

Modified thermo-ecological optimization for refrigeration systems and an application for irreversible four-temperature-level absorption refrigerator

Emin Acikkalp^{1*}

* Corresponding author

Email: acikkalp@gmail.com

¹ Department of Mechanical and Manufacturing Engineering, Engineering Faculty, Bilecik S.E. University, Bilecik, Turkey

Abstract

In this study, a modified ecological function is presented for four-temperature-level absorption refrigerators, and an absorption refrigerator has been optimized using this function. An equivalent system was initially identified, followed by the application of the first and second laws of thermodynamics. Next, the optimization results were interpreted. The results of the optimization process, optimum point of x_H , x_R , a_H , and a_R , were defined as 0.87, 0.58, 2.41, 0.25, respectively. In addition to that, the effects of x_H , x_R , a_H , a_R , T_a , T_c , T_e , and T_g were shown according to exergy destruction, ecological function, coefficient of performance, and cooling load. As a result, it was understood that especially the x_H , parameter, and the generator temperatures are the variables of the system that require special attention because, at some points, they cause that generator and absorber transfer heat in the opposite direction they need to do.

Keywords

Absorption system, Optimization, Irreversibility, Refrigeration, Ecological

Background

Absorption refrigerators may provide a cooling effect using various energy resources in the system such as waste heat and renewable energy. However, the factor that makes them attractive is not only the diversity of energy resources used, but also their economic aspects and minimal pollution effect on the environment. In addition to these, it was demonstrated that absorption refrigerators are more economic when they are integrated with the combined heat and power stations. Finite-time thermodynamics have been used for the optimization of thermal systems since the 1970s. The ecological criterion [1] suggested by Angulo-Brown is another optimization method. This criterion is shown as $\dot{E} = \dot{W} - T_L \dot{\sigma}$ for the finite-time Carnot machines. Here, T_L is the cold reservoir temperature, and $\dot{\sigma}$ is the entropy production of the system. However, Yan demonstrated that showing the ecological criterion as $\dot{E} = \dot{W} - T_0 \dot{\sigma}$ would be more reasonable [2]. Here, T_0 is the ambient temperature. There are many studies in the literature for the optimization of irreversible absorption refrigerators.

Bhardwaj et al. optimized absorption refrigerators using finite-time thermodynamics [3,4]. Huang et al. examined the surface area, coefficient of performance (COP), and ecological criteria for the irreversible four-temperature-level absorption refrigerator and used an ecological function based on energy analysis [5]. The ecological function they used is $E^* = \dot{Q}_L - \varphi_C T_L \dot{\sigma}$. Here, φ_C is the Carnot COP of the system, T_L is the cold reservoir temperature, \dot{Q}_L is the cooling load, and $\dot{\sigma}$ is the entropy production. Chen offered models for the optimization of absorption refrigerators, heat pumps, and cooling machines [6,7]. Sun et al. examined the optimization of the relation between heating load and entropy production of the internal heating heat exchanger [8]. Chen et al. offered a model for the analysis of four-temperature-level absorption refrigerator cycles [9]. Chen and Yan made an optimization on the irreversible absorption heat exchangers for ecological criterion [10]. Yan and Chen [11] and Yan and Lin [12] optimized the three-temperature-level absorption cycles for the ecological criterion on the basis of energy and exergy output of the irreversible Carnot cycles. Fathi et al. offered a model for irreversible and solar absorption cooling [13]. In addition to these, except for the ones stated here, there are many performance optimization studies for absorption refrigerators [14-17]. In addition, new optimization methods called ‘advanced exergy-based analysis methods’ [18,19] or ‘exergy-entropy relationship’ [20] of the system can be investigated for the future studies. Contrary to other studies in which cooling load is the only evaluation criteria, this study aimed to provide maximum cooling load, minimum operation input, and minimum entropy production in the system. Due to this reason, a modified ecological function was presented. Considering the energy requirements of the world, the purpose of this study is to contribute to the more effective operation of absorption refrigerators by presenting a modified ecological function.

Methods

In this study, a four-temperature-level absorption refrigerator was optimized using a modified ecological function, and the results were presented. A vapor absorption refrigeration system consists of four components as generator, absorber, evaporator, and condenser. Working fluid in the system is made of a refrigerant and a solution that absorbs refrigerant. Refrigerant vapor comes from the evaporator and goes to the absorber, and it is absorbed in a liquid. The liquid is sent to the generator and is boiled out of the solution, taking the generator’s heat reservoir. It is sent to the condenser and becomes a rejected heat to the environment. Finally, working fluid comes back to the evaporator, and the cycle is completed. The system is the general absorption refrigerator (seen in Figure 1). Temperatures of heat reservoirs are T_A , T_C , T_E , and T_G in absorber, condenser, evaporator, and generator, respectively. T_a , T_c , T_e , and T_g are working fluid temperatures in absorber, condenser, evaporator, and generator, respectively. In Figure 2, an equivalent system is defined. In the equivalent system, the absorber-generator assembly can be considered as heat engine part, and similarly, condenser-evaporator part can be considered as refrigerator part.

Figure 1 Generalized four-temperature-absorption heat pump system [3].

Figure 2 An equivalent system for generalized four-temperature-absorption heat pump system [3].

Thermodynamic analysis of system

Thermodynamic analysis and optimization were established for the equivalent system, and previous formulas offered by Chen et al. were used in the calculations [21]. The ambient temperature was assumed as 300 K.

Thermodynamic analysis of the absorber-generator assembly (heat engine part)

According to the first law of thermodynamics,

$$\dot{Q}_{GE} - \dot{Q}_{AB} - \dot{W} = 0. \quad (1)$$

Here, \dot{Q}_{GE} (heat input from the generator heat source) and \dot{Q}_{AB} (heat output to the absorber heat sink) may be defined as

$$\dot{Q}_{GE} = \alpha A_{GE}(T_G - T_g) \text{ and } \dot{Q}_{AB} = \beta A_{AB}(T_a - T_A). \quad (2)$$

According to the second law of thermodynamics,

$$\frac{\dot{Q}_{GE}}{T_g} - \frac{\dot{Q}_{AE}}{T_a} \leq 0 \text{ or } \frac{\dot{Q}_{AB}}{T_a} = I_H \frac{\dot{Q}_{GE}}{T_g}. \quad (3)$$

Here, I_H is the internal irreversible parameter for absorber-generator assembly. $I_H > 0$ for the irreversible cycles, and $I_H = 0$ for the reversible cycles. Working fluid temperature ratio (x_H), heat transfer surface area ratio (a_H), and total heat transfer surface area ($A_{T,H}$) for the absorber-generator assembly can be defined as following:

$$x_H = \frac{T_a}{T_g}, \quad a_H = \frac{A_{AB}}{A_{GE}}, \quad \text{and } A_{T,H} = A_{GE} + A_{AB} \text{ are defined.} \quad (4)$$

T_g , T_a , and \dot{Q}_{GE} can be defined using Equations (1) to (3):

$$T_g = \frac{I_H \alpha T_G x_H + a_H \beta T_A}{x_H (a_H \beta + I_H \alpha)} \quad (5)$$

$$T_a = \frac{I_H \alpha T_G x_H + \beta a_H T_A}{(\beta a_H + I_H \alpha)} \quad (6)$$

$$\dot{Q}_{GE} = \frac{\alpha A_{T,H}}{1 + a_H} \left(\frac{\beta a_H (T_G x_H - T_A)}{x_H (\beta a_H + I_H \alpha)} \right). \quad (7)$$

Thermodynamic analysis of the condenser-evaporator assembly (refrigerator part)

According to the first law of thermodynamics,

$$\dot{Q}_{CO} - \dot{Q}_{EV} - \dot{W} = 0. \quad (8)$$

Here, \dot{Q}_{CO} (heat rejection from the condenser) and \dot{Q}_{EV} (heat input to the evaporator) are

$$\dot{Q}_{CO} = \mu A_{CO}(T_c - T_c) \text{ and } \dot{Q}_{EV} = \theta A_{EV}(T_E - T_e). \quad (9)$$

According to the second law of thermodynamics,

$$\frac{\dot{Q}_{CO}}{T_c} - \frac{\dot{Q}_{EV}}{T_e} \leq 0 \text{ or } \frac{\dot{Q}_{CO}}{T_c} = I_R \frac{\dot{Q}_{EV}}{T_e} \quad (10)$$

Here, I_R is the internal irreversible parameter for condenser-evaporator assembly. $I_R > 0$ for the irreversible cycles, and $I_R = 0$ for the reversible cycles. Working fluid temperature ratio (x_R), heat transfer surface area ratio (a_R), and total heat transfer surface area ($A_{T,R}$) for the condenser-evaporator assembly can be defined as following:

$$x_R = \frac{T_e}{T_c} \quad (11)$$

$$a_R = \frac{A_{CO}}{A_{EV}}$$

$$A_{T,R} = A_{CO} + A_{EV}.$$

T_c , T_g , \dot{Q}_{CO} , and \dot{Q}_{EV} can be defined using Equations (8) to (11):

$$T_c = \frac{T_c \mu x_R a_R + \theta T_E I_R}{x_R (\theta I_R + \mu a_R)} \quad (12)$$

$$T_e = \frac{T_E \theta I_R + \mu x_R a_R T_c}{\mu a_R + \theta I_R} \quad (13)$$

$$\dot{Q}_{CO} = \frac{\mu A_{T,R} a_R}{1 + a_R} \left(\frac{\theta I_R (T_E - T_c x_R)}{x_R (\theta I_R + \mu a_R)} \right) \quad (14)$$

$$\dot{Q}_{EV} = \frac{\theta A_{T,R}}{1 + a_R} \left(\frac{\mu a_R (T_E - x_R T_c)}{\mu a_R + \theta I_R} \right) \quad (15)$$

Optimum ecological function of the system

The modified ecological function, including the values of maximum cooling load in the absorption cooling system, maximum power output for the heat engine, and maximum energy destruction of the system is defined below:

$$\dot{E} = (\dot{Q}_{EV}) + (\dot{Q}_{GE}) - (\dot{Q}_{AB}) - T_o \left(\frac{\dot{Q}_{AB}}{T_A} - \frac{\dot{Q}_{GE}}{T_G} + \frac{\dot{Q}_{CO}}{T_c} - \frac{\dot{Q}_{EV}}{T_E} \right). \quad (16)$$

Substituting (3) to (15) in (16) and setting Equation (16)'s derivation to zero with respect to $a_H \left(\frac{\partial \dot{E}}{\partial a_H} = 0 \right)$, the value for which the optimum heat exchanger area was detected for the assembly of the absorber-generator and which corresponds to such area is as below:

$$a_{o,H} = \sqrt{\frac{I_H \alpha}{\beta}}. \quad (17)$$

Similarly, the optimum heat rate of the operation fluids is detected by decreasing the derivative of the ecological function to $x_H \left(\frac{\partial \dot{E}}{\partial x_H} = 0 \right)$ for the assembly of the absorber-generator, and the value corresponding to this rate is as below:

$$x_{o,H} = \sqrt{\frac{T_A T_a (T_G + T_0)}{T_G I_H T_g (T_A + T_0)}} \quad (18)$$

By applying the same calculations for the assembly of the condenser-evaporator $a_R \left(\frac{\partial \dot{E}}{\partial a_R} = 0 \right)$,

$x_R \left(\frac{\partial \dot{E}}{\partial x_R} = 0 \right)$ values are detected, and these values are as below,

$$a_{o,R} = \sqrt{\frac{I_R \theta}{\mu}} \quad (19)$$

$$x_{o,R} = \sqrt{\frac{T_E I_R T_e T_0}{T_C T_c (T_E + T_0)}} \quad (20)$$

Substituting (17) to (20) in (16), the optimized ecological function is calculated as

$$\begin{aligned} \dot{E}_o = & \frac{\mu \theta A_{T,R} (I_R T_C T_E T_0 + T_C T_e (T_E + T_0) - 2 \sqrt{I_R T_E T_e T_0 T_C T_c (T_E + T_0)})}{(\sqrt{\theta I_R} + \sqrt{\mu})^2 T_E T_C} \\ & + \frac{\alpha A_{T,H} \beta (I_H T_a T_G (T_A + T_0) + T_A T_g (T_G + T_0) - 2 \sqrt{I_H T_g T_G T_a T_A (T_G + T_0) (T_A + T_0)})}{(\sqrt{\beta} + \sqrt{\alpha I_H})^2 T_A T_G} \end{aligned} \quad (21)$$

Optimum COP of the system

The COP of the absorption system can be defined as

$$\phi = \frac{\dot{Q}_{EV}}{\dot{Q}_{GE}} \quad (22)$$

The COP of the system is described by substituting Equations (7) and (15) into (22)

$$\phi = \frac{\dot{Q}_{EV}}{\dot{Q}_{GE}} = \frac{\theta A_{T,R} a_R (\mu a_R (T_E - x_R T_C))}{\frac{1 + a_R}{1 + a_H} \left(\frac{\mu a_R + \theta I_R}{x_H (\beta a_H + I_H \alpha)} \right)} \quad (23)$$

The optimized COP value is calculated as below, by putting the values in their places in Equation (23):

$$\phi_{EC} = \frac{A_{T,R} \mu \theta (\sqrt{\beta} + \sqrt{\alpha} \sqrt{I_H})^2 \sqrt{T_A T_a (T_G + T_0)} (T_c \sqrt{I_R T_E T_e T_0} - T_e \sqrt{T_C T_c (T_E + T_0)})}{A_{T,H} \alpha \beta (\sqrt{\mu} + \sqrt{\theta I_R})^2 \sqrt{T_C T_c (T_E + T_0)} \left(T_a \sqrt{I_H T_g T_g (T_A + T_0)} - T_g \sqrt{T_a T_A (T_G + T_0)} \right)} \quad (23)$$

Optimum cooling load of the system

The optimum cooling load of the system is calculated by putting the optimized values in their places in Equation (15):

$$\dot{Q}_{EV,EC} = \frac{A_{T,R}\mu\theta(T_e\sqrt{T_c T_c(T_E+T_o)} - T_c\sqrt{I_R T_E T_e T_o})}{(\sqrt{\mu} + \sqrt{\theta I_R})^2 \sqrt{T_c T_c(T_E+T_o)}} \quad (24)$$

Results and discussion

This section presents an implementation for the absorption. The performance characteristics used for the numerical calculations are given in Table 1. The calculated optimum parameters for the system are listed in Table 2.

Table 1 Fixed values for cycle

Data	Unit	Value
α^a	$\text{kW m}^{-2} \text{K}^{-1}$	2.30
β^a	$\text{kW m}^{-2} \text{K}^{-1}$	0.4
μ^a	$\text{kW m}^{-2} \text{K}^{-1}$	3.265
θ^a	$\text{kW m}^{-2} \text{K}^{-1}$	0.195
A_{AB}	m^2	60
A_{CO}	m^2	20
A_{EV}	m^2	150
A_{GE}	m^2	45
I_H	kW K^{-1}	1.05
I_R	kW K^{-1}	1.05
T_a	K	290
T_c	K	360
T_e	K	250
T_g	K	354
T_A	K	315
T_C	K	300
T_E	K	285
T_G	K	365

^aData from [22]

Table 2 Calculated optimum parameters for the system

Data	Unit	Value
$a_{H,o}$	Dimensionless	2.41
$a_{R,o}$	Dimensionless	0.25
$x_{H,o}$	Dimensionless	0.87
$x_{R,o}$	Dimensionless	0.58

The ecological function and exergy destruction of the absorption cooling system and its change according to COP a_H are given in Figure 3. Here, it can be seen that as the COP value a_H increases, it decreases logarithmically, and it decreases to the value of 2.6 for the optimum a_H value. However, this value is quite high for the currently used absorption cooling systems. In addition, it can be seen that the a_H value is not very effective for the exergy destruction. While the a_H value changes between 0.1 and 0.3, the increase seen for the exergy destruction is 3 kW.

Figure 3 Effect of a_H ($a_{R,E} = 0.25$, $x_{H,E} = 0.87$, $x_{R,E} = 0.58$) on the system. E , ecological function; ExD, exergy destruction; Φ , COP.

The ecological function, exergy destruction, and cooling load of the absorption cooling system and its change according to COP a_R are given in Figure 4. Here, it can be seen that the ecological function, exergy destruction, and cooling load increase logarithmically until optimum point with the increase of the a_R parameter.

Figure 4 Effect of a_R ($a_{H,E} = 2.41$, $x_{H,E} = 0.87$, $x_{R,E} = 0.58$) on the system. E , ecological function; ExD, exergy destruction; Φ , COP; Q_E , cooling load.

In Figure 5, the change of absorption cooling system according to x_H is evaluated for the ecological function, exergy destruction, and COP. For the values of x_H lower than 0.82, it can be seen that the generator radiates heat to the environment instead of removing heat from the environment, and the absorber removes heat from the environment instead of radiating heat to the environment. In other words, the heat machine part consumes power instead of producing power, and the system becomes unusable with these values. The COP of the system decreases with the increase of x_H , and it requires an approximate value of 2.6 for the optimum x_H . Exergy destruction also increases with the increase of x_H , and an increase of 300 kW is seen with an exergy destruction in x_H with a value of 0.2.

Figure 5 Effect of x_H ($a_{H,E} = 2.41$, $a_{R,E} = 0.25$, $x_{R,E} = 0.58$) on the system. E , ecological function; ExD, exergy destruction; Φ , COP.

In Figure 6, the change of the absorption cooling system according to x_R is evaluated for the ecological function, exergy destruction, cooling load, and COP. The COP of the system decreases with the increase of x_R , and it approximately takes a value of 2.4 for the optimum x_R . Exergy destruction and ecological function change drastically together with x_R . The cooling load decreases with the increase of the value of x_R . While the cooling capacity is 4917.49 kW when the x_R value is 0.05, it decreases to 339.317 kW when the x_R value is 0.65.

Figure 6 Effect of x_R ($a_{H,E} = 2.41$, $a_{R,E} = 0.25$, $x_{H,E} = 0.87$) on the system. E , ecological function; ExD, exergy destruction; Φ , COP; Q_E , cooling load.

The effects of the absorber temperature on the system are given in Figure 7. As the absorber temperature increases, it can be seen that while the ecological function and exergy destruction values decrease, the COP value increases.

Figure 7 Effect T_a ($a_{H,E} = 2.41$, $a_{R,E} = 0.25$, $x_{H,E} = 0.87$, $x_{R,E} = 0.58$) on the system. E , ecological function; ExD , exergy destruction; Φ , COP.

The effect of the generator temperature on the absorption cooling system is given in Figure 8. For temperatures lower than 335 K, the generator radiates heat to the environment instead of removing heat from the environment, so it cannot be used in lower temperatures. In addition to this, while the increase of the generator temperature increases ecological function and exergy destruction, it decreases the COP.

Figure 8 Effect of T_g ($a_{H,E} = 2.41$, $a_{R,E} = 0.25$, $x_{H,E} = 0.87$, $x_{R,E} = 0.58$) on the system. E , ecological function; ExD , exergy destruction; Φ , COP.

The effect of a condenser on the system is given in Figure 9. As seen here, as the condenser temperature increases, the ecological function, exergy destruction, cooling load, and COP values decrease.

Figure 9 Effect of T_c ($a_{H,E} = 2.41$, $a_{R,E} = 0.25$, $x_{H,E} = 0.87$, $x_{R,E} = 0.58$) on the system. E , ecological function; ExD , exergy destruction; Φ , COP; Q_E , cooling load.

In Figure 10, COP, ecological function, exergy destruction, and cooling load increase linearly with the increase of evaporator temperature.

Figure 10 Effect of T_e ($a_{H,E} = 2.41$, $a_{R,E} = 0.25$, $x_{H,E} = 0.87$, $x_{R,E} = 0.58$) on the system. E , ecological function; ExD exergy destruction; Φ , COP.

Conclusions

The optimization of four-temperature-level irreversible absorption refrigerator was researched for modified ecological function in this study. By this method, it was attempted to establish a balance between the cooling load, the load supplied to the system, and entropy production. The effects of a_H , a_R , x_H , and x_R parameters and operation fluid temperatures on the system were studied, and the results were shown in Figures 3, 4, 5, 6, 7, 8, 9, and 10. Considering these results, the criteria that should be observed in designing an absorption cooling cycle were determined, and the parameters that should be observed were shown. This study will hopefully be a model for absorption refrigerator designs.

Abbreviations

Nomenclature

a , ratio of heat exchanger surface areas; A , heat exchanger surface area (m^2); COP, coefficient of performance; E , ecological function (kW); ExD, exergy destruction (kW); I , internal irreversibility parameter; m , ratio of environment temperature to absorber; n , ratio of environment temperature to generator; Q , heat (kW); T , temperature (K); W , work (kW); x , ratio of temperatures; y , ratio of environment temperature to condenser; z , ratio of environment temperature to evaporator.

Subscripts

0, environment; 1, inlet, 2, outlet; a, absorber working fluid temperature; A, mean absorber heat source temperature; AB, evaporator; c, condenser working fluid temperature; C, mean condenser heat source temperature; CO, condenser; e, evaporator working fluid temperature; EC, ecological; E, mean evaporator heat source temperature; EV, evaporator; g, generator working fluid temperature; G, mean generator heat source temperature; GE, generator; H, heat engine part; R, refrigerator part; o, optimum; T, total.

Greek letters

α , total heat conductance of generator ($\text{kW m}^{-2} \text{K}^{-1}$); β , total heat conductance of absorber ($\text{kW m}^{-2} \text{K}^{-1}$); Φ , coefficient of performance of system (COP); μ , total heat conductance of condenser ($\text{kW m}^{-2} \text{K}^{-1}$); θ , total heat conductance of evaporator ($\text{kW m}^{-2} \text{K}^{-1}$); σ , entropy generation (kW K^{-1}).

Competing interests

The authors declare that they have no competing interests.

Authors' information

EA is a research assistant in the Faculty of Engineering at Bilecik S.E. University. He received his BSc and MSc in the Engineering and Architecture Faculty in Eskisehir Osmangazi University in 2000 and 2004, respectively.

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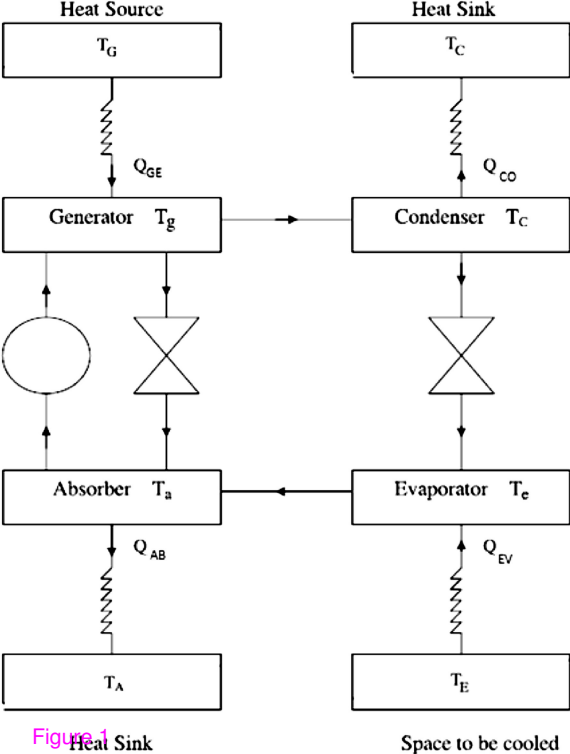
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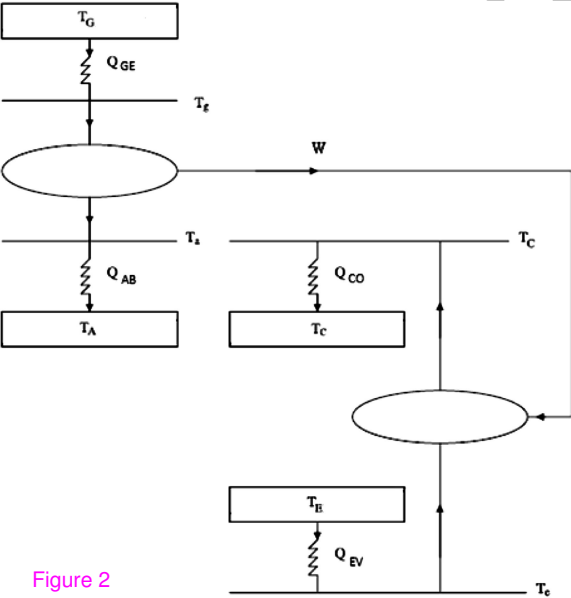


Figure 2

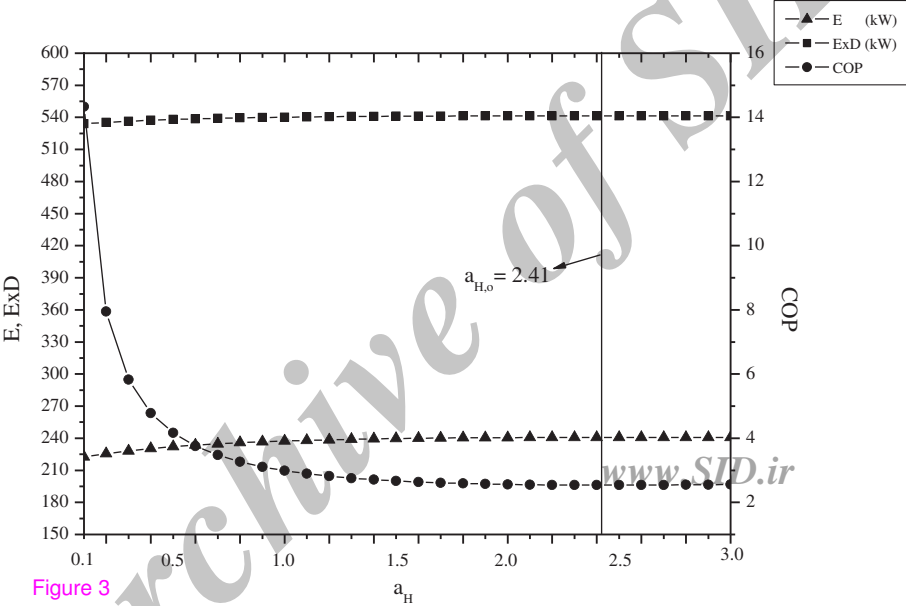


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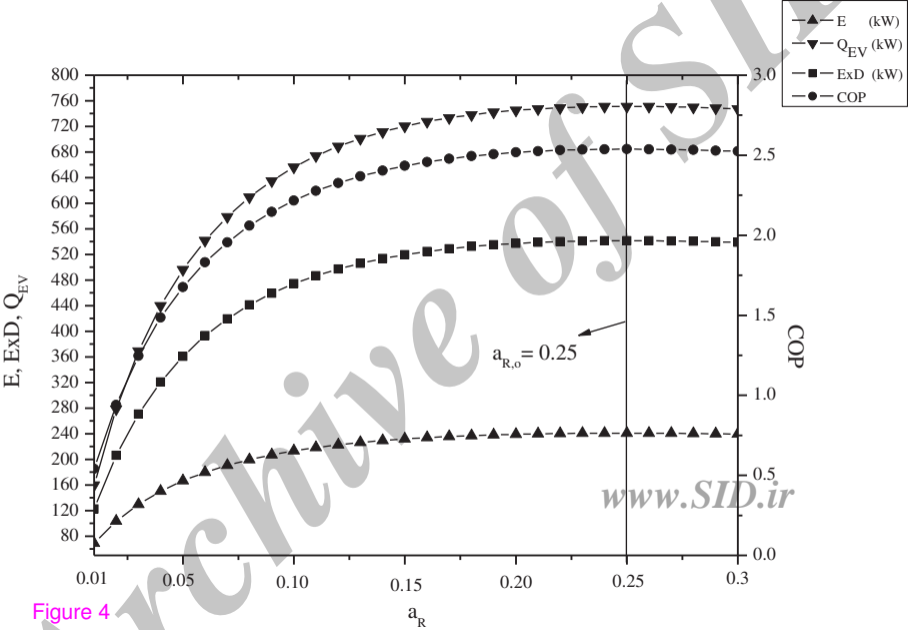


Figure 4

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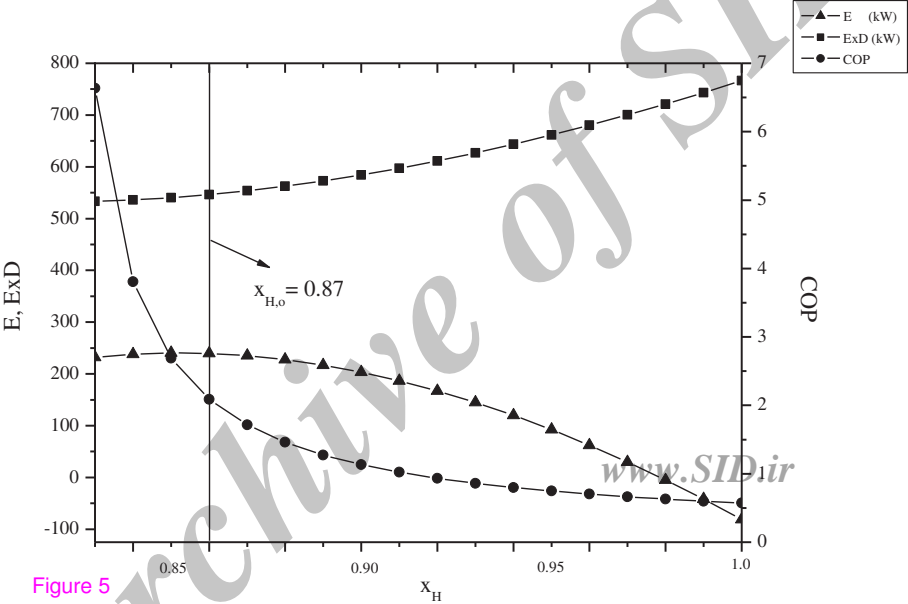


Figure 5

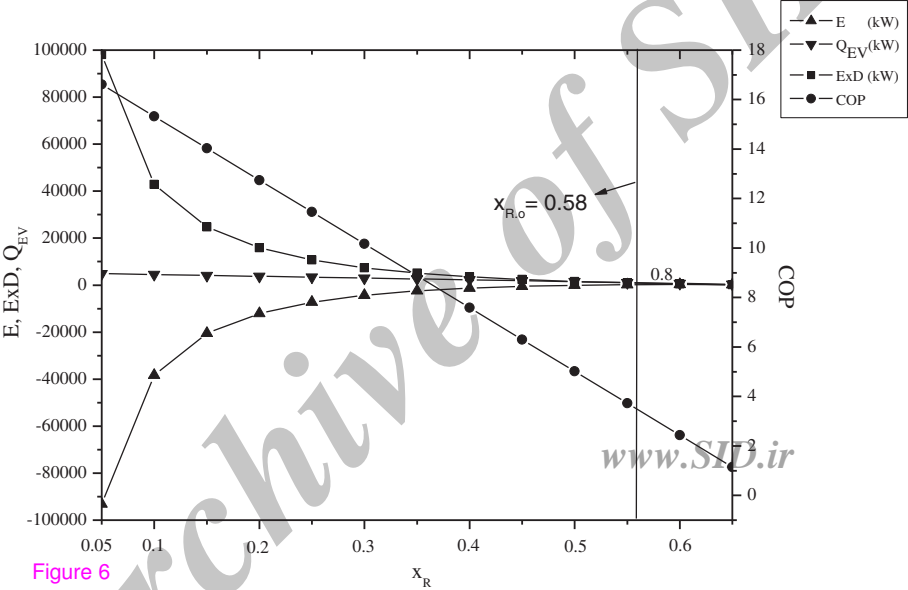


Figure 6

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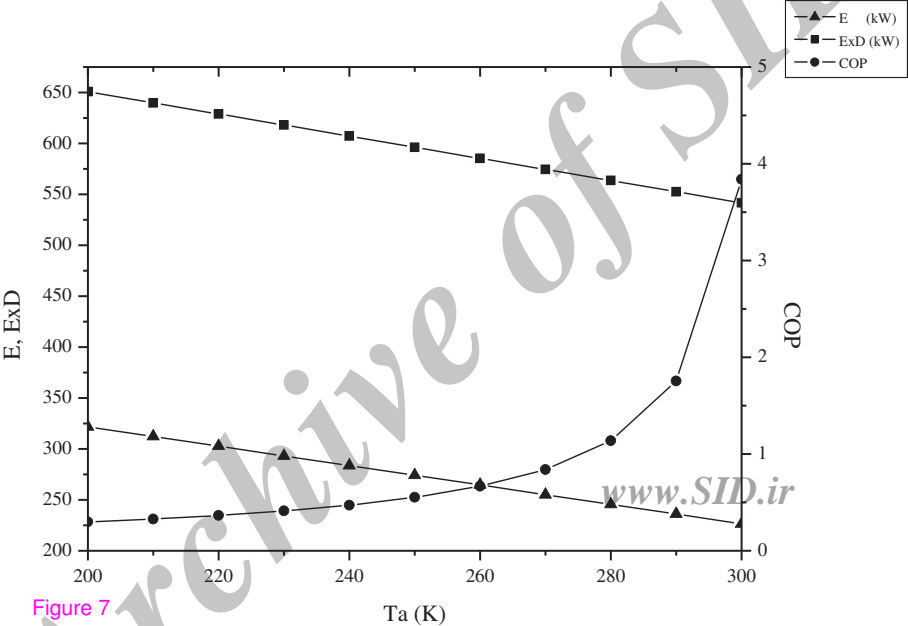


Figure 7

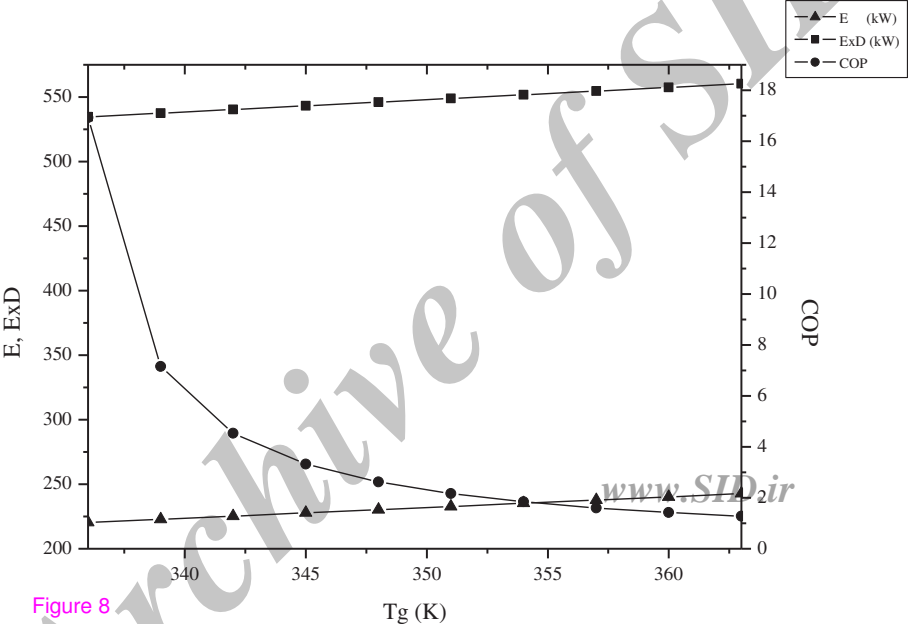


Figure 8

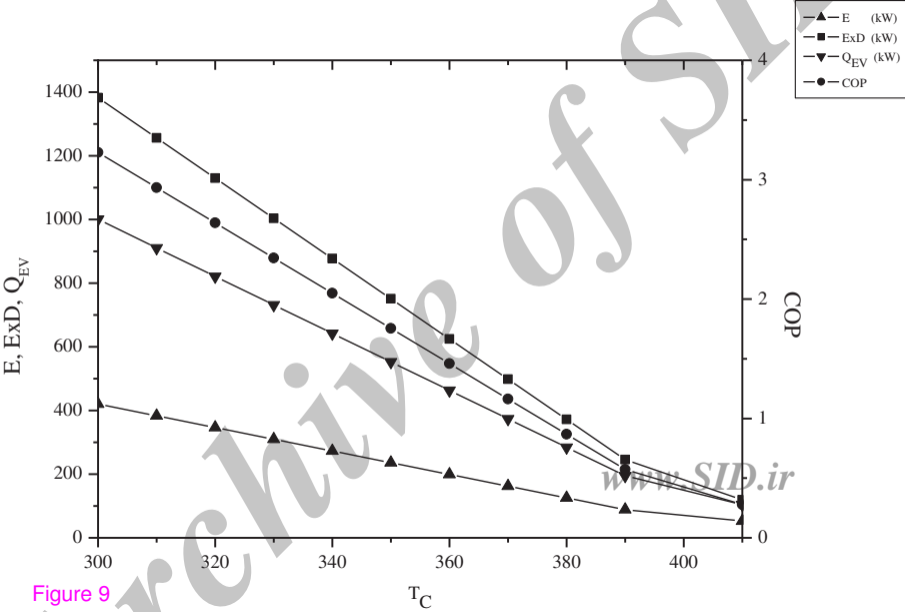


Figure 9

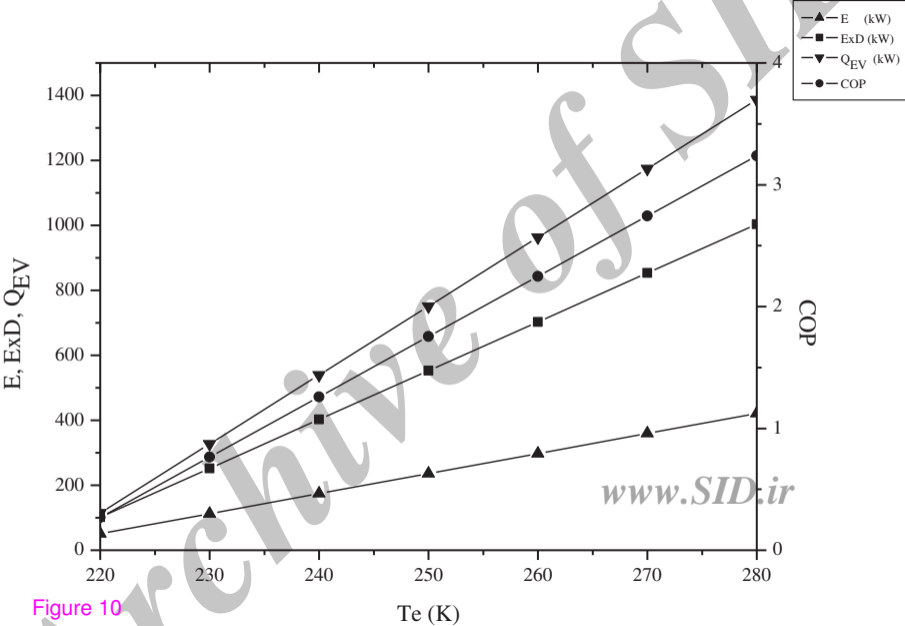


Figure 10