OPTIMIZATION OF A DOUBLE EFFECT LiBr-H₂O ABSORPTION SYSTEM

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ABSTRACT

A computer program is developed to optimize a double-effect LiBr-H₂O absorption refrigeration system. Optimum coefficient of performance (COP) of the refrigeration system is sought for a fixed cooling load capacity of a system and fixed upper cycle pressure (pressure of Generator 1) and lower cycle pressure (pressure of Evaporator/Absorber). For any fixed upper cycle pressure and lower cycle pressure, the intermediate pressure (pressure of Condenser/ Generator 2) of the system is varied until the optimum COP is obtained.

The effect of increasing of the temperature of Generator 1 on the COP of the system is also analysed. Pure water vapor is considered to be generated in both generators at the respective prevailing pressure and temperature combinations. Specific enthalpies of pure water vapor at outlet from the respective generators and the condensate water leaving the Condenser are automatically determined by the developed programme at the prevailing saturation temperatures. Concentrations at all state points in the cycle are also obtained from the programme at the respective state point pressure and temperature values. In the analysis, steady-state conditions and no pressure losses are assumed. The effect of the effectiveness of the heat exchangers on the COP of the system are also analysed using the developed program.

INTRODUCTION

One possible way of greatly increasing the COP of an absorption refrigeration system is incorporating a second-effect or second-stage generator. Due to its higher COP, the double-effect generation system has an improved potential for applications in absorption cooling systems than the single-effect system.

In the double-effect LiBr-H₂O absorption refrigeration cycle, weak solution leaving the Absorber is pumped to the pressure of Generator 1, Fig. 1. External heat supplied to the weak solution in Generator 1 releases water vapor from the solution. To reduce the quantity of external heat required by Generator 1 and thus improve the performance of the system, two heat exchangers (Heat Exchanger 1 and Heat Exchanger 2) are introduced as indicated. Heat transfer in Heat Exchanger 1 to the weak solution pumped to Generator 1 cools the strong solution entering the Absorber. The lower the temperature of the solution, the better the absorption rate and the lower will be the heat of absorption developed in the Absorber.

More refrigerant (water vapor) is vaporized in Generator 2 by the heat of condensation of the refrigerant vapor generated in Generator 1. The condensate leaving the Condenser is throttled to the Evaporator pressure. After taking the cooling load in the Evaporator, the refrigerant vapor enters the Absorber to complete the cycle.

In the thermal modeling of the double-effect generation cycle the following assumptions are considered. The intermediate pressure of the system is in equilibrium pressure corresponding to the temperature of the Condenser while the low pressure and high pressure of the system correspond to the equilibrium saturation temperatures of the Evaporator and Generator 1, respectively. These assumptions along with the mass, material and energy conservations and the thermodynamic state equations lead to the complete analysis and optimization of the absorption cooling cycle.

The thermodynamic analysis of the cycle has been carried out using state equations for LiBr-H₂O system available in the literature and generated expressions. At each operating point, the LiBr-H₂O system is assumed to be in steady state at the prevailing conditions in the respective components of the cycle.

The analysis of the cycle is started at the Condenser where temperature $T_C$ is used to determine the Condenser saturation pressure $p_C$ and thus the pressure of Generator 2, $p_{G2}$. The temperature at the Evaporator gives the low side pressure in the Evaporator and the Absorber. From the Absorber temperature and the low pressure of the system, the equilibrium weak solution concentration is evaluated. The temperature in Generator 1 and the high-pressure of the system furnish the concentration of the strong solution leaving Generator 1. The intermediate (medium) pressure $p_{G2}$ and temperature $T_{G2}$ of Generator 2 are used to evaluate the concentration of the strong solution leaving Generator 2.

The acquired concentrations, pressures and temperatures at different state points, the mass, material and heat balances for each component along with the effectiveness expressions of the heat exchangers provide mass flow rates and enthalpies at each state point.

**Concentrations at Various Points in the Cycle**

Prevailing pressure and temperature values at each state point of the cycle are used to determine the respective concentrations. Concentration of the weak solution between state 1 to state 4 remains constant. Similarly, concentrations of the strong solution between state points 5 to state point 7 and state point 8 to state point 10 are, separately, constant. These are indicated in Eqs. (1) to (3).

Concentration of the weak solution leaving the Absorber is determined at the pressure $p_A$ and temperature $T_A$ of the Absorber:

$$\xi_7 = \xi(p_A, T_A) = \xi_2 = \xi_4 = \xi_w$$

(1)

Concentration of the strong solution leaving Generator 1 at state point 5 is obtained at $p_{G1}$ and temperature $T_{G1}$ of Generator 1:

$$\xi_5 = \xi(p_{G1}, T_{G1}) = \xi_6 = \xi_7 = \xi_{s1}$$

(2)

Similarly, the concentration of the stronger solution leaving Generator 2 at state point 8 is obtained at $p_{G2}$ and temperature $T_{G2}$ of Generator 2:

$$\xi_8 = \xi(p_{G2}, T_{G2}) = \xi_9 = \xi_{s2}$$

(3)

where: $\xi_w$, $\xi_{s1}$ and $\xi_{s2}$ are concentrations of weak solution leaving the Absorber, strong solution leaving Generator 1 and stronger solution leaving Generator 2, respectively.

**Specific Enthalpies at the Various State Points**

Specific enthalpy values at the various state points are determined as follows.

a) **Saturated Liquid (Refrigerant)**

The refrigerant leaving the Condenser at state point 15 is assumed to be saturated liquid at the Condenser temperature.
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The vapor refrigerant leaving Generator 1 (state 11) has to be completely condensed on leaving Generator 2 (state 12), if the double-effect generation absorption system is to yield the maximum COP. Accordingly,

\[ h_{12} = h_{\text{sat}} \bigg|_{T_{G1}} = h_{13} \]  

(5)

b) Saturated Vapor (Refrigerant)

Refrigerants leaving the Evaporator, Generator 1 and Generator 2, at state points 17, 11 and 14, respectively, are assumed to be saturated vapor at the temperature of the corresponding component.

\[ h_{17} = h_{\text{sat}} \bigg|_{T_E}, \quad h_{11} = h_{\text{sat}} \bigg|_{T_{G1}}, \quad \text{and} \quad h_{14} = h_{\text{sat}} \bigg|_{T_{G2}} \]  

(6)

c) LiBr-H₂O Solution

The specific enthalpies of the LiBr-H₂O solution at the various state points are determined at the respective prevailing temperatures and concentrations.

\[ h_{1} = h_{\text{sol}} \bigg|_{T_{A1}, \xi_{w1}} = h_{1} \]  

(7)

where:

\[ h_{\text{sol}} \bigg|_{T_{A1}, \xi_{w1}} = \sum_{i=0}^{4} a_i \xi_{w1}^i + T \sum_{i=0}^{4} b_i \xi_{w1}^i + T^2 \sum_{i=0}^{4} c_i \xi_{w1}^i \]  

(8)

where: \( T \) in °C and \( \xi \) in % LiBr

\( a_i = -2024.33 \) \( b_i = 18.2829 \) \( c_i = -3.7006214E-2 \)
\( A_i = 151.309 \) \( B_i = -1.1694757 \) \( C_i = 2.8877566E-3 \)
\( A_i = -4.88161 \) \( B_i = 3.248041E-2 \) \( C_i = -8.141105E-5 \)
\( A_i = 6.302948E-2 \) \( B_i = -4.034184E4 \) \( C_i = 9.9116628E-7 \)
\( A_i = -2.913701E-4 \) \( B_i = 1.8520696E-6 \) \( C_i = -4.444120E-9 \)

Concentration range: 40 < \( \xi \) < 70 % LiBr
Temperature range: 15 < \( T < 165 \) °C

\[ h_{15} = h_{\text{sol}} \bigg|_{T_{G1}, \xi_{w1}} \]  

(9)

\[ h_{18} = h_{\text{sol}} \bigg|_{T_{G2}, \xi_{w1}} \]  

(10)

\[ h_{20} = h_{8} - e_1 \left( h_{9} - h_{92} \right) = h_{10} \]  

(11)

where: \( e_1 \) = Effectiveness of Heat Exchanger 1

\[ e_1 = \frac{h_{9} - h_{92}}{h_{9} - h_{92}} \]  

(12)

and \( h_{92} \) = LiBr-H₂O solution enthalpy at concentration of state point 9 and temperature of state point 2

\[ h_{92} = h_{\text{sol}} \bigg|_{T_{A2}, \xi_{w2}} \]  

(13)

Energy balance for Heat Exchanger 2 yields:

\[ h_{3} = h_{2} + \left( \frac{m_2}{m_1} \right) [ h_{9} - h_{92} ] \]  

(14)

\[ h_{9} = h_{2} - e_2 \left( h_{9} - h_{92} \right) = h_{93} \]  

(15)

where: \( e_2 \) = Effectiveness of Heat Exchanger 2

\[ e_2 = \frac{h_{9} - h_{92}}{h_{9} - h_{93}} \]  

(16)

and \( h_{93} \) = LiBr-H₂O solution enthalpy at concentration of state point 6 and temperature of state point 3

\[ h_{93} = h_{\text{sol}} \bigg|_{T_{A3}, \xi_{w3}} \]  

(17)

\[ h_{94} = h_{3} + \left( \frac{m_3}{m_2} \right) \left[ h_{5} - h_{6} \right] \]  

(18)

Mass Flow Rates

The mass flow rate of the refrigerant through the Evaporator can be obtained from the given load of the system and the specific enthalpy values at the Evaporator, as:

\[ m = m_{15} = m_{18} = m_{10} = \frac{Q_{E}}{h_{15} - h_{10}} \]  

(19)

Mass flow rate of weak LiBr-H₂O solution at inlet of Generator 1 is obtained from:

\[ m_{1} = m_{2} = m_{3} = m_{4} = \left[ 1 - \left( \frac{\xi_{w1}}{\xi_{w2}} \right) \right] m \]  

(20)

The following relations are obtained from material balance at Generator 1:

\[ m_{11} = m_1 \left( \frac{\xi_2}{\xi_3} \right) = m_0 = m_1 \]  

(21)

\[ m_{12} = m_1 \left( 1 - \frac{\xi_2}{\xi_3} \right) = m_{12} = m_{13} \]  

(22)

Material balance at Generator 2 yields:

\[ m_{13} = m_2 \left( \frac{\xi_4}{\xi_5} \right) = m_0 = m_2 \]  

(23)

Using mass and energy balances at Generator 2, the following relations are obtained:

\[ m_{14} = m_4 \left( \frac{\xi_6}{\xi_7} \right) - m_6 = m_{10} \]  

(24)

Mass balance at the Condenser gives:

\[ m_{15} = m_r = m_4 \left( 1 - \frac{\xi_6}{\xi_7} \right) = m_{15} = m_{17} \]  

(25)

**Pressure-Temperature-Concentration Relations**

The following expressions are applied for the determination of the pressure, temperature and concentration of LiBr-H₂O solution:

\[ t = \sum_{i=0}^{\infty} A_i \xi^i + t' \sum_{i=0}^{\infty} A_i \xi^i; \quad \text{Solution temperature [°C]} \]  

\[ t' = \frac{1 - \xi}{\xi} \sum_{i=0}^{\infty} A_i \xi^i; \quad \text{Refrigerant temperature [°C]} \]  

(26)

\[ \log P = C + D/T' + D/T' ; \quad \text{Pin [kPa] and } T' \text{ in [K]} \]

\[ T' = \frac{-2E}{D + 4E(C - \log P)} \]

where:

\[ A_1 = -2.00755 \quad B_1 = 124.937 \quad C = 7.05 \]

\[ A_2 = 0.16976 \quad B_2 = -7.71649 \quad D = -1596.49 \]

\[ A_3 = -3.133362E-3 \quad B_3 = 0.152286 \quad E = -104095.5 \]

\[ A_4 = 1.97668E-3 \quad B_4 = -7.95090E-4 \]

where:

- Concentration range: \( 45 < \xi < 70 \% \text{ LiBr} \)
- Temperature range: \( 5 < t < 175 \degree C \)
- \( -15 < t' < 110 \degree C \)

**Heat Transfer Duties and COP of the System**

The heat transfer duties of the various components of the double-effect LiBr-H₂O system are determined from application of energy balance.

**a) Absorber:**

The heat of absorption in the Absorber is given by:

\[ Q_A = m_1 h_1 + m_0 m_1 h_5 - m_r h_4 \]  

(27)

**b) Generator 1:**

Energy balance at Generator 1 gives:

\[ Q_{G1} = m_1 h_1 + m_0 h_4 - m_4 h_2 \]  

(28)

**c) Generator 2:**

The quantity of heat released by the vapor leaving Generator 1 and absorbed by solution in Generator 2 is given by:

\[ Q_{G2} = m_{11} (h_{11} - h_{12}) \]  

(29)
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\section*{d) Condenser:}

Energy balance at the Condenser gives:

\[ Q_C = m_{15} h_{15} + m_{14} h_{14} - m_{13} h_{13} \]  

(30)

The coefficient of performance (COP) of the double effect absorption cycle is given by:

\[ \text{COP} = \frac{Q_C}{\dot{Q}_H} \]  

(31)

A double-effect generation LiBr-H₂O absorption refrigeration cycle is represented on a \( P-T \) in Fig. 2. The numbers on this figure correspond to the state points indicated on the double-effect absorption cycle, Fig. 1.

\section*{System Optimization Flow Chart}

A flow-chart illustrating the optimization process of the double-effect LiBr-H₂O absorption refrigeration model is given in Fig. 3.

\section*{Discussion of Results}

The LiBr-H₂O absorption refrigeration cycle model is analyzed based on values chosen from the actual ranges over which the machine is expected to operate. Accordingly, the following data are considered for the analysis.

\[ \begin{align*}
Q_i &= 10 \text{kW}; \quad p_s = p_a = 2.0 \text{kPa}; \\
p_c &= p_{r2} = 5 \text{kPa}; \quad p_{g2} = 15 \text{kPa}; \\
\epsilon_t &= \epsilon_s = 90\%; \quad \xi_{w} = 43\%; \quad \xi_{v1} = 59\%; \\
\xi_{v2} &= 55\%.
\end{align*} \]

\textbf{Figure 2} \( P-T \) Representation of Double-Effect Generation LiBr-H₂O Absorption Refrigeration Cycle

Using the steady state equations for the thermodynamic properties of LiBr-H₂O, the effects
varying parameters on the coefficient of performance (COP) of the system were analysed. The maximum COP of the double effect absorption refrigeration system of the example considered (1.855) is about twice that of the COP of a single effect absorption refrigeration system (0.901). The effect of increasing the temperature of Generator 2 and thus its pressure and the pressure of the condenser, keeping the concentration of the strong solution leaving it constant at \( \xi_{s2} = 55\% \), increases the COP of the system until the heat of condensation of the refrigerant vapor generated in Generator 1 is equal to the energy absorbed by the solution in Generator 2, Fig. 4. The COP of the double-effect absorption refrigeration is optimum, for the specified fixed upper cycle pressure (pressure of Generator 1) and lower cycle pressure (pressure of Evaporator/Absorber), at about \( T_{C1} = 60.5^\circ\text{C} \).

Figure 3 Flow Chart for Computer Simulation of the Double-Effect Generation LiBr-H2O Absorption Refrigeration Cycle
Figure 3 Flow Chart for Computer Simulation of the Double-Effect Generation LiBr-H2O Absorption Refrigeration Cycle (Continued)
The effect of increasing the temperature of Generator 1 on the COP of the system, for fixed evaporator and condenser pressures, is shown in Fig. 5. Increasing the temperature of Generator 1 increases the generator capacity. And as may be expected (Eq. (31)), for a fixed cooling load, increasing the generator capacity reduces the COP of the system. The effect of increasing the temperature of Generator 1 is performed for two cases. In the first case, the temperature of Generator 1 and its corresponding pressure are varied while the concentration of the solution leaving Generator 1 is maintained constant. In the second case, the pressure of Generator 1 is maintained constant and the temperature of Generator 1 and concentration of the solution leaving Generator 1 are varied. The latter case shows a drastic drop in the COP of the system while the drop in COP is insignificant for the first case, Fig. 5.

The effect of varying the cooling load on the COP of the system is observed with increasing values of Evaporator pressure $P_E$, for fixed pressures of Condenser and Generator 1, Fig. 7.

A decrease in the COP of the system is observed with increasing values of Evaporator pressure $P_E$, for fixed pressures of Condenser and Generator 1, Fig. 7.

The effect of solution heat exchangers effectiveness is depicted in Fig. 8. The effectiveness of heat exchangers markedly affect the COP of the system. Keeping all other parameters fixed and varying the effectiveness of Heat Exchanger 1 from the original value of $\varepsilon_1 = 99\%$ to $\varepsilon_1 = 65\%$ reduces the COP of the system from 1.851 to 1.825. The effect of Heat
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Exchanger 2 on the COP of the system is more pronounced as its value reduces from 1.851 to 1.755 for the same range of effectiveness drop, \( \varepsilon_2 = 90\% \) to \( \varepsilon_2 = 65\% \). The reduction in the COP of the system would even be lower if the effectiveness of both heat exchangers is simultaneously reduced from \( \varepsilon_2 = 90\% \) to \( \varepsilon_2 = 65\% \).

\[
P_c = 5 \text{ kPa}
\]

\[
P_e = 2 \text{ kPa and } P_{GL} = 15 \text{ kPa}
\]

\[
P_C = 5 \text{ kPa}
\]

\[
\varepsilon_1, \varepsilon_2 = \text{ Effectiveness of Heat Exchanger 1 and Heat Exchanger 2, respectively}
\]

\[
T_A = \text{ Absorber temperature} \ [\degree C]
\]

\[
T_C = \text{ Condenser temperature} \ [\degree C]
\]

\[
T_E = \text{ Evaporator temperature} \ [\degree C]
\]

\[
T_{GI} = \text{ Generator 1 temperature} \ [\degree C]
\]

\[
T_{G2} = \text{ Generator 2 temperature} \ [\degree C]
\]

\[
\xi = \text{ Concentration of LiBr-H}_2\text{O solution} \ [%]
\]

\[
\xi_{G1} = \text{ Concentration of solution leaving Generator 1} \ [%]
\]

\[
\xi_{G2} = \text{ Concentration of solution leaving Generator 2} \ [%]
\]

\[
\xi_w = \text{ Concentration of weak solution leaving Absorber} \ [%]
\]

\[
\varepsilon_1, \varepsilon_2 = \text{ Effectiveness of Heat Exchanger 1 and Heat Exchanger 2, respectively}
\]

REFERENCES


