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Thermoeconomic Analysis and Multi-Objective Optimization of a LiBr-Water Absorption Refrigeration System

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ABSTRACT: Performance optimization of a single-effect lithium bromide water absorption refrigeration system is the subject of this paper. First, the thermodynamic model of the system wasderived based on the first and second law analysis of an absorption refrigeration cycle with LiBr-water as the working fluid pair. Then, the effects of different design parameters such as the generator inlet hot water temperature, evaporator inlet chilling water temperature and absorber and condenser inlet cooling water temperatures on the performance of the systemwere investigated. In order that, by defining the coefficient of performance (COP), exergy efficiency (Second-law efficiency) and total cost function of the system as the objective functions, the genetic algorithm optimization technique was implemented to evaluate these performance indexes. Finally, the optimal values of design parameters and objective functions were found and compared to the initial values. Results show significant improvement in system COP (about 75 %), exergy efficiency (47 %) and total cost (12 %).

Key words: Absorption, Refrigeration, Thermoeconomic analysis,Exergy analysis, Genetic algorithm, Optimization

INTRODUCTION

There has been a growing interest in utilizing absorption refrigeration systems for cooling applications in recent years. Since these systems are thermally activated, they need lower input power compared to conventional cooling systems. Also, absorption refrigeration systems can be activated by solar and geothermal energies which are mostly free with low exergy level. Furthermore, these systems can provide reliable and quiet cooling, where there is excessive heat and natural gas available instead of expensive or unavailable electricity (Florides *et al.*, 2002). Considering environmental restrictions of using chlorofluorocarbons refrigerants (CFCs) in air conditioning industry due to depletion of the ozone layer, absorption chillers are charging with environmentally friendly solutions like LiBr-water or ammonia-water. Moreover, these two pairs offer good thermodynamic performance, which makes them more attractive for engineering applications (Chua *et al.*, 2000).

The science of Thermodynamics is built primarily based on the first and second laws. First law analysis is a prevalent method to analyze thermal systems. However, the first law of thermodynamics is simply an expression of the conservation of energy principle, and places no restrictions on the direction of processes. On the other hand, second law analysis is key tool in design, optimization and performance assessment of energy systems (Yumrutas*et al.*, 2002); this is carried out by the concept of exergy which indicates the useful work potential of a given amount of energy at a specified state (Kotas, 1995).

During recent years, many studies have been conducted by various researchers in thermodynamic and exergy analysis of absorption refrigeration systems. Bejan performed a theoretical analysis of the systems based on entropy generation minimization (Bejan, 1996). Thermodynamic analysis of LiBr-water absorption system for cooling and heating applications based on first and second law analysis was carried out by Lee and Sherif.(Lee and Sherif, 2001). Talbi et al., also Sencan et al. performed exergy analysis of LiBr-water absorption refrigeration systems (Talbi and Agnew, 2000, Sencan *et al.*, 2005).

Gebreslassie *et al.* selected the best design from a set of design alternatives for the absorption cycles by using a non-linear programming based optimization study for minimization of the absorption chiller (Gebreslassie *et al.*, 2009). Gebreslassie *et al.* presented a multi-objective for optimization of sustainable single-

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effect Water/Lithium bromide absorption cycle (Gebreslassie *et al.*, 2012). Godarzi *et al*, mployed the Exergoeconomic analysis to improve the cost effective performance (Godarzi *et al.*, 2013). Mazzei *etal.* developed a non-linear mathematical model for optimization of a single effect absorption refrigeration system operating with lithium bromide-water solution. (Mazzei *et al.*, 2014). Gogoi and Talukdar employed a parametric analysis for thermodynamic optimization of a combined water–LiBr vapor absorption refrigeration system and reheat regenerative steam turbine based power cycle. (Gogoi and Talukdar, 2014). In this paper, in order to optimize the performance of a LiBr-water absorption refrigeration system, COP, exergy efficiency and total cost function of the system were defined as objective functions and optimized simultaneously using genetic algorithm. For given ranges, system design parameters including generator inlet hot water temperature, absorber and condenser inlet cooling water temperatures and evaporator inlet chilling water temperature were computed at the optimum design conditions. Finally, the optimal values of objective functions were estimated and compared to the initial amounts.

MATERIALS & METHODS

The absorption cycle is a process in which the refrigeration effect is produced through the use of refrigerant and absorbent as a working fluid pair. In the studied system, LiBr acts as an absorbent that circulates and absorbs the water as a refrigerant which is vaporized in the evaporator. A schematic representation of the single-stage absorption cycle is shown in Fig.1.

The cycle consists of an evaporator, a condenser, a generator, an absorber, a heat exchanger, a solution pump and two throttling valves. The cycle performs as follows (Kizilkan *et al.*, 2007):

The strong solution (a mixture strong in refrigerant) which consists of the refrigerant (water) and absorbent (LiBr), is heated up to the high-pressure section of the system, which is a generator. This drives refrigerant vapor off the solution. The hot refrigerant vapor is cooled in the condenser until it condenses. Then, the refrigerant liquid passes through a throttling valve into the low-pressure section of the system, the evaporator. This pressure reduction facilitates the vaporization of the water, which effects the heat removal from the medium. The desired refrigeration effect is then provided accordingly. The weak solution flows down through a throttling valve to the absorber. After the evaporator, the cold refrigerant comes to the absorber and is absorbed by this weak solution. The strong solution is then obtained and is pumped by a solution pump to the generator, where it is again heated, and the cycle continues.

As mentioned before, thermodynamic analysis of absorption refrigeration system refers to the first and second laws. Therefore, each component in the system is considered as a control volume consisting input and output flows and heat and work transfer through the boundaries. It should be noted that, in absorption refrigeration system, there are two fluids (refrigerant and absorbent), make a working fluid and their composition at different points is different, particularly in absorber and generator. Thus, mass balance equations should be written for those two components and the whole mass. Referring to Fig.1, for a steady-

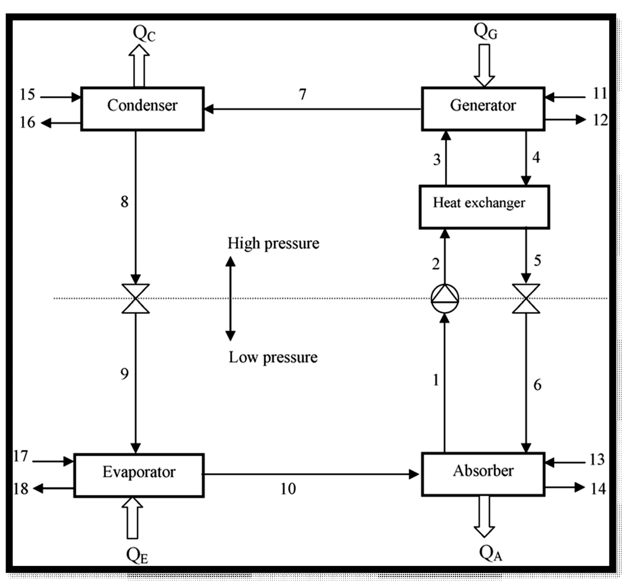

Fig. 1. Schematic of a single-effect absorption refrigeration system (Kizilkan *et al.*, 2007)

state steady flow (SSSF) process, mass balance equations are as follows (Cengel and Boles, 2010):

$$\sum \dot{m}_i - \sum \dot{m}_o = 0 \quad (1)$$

$$\sum (\dot{m}.x)_i - \sum (\dot{m}.x)_o = 0 \quad (2)$$

in which, $\dot{m}$ and x are mass flow rate and mass fraction of LiBr in the solution respectively.

In addition, energy balance equation or first law of thermodynamics is written as follows (Cengel and Boles, 2010):

$$\sum (\dot{m}.h)_i - \sum (\dot{m}.h)_o + \left[\sum Q_i - \sum Q_o + W \right] = 0 \quad (3)$$

The COP of the system is defined as the ratio of the evaporator cooling load (which provides refrigeration effect) to the generator heating load. Since the pumping work is negligible relative to the input heat at the generator, COP of the overall system becomes:

$$COP_{cooling} = \frac{Q_E}{Q_G} = \frac{\dot{m}_{10} h_{10} - \dot{m}_9 h_9}{\dot{m}_4 h_4 + \dot{m}_7 h_7 - \dot{m}_3 h_3} = \frac{\dot{m}_{17} (h_{17} - h_{18})}{\dot{m}_{11} (h_{11} - h_{12})} \quad (4)$$

where, $\dot{m}$ and h are mass flow rate and specific enthalpy of working fluid for the state points, respectively.

As mentioned before, exergy analysis is based on second law of thermodynamics. Neglecting chemical term, specific exergy of a flow is expressed as follows (Kotas, 1995):

$$\varepsilon = (h - T_0 s) + \frac{1}{2} V^2 + gz - (h_0 - T_0 s_0) \quad (5)$$

Also, neglecting kinetic and chemical exergy terms,specific exergycan be stated as below:

$$\varepsilon = (h - T_0 s) - (h_0 - T_0 s_0) \quad (6)$$

Exergy efficiency (second-law efficiency) of a system is expressed as a ratio of product exergy to fuel exergy (Kotas, 1995). In absorption refrigeration system, evaporator cooling load takes product role, and the input heat to the generator from external source, performs as a fuel. Therefore, exergy efficiency of absorption refrigeration system is defined by equation (7):

$$\Psi = \frac{\dot{E}_P}{\dot{E}_F} + \frac{\dot{E}_E}{\dot{E}_G} = \frac{\dot{m}_{17} \left[(h_{17} - h_{18}) - T_0 (s_{17} - s_{18}) \right]}{\dot{m}_{11} \left[(h_{11} - h_{12}) - T_0 (s_{11} - s_{12}) \right]} \quad (7)$$

Exergy balance equation fora steady-state steady flow condition is written as follow (Kotas, 1995):

$$\dot{E}_W = \sum \dot{E}_Q + \sum (\dot{m} \varepsilon)_i - \sum (\dot{m} \varepsilon)_0 + T_0 \dot{S}_{gen} \quad (8)$$

The last term in the above equation $(T_0 \dot{S}_{gen})$ is called "irreversibility" and is expressed as (Kizikan *et al.*, 2007):

$$\dot{I} = T_0 \dot{S}_{gen} \quad (9)$$

Now, irreversibility equation (Eq. (9)) is applied to each component of the system with respect to Fig. 1 as follows:

- Solution pump:

$$\dot{I}_{pump} = \dot{m}_1 T_0 (s_2 - s_1) \quad (10)$$

- Solution throttling valve:

$$\dot{I}_{STV} = \dot{m}_5 T_0 (s_6 - s_5) \quad (11)$$

- Solution heat exchanger:

$$\dot{I}_{SHE} = T_0 \left[\dot{m}_3 (s_3 - s_2) - \dot{m}_5 (s_4 - s_5) \right] \quad (12)$$

- Generator:

$$\dot{I}_G = T_0 \left[\dot{m}_4 s_4 - \dot{m}_3 s_3 - \dot{m}_{11} (s_{11} - s_{12}) + \dot{m}_7 s_7 \right] \quad (13)$$

- Condenser:

$$\dot{I}_C = T_0 \left[\dot{m}_8 (s_8 - s_7) - \dot{m}_{15} (s_{15} - s_{16}) \right] \quad (14)$$

- Refrigerant throttling valve:

$$\dot{I}_{RTV} = \dot{m}_8 T_0 (s_9 - s_8) \quad (15)$$

- Evaporator:

$$\dot{I}_E = T_0 \left[\dot{m}_9 (s_{10} - s_9) - \dot{m}_{17} (s_{17} - s_{18}) \right] \quad (16)$$

- Absorber:

$$\dot{I}_A = T_0 \left[\dot{m}_1 s_1 - \dot{m}_6 s_6 - \dot{m}_{13} (s_{13} - s_{14}) + \dot{m}_{10} s_{10} \right] \quad (17)$$

Total irreversibility of a system, is the summation of all components' irreversibility (Eq. 10 to 17), which result in:

$$\dot{I}_{tot} = T_0 \left[\dot{m}_{11} (s_{12} - s_{11}) - \dot{m}_{15} (s_{16} - s_{15}) + \dot{m}_{17} (s_{18} - s_{17}) - \dot{m}_{13} (s_{14} - s_{13}) \right] \quad (18)$$

Considering Eq. (18), expression $(s_A - s_B)$ for liquids is calculated from the following equation (Bergman, 2011):

$$s_A - s_B = c_p \ln \frac{T_A}{T_B} \quad (19)$$

Total cost function of the system is equal to sum of cost rate of fuel, capital cost and operation cost. Total cost function of the system is defined by equation (20)

$$\dot{C}_{P,tot} = \dot{C}_{F,tot} + \dot{Z}^{CI}_{tot} + \dot{Z}^{OM}_{tot} \quad (20)$$

Capital cost and operation cost of the equation (20) can be inserted into equation (21):

$$\dot{Z} = \dot{Z}^{CI}_{tot} + \dot{Z}^{OM}_{tot} = \sum \dot{Z}_k \quad (21)$$

Now, a unit cost can be taken into account for each exergy flow for example work, heat or exergy flows due to entering or leaving the materials through the system. This consideration can be stated as below:

$$\dot{C}_i = c_i\dot{E}_i = c_i(\dot{m}_i e_i) \quad (22)$$

$$\dot{C}_e = c_e\dot{E}_e = c_e(\dot{m}_e e_e) \quad (23)$$

$$\dot{C}_w = c_w\dot{W} \quad (24)$$

$$\dot{C}_Q = c_Q\dot{E}_Q \quad (25)$$

Where c_i ,c_e ,c_w and c_q are average costs per unit of exergy. Equation (20) can be reformed as below:

$$\sum_e \dot{C}_{e,k} + \dot{C}_{W,k} = \dot{C}_{Q,k} + \sum_i \dot{C}_{i,k} + \dot{Z}_k \quad (26)$$

Substituting equations (22) through (25) into equation (26) will lead to

$$\sum_e (c_e\dot{E}_e)_k + c_{w,k}\dot{W}_k = c_{q,k}\dot{E}_{q,k} + \sum_i (c_i\dot{E}_i)_k + \dot{Z}_k \quad (27)$$

where $\dot{E}_i$,$\dot{E}_e$,$\dot{W}$ and $\dot{E}_q$ are calculated from the first law of thermodynamics or the energy balance equation.

Equation (27) can be solved for all components simultaneously and then it is possible to determine the product cost with knowing the fuel price.

Total cost function is defined as the sum of the operational cost rate and the rate of capital cost. The former depends on the fuel price and the later stands for the capital investment and maintenance expenses. Therefore, total cost function calculates the total cost rate of a plant in terms of dollar per unit of time as below:

$$\dot{C}_T = c_F\dot{m}_F LHV + \sum \dot{Z}_k \quad (28)$$

Where c_F is the local fuel price per unit of energy, $\dot{m}_F$ is the fuel mass flow rate, and LHV is the lower heating value of fuel. In the present work, some modifications in the cost functions of that of reference (Cengel and Boles, 2010) are made to consider the regional conditions in Iran and also the annual inflation rate is taken into account. To calculate the cost rate from capital investment, the following equation can be used:

$$\dot{Z}_k = CRF \times \frac{\Phi_r}{(N \times 3600)} \times PEC_k = (\dot{Z}^{CI} + \dot{Z}^{OM}) \quad (29)$$

where PEC_k is the purchase cost function of the kth component, which is stated in terms of the system thermodynamic design factors (Cengel and Boles, 2010). The term PEC in equation (29) is obtained from manufacturer catalogs.

"Φ_r" is the maintenance factor and "CRF" is used to determine the annual cost of components, with assumption of 15 years operation, for system (n=15) and %12 interesting factor (i = .12); CRF is given by equation (1):

$$CRF = \frac{i(1+i)^n}{(1+i)^n - 1} \quad (30)$$

Thus the objective function ($\dot{C}_T$) is defined by equation (31):

$$\dot{C}_T = c_F\dot{m}_F LHV + \sum \frac{i\,(1+i)^n}{(1+i)^n - 1} \times \frac{\Phi_r}{(N \times 3600)} \times PEC_k \quad (31)$$

Economic parameters and assumptions are presented in Table1.

Four independent parameters were selected as decision variables, which are listed in Table2. Also, specified ranges for design parameters were applied as constraints to optimization problem which are shown in the following table.

Optimization procedure is performed in energy, exergy and cost approaches. Genetic algorithm is a general-purpose search method and non-deterministic optimization technique based upon the principles of evolution observed in nature. It combines selection, crossover, and mutation operators with the goal of finding the best solution to a problem. Potential solution are repeatedly graded on fitness and combined to produce new and potentially better solutions. GA searches for the optimal solution until a specified termination criterion is met (Florides *et al.*, 2003).

In this specific problem, number of generations were selected to be 1000, since the results for objective functions values (COP, Ψ and $\dot{C}_T$) after the 1000th generation remained unchanged. This is also set to be the termination criterion. The following important properties were considered in GA optimization method:

- Number of generations: 1000.
- Size of population: 100.
- Mutation type: Randomly.
- Maximum run time: 3600 Seconds.

Relations for objective functions are presented in Table3.

Because of the time-consuming optimization process and massive computations, a computer code was developed in Visual Basic 6.0 based on the

Table1. Economic parameters and assumptions

	Parameters	Value
1	N(hr)	7500
2	i	12%
3	n(year)	15
4	CRF	0.147
5	φ	1.2
6	$c_f\,(\$/GJ)$	8.588

presented model using GA optimization to obtain the optimal values of objective functions. The optimization procedure is shown in Fig.2. Regarding to Eqs. (4), (7) and (28), the optimization process is started by selecting design parameters with the applied constraints. The optimized values for decision variables are obtained when $COP\ and\ \Psi$, maximize and $\dot{C}_T$ minimize simultaneously.

RESULTS & DISCUSSION

The effects of various important design parameters on system COP and exergy efficiency have been studied here. COP variations with generator inlet hot water temperature (T_{11}), and evaporator chilling water temperature (T_{17}) are shown in Fig. 3 and Fig. 4 respectively. It can be obviously seen from the corresponding figures that, increasing and cause sensible increase in system COP. These can be interpreted as follows:

Increasing generator inlet hot water temperature, T_{11}, causes more refrigerant to vaporize and separates from absorbent which leads to high quality refrigerant vapor and, subsequently improved COP. Likewise, improvement in system COP is achieved by increasing evaporator chilling water temperature, T_{17}. This is mainly due to increased refrigerant potential, to extract heat from the refrigerated space (i.e. refrigeration effect) with higher chilling water temperature. This can be also understood with respect to Eq. (4), which expresses that, increasing Q_E leads to a higher COP. The effect of generator inlet hot water temperature (T_{11}) on exergy efficiency (Ψ), is shown in Fig. 5. Results show that the Ψ values, decreases sharply with increase in T_{11}. A heat source with higher temperature, provide hotter supply water for generator; however, the input exergy and subsequent exergy dissipation would be greater through the heat transfer process in the generator. This leads to a significant drop in exergy efficiency. Furthermore, Fig. 6 shows variation of exergy efficiency with evaporator chilling water temperature, T_{17}. Results show that, an absorption system with lower chilling water temperature has higher second-law efficiency. In fact, less input power is required to provide specified refrigeration effect, when chilling water enters to the evaporator with cooler temperature. As a result, exergy efficiency improves with decreasing T_{17}. The main reason to cause irreversibility in absorption system is undesirable heat transfer in system heat exchangers (Sencan *et al.*, 2005). Figs. 7-10 show changes in system total irreversibility (I_{tot}), with generator inlet hot water temperature (T_{11}), absorber inlet cooling water temperature (T_{13}), condenser cooling water temperature (T_{15}) and evaporator chilling water temperature (T_{17}), respectively. There is a common specification in all these figures which shows increase in system total irreversibility (I_{tot}) with rising temperatures. In fact, rising temperature

Table2. Design parameters and their ranges

	Design parameters	Ranges, ℃
1	Generator inlet hot water temperature, ??11	60-120
2	Absorber inlet cooling water temperature, ??13	20-40
3	Condenser inlet cooling water temperature, ??15	20-40
4	Evaporator inlet chilling water temperature, ??17	10-20

Table3. Objective functions

	Objective functions	Relations
1	$Objective\ function\ I = COP_{cooling}$	$\frac{Q_E}{Q_G} = \frac{\dot{m}_{17}(h_{17} - h_{18})}{\dot{m}_{11}(h_{11} - h_{12})}$
2	$Objective\ function\ II = \Psi$	$\frac{\dot{E}_P}{\dot{E}_F} + \frac{\dot{E}_E}{\dot{E}_G} = \frac{\dot{m}_{17}\left[(h_{17} - h_{18}) - T_0\left(s_{17} - s_{18}\right)\right]}{\dot{m}_{11}\left[(h_{11} - h_{12}) - T_0\left(s_{11} - s_{12}\right)\right]}$
3	$Objective\ function\ III = \dot{C}_T$	$c_f\,\dot{m}_f\,LHV + \sum \dot{Z}_k$

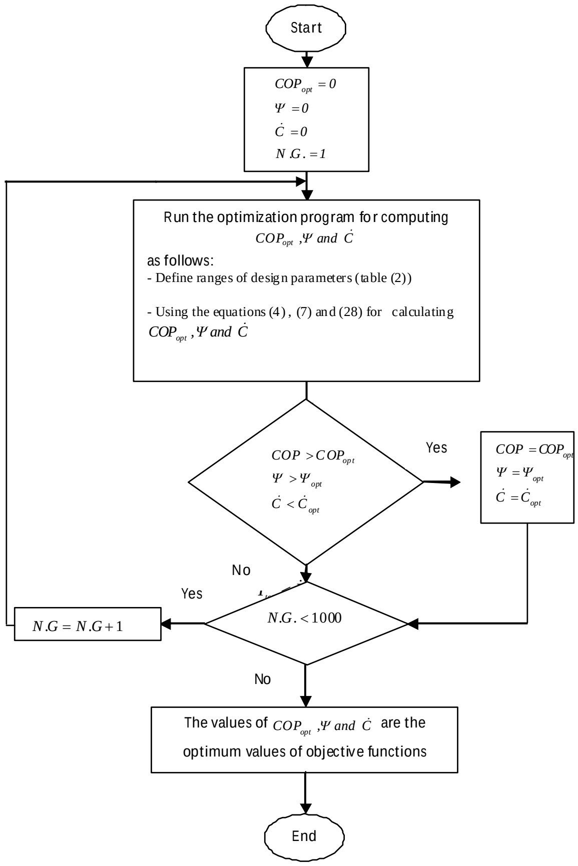

Fig. 2. Optimization process flowchart

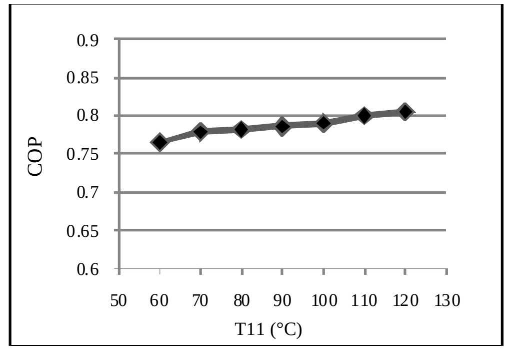

Fig. 3. COP variation with generator inlet hot water temperature (T_{11})

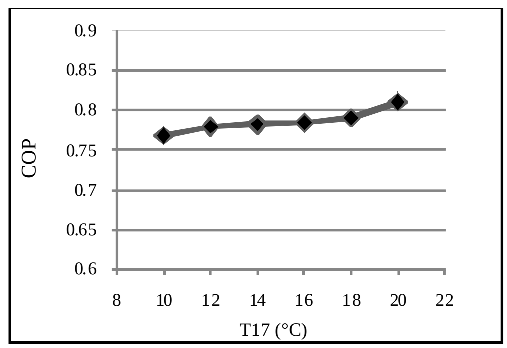

Fig. 4. COP variation with evaporator chilling water temperature (T_{17})

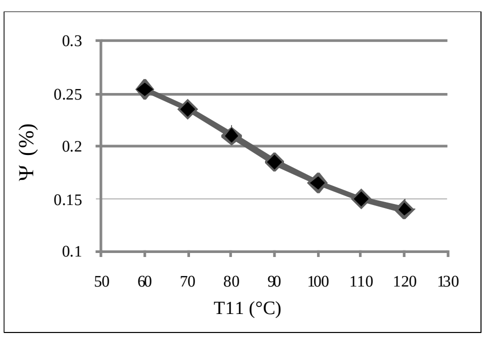

Fig. 5. Exergy efficiency variation with generator inlet hot water temperature (T_{11})

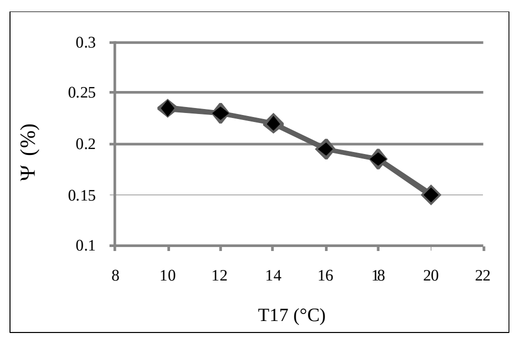

Fig. 6. Exergy efficiency variation with evaporator chilling water temperature (T_{17})

causes heat transfer rate to be increased which itself leads to more irreversibility. After thermodynamic modeling and optimization of the system were performed, optimal values for objective functions were found using GA optimization technique. As is shown in Table4, there is considerable progress in system COP, exergy efficiency and total cost function after the optimization was carried out. There is about 75 % boost in system COP, 47 % in Ψ and 12% in total cost which represent significant improvements. Considering common decision variables, COP value for absorption chillers is about 0.5 and exergy efficiency is about 14 %. However, in the current study, objective function values have been optimized by optimizing the decision variables in operational working range for the system. The values shown in Table 4 are calculated based on the decision variables that are in the logical range.

CONCLUSIONS

Energy and exergy analysis of a LiBr absorption refrigeration system were performed and as a result,

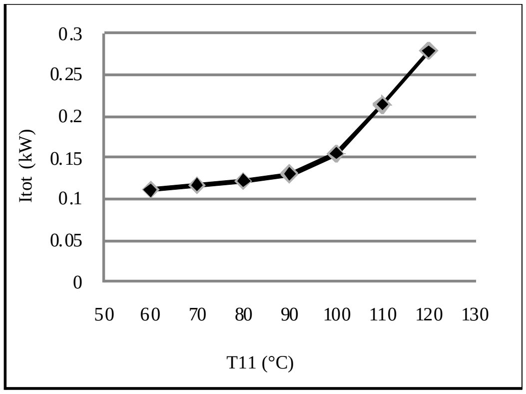

Fig. 7. Change in system total irreversibility (I_{tot}), with generator inlet hot water temperature (T_{11})

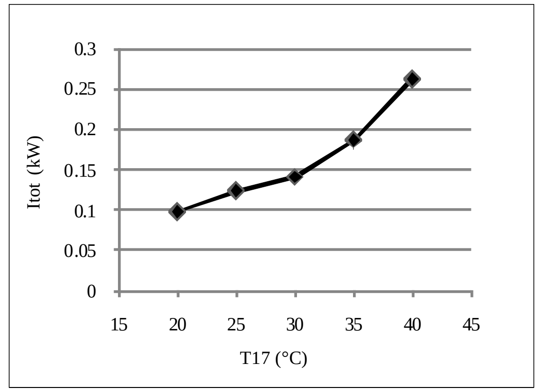

Fig. 8. Change in system total irreversibility (I_{tot}), with absorber inlet cooling water temperature (T_{13})

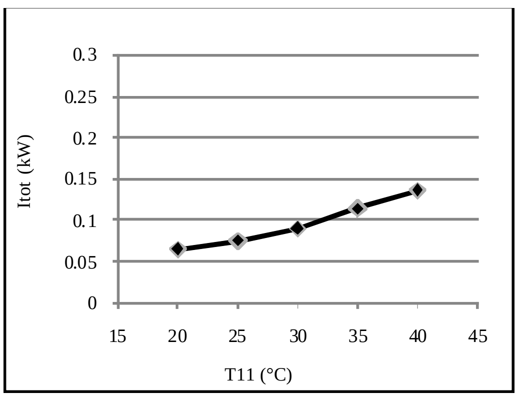

Fig. 9. Change in system total irreversibility (I_{tot}), with condenser cooling water temperature (T_{15})

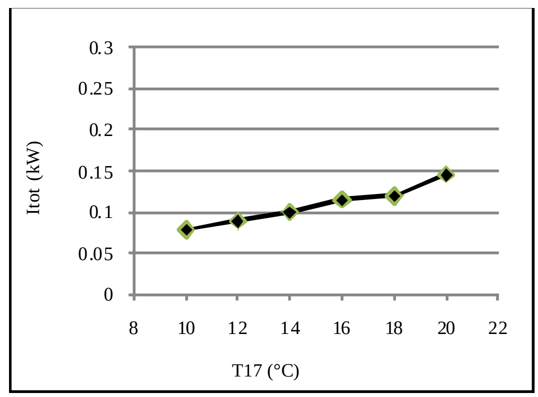

Fig. 10. Change in system total irreversibility (I_{tot}), with evaporator chilling water temperature (T_{17})

Table 4. Values for objective functions before and after optimization

Objective functions	Before optimization	After optimization (present work)	Improvement percentage
COP	0.451	0.787	74.5
$\Psi\,(\%)$	13.4	19.7	47
$\dot{C}_T\ (\$ / s)$	0.471	0.414	12

thermodynamic model of a system was presented. Then, optimization process was carried out in energy, exergy and cost approaches by using concepts of COP and second-law efficiency. Genetic algorithm method was applied to achieve optimum design. Also, the effects of various design parameters on system optimum performance were investigated. The results showed:

• COP of the system increases with rise in generator inlet hot water temperature and evaporator chilling water temperature.

• Exergy efficiency of the system increases in favor of rising generator inlet hot water temperature, while treats in the reverse direction with increasing evaporator chilling water temperature.

• Thermodynamic optimization of the system leads to dramatic improvement in system performance.

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Nomenclature	
c_P	Specific Heat Capacity (J/kgK)
COP	Coefficient Of Performance
CRF	Capital Recovery Factor
$\dot{C}$	Cost of Exergy (\$)
c	Unit Cost of Exergy $(\$GJ^{-1})$
$\dot{E}$	Rate of Exergy (kW)
g	Gravitational Acceleration (m/s^2)
h	Specific Enthalpy (kJ/kg)
I	Interest Rate
$\dot{I}$	Rate of Irreversibility (kW)
$LiBr$	Lithium Bromide
$\dot{m}$	Mass Flow Rate *(Kg/S)*
N	Total Operating Period
$N.G.$	Number of Generation
P	Pressure (Pa)
PEC	Purchased Equipment Cost (\$)
$\dot{Q}$	Heat Load (kW)
s	Specific Entropy (kJ/kgK)
S	Entropy (kJ/K)
T	Temperature (K)
V	Bulk Velocity of The Stream (m/s)
$\dot{W}$	Work (kW)
x	Mass Fraction of Lithium Bromide (%)
z	Altitude of The Stream (m)
$\dot{Z}$	Capital Cost Rate $(\$s^{-1})$

Greek Abbreviations	
ε	Specific Exergy (kJ/kg)
ψ	Exergetic Efficiency
Φ_r	Maintenance Factor

Subscripts	
A	Absorber
C	Condenser
E	Evaporator
EV	Expansion Valve
F	Fuel
G	Generator
gen	Generated
i	Inlet Stream
o	Outlet Stream
opt	Optimum State
P	Product
$Pump$	Pump
Q	Heat Load
SHE	Solution Heat Exchanger
tot	Total
W	Work
0	AmbientCondition

REFERENCES

Bejan, A. (1996). Entropy generation minimization. (New York: CRC Press)

Bergman, T. L., Lavine,A. S., Incropera, F. P. and DeWitt, D. P. (2011). Fundamentals of Heat and Mass Transfer. New York: JohnWiley &Sons.

Cengel, A.Y. and Boles, A.M. (2010). Thermodynamics: an Engineering Approach. New York: McGraw-Hill.

Chua, H.T., Toh, H.K., Malek, A. and Sirinivasan,K. (2000).A general thermodynamic framework forunderstanding the behavior of absorption chillers.International Journal of Refrigeration, **23**, 491–507.

Florides, G.A., Kalogirou, S.A., Tassou, S.A. and Wrobel, L.C. (2002). Modeling simulationand warming impact assessment of adomestic-size absorption solarcooling system. Applied Thermal Engineering, **22**, 1313–1325.

Florides, G.A., Kalogirou, S.A., Tassou, S.A. and Wrobel, L.C. (2003). Design and construction of a LiBr–water absorptionmachine.Energy Conversion and Management, **44 (15)**, 2483–508.

Gebreslassie, B.H., Guille´n-Gosa´ lbez, G., Jime´nez, L. and Boer, D. (2009). Design of environmentally conscious absorption cooling systems via multi-objective optimization and life cycle assessment. Appl. Energ, **86 (9)**, 1712-1722.

Gebreslassie, B.H., Groll, E.A. and Garimella, S.V. (2012). Multi-objective optimization of sustainable single-effect Water Lithium bromide absorption cycle. Renew. Energy. **46**, 100-110.

Godarzi, A.A., Jalilian, M., Samimi, J., Jokar, A. and Vesaghi, M.A. (2013). Design of a PCM storage system for a solar absorption chiller based on exergoeconomic analysis and genetic algorithm. International Journal of Refrigeration, **36 (1)**, 88-101.

Gogoi, T.K. and Talukdar, K. (2014). Thermodynamic analysis of a combined reheat regenerative thermal power plant and water–LiBr vapor absorption refrigeration system. Energy Conversion and Management, **78**, 595–610.

Kilikan, O., Sencan, A. and Kalogirou, S.A. (2007). Thermoeconomic optimization of a LiBr absorption refrigeration system. Chemical Engineering and Processing, **46**, 1376–1384.

Kotas, T. J. (1995) .The Exergy Method of Thermal Plant Analysis. Florida: Krieger PublishingCompany.

Lee, S.F. and Sherif, S.A. (2001). Thermodynamic analysis of a lithium bromide/water absorption system for cooling and heating applications. International Journal of Energy Research, **25**, 1019–31.

Mazzei, M.S. , Mussati, M.C. Mussati, S.F. (2014). NLP model-based optimal design of LiBr-H2O absorption refrigeration systems. International Journal of Refrigeration, **38**, 58-70

Sencan, A., Yakut, K.A. and Kalogirou, S.A. (2005). Exergy analysis of lithium bromide/water absorption systems. Renewable Energy, **30**, 645–657.

Talbi, M.M. and Agnew, B. (2000). Exergy analysis of an absorption refrigerator using lithium bromide and water asworking fluids. Applied Thermal Engineering, **20**, 619–30.

Yumrutas, R., Kunduz, M. and Kanoglu, M. (2002). Exergyanalysis ofvapor compression refrigeration systems. Exergy, An International Journal, **2**,266–272.