature of the absorber.

IV. MODEL SIMPLIFICATION

In the previous half of this article, the sub-models of the components making up the absorption chiller have been built. The overall model of the chiller can be then preliminarily established after integrating the sub-models. Obviously, the initial system model includes tens of equations and is fairly complex. The aim of the following work is to reduce the complexity of the model by model reduction.

Firstly, the overall mass of the absorbent (M_{LiBr}) and the overall mass of the refrigerant and the absorbent (M_{total}) within the chiller are determined while the chiller was manufactured and thus are constant values. If the length of the connecting pipes between the main components are further neglected, or in other words, the connecting pipes are treated as part of the main components, the mass conservation equation of the absorbent and the overall mass conservation equation within the chiller can be respectively written as:

$$M_{gen}X_{strong} + M_{abs}X_{weak} = M_{LiBr} \tag{25}$$

$$M_{gen} + M_{con} + M_{eva} + M_{abs} = M_{total} \tag{26}$$

It can be observed from Eq. (25) and (26) that while any four of the six variables in the left side of the equations are determined, the rest two can be calculated and need to be eliminated as intermediate variables in the modeling process. In this paper, the accumulative mass in the absorber (M_{abs}) and the accumulative mass in the evaporator (M_{eva}) are chosen as intermediate variables.

$$M_{abs} = \left(M_{LiBr} - M_{gen}X_{strong}\right)/X_{weak} \tag{27}$$

$$\begin{aligned} M_{eva} &= M_{total} - M_{gen} - M_{con} \\ &\quad - \left(M_{LiBr} - M_{gen}X_{strong}\right)/X_{weak} \end{aligned} \tag{28}$$

Secondly, the density (ρ_{sol}), enthalpy (h_{sol}), specific heat capacity ($c_{p,sol}$), and surface vapor pressure (P_{sol}) of the LiBr solution are functions of the temperature and the concentration of the solution.

$$\rho_{sol} = \rho\left(T_{sol}, X_{sol}\right) \tag{29}$$

$$h_{sol} = h\left(T_{sol}, X_{sol}\right) \tag{30}$$

$$c_{p,sol} = c\left(T_{sol}, X_{sol}\right) \tag{31}$$

$$P_{sol} = P\left(T_{sol}, X_{sol}\right) \tag{32}$$

Moreover, the refrigerant water within the chiller exists in three states, saturated liquid phase, saturated vapor phase and superheated vapor phase. The refrigerant in the state of superheated vapor phase only exists in the generator. For the refrigerant in the saturated state, both the liquid phase and vapor phase, the physical parameters can be calculated from the condensing pressure and evaporating pressure. As for the calculation of the superheated vapor phase, the condensing pressure and the average temperature of the vapor in the generator are used. The condensing pressure and evaporating pressure respectively equal to the surface vapor pressure of the strong solution inside the generator and weak solution in the absorber.

As a consequence of previous analysis, the state of the chiller can totally be determined by six variables, the accumulative mass in the generator (M_{gen}), the average temperature in the generator (T_{gen}), the concentration of the strong solution in the generator (X_{strong}), the accumulative mass in the condenser (M_{con}), the average temperature in the absorber (T_{abs}) and the concentration of the weak solution in the absorber (X_{abs}). The six variables are selected as the state variables of the state-space model.

$$x = \left[M_{gen} \;\; T_{gen} \;\; X_{strong} \;\; M_{con} \;\; T_{abs} \;\; X_{weak}\right]^T \tag{33}$$

Considering that the latent heat of the refrigerant water is much larger than the sensible heat, the heat transfer in the condenser and evaporator can be treated as a process of a phase change of the refrigerant. Then, Eq. (10) and (12) can be simplified as the mass flow rates of the refrigerant condensed in the condenser and evaporated in the evaporator.

$$\dot{m}_{v,gen} = \dot{Q}_c / \left(h_{v,gen} - h_{l,c}\right) \tag{34}$$

$$\dot{m}_{eva} = \dot{Q}_{eva} / \left(h_{v,abs} - h_{l,c}\right) \tag{35}$$

Then, the simplified model of the absorption system only consists of six differential equations, Eq. (1), (5), (4), (9), (8)and (7). The system model can be rewritten in the following state-space form ($\dot{\boldsymbol{x}} = f(\boldsymbol{x}, \boldsymbol{u})$).

The input vector for the model is:

$$u = \left[\dot{m}_{hw} \;\; \dot{m}_{clw} \;\; \dot{m}_{cw} \;\; T_{hw,i} \;\; T_{clw,i} \;\; T_{cw,i} \;\; f_p\right]^T \tag{37}$$

where $\dot{m}_{hw}$, $\dot{m}_{clw}$, $\dot{m}_{cw}$, $T_{hw,i}$, $T_{clw,i}$ and $T_{cw,i}$ are the mass flow rates and inlet temperatures of the external water circuits.

V. VALIDATION OF THE MODEL

The system model is implemented in Matlab/Simulink, a general purpose simulation software, to verify the accuracy of the proposed model. The physical parameters of the LiBr solution used in the simulation process, including density, enthalpy, specific heat capacity and surface vapor pressure, are calculated by the fitting formulas proposed by Kaita [14]. The valid range of the formulas, from concentrations of 40-65% and temperatures of 40-210° C, can completely cover the operating conditions of a single-effect absorption chiller. The thermodynamic properties of the refrigerant water are calculated by REFPROP, a refrigerant property parameter query software developed by the national institute of standards and technology (NIST).

A small-sized single effect LiBr/H$_2$O absorption chiller with a refrigerating capacity of 6.5kW installed in the laboratory of Tianjin University is used to validate the model. The on-design parameters of the absorption chiller supplied by the manufacturer, including the overall mass of the absorbent, the overall mass of the filled solution, the heat transfer coefficients of the main components and the coefficient of the solution heat exchanger, as listed in Table 1, are used to determine the parameters of the model.

TABLE 1. ON-Design parameters.

Symbol	Description	Value
M_{LiBr}	Overall mass of LiBr (kg)	32.5
M_{total}	Overall mass of LiBr and H$_2$O (kg)	67.5
$(UA)_{gen,d}$	Generator heat transfer coefficient (kW/K)	1.37
$(UA)_{con,d}$	Condenser heat transfer coefficient (kW/K)	3.20
$(UA)_{eva,d}$	Evaporator heat transfer coefficient (kW/K)	2.03
$(UA)_{abs,d}$	Absorber heat transfer coefficient (kW/K)	1.82
$z_{gen,abs}$	Generator-absorber height difference (m)	0.2
$z_{con,eva}$	Condenser-evaporator height difference (m)	0.2
η_{she}	Solution heat exchanger coefficient (-)	0.6

TABLE 2. Input parameters under on-design condition.

Symbol	Description	Value
$\dot{m}_{hw}$	Hot water mass flow rate (kg/s)	0.32
$\dot{m}_{clw}$	Cooling water mass flow rate (kg/s)	0.65
$\dot{m}_{cw}$	Chilled water mass flow rate (kg/s)	0.26
f_p	Solution pump frequency (Hz)	30.8
$T_{hw,i}$	Inlet hot water temperature (°C)	95
$T_{clw,i}$	Inlet cooling water temperature (°C)	32
$T_{cw,i}$	Inlet chilled water temperature (°C)	15

Firstly, a comparison of key parameters between the steady simulation results of the proposed model and the designed parameters, including the concentrations of the solution, the refrigerating capacity, the outlet temperature of the cooling water and the coefficient of performance (COP) of the chiller, are carried out under on-design condition. The input parameters of the simulation are listed in Table 2. The initial values of the state variables for the simulation are set at random within a reasonable range.

$$\boldsymbol{x}_0 = \left[20 \;\; 50 \;\; 55 \;\; 5 \;\; 55 \;\; 39\right]^T \tag{38}$$

The comparison results, as shown in Table 3, indicate that the simulation results exhibit good consistency with the designed parameters under the on-design condition, with a maximum relative error of 2.15%.

In addition, an experiment has been designed to test the dynamic performance of the model. The schematic diagram of the experimental system is shown in Fig. 2. The heater mainly consists of a series of solar hot water pipes. Beside, an auxiliary electric boiler is equipped to avoid shortage of

$$\begin{bmatrix} \dot{M}_{gen} \\ \dot{T}_{gen} \\ \dot{X}_{strong} \\ \dot{M}_{con} \\ \dot{T}_{abs} \\ \dot{X}_{weak} \end{bmatrix} = \begin{bmatrix} \dot{m}_{weak} - \dot{m}_{strong} - \dot{m}_{v,gen} \\ \left(\dot{m}_{weak}\left(h_{weak,gen} - h_{strong}\right) - \dot{m}_{v,gen}\left(h_{v,gen} - h_{strong}\right) + \dot{Q}_{gen}\right) / \left(M_{gen} c_{p,strong}\right) \\ \left(\left(\dot{m}_{v,gen} - \dot{m}_{weak}\right) X_{strong} + \dot{m}_{weak} X_{weak}\right) / M_{gen} \\ \dot{Q}_{con} / \left(h_{v,gen} - h_{l,c}\right) - \dot{m}_{l,c} \\ \dot{m}_{strong}\left(h_{strong,abs} - h_{weak}\right) + \dot{m}_{v,abs}\left(h_{v,abs} - h_{weak}\right) - \dot{Q}_{abs} / \left(M_{abs} c_{p,weak}\right) \\ \left(\dot{m}_{strong} X_{strong} - \left(\dot{m}_{strong} + \dot{m}_{weak}\right) X_{weak}\right) / M_{abs} \end{bmatrix} \tag{36}$$

TABLE 3. Comparison results of key parameters under on-design condition.

Symbol	Description	Simulation results	Design value	Relative error (%)
X_{strong}	Strong solution concentration (%)	60.77	61	0.38
X_{weak}	Weak solution concentration (%)	54.9	55	0.18
$T_{cw,o}$	Outlet chilled water temperature(°C)	9.077	9	0.86
COP	Coefficient of performance (-)	0.807	0.79	2.15

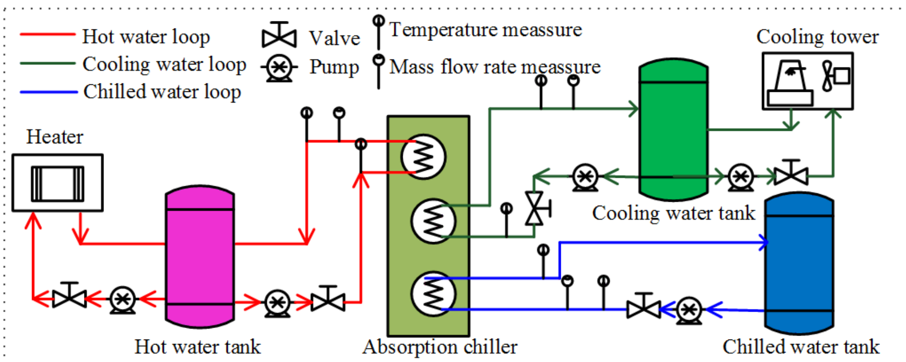

FIGURE 2. Schematic diagram of the experimental system.

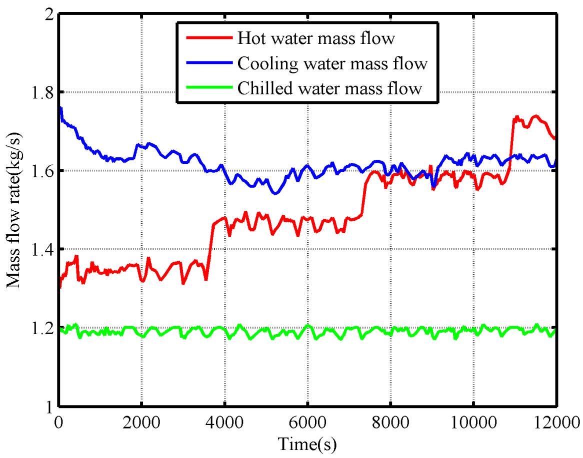

FIGURE 3. Mass flow rates of the external water circulations.

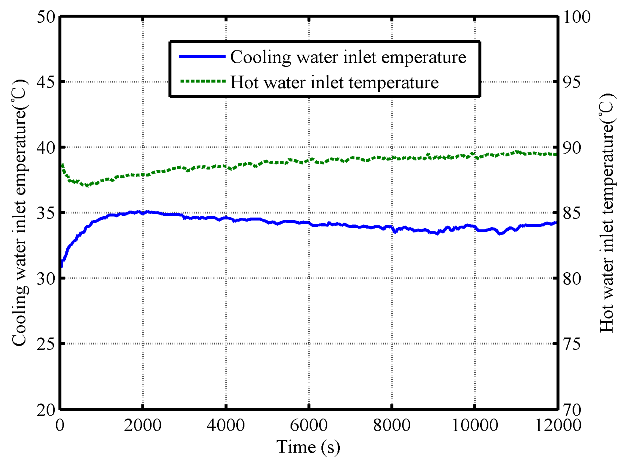

FIGURE 4. Inlet temperatures of the hot water and cooling water.

heat. The cooling water is cooled by an air-cooled cooling tower. All the external water are firstly stored in three tanks and then supplied to the chiller in order to reduce sudden changes in temperature. Specially, the chilled water tank is directly used to simulate the load of the chiller, so that the temperature of the chilled water drops along with the operation of the system. Due to the enclosed structure, it is impossible to measure the parameters inside the chiller directly. Data collected in this paper are the mass flow rates, inlet temperatures and outlet temperatures of the external water circulation. The data was recorded once per thirty seconds and the experiment lasted for about two hours.

The collected mass flow rates and inlet temperatures are adopted as inputs of the model simulation, shown in Fig 3 and Fig 4. The collected outlet temperature of the chilled water is compared to the model simulation results under the same inputs, shown in Fig 5. As shown in Fig 3 and Fig 4, during the experimental data was collected, the mass flow rate of the chilled water has been basically kept constant, and the mass flow rates of the hot water and cooling water has been adjusted several times. The inlet temperatures of the hot water and cooling water are respectively kept around 87 °C and 32 °C, with a variation range of about 4 °C. The inlet temperature of the chilled water drops along with time because of the cold accumulation in the cold water tank. From Fig 5, it can be observed that the simulation results are in good agreement with the experimental data except for the beginning. Cause for the big difference at the beginning is that the values of the state variables cannot be measured in the experiment and therefore are set at random within a reasonable range in the simulation process.

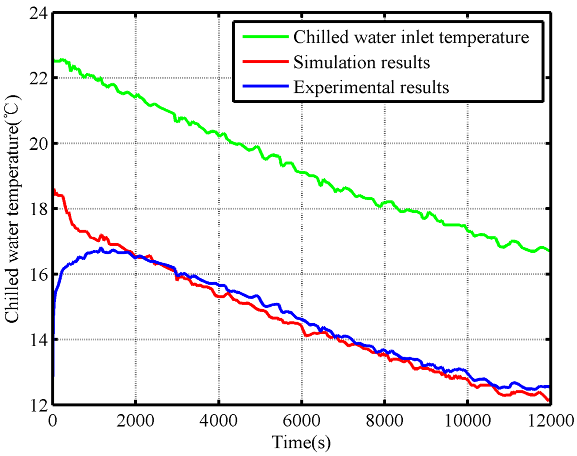

FIGURE 5. **Comparison of chilled water outlet temperatures.**

VI. CONCLUSION

In this study, a simplified six-order state-space model has been developed, which is able to simulate the transient behavior of a typical single-effect absorption refrigeration chiller driven by hot water. Sub-models of each components composing the system are firstly established and then combined to achieve an initial overall system model of the chiller. Considering the complexity of the preliminary model, further simplification is done through reducing variables and order of the model. Finally, two approaches are carried out to validate the accuracy of the model, one is to compare the steady-state simulation results of the proposed model with the design parameters under the on-design condition, and the other one is to compare the model simulation results with the experimental data under dynamic inputs. Results of both steady and dynamic comparison prove good correctness of the proposed model. The establishment of the model would contribute to the design of control strategy and improvement of the system efficiency.

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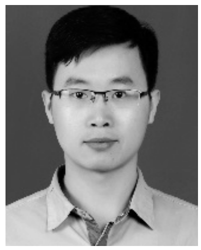

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