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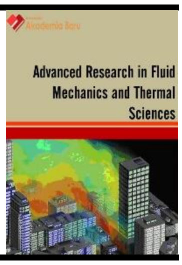

Modelling and Optimization Study of an Absorption Cooling Machine using Lagrange Method to Minimize the Thermal Energy Consumption

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ABSTRACT

In this paper, the development of a mathematical model has been studied in order to optimize the absorption cooling machine by using the Lagrange multiplier method to the thermal model. The objective of this study is to minimize the thermal energy consumption in the generator, in the frame of a cooling capacity imposed in the evaporator. The optimization is based on the objective function taking into account two constraints; the entropy and the physical dimension constraint. The results show that the temperature of the heat source at the input of the generator has a significant effect on the performance of the absorption cooling machine. In order to minimize the thermal power of the generator, It is found that the thermal power in the generator decreases from 10.52 kW to 7.61 kW when the inlet temperature in the generator varies from 120 ° C to 130 ° C.

1. Introduction

Currently, a continuous increase in energy consumption is observed. According to the international energy outlook [1], world energy consumption increases by 28% between 2015 and 2040. For this reason, more attention has been paid on the various applications of absorption cooling machine in buildings. This cooling technology appears to be a promising alternative to the mechanical vapor compression system especially in the field of air conditioning. The current research focuses on the development of absorption cooling systems to improve their energy efficiency. Absorption cooling systems have been considered as excellent alternatives for vapor compression and vapor absorption systems due to some of their distinguishing features [2-5]. These systems use environmentally friendly refrigerant-absorbent mixtures such as $\mathrm{NH_3 - H_2O}$ and $\mathrm{H_2O - LiBr}$, which

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are respectful to the ozone and have very low global warming potential [6]. Absorption cooling systems can be powered by heat generated by industrial processes; solar energy or geothermal energy. These types of machines have great potential to reduce environmental thermal pollution [7]. Many researchers have presented modeling and simulation studies of solar cooling and air-conditioning systems in which several mathematical models were used to describe the characteristics of absorption cooling machines' performance have been developed.

Joudi and Lafta [8] have developed a steady state computer simulation model capable of predicting the performance of an absorption cooling system using water and lithium bromide ($\mathrm{H_2O-LiBr}$) as working fluid and operating in steady-state conditions. Patek and Klomfar [9] presented a set of highly efficient formulations for calculating the thermodynamic properties of $\mathrm{H_2O-LiBr}$ solution at the vapor-liquid equilibrium. The effects of operating temperatures, the effectiveness of the solution heat exchangers and the choice of the working fluid pair on the performance of the system were examined by Karamangil *et al*., [10]. It was found that cycle performance improved with increasing generator and evaporator temperatures, but decreased with increasing condenser and absorber temperature. In the same way, Touaibi *et al*., [11] studied the energy and exergy analysis of the solar water-lithium bromide absorption cooling system. The results show that the distribution of exergy destroyed in the system between components is highly dependent on working temperatures.

In recent years (thermodynamics in finite time, endoreversible thermodynamics, minimization of entropy generation or thermodynamic optimization) have been applied in the study of the performance of absorption refrigerators. The obtained results were different from those of using the classical thermodynamics [12]. Bhardwaj *et al*., [13] presented a finite time thermodynamic optimization of a simple vapour absorption refrigeration system affected by both external and internal irreversibility. External heat reservoirs (heat source/sink) have been considered of finite heat capacity, the optimal bounds for the coefficient of performance COP and working fluid temperatures of the absorption system were determined at the maximum cooling capacity. Moreover, the effects of the cycle parameters on the COP and the cooling capacity of the cycle were studied by Fellah *et al*., [14], where the results obtained presented the importance to the optimal design and performance improvement of an absorption refrigeration cycle. Many researchers studied the optimization of endo-reversible absorption refrigeration machines [15-17], while others studied irreversible machines [18, 19]. Jain *et al*., [20, 21] developed a thermodynamic model for cascaded vapor compression-absorption systems, which consist of a vapor compression system using $\mathrm{R_{22}}$ or $\mathrm{R_{410a}}$ as refrigerant and a single-effect absorption system using $\mathrm{H_2O-LiBr}$ as working pair. Chen *et al*., [22] analyzed an absorption-compression refrigeration system coupled of an absorption refrigeration sub-cycle and a mechanical compression refrigeration sub-cycle. Chen *et al.,* proposed a new thermal compressor model with a key parameter of boost pressure ratio to optimize the absorption-generation process [23]. The ultimate generation pressure and boost pressure ratio are used to represent the potential and operating conditions of the thermal compressor, respectively. The proposed thermal compressor model is presented as an effective method in order to simplifying the optimization of the thermodynamic systems involving an absorption-generation process.

In this study, the optimization of the absorption cooling machine will be studied by using the Lagrange multiplier method. The main objective is to minimize thermal energy consumption by taking into account entropy and dimensional constraints based on the input temperature of the generator.

2. Materials and Methods

2.1 Machine Description

As shown in Figure 1, the main components of the single effect absorption cooling machine are the generator (G), absorber (A), condenser (C), evaporator (E), pump (P), refrigerant expansion valve (V 1) and solution expansion valve (V 2). $\dot{Q}_G$ is the input heat rate from the heat source to the generator, $\dot{Q}_C$ and $\dot{Q}_A$ are the rejection heat rates from condenser and absorber to the heat sinks, respectively, and $\dot{Q}_E$ is the input heat rate from the cooling load to the evaporator. The refrigerant vapor coming from the evaporator is absorbed by liquid solution. This liquid solution is then pumped to the generator at high pressure. The refrigerant is boiled out of the solution by the addition of heat. Subsequently, the refrigerant goes to the condenser and to the evaporator through the refrigerant expansion valve (V 2). Finally, the strong lithium liquid solution returns back to the absorber through the solution expansion valve (V 2) [11].

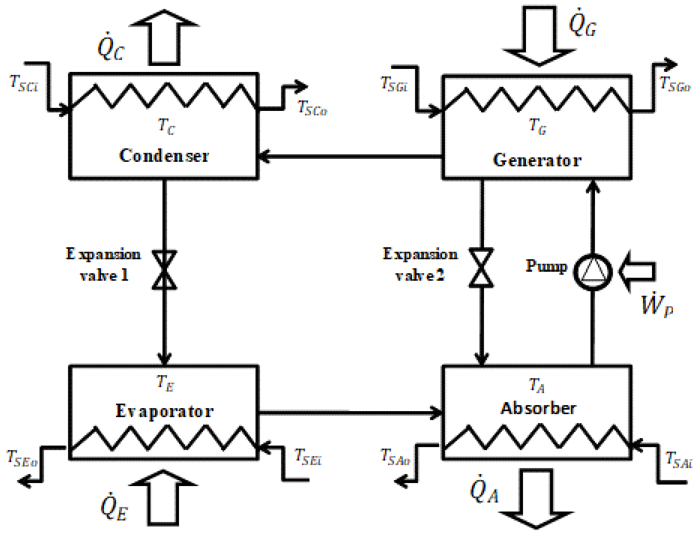

Fig. 1. Scheme of the absorption cooling machine

2.2 Modelling Process

The mathematical model is based on the first and second laws of thermodynamics applied to the entire machine and the heat transfer equations at the sources and sinks.

The calculation process is performed by taking into account the following assumptions:

I. Machine components are assumed to be at steady state and steady flow process.
II. The absorption cooling machine is considered adiabatic.
III. Changes in the potential and the kinetic energy of the components are negligible.

Using the first law of thermodynamic, the energy balance equation of the absorption cooling machine is written as follows [24, 25]:

$$\dot{Q}_G + \dot{Q}_E - \dot{Q}_A - \dot{Q}_C + \dot{W}_P = 0 \tag{1}$$

By using the second law of thermodynamics, the entropy balance of the circulating working fluids can be expressed as follows [24].

$$\frac{\dot{Q}_G}{T_G}+\frac{\dot{Q}_E}{T_E}-\frac{\dot{Q}_A}{T_A}-\frac{\dot{Q}_C}{T_C}+\dot{S}_i=0 \quad (2)$$

When the internal entropy generation $\dot{S}_i$ is assumed constant, the number of transfer units (NTU) method is used to calculate the rate of heat transfer in each heat exchanger of the system as follows [26].

$$\dot{Q}_j=\varepsilon_j\dot{C}_j(T_{Sji}-T_j) \quad (3)$$

Where $\dot{C}_j$ is the heat capacity rate of sources and sinks fluid, ε_j is the effectiveness of each heat exchanger (*i* is the inlet parameter subscript and *j* is the component subscript).

The thermal energy exchanged in each source can be expressed by the (NTU) method with $a_j=\varepsilon_j\dot{C}_j$ as follows.

$$\dot{Q}_G=a_G(T_{SGi}-T_G) \quad (4)$$

$$\dot{Q}_E=a_E(T_{SEi}-T_E) \quad (5)$$

$$\dot{Q}_A=a_A(T_A-T_{SAi}) \quad (6)$$

$$\dot{Q}_C=a_C(T_C-T_{SCi}) \quad (7)$$

where a_G, a_E, a_A and a_C are the intermediate variables of the generator, evaporator, absorber and condenser respectively.

Eq. (4) to Eq. (7) are used to determine the temperatures of the working fluid in the generator, evaporator, absorber and condenser T_G, T_E, T_A and T_C respectively; by taking into consideration the Eq. (2) of the second law of thermodynamics, as below:

$$\frac{\dot{Q}_G}{T_{SGi}-\frac{\dot{Q}_G}{a_G}}+\frac{\dot{Q}_E}{T_{SEi}-\frac{\dot{Q}_E}{a_E}}-\frac{\dot{Q}_A}{T_{SAi}+\frac{\dot{Q}_A}{a_A}}-\frac{\dot{Q}_C}{T_{SCi}+\frac{\dot{Q}_C}{a_C}}+\dot{S}_i=0 \quad (8)$$

2.2.1 Objective function

The absorption cooling machine exchanges heat with four thermal reservoirs (two heat sources, two heat sinks) at temperatures T_{SGi},T_{SEi} ,T_{SAi} and T_{SCi} using the first law of thermodynamics, the objective function of the thermal power consumption takes the following form.

$$\dot{Q}_G=\dot{Q}_A+\dot{Q}_C-C_1 \quad (9)$$

where $C_1=\dot{Q}_E+\dot{W}_P$; $\dot{Q}_E$ is the cooling capacity.

In this case the objective function consists of three intermediate variables ($\dot{Q}_G$,$\dot{Q}_A$,$\dot{Q}_C$) and two parameters ($\dot{Q}_E$,$\dot{W}_P$).

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2.2.2 Constraints

In this model two constraints are presented; the entropy constraint (8) and the size constraint; once Eq. (9) introduced in Eq. (8), the entropy constraint takes the following expression.

$$\frac{\dot{Q}_A+\dot{Q}_C-C_1}{T_{SGi}-\frac{\dot{Q}_A+\dot{Q}_C-C_1}{a_G}}+\frac{\dot{Q}_E}{T_{SEi}-\frac{\dot{Q}_E}{a_E}}-\frac{\dot{Q}_A}{T_{SAi}+\frac{\dot{Q}_A}{a_A}}-\frac{\dot{Q}_C}{T_{SCi}+\frac{\dot{Q}_C}{a_C}}+\dot{S}_i=0 \quad (10)$$

Two intermediate variables ($\dot{Q}_A$,$\dot{Q}_C$) and seven parameters (T_{SGi},T_{SEi},T_{SAi},T_{SCi},$\dot{Q}_E$,$\dot{W}_P$, $\dot{S}_i$) are defined in the model.

The size constraint is related to the a_i variables as follows.

$$a_G+a_E+a_C+a_A-a_T=0 \quad (11)$$

There are four intermediate variables (a_G, a_E,a_A,a_C) and only one parameter (a_T). This constraint is related to the size of all heat exchangers.

The coefficient of performance of the absorption cooling machine is given by the expression as follows.

$$COP=\frac{\dot{Q}_E}{\dot{Q}_G+\dot{W}_P} \quad (12)$$

3. Mathematical Modelling

The mathematical model is based on the objective function defined above by the Eq. (9) and taking into account the two previous constraints (10, 11), using the method of Lagrange multipliers. The Lagrangian function (L) takes the following form.

$$L(\dot{Q}_A,\dot{Q}_C,a_G,a_E,a_A,a_C,\lambda_1,\lambda_2)=\dot{Q}_A+\dot{Q}_C-C_1+\lambda_1\left[\frac{\dot{Q}_A+\dot{Q}_C-C_1}{T_{SGi}-\frac{\dot{Q}_A+\dot{Q}_C-C_1}{a_G}}+\frac{\dot{Q}_E}{T_{SEi}-\frac{\dot{Q}_E}{a_E}}-\frac{\dot{Q}_A}{T_{SAi}+\frac{\dot{Q}_A}{a_A}}-\frac{\dot{Q}_C}{T_{SCi}+\frac{\dot{Q}_C}{a_C}}+\dot{S}_i\right]+\lambda_2(a_G+a_E+a_A+a_C-a_T) \quad (13)$$

The partials derivatives of the Lagrangian function (L) with the two Lagrange multipliers factors λ_1 and λ_2 with the variables ($\dot{Q}_A$, $\dot{Q}_C$, a_G, a_E, a_A, a_C) forms the system of equations describing the behavior of the designed machine. The partials derivatives of the Lagrangian function (L) relative to the thermal heat rate of the absorber and the condenser respectively are given by following expressions:

$$\frac{\partial L}{\partial \dot{Q}_A}=1+\lambda_1\left[\frac{\dot{Q}_A}{a_A\left(T_{SAi}+\frac{\dot{Q}_A}{a_A}\right)^2}-\frac{1}{T_{SAi}+\frac{\dot{Q}_A}{a_A}}+\frac{\dot{Q}_A+\dot{Q}_C-C_1}{a_G\left(T_{SGi}-\frac{\dot{Q}_A+\dot{Q}_C-C_1}{a_G}\right)^2}+\frac{1}{T_{SGi}-\frac{\dot{Q}_A+\dot{Q}_C-C_1}{a_G}}\right] \quad (14)$$

$$\frac{\partial L}{\partial \dot{Q}_C}=1+\lambda_1\left[\frac{\dot{Q}_C}{a_C\left(T_{SCi}+\frac{\dot{Q}_C}{a_C}\right)^2}-\frac{1}{T_{SCi}+\frac{\dot{Q}_C}{a_C}}+\frac{\dot{Q}_A+\dot{Q}_C-C_1}{a_G\left(T_{SGi}-\frac{\dot{Q}_A+\dot{Q}_C-C_1}{a_G}\right)^2}+\frac{1}{T_{SGi}-\frac{\dot{Q}_A+\dot{Q}_C-C_1}{a_G}}\right] \quad (15)$$

The partials derivative of the Lagrangian function (L) to the intermediate variable $a_j = \varepsilon_j \dot{C}_j$ of generator, evaporator, absorber and the condenser are presented as follows.

$$\frac{\partial L}{\partial a_G} = \lambda_2 - \lambda_1 \left[\frac{(\dot{Q}_A + \dot{Q}_C - C_1)^2}{a_G{}^2 \left(T_{SGi} - \frac{\dot{Q}_A + \dot{Q}_C - C_1}{a_G} \right)^2} \right] \tag{16}$$

$$\frac{\partial L}{\partial a_E} = \lambda_2 - \lambda_1 \left[\frac{\dot{Q}_E{}^2}{a_E{}^2 \left(-\frac{\dot{Q}_E}{a_E} + T_{SEi} \right)^2} \right] \tag{17}$$

$$\frac{\partial L}{\partial a_A} = \lambda_2 - \lambda_1 \left[\frac{\dot{Q}_A{}^2}{a_A{}^2 \left(\frac{\dot{Q}_A}{a_A} + T_{SAi} \right)^2} \right] \tag{18}$$

$$\frac{\partial L}{\partial a_C} = \lambda_2 - \lambda_1 \left[\frac{\dot{Q}_C{}^2}{a_C{}^2 \left(\frac{\dot{Q}_C}{a_C} + T_{SCi} \right)^2} \right] \tag{19}$$

The partials derivatives of the Lagrangian function (L) to the Lagrange multipliers factors λ_1 and λ_2 are given as follows.

$$\frac{\partial L}{\partial \lambda_1} = \frac{\dot{Q}_A + \dot{Q}_C - C_1}{T_{SGi} - \frac{\dot{Q}_A + \dot{Q}_C - C_1}{a_G}} + \frac{\dot{Q}_E}{T_{SEi} - \frac{\dot{Q}_E}{a_E}} - \frac{\dot{Q}_A}{T_{SAi} + \frac{\dot{Q}_A}{a_A}} - \frac{\dot{Q}_C}{T_{SCi} + \frac{\dot{Q}_C}{a_C}} + \dot{S}_i \tag{20}$$

$$\frac{\partial L}{\partial \lambda_2} = a_G + a_C + a_A + a_E - a_T \tag{21}$$

This equations system contains eight equations and eight variables corresponding to its optimum conditions. In the following section, the sensitivity of the optimal variable values will be studied by taking into account the objective function $\dot{Q}_G$ for the imposed cooling capacity at the evaporator with respect to various parameters. The aim of this study is to determine which parameters have the largest effect on the optimum operating condition of the absorption machine. One parameter will be varied and keeping all other parameters at their central point values. The parameters are the input temperatures at the entrance of each heat exchanger T_{SGi}, T_{SEi}, T_{SAi} and T_{SCi}, the total size parameter a_T, the generated internal entropy $\dot{S}_i$ and the power pump $\dot{W}_P$. The central point is illustrated in Table 1 using the parameters of absorption cooling machine [9].

Table 1
The values of Reynolds number and velocity

Parameter	Value	Unit
$T_{SG\,i}$	373.15	K
$T_{SE\,i}$	278.15	K
$T_{SA\,i}$	298.15	K
$T_{SC\,i}$	298.15	K
$\dot{Q}_E$	10	kW
$\dot{W}_P$	1.2	kW
$\dot{S}_I$	0.007	$kW.K^{-}$
a_T	11	$kW.K^{-}$

4. Results and Discussion

Results are obtained by solving the equations of the mathematical model developed in this study using the EES software [27] with taking into account the input parameters of the cooling machine. In Table 2 the predicted results are presented.

Table 2
Predicted results through the mathematical model based on Lagrange multipliers method

Parameter	Value	Unit
$\mathbf{T_G}$	368.6	K
$\mathbf{T_E}$	274.7	K
$\mathbf{T_C}$	301.9	K
$\mathbf{T_A}$	301.9	K
$\mathbf{a_G}$	2.294	$kW.K^{-1}$
$\mathbf{a_E}$	2.925	$kW.K^{-1}$
$\mathbf{a_A}$	2.887	$kW.K^{-1}$
$\mathbf{a_C}$	2.894	$kW.K^{-1}$
$\dot{\mathbf{Q}}_\mathbf{G}$	10.52	kW
$\dot{\mathbf{Q}}_\mathbf{A}$	10.85	kW
$\dot{\mathbf{Q}}_\mathbf{C}$	10.87	kW
COP	0.8533	-

The effect of the temperature source (T_{SGi}) at the generator inlet on the thermal power consumption ($\dot{Q}_G$) predicted by the developed mathematical model, as function of the cooling capacity ($\dot{Q}_E$) is shown in Figure 2 to Figure 4. The Results illustrated in these figures show that the thermal power for the three components (generator, absorber and condenser) is influenced by the increase of the cooling capacity ($\dot{Q}_E$) and it decreases by an increasing the temperature of the heat source at the entrance to the generator. From the obtained results, it can be noted that the temperatures at the input of the generator decrease the thermal energy consumption, which improves the performance of the studied machine.

Through the Figure 2, it observed that for the cooling capacity of 25 kW, the thermal energy consumption at the generator reaches a high value of 32 kW for a temperature of 100 °C, when the temperature of the source is at 130 °C, the thermal power consumption decreases to 18.95 kW. The temperature of the source at the input of the generator increases with decreasing of the thermal power of the generator: For a cooling capacity of 10 kW, the thermal power consumed by the generator decreases from 10.52 kW to 7.61 kW when T_{SGi} goes from 100 °C to 130 °C.

These results show that the temperature of the source at the input of generator has a great effect on improving the performance of the absorption refrigerating machine.

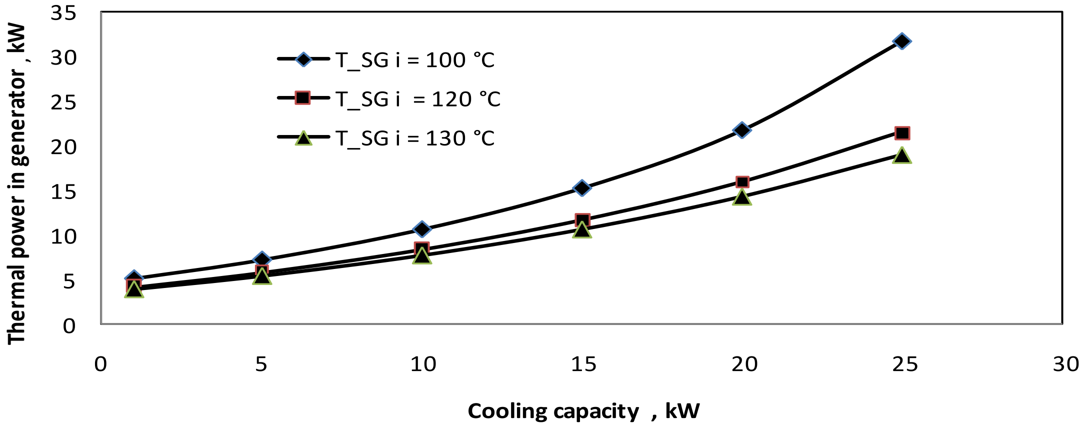

Fig. 2. The variation of the thermal power in generator with cooling capacity for different values of the input temperature source T_{SGi}

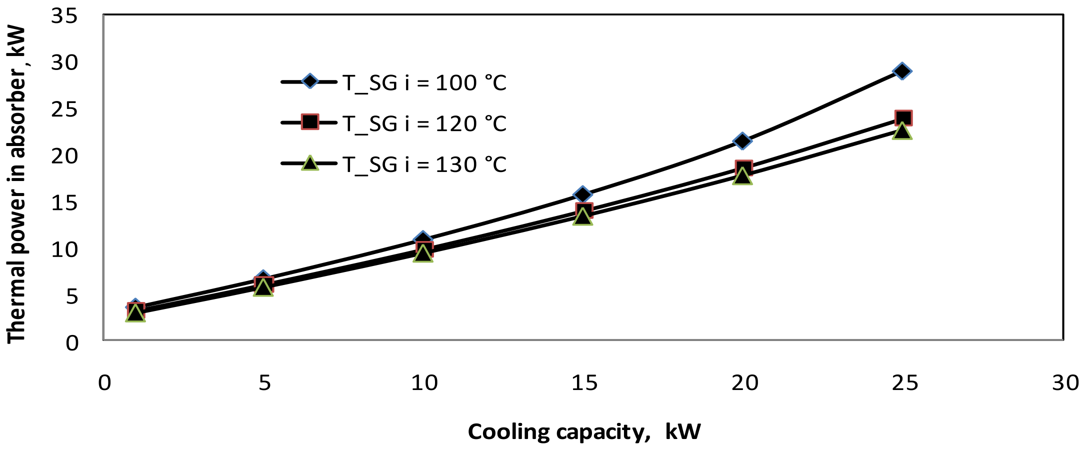

Fig. 3. The variation of the thermal power in absorber with cooling capacity for different values of the input temperature source T_{SGi}

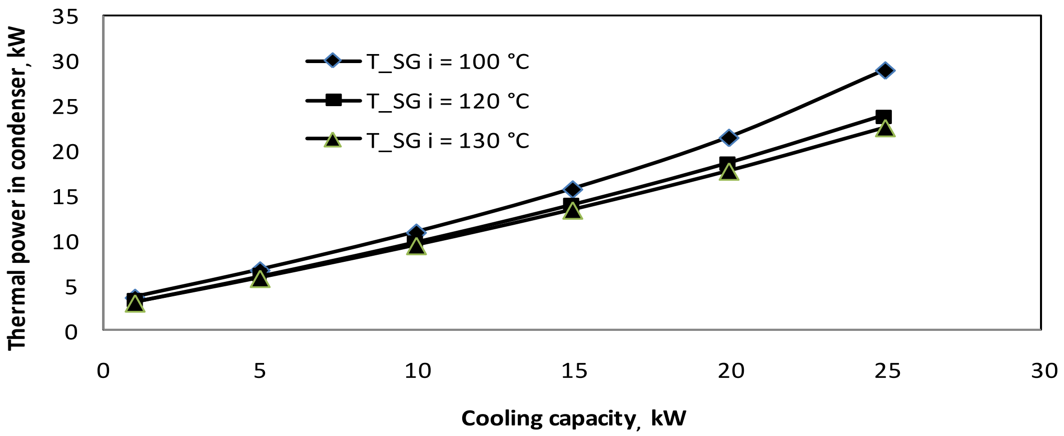

Fig. 4. The variation of the thermal power in condenser with cooling capacity for different values of the input temperature source T_{SGi}

Figure 5 to Figure 8 show the results of the temperature effect of the source at the input of the generator and the variation of the intermediate variable (a) which represents the product of ε times $\dot{C}$ for each component of the absorption cooling machine (generator, evaporator, absorber and condenser) as function of the cooling capacity ($\dot{Q}_E$). The intermediate variable (a) decreases to a minimum value by the increase of the cooling capacity ($\dot{Q}_E$) for the generator, absorber and condenser, but it increases to a maximum value in the case of the evaporator when the cooling capacity increases.

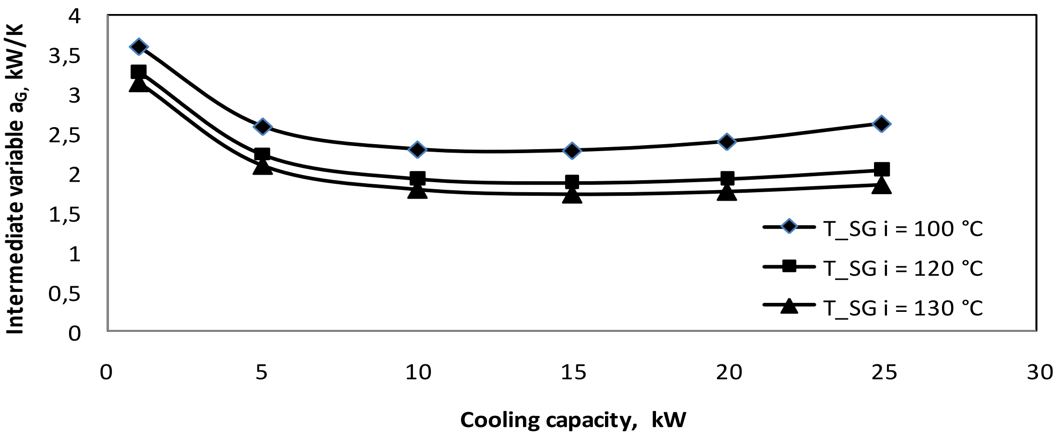

Fig. 5. The variation intermediate variable a_G with cooling capacity for different values of the input temperature source at the generator T_{SGi}

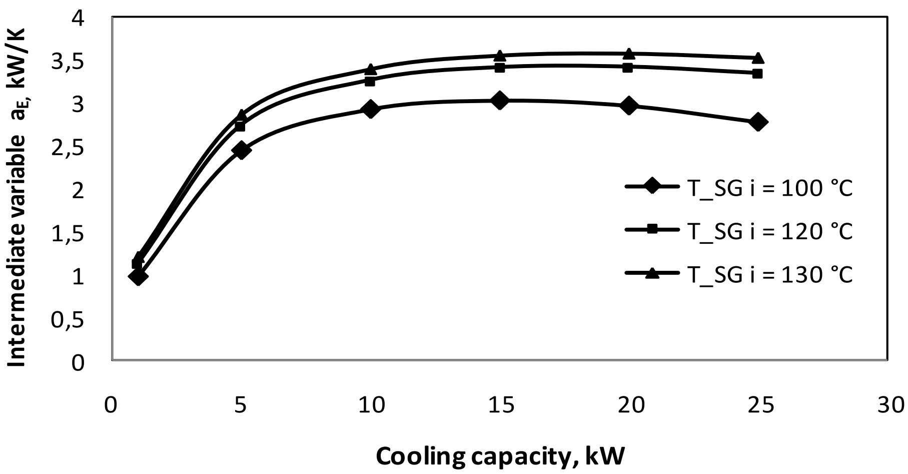

Fig. 6. The variation intermediate variable a_E with cooling capacity for different values of the input temperature source at the generator T_{SGi}

The optimization model developed in this study allowed us to find the optimal point for a cooling capacity of 10 kW (appears in Figure 5) as follows: $a_G = 1{,}782$ kW/K, $a_E = 3.392$ kW/K, $a_A = 2.913$ kW/K and $a_C = 2.913$. The other optimal point appears in Figure 6 at a value of $a_E = 3.574$ kW/K, this value corresponds to a cooling capacity of 20 kW, in which the optimum point is as follows: $Q_E = 20$kW, $a_G = 1{,}756$ kW/K, $a_E = 3.574$ kW/K, $a_A = 2.835$ kW/K and $a_C = 2.835$ kW/K.

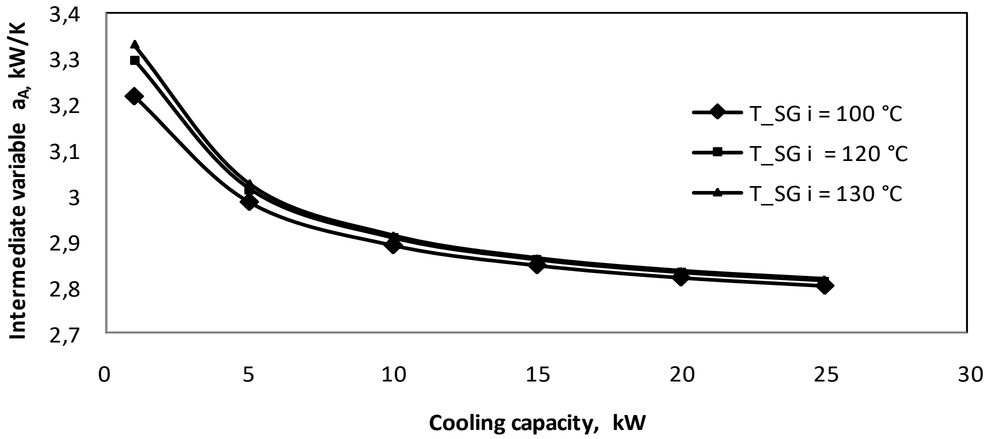

Fig. 7. The variation intermediate variable a_A with cooling capacity for different values of the input temperature source at the generator T_{SGi}

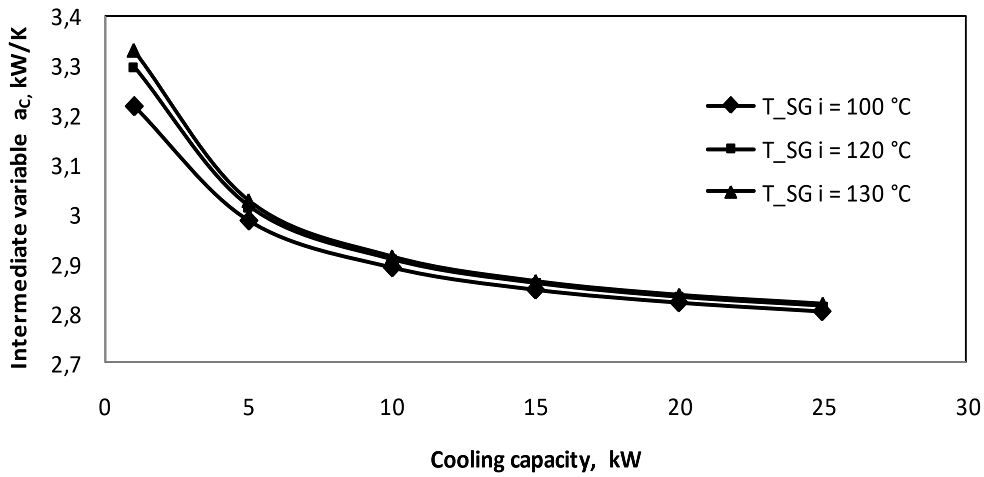

Fig. 8. The variation intermediate variable a_C with cooling capacity for different values of the input temperature source at the generator T_{SGi}

Another important note of this study is that the developed model gave us the same results for both components of the absorber and the condenser as shown in Figure 7 and 8 because they are subjected to the same working conditions.

The results of Figure 9 also show that the temperature of the heat source at the input of the generator is very important to improve the performance of the absorption cooling machine, when the temperature of the heat source is high, the coefficient of performance of the refrigerating machine is better. The results show that the generator thermal power goes from 21.66 to 14.24 kW when the coefficient of performance goes from 0.87 to 1.29 over a temperature range of 120 ° C to 130 ° C.

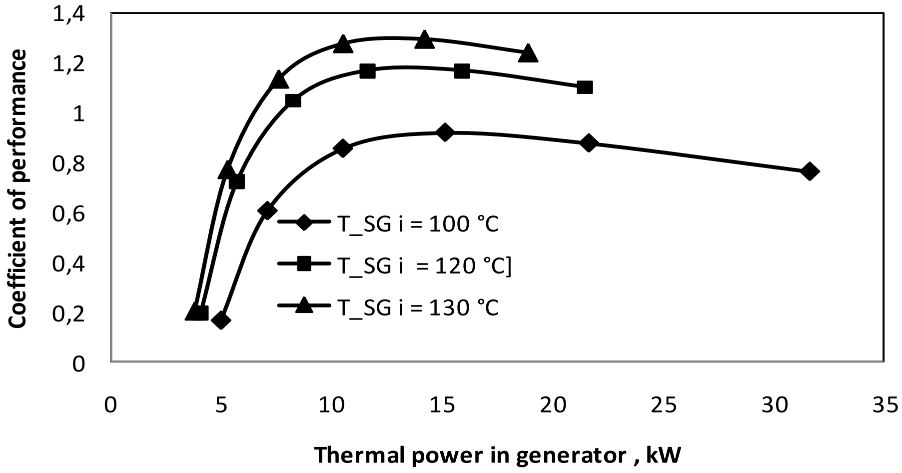

Fig. 9. The variation coefficient of performance with thermal power in generator for different values of the input temperature source at the generator T_{SGi}

5. Conclusion

This article proposes a new thermodynamic model to optimize the absorption cooling machine using the Lagrange multiplier method. Modeling and optimization of the absorption cooling machine were performed by analyzing the sensitivity of the parameters. In this regard, we studied the effect of the operating parameters of the heat source, in particular the temperature of the heat source at the input of the generator. The parameters that have a significant effect on improving the performance of the absorption cooling machine are identified. The simulation was performed to determine the effect of the source temperature on the optimal performance of the cooling machine. The thermal power consumed at the generator was characterized. The optimization model developed in this study allowed us to find the optimal functioning. The results show that the temperature of the heat source at the input of the generator is very important to improve the performance of the absorption cooling machine. Any increase in the temperature of the heat source at the input of the generator causes a decrease in the thermal energy consumption in the generator. The thermal power of the generator goes from 21.66 to 14.24 kW, which will allow the coefficient of performance to go from 0.87 to 1.29 on a temperature of the heat source at the entrance of the beach 120 ° C to 130 ° C. From the obtained results, it was concluded that the generator input temperatures has a significant effect on decreasing the thermal energy consumption, which improves the Performance of the absorption cooling machine.

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