

Simulation Model for Dynamic Operation of Double-Effect Absorption Chillers

Mojahid Sid Ahmed Mohammed Salih Ahmed^{1,a}, Syed Ihtsham Ul-Haq Gilani¹

¹Department of Mechanical Engineering, Universiti Teknologi PETRONAS
Bandar Seri Iskandar, 31750, Tronoh, Perak, MALAYSIA

Abstract. The development in the field of refrigeration and air conditioning systems driven by absorption cycles acquired a considerable importance recently. For commercial absorption chillers, an essential challenge for creating chiller model certainly is the shortage of components technical specifications. These kinds of specifications are usually proprietary for chillers producers. In this paper, a double-effect parallel-flow-type steam absorption chiller model based on thermodynamic and energy equations is presented. The chiller studied is Lithium bromide-water with capacity of 1250 RT (Refrigeration Tons). The governing equations of the dynamic operation of the chiller are developed. From available design information, the values of the overall heat transfer coefficients multiplied by the surface area are computed. The dynamic operation of the absorption chiller is simulated to study the performance of the system. The model is able to provide essential details of the temperature, concentration, and flow rate at each state point in the chiller.

1 Introduction

Considering the energy depletion and environmental problems associated with burning fossil fuels, absorption cooling cycles are becoming attractive in the last decades. Compared to vapor compression cycles, absorption cycles are utilized in expensive thermal energy. Furthermore, absorption cycles use environmental friendly working fluids that do not cause global warming and do not contribute to ozone depletion [1]. Many researchers studied the dynamic behavior of absorption chillers. Most of the previous works dealt with the absorption chillers were for small capacity or experimental single-effect [2-4], and double-effect chillers [5-7]. Nearly all of the works done in double-effect LiBr-H₂O absorption chillers simulation depend totally in detailed chiller specifications and geometry. Shin, et al.[5] developed a dynamic simulation model to simulate the dynamic operation of a double-effect absorption chiller. The dynamic performance of the model was compared with the data of a medium size chiller. The model results showed a good agreement with the experimental data except for the first 83 minutes. Kohlenbach and Ziegler developed a single effect absorption chiller dynamic model, a sensitivity analysis using variation on the model variables is carried out. In comparison to their delicate modeling efforts, validation using experimental data were not sufficient [2-3]. Bian [4] developed a transient simulation model for single effect direct-fire LiBr-H₂O absorption chiller. By using the above model, reasonable transient trends of the start-up process

^a Corresponding author : mojahid76@yahoo.com

have been achieved. A dynamic simulation program for predicting the transient behavior of a double-effect absorption chiller is developed by Matsushima et al. [8]. The simulated results agree with the measured data. Seo.J et al. [6] stated that the solution level control in HTG requires a good flow rate prediction model. They found that the solution flow rate can be approximated as a function of HTG solution temperature and cooling water inlet temperature.

2 Description of the chiller cycle

Figure 1 shows the schematic of a double-effect steam absorption chiller. The circulation of the solution inside the system is guaranteed via the solution pump. The weak solution leaving the absorber is pumped through the heat exchangers to the high and low temperature generators, HTG & LTG respectively. In HTG heat is added to the solution from steam and water vapor is given off by the solution. The vapor is in superheated state and used to heat the solution in the LTHX. The vapor generated by LTG is condensed in the condenser and that from HTG after condensing in LTG is throttled to the condenser pressure [9]. The concentrated solutions from the HTG & LTG are reunited at the solution heat exchanger, transferring heat to the weak solution pumped from the absorber, the united strong solution stream enters the absorber where it is sprayed onto the absorber tubes, thus facilitating the absorption of the refrigerant vapor from the evaporator [10].

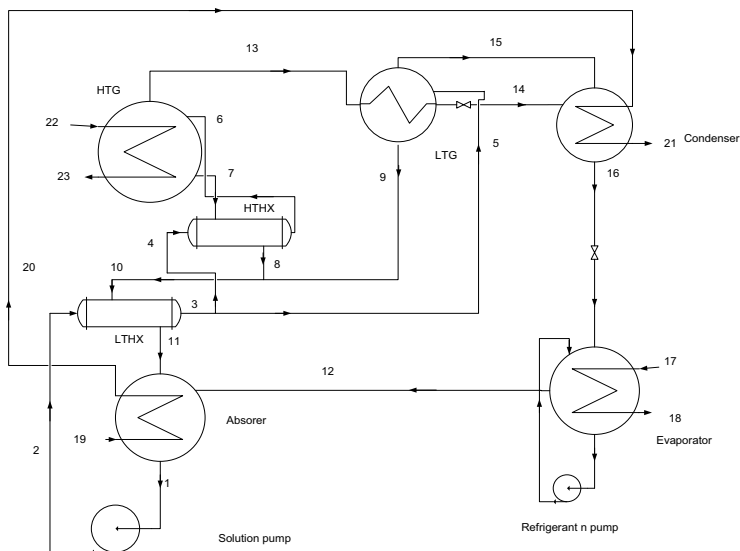


Figure 1. Schematic of a double-effect steam absorption chiller.

3 Methodology

The dynamic equations that represent the chiller components are developed. Using energy balance, mass balance, and heat transfer equations. The derivatives of temperatures, mass stored, and concentrations are driven. For initialization all temperatures are set to ambient temperature and the concentration is set to 55% weight. The developed equations are first order differential equations. The derivatives of temperature, concentration, and mass stored are calculated based on the time interval. Then the values of temperature, concentration, and masses in generators and absorber are updated. For heat transfer calculation, all chiller components are considered as shell and tube heat exchangers. Empirical equations are used to calculate the overall heat transfer coefficient multiplied by heat transfer surface area (UA). The simulation model is built in MATLAB environment.

Modeling assumption:

The assumptions below were used to properly represent dynamic model of the chiller:

1. The solution leaving the absorber is saturated.
2. The enthalpy change across the pump is small and can be neglected.
3. Perfectly insulated components.
4. The temperatures of the vapor generated in generators equal the average generator temperature.
5. The water entering the evaporator is saturated.
6. Isenthalpic expansions.
7. Valves are treated ideally such that there's no time delay.

The Equations of the Double Effect steam Absorption Chiller:

Applying energy balance, mass balance, heat transfer equations and equation of state for the LiBr-H₂O solution for each component of double effect steam absorption chiller the following set of equations are obtained.

Mass balance:

Generally, the total mass balance equation for refrigerant, chilled and cooling water, and lithium bromide solution can be stated as:

$$\sum \dot{m}_{in} = \sum \dot{m}_{out} \quad (1)$$

where, \dot{m} = mass flow rate. Subscripts in & out =the stream entering and leaving the components. The mass balance of LiBr-H₂O streams

$$\sum \dot{m}_{in} * X_{in} = \sum \dot{m}_{out} * X_{out} \quad (2)$$

Where, X= the concentration of LiBr solution.

Energy balance:

The energy balance for each chiller components is expressed as:

$$\dot{Q} + \sum \dot{m}_j * h_j = 0 \quad (3)$$

Where: h = enthalpy of each stream, \dot{Q} = the quantity of heat transfer to or from the system.

For absorption process in the absorber

$$\dot{Q} = \dot{m}_r h_r + \dot{m}_c h_c - \dot{m}_d h_d \quad (4)$$

Where : r = for refrigerant, c =concentrate solution, d =dilute solution

Heat transfer equations:

The logarithmic mean temperature difference (LMTD) method is used to evaluate the (UA) values.

$$UA = \frac{\dot{Q}}{\Delta T_{LMTD}} \quad (5)$$

The thermal properties of LiBr solutions and steam used in the calculations are obtained from refs [11-12]. The value of UA at off design conditions can be estimated according to empirical equation[13]. From the above equations the drivetives of temperatures, concentrations and mass stored are driven as indicated in Eq 6 to Eq18.

$$\frac{dT_{g1}}{dt} = (\dot{Q}_{g1_out} - \dot{Q}_{g1_in}) / M_{g1} \cdot Cp_{g1} \quad (6) \quad \frac{dT_{g2}}{dt} = (\dot{Q}_{g2_out} - \dot{Q}_{g2_in}) / Cp_{g2} \cdot M_{g2} \quad (7)$$

$$\frac{dT_c}{dt} = (\dot{Q}_{c_in} - \dot{Q}_{c_out}) / Cp_c \cdot M_c \quad (8) \quad \frac{dT_e}{dt} = (\dot{Q}_{e_in} - \dot{Q}_{e_out}) / Cp_e \cdot M_e \quad (9)$$

$$\frac{dT_{abs}}{dt} = (\dot{Q}_{abs_in} - \dot{Q}_{abs_out}) / Cp_{abs} \cdot M_{abs} \quad (10) \quad \frac{dT_{hx1}}{dt} = (\dot{Q}_{hex1_s} - \dot{Q}_{hex1_w}) / (M_{hx1} Cp_{hx1}) \quad (11)$$

$$\frac{dT_{\text{hx2}}}{dt} = (\dot{Q}_{\text{hex2}_s} - \dot{Q}_{\text{hex2}_w}) / (M_{\text{hx2}} C_{p_{\text{hx2}}}) \quad (12) \quad \frac{dX_7}{dt} = (\dot{m}_6 X_6 - \dot{m}_7 X_7) / M_{g1} \quad (13)$$

$$\frac{dX_9}{dt} = (\dot{m}_5 X_5 - \dot{m}_9 X_9) / M_{g2} \quad (14) \quad \frac{dX_1}{dt} = (\dot{m}_5 X_5 - \dot{m}_9 X_9) / M_a \quad (15)$$

$$\frac{dM_{g1}}{dt} = \dot{m}_6 - \dot{m}_{13} - \dot{m}_7 \quad (16) \quad \frac{dM_{g2}}{dt} = \dot{m}_5 - \dot{m}_{15} - \dot{m}_9 \quad (17)$$

$$\frac{dM_a}{dt} = \dot{m}_{11} + \dot{m}_{12} - \dot{m}_1 \quad (18)$$

4 Results and discussion

The simulation result for the cooling production is illustrated in Fig. 2. The chiller starts to produce some cooling effect 25 minutes after the start up due to consume of the heat input by the thermal mass of the chiller. The results show that the chiller attains the steady state operation in a round 55 minutes. After 900 minutes the chiller is shut-down and the cooling capacity dropped sharply when the steam flow to the chiller is stopped. The cooling production stopped in about 15 minutes.

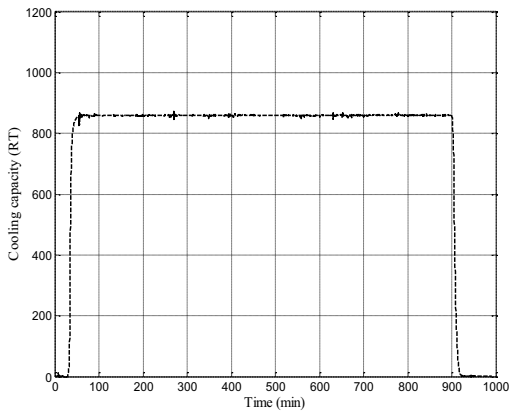


Figure 2. chiller cooling capacity

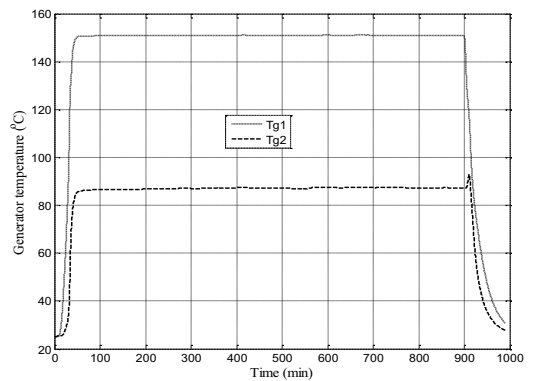


Figure 3. high and low temperature generators temperature

Fig. 3 shows the generators temperature which were initialized as ambient temperature 25°C. Once the steam is added to the HTG, the temperature starts to increase sharply due to the sensible heat transfer. once the the solution starts to boil, the generators temperature trends flattened because of the heat consumed by vaporization of the refrigerants vapor. For solution concentration, the simulation started with an initial concentration for all components of 55%. At the first 25 minutes the heat is absorbed by the system thermal mass, the HTG started to generate refrigerant vapor after reaching the boiling temperature of solution. Thus, the concentration of strong solution leaving HTG (X_7) started to increase sharply as shown in Fig. 4. For the concentration of the solution leaving the LTG (X_9), the concentration change started 7 minutes later than the HTG due to the fact that the refrigerant vapor from HTG is used as heat input for the LTG. Fig 5 shows the inlet and outlet cooling water temperatures at absorber and condenser. During the start up phase the temperatures of the cooling water leaving absorber and condenser initially decreased since the cooling water flow in the system without heat load in these components. After that the temperature rises as refrigerant vapor and hot solution begin to come in condenser and absorber respectively.

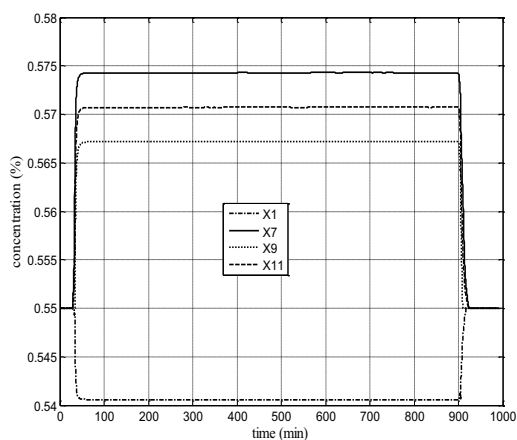


Figure 4. Solution concentration in the generators and the absorber.

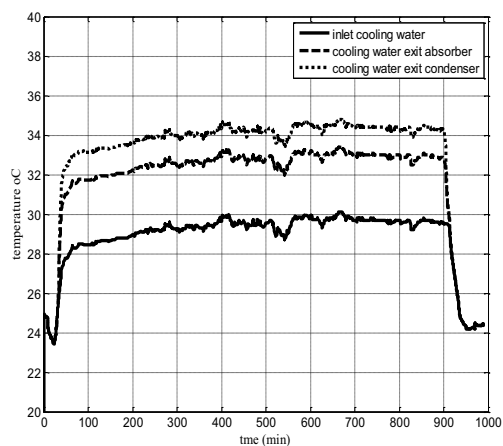


Figure 5 Inlet & outlet cooling water temperatures to the chiller

5 Conclusions

The governing equations of the dynamic behavior of the double-effect chiller are developed. From the available design data the values of the UA are calculated. The derivatives of temperatures, mass stored, and concentrations are driven. For initialization all temperatures are set to ambient temperature and the concentration is set to 55% weight. The model represent accurately the inside operation of the chiller beside the overall performance and provides important information for chiller optimization as well as a base for technical support for alternative control of absorption chiller.

Acknowledgment

The authors would like to acknowledge Universiti Teknologi PETRONAS for their support.

References

1. R. Gomri and R. Hakimi, *Energy conversion and management*, **49**, 3343-3348, (2008).
2. P. Kohlenbach and F. Ziegler, *International Journal of Refrigeration*, **31**, 217-225, (2008).
3. P. Kohlenbach and F. Ziegler, *International Journal of Refrigeration*, **31**, 226-233, (2008).
4. J. Bian, Master, Maryland, (2005).
5. Y. Shin, J. Seo, H.W Cho, S.C.Nam, J.H.Jeong, *Applied Thermal Engineering*, **29**, 2718-2725, (2009).
6. J. A. Seo, Y. Shin, J. Chung, *International Journal of Refrigeration*, **35**, 1123-1129, (2012).
7. L. Xiang, in *Power and Energy Engineering Conference*, 1-4., (2009).
8. F. Matsushima, T. Komatsu, A. Nishiguchi, *International Journal of Refrigeration*, **33**, 259-268, (2010).
9. J. M. Gordon and K. C. Ng, *Cool thermodynamics*: Cambridge International Science Publishing Cambridge, (2000).
10. M. Ahmed and S. I. U.-H. Gilani, in *American Institute of Physics Conference Series*, , 870-879, (2012).
11. K. E. Herold, Radermacher, R., Klein, S.A., *Absorption chillers and heat pumps*: CRC, (1996).
12. H. Chua, H.K. Toh, A. Malek, K.C. Ng, K. Srinivasan, *International Journal of Refrigeration*, **23**, 412-429, (2000).
13. M. Divid, *NASA technical paper*(1978).