Top-down Plantwide Control of a Parallel Double Effect Aqueous Lithium Bromide Absorption Chiller

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Abstract—An alternative control structure for a parallel double effect aqueous lithium bromide absorption chiller has been designed using top-down plantwide control structure analysis. Concept of self-optimizing control and location of throughput manipulator (TPM) have been utilized to improve performance of the control system economically. As the results, while a combination of liquid flowrate out of each generator is used as a self-optimizing measurement to control a spiting ratio, coefficient of performance (COP) can be increased up to 0.5% while heat source is supplied less than nominal value. In addition, by changing the position of TPM from chilled water flowrate to heat source flowrate, crystallization condition can be prevented more effectively, resulted in less back-off required and higher plant capacity by approximately 2.39%.

Index Terms—Absorption chiller, Plantwide control, Control structure design

I. INTRODUCTION

An absorption chiller is one of the oldest and widespread cooling technology, currently used in many industries as well as large public buildings like sport arenas and airports. Its prominent point is the feasibility of energy sources as it is able to use low quality heat from other processes and solar power as its input. In case of medium quality energy, for instance $100 - 150 \ C$ steam, a double effect absorption chiller can be utilized to achieve higher coefficient of performance (COP). Typically, parallel arrangement double effect chiller has higher COP than its series arrangement counterpart but harder to control [1].

Despite the ubiquity of the system, few effort has been spent into improving its control structure. Presently, due to market demand, the competition to improve production efficiency is much more intense. The control structure role can no longer be treated as only ensuring plant stability and smoothing transients but have to take into account of the economic profitability as well. Thus, economic focus plantwide control design has been received much more attention nowadays [2].

Self-optimizing control is one of the new concept introduced in recent years. Basically, in the face of disturbances, model uncertainties and changing of plant operating conditions, choosing different measurements as control variables may lead to different outcomes even though all degrees of freedom are consumed due to the difference in sensitivity of each measurement [3]. By selecting the suitable measurements, plant can be controlled at near – optimal states economically even while set points are held constant, so complicate systems such as real-time optimization (RTO) are not necessary.

In order to find suitable control variables, many methods have been introduced. In this work, 2 methods were used. Firstly, suitable candidates were selected by Minimum Singular Value (MSV) rule then, Null-space method was used to combine them to the control variable [4]. MSV is one of the simplest ways to find a measurement that yield minimal loss in the worst case scenario. However, it assumes that each error is independent of each other which rarely satisfied for complex plants. So, this rule is used only to identify the potential candidates. On the other hand, Null-space method is method for finding the optimal measurement combination considering those interaction between each variable. Under assumptions that there is no measurement error and number of the measurement candidates is higher than number of inputs and disturbances combined, the combination of measurements provided by this method can theoretically give zero loss in linear region.

A location of throughput manipulator is another noteworthy concept. Mathematically, for a process to operate at the maximum capacity, its working condition has to be at active constraints. However, due to disturbances and model uncertainties, in practice, a process has to be operated at lower capacity to create a

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back-off as a safety factor. This back-off can be reduced by placing TPM at the bottom neck to limited effect of the uncertainties [5]. The location of TPM also affects a design of entire control structures due to a radiation rule [6]. In this work, both concepts have been utilized to create an alternative control structure for a parallel double effect aqueous lithium bromide absorption chiller. The obtained structure has been evaluated using a dynamic model simulated in MATLAB – Simulink.

II. PROCESS DESCRIPTION

A parallel double effect absorption chiller system is illustrated in Fig. 1. The process consists of an absorber, an evaporator and a couple of generators, condensers and heat exchangers. It is operated at 3 pressure level depended on temperature of desired chilled water, room temperature and temperature of heat source. The cvcle starts when the weak aqueous lithium bromide is pumped from an absorber through a solution heat exchanger then spite to each generator. In the generators, heat is supplied and water is boiled off. After the partial evaporation, steam is sent to condensers and the concentrated solution is flowed back, through solution heat exchangers and solution expansion valves, to the absorber. In condensers, steam is liquefied. Then the liquid water is throttled via a refrigerant expansion valve and sent to an evaporator at under atmosphere pressure. In the evaporator, the water acts as refrigerant to cooling produced chilled water as it consumes heat to complete the evaporation process. In the end of the cycle the vapor flows back to the absorber and absorbed by concentrated solution. There is also a heat exchange between a high pressure condenser and a low pressure generator, which provides heat for the low pressure cycle.

The mathematical model of the chiller is simulated following mass and energy balances of each consisting units. In order to simplify the process, the solution that exits the absorber and generator, the water that exits condensers and vapor that exists the evaporator are assumed saturated [7]. Thermophysical properties of aqueous lithium bromide solution are calculated using data from 2009 ASHRAE handbook [8] and equations for each unit are listed as follows

Evaporator

$$\frac{dM_{l,e}}{dt} = m_9 - m_{evap} \tag{1}$$

$$\frac{dM_{v,e}}{dt} = m_{evap} - m_{10} \tag{2}$$

$$\frac{dU_{e}}{dt} = m_{9}h_{9} - m_{10}h_{10} - Q_{e}$$
(3)

Absorber

$$\frac{dM_{l,a}}{dt} = m_{abs} + m_6 - m_1 \tag{4}$$

$$\frac{dM_{v,a}}{dt} = m_{10} - m_{abs} \tag{5}$$

$$\frac{dM_{1,a}X_1}{dt} = m_6 X_6 - m_1 X_1 \tag{6}$$

$$\frac{dU_a}{dt} = m_6 h_6 + m_{10} h_{10} - m_1 h_1 - Q_a \tag{7}$$

Low pressure generator

$$\frac{dM_{l,d1}}{dt} = m_3 - m_{4\#} m_{es1}$$
(8)

$$\frac{dM_{v,d1}}{dt} = m_{des1} - m_7 \tag{9}$$

$$\frac{dM_{I,d1}X_4}{dt} = m_3X_3 - m_4X_4 \tag{10}$$

$$\frac{dU_{d1}}{dt} = m_3 h_3 - m_4 h_4 - m_7 h_7 + Q_{d1}$$
(11)

High pressure generator

d

$$\frac{M_{1,d2}}{dt} = m_{13} - m_{14\frac{m}{44}} m_{es2}$$
(12)

$$\frac{dM_{v,d2}}{dt} = m_{des2} - m_{17}$$
(13)

$$\frac{dM_{_{1,d_2}}X_{_{14}}}{dt} = m_{_{13}}X_{_{13}} - m_{_{14}}X_{_{14}}$$
(14)

$$\frac{dU_{d2}}{dt} = m_{13}h_{13} - m_{14}h_{14} - m_{17}h_{17} + Q_{c1}$$
(15)



Figure 1. The parallel double effect absorption chiller system.

Low pressure condenser

$$\frac{dM_{I,c1}}{dt} = m_{con1} - m_8$$
(16)

$$\frac{dM_{v,c1}}{dt} = m_7 - m_{con1} \tag{17}$$

$$\frac{dU_{c1}}{dt} = m_{\gamma}h_{\gamma} - m_{8}h_{8} - Q_{c1}$$
(18)

High pressure condenser

$$\frac{dM_{_{l,c2}}}{dt} = m_{_{con2}} - m_{_{18}}$$
(19)

$$\frac{dM_{v,c2}}{dt} = m_{17} - m_{con2}$$
(20)

$$\frac{dU_{c2}}{dt} = m_{17}h_{17} - m_{81}h_{18} - Q_{c2}$$
(21)

where $M_{i,i}$, $M_{v,i}$, U_i and Q_i are mass of liquid, mass of vapor, internal energy and heat transfer rate in equipment *i* whereas *m*, *X* and *h* is mass flow, mass fraction and specific enthalpy respectively.

$$Q = UA\Delta T_{LMTD}$$
(22)

$$PV = nRT \tag{23}$$

Where U, A, T_{LMTD} , P, V, n, R and T are specific heat transfer coefficient, heat transfer area, logarithmic mean temperature difference, pressure, gas volume, mole, gas constant and gas temperature respectively. In this work, the specific heat transfer coefficients are assumed to be constant. Prominent design specifications are listed as follows

TABLE I. DESIGN SPECIFICATION

Variable	Value	Unit
mass flowrate of LiBr solution (m_1)	1	kg/s
mass flowrate of chilled water (m_{24})	3.94	kg/s
Temperature of chilled water (T_{24})	9	C
Room Temperature	31	C
Temperature of heat source (T_{21})	145	°C

III. SELF-OPTIMIZING CONTROL

In order to optimize the process, a spiting ratio between solution entering the low pressure generator and high pressure generator (m_{20}/m_{13}) is used as a manipulated variable. By a steady state analysis, the maximum COP at 1.362 is achieved when the ratio is equal to 0.927.

For the MSV method, gain matrix is defined as Eq. 24 and loss in the worst case is given as Eq. 25. Since $J_{uu}^{-1/2}$ is not depended on choices of a control variable, suitable candidates are the ones with the highest $S_{c}G$.

$$G' = S_c G J_{uu}^{-1/2} \tag{24}$$

$$L_{\text{werr}} = \frac{1}{2\sigma(G')^2}$$
(25)

where $S_{c} = diag \left\{ \frac{1}{\left| n_{i}^{c} \right| + \left| \Delta c_{i,opt} \left(d \right) \right|} \right\}$, $\Delta c_{i,opt}$ is deviation

from the optimal value of a control variable c_i due to disturbances d at the worst case scenario, n_i is a

measurement error and G is a gain from a manipulated variable to a candidate.

For this case, n_i was assumed to be zero. Worst case disturbances were mass flow rate of heat source at $\pm 10\%$ and room temperature at ± 1 °C. The measurement candidates were selected from mass flowrate and temperature throughout the process ($m_{4?}, m_{7?}, m_{14}, m_{17?},$ $T_{3?}, T_4, T_5, T_7, T_{13}, T_{14}, T_{15}$ and T_{17}). From the calculation shown in Table II, the most suitable control variables were m_4 and m_{14} which values at operating condition were equal to 0.4126 and 0.4347 kg/s respectively.

For Null-space method, the combined control variable (*H*) was selected so that

1

$$HF = 0 \tag{26}$$

where
$$F = \frac{\partial y^{opt}}{\partial d} = \left[\frac{\partial m_4^{opt}}{\partial d}, \frac{\partial m_{14}^{opt}}{\partial d}\right]^T$$

Since there were only 2 suitable candidates, only one disturbance could be nullified. In this case, the mass flow rate of heat source was chosen to be the disturbance then *F* and *H* could be calculated as $[0.151, -0.209]^{T}$ and [1, 0.7225] respectively. Thus, the selected control variable for the spiting ratio was $m_{4?}$ + 0.7225 m_{14} with a set point at 0.7267 kg/s.

TABLE II. MINIMUM SINGULAR VALUE CALCULATION

Variable	G	$\Delta c(Flow)$	$\Delta c(T)$	$S_{c}G$
<i>m</i> _{4?}	60.87	0.1242	0.2035	255.3
$m_{_{7?}}$	0.8693	0.1242	0.2035	3.646
<i>m</i> ₁₄	60.79	0.0074	0.2769	219.4
<i>m</i> _{17?}	0.7901	0.0070	0.2769	2.852
$T_{_{3?}}$	19.91	0.5949	0.2255	31.30
T_4	41.86	1.0119	0.4639	37.61
<i>T</i> _{5?}	15.07	0.3643	0.1670	37.61
T_7	50.51	1.6605	1.1022	25.34
<i>T</i> _{13?}	42.90	0.2932	0.9641	42.57
$T_{_{14}}$	38.21	0.9521	0.5214	35.20
<i>T</i> ₁₅	12.74	0.3807	0.1443	31.30
<i>T</i> ₁₇	48.45	1.4201	0.9015	28.80

IV. LOCATION OF TPM SELECTION

One of the emphatic limitations of an aqueous lithium bromide absorption chiller is the crystallization limit of the solution which is a strong function of mass fraction and temperature, and a weak function of pressure. The critical LiBr mass fraction can be approximated by Eq. 27 [9].

$$X_c = 8.09 \times 10^{-4} T + 0.61341$$
 (27)

In this process, the closest working condition to the limitation took place at a stream leaving the low pressure heat exchanger with LiBr mass fraction (X_5) equaled 0.618, whereas the limit was around 0.652. This difference between the values was a designed back-off of the base case condition. In order to compare locations of TPM and its influence to the back-off, 2 control structures were simulated, one used chilled water flow rate as TPM (CV1) and the other used mass flow rate of heat source which was located closer to the limitation streams (CV2). The room temperature was used as a primary disturbance. Changing of the control structure due to location of TPM and radiation rule was shown as Table III.

TABLE III. MANIPULATED AND CONTROL VARIABLES PAIRING

Manimulated Maniahla	Control Variable		
Manipulated Variable	CV1	CV2	
<i>m</i> ₂₇	TPM	T ₂₈	
$m_{_{8?}}$	T ₂₈	M _{L.con2}	
<i>m</i> ₃₁	M _{L.con2}	P _{con2}	
<i>m</i> _{17?}	P _{con2}	Pgen2	
<i>m</i> ₂₁	P _{gen2}	TPM	

V. RESULTS AND DISCUSSION

Fig. 2 showed a comparison of COP step responses between a structure with the self-optimized control variable and structure with the spiting ratio held constant at 0.927. While decreasing mass flow rate of heat source by 10%, at first COP went up for both structures due to instantaneous change of the denominator. After that, while COP of the fixed ratio structure responded slightly faster, its final state COP was 1.3491 whereas the COP of thr structure with a self-optimized control variable ended up at 1.3558 which was around 0.5% higher.

The step response when mass flow rate of heat source was increased by 10% shown in Fig. 3. Due to an equipment constant, heat transfer area between a high pressure condenser and a low pressure generator, was met, the COP was decreased more dramatically than the previous case. This limitation also decreased the effectiveness of the self-optimized control structure as it performed better than its counterpart by only 0.06%. In Fig. 4, when the room temperature was increased by 1 C, both structures provided more or less identical responses; they finished up at almost the same COP with only 0.02% difference.



Figure 2. Coefficient of performance respond while mass flow rate of heat source is decreased.



Figure 3. Coefficient of performance respond while mass flow rate of heat source is increased.



Figure 4. Coefficient of performance respond while room temperature is increased.

Fig. 5 showed a comparison of step responses of mass fraction in stream entering an absorber between a structure using chilled water flow rate as TPM (CV1) and a structure using mass flow rate of heat source as TPM (CV2) with room temperature increased by 1 °C. In the former case, the disturbance increased the mass ratio from 0.6178 to 0.6203 while it rarely effected in the latter. It provided that by changing position of TPM, effect of the room temperature could be neglected, thus throughput could be set higher. As mass ratio could be set at 0.6203 instead of 0.6178, for instance, the nominal production rate calculated from steady state would be higher by 2.39%. Moreover, a side advantage of CV2 structure was the temperature of chilled water could be better controlled as shown in Fig. 6.



Figure 5. LiBr mass fraction of a stream leaving a low pressure heat exchanger while room temperature is increased.



Figure 6. Chilled water temperature while room temperature is decreased.

VI. CONCLUSION

The alternative control structure of a parallel double effect aqueous lithium bromide absorption chiller was successfully designed and simulated. By using the selfoptimizing control variable, improving in system performance was apparent when heat input was supplied less than the design specification which could happen regularly considering that the heat source would be from other process or solar energy. Furthermore, when the position of TPM was changed to the flowrate of heat source, the effect of room temperature to the crystallization condition was negligible. So, the working condition could be set at higher capacity, though the back-off could not be removed completely due to other disturbances such as the flowrate of the heat source itself.

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