

## A COMPARISON BETWEEN THE NOVEL ABSORPTION REFRIGERATION CYCLE AND THE CONVENTIONAL AMMONIA-WATER ABSORPTION REFRIGERATION CYCLE

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### ABSTRACT

One of the recently proposed systems that are capable of reaching to the low temperature of  $-40\text{ }^{\circ}\text{C}$  is the novel absorption refrigeration cycle with expander and compressor that works as a co-generation or cooling/power system. This cycle provides better COP than other co-generation low temperature cycles when the refrigerating temperature is low enough. By increasing the evaporation temperature to more than  $253\text{ }^{\circ}\text{C}$  the conventional refrigeration cycle shows better COP and exergetic efficiency in comparison to the novel absorption refrigeration cycle. The purpose of the current study was to analyze the novel absorption refrigeration cycle from the viewpoint of exergy or second law in order to evaluate whether the cycle is practically efficient or not. The exergy destruction rate and the exergetic efficiency for each component were calculated and presented in the corresponding graphs to recognize the components with lower efficiency or higher exergy destruction rate. The influences of an expander in COP and exergetic efficiency were identified by ignoring the expander and super heater from the configuration of the novel absorption refrigeration cycle. Furthermore, energy and exergy balances of a conventional ammonia-water absorption refrigeration cycle were formulated to assess the performance of the system in low and high evaporation temperatures. Our results showed that the novel absorption refrigeration cycle obtains a COP of 0.1109 and exergetic efficiency of 0.1156 when the generation temperature is 360 K and evaporation temperature is 233 K. At the same situation, this cycle without using an expander obtains a COP of 0.0525. In addition, the lowest temperature that can be provided by a conventional refrigeration cycle is 252 K but with a much higher COP. Our results approved that it is possible to choose an absorption refrigeration cycle for a special application without wasting time and consuming extra exergy.

### INTRODUCTION

The energy and environment norms regarding the ozone layer depletion and global warming are becoming more and more stringent in almost all countries [8]. In addition, there are

abundant geothermal resources and unlimited solar energy available in the world along with considerable waste heats from power plants. A challenging issue is how to replace the traditional energy sources such as the harmful fossil fuels with these low graded resources. In this scenario, absorption refrigeration systems using the binary mixture of ammonia and water are gaining popularity, because of the use of zero global warming and ozone depletion fluids as refrigerants. Moreover, these systems are heat operated and they need very little electricity. However, these systems show a lower COP and are costlier than the conventional vapour compression systems. Therefore, these systems need to be optimized in order to raise their COP and exergetic efficiency and to produce a lower refrigeration temperature as the conventional absorption refrigeration systems are rarely capable to provide temperatures below than  $-20\text{ }^{\circ}\text{C}$ .

A topic of recent interest is the idea of combined power and cooling absorption refrigeration cycles especially those who utilize these waste heats and renewable energies for their potential in reducing fossil fuel consumption and alleviating environmental problems so that the overall energy conversion efficiency of system improves. In this regard, the GAX absorption refrigeration cycle was introduced in 1911 by Altenkrich and Tenckoff [9] in which slug flow absorber was used to solve the problem of poor wettability of the liquid film in a conventional film flow type absorber [10-12]. This cycle can provide relatively higher COP value than that of the conventional single effect absorption cycle, but it needs a higher grade heat source. The double-effect absorption refrigeration cycle introduced in 1956, can also improve the performance of the absorption refrigeration cycle by generating additional refrigerant [13]; however; this cycle can only be used in case of high temperature heat sources. In 1985, two-stage absorption system was proposed by the community advisory council (CAC) [14]. Compared to conventional single effect absorption refrigeration cycle, this cycle can make refrigeration at lower temperature but its COP is much lower than that of single-effect cycle. In 1998 a new absorption refrigeration cycle with two absorbers was introduced by Mostofizadeh [15].

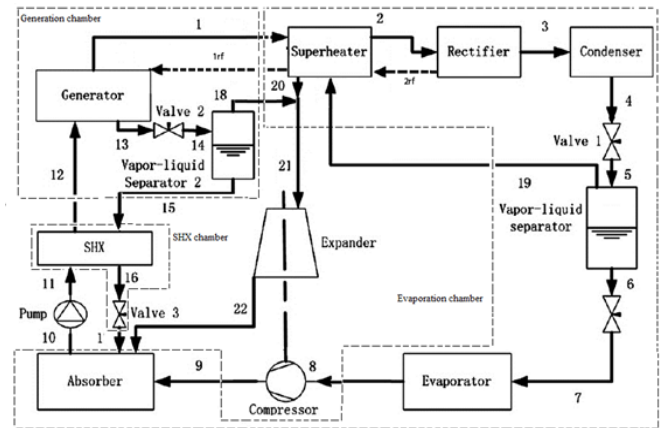
Although this cycle is proposed as a heat pump, it can also be used to make refrigeration. The COP of this cycle is slightly lower than that of two-stage refrigeration cycle but it has a simpler configuration. In 2002 [16] an auto-cascade absorption refrigeration cycle was designed capable of making refrigeration at much lower temperature; however, it must use a mixture as the refrigerant. Compression/absorption cycle is another advanced cycle developed to improve the performance of absorption refrigeration cycles [17-19]. By recovering the sensible heat of the ammonia outlet of the compressor the COP of the cycle becomes higher than that of individual compression or absorption cycles; however, this cycle has to consume additional electric power to drive the compressor. Combined ejector-absorption refrigeration cycle and combined expander-absorption refrigeration cycle were introduced in last centuries [20-21]. The above combined ejector-absorption refrigeration cycles were proposed to improve the performance of absorption refrigeration cycle but most of them were not able to make use of the low grade heat source to make refrigeration at low temperature because of the low efficiency of the ejector [14,22]. Although an expander has a much higher efficiency than that of an ejector, combined expansion-absorption refrigeration cycles were proposed as combined power/cooling cycles. Following these researches, in 2010 a novel absorption refrigeration cycle was proposed by Daliang Hung [3] which provides a low refrigeration temperature (above  $-40\text{ }^{\circ}\text{C}$ ) and also makes use of low graded heat sources (above  $67\text{ }^{\circ}\text{C}$ ). The power needed to drive the compressor is completely provided by the expander. The cycle showed a higher COP than single-effect and double-effect refrigeration cycles. In the current study, after evaluating the system from the exergetic viewpoint, we provide a comparison between the novel absorption refrigeration cycle and a conventional absorption refrigeration cycle. We also identified the temperature regions at which each of them is more efficient. In addition we compared the cycle at which the expander and related power generation components are neglected from the novel absorption refrigeration cycle configuration to evaluate the effect of expander on the first and second low efficiencies for the novel absorption refrigeration cycle.

## NOMENCLATURE

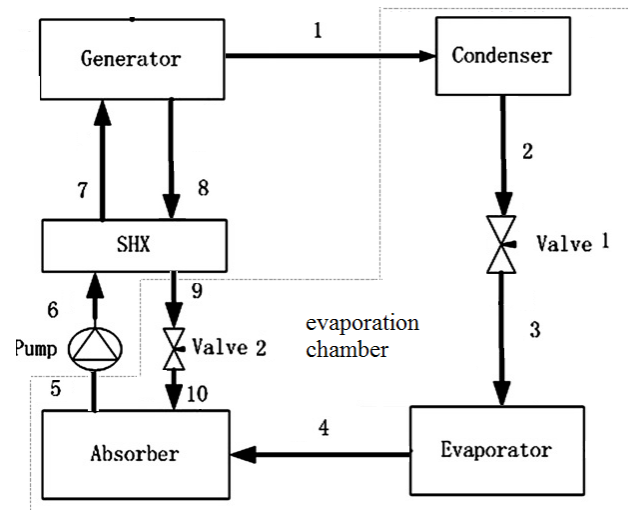
$\dot{E}$	[kJ/s]	exergy transfer rate
$h$	[kJ/kg]	specific enthalpy
$\dot{m}$	[kg/s]	mass flow rate
$M$	[kg/kmol]	molar mass
$P$	[bar]	pressure
$Q$	[kW]	heat transfer rate
$s$	[kJ/kg.k]	specific entropy
$T$	[k]	temperature
$X$		ammonia concentration
$Y$		exergy destruction ratio
$Y^*$		exergy destruction ratio

Subscripts	
$a$	ammonia
$Abs.$	absorber
$E$	exergy
$CH$	chemical component
$CV$	control volume
$Con.$	condenser

$D$	destruction
$e$	exiting stream
$Evap.$	evaporator
$Gen.$	generator
$i$	entering stream
$L$	loss
$PH$	physical component
$Rec.$	rectifier
$rf$	reflux
$s$	surface
$w$	water
$o$	dead state



**Figure 1** Schematic diagram of the novel absorption refrigeration cycle



**Figure 2** Schematic diagram of a conventional absorption refrigeration cycle

## SYSTEM DESCRIPTIONS AND ASSUMPTIONS

Schematics of the novel absorption refrigeration cycle and a conventional absorption refrigeration cycle are shown in Figure 1 and Figure 2 respectively. Compared with the conventional system, the novel system has some extra components which raise the system complexity. The super heater preheats the solution which is entering the expander and does the partial rectification simultaneously. Then the rectifier completes the

rectification process to provide pure ammonia for the refrigeration process. The second valve after the condenser decreases the pressure and the temperature of the refrigerant at the evaporator entrance. After that the compressor increases the exiting saturated vapour pressure to the level that equals the absorption pressure. Two vapour-liquid separators are stated in the configuration in order to provide vapour at intermediate pressure to generate power in the expander. It should be noted that there are four pressure regions for the novel absorption refrigeration cycle: high pressure, intermediate pressure, absorption, and low or evaporation pressure while the conventional system only includes two of them: low and high pressure regions.

The following assumptions are made for this study:

- The systems operate in the steady state condition.
- The pressure drop along the pipe lines is neglected.

Novel system:

- The compression in the compressor and the expansion in the expander are assumed to be adiabatic processes.
- A temperature difference of 10 K is assumed between state 1 and state 20 ( $T_1 - T_{20} = 10$  K).
- A temperature difference of 10 K is assumed between state 16 and state 11 ( $T_{16} - T_{11} = 10$  K).
- The condensation temperature (state 4) and the temperature of the weak solution leaving the absorber (state 10) are assumed to be equal.
- Both the outlet state from the condenser and the outlet state from the evaporator are saturated-vapour.
- The rectification and condensation temperatures are assumed equal.
- The refrigerant vapour from the rectifier has a concentration of 0.999.

Conventional system:

- The refrigerant from the generator has a concentration of 0.90.
- The outlet states of the evaporator and the generator are assumed to be in a saturation condition.
- The outlet reflux from the generator is saturated liquid.
- A partial rectification process is done in the generator.

The input data that are employed for simulating the cycles are represented in the Table 1:

**Table 1** The input data assumed for simulating the cycles

Ambient and dead state temperature ( $T_o$ ) [K]	300
Ambient and dead state pressure ( $P_o$ ) [bar]	1
<b>Novel system</b>	
Condenser, absorber and rectifier temperature [K]	313
Isentropic efficiency of pump and compressor	0.75

Isentropic efficiency of expander	0.80
Intermediate Pressure [bar]	9.09
<b>Conventional system</b>	
Condenser and absorber temperature [K]	300
Heat exchanger efficiency	0.70

## EXERGETIC ANALYSIS

Exergy is defined as the maximum possible reversible work which is obtainable in bringing the state of a system to equilibrium with that of environment [6]. In the absence of magnetic, electrical, nuclear, and surface tension effects and along with considering that the system is at rest relative to the environment, the total exergy of a system is calculated as the summation of physical exergy and chemical exergy components [27]:

$$E = E_{PH} + E_{CH} \quad (1)$$

The physical exergy component is calculated using the following relations:

$$E_{PH} = (h - h_o) - T_o \cdot (s - s_o) \quad (2)$$

$$E_{CH} = \frac{x_n - ch}{M_a} e_a + \frac{1 - x_n - ch}{M_w} e_w \quad (3)$$

The calculation procedures of the chemical exergy of various substances based on standard chemical exergy values of respective species are widely discussed by Bejan et al, [6], Ahrendts [4], and Szargut [5].

Exergy balance at the steady state for a control volume reads:

$$\sum_i \dot{m}_i e_i + \sum_j (1 - \frac{T_o}{T_j}) \cdot \dot{Q}_j = \sum_e \dot{m}_e e_e + \dot{W}_{CV} + \dot{E}_D \quad (4)$$

A detailed exergy analysis includes calculation of exergy destruction rate ( $\dot{E}_D$ ), exergy loss ( $\dot{E}_L$ ), exergetic efficiency ( $\epsilon$ ), and two exergy destruction ratios in each component of the system ( $Y, Y^*$ ) along with the overall system and second law efficiency for the cycle ( $\eta_2$ ).

Mathematically these are expressed described as follows:

$$\dot{E}_D = \dot{E}_F - \dot{E}_P - \dot{E}_L \quad (5)$$

$$\epsilon = \frac{\dot{E}_P}{\dot{E}_F} = 1 - \left[ \frac{\dot{E}_D + \dot{E}_L}{\dot{E}_F} \right] \quad (6)$$

$$Y_D = \frac{\dot{E}_D}{\dot{E}_F} \quad (7)$$

$$Y_D^* = \frac{\dot{E}_D}{\dot{E}_{D, total}} \quad (8)$$

$$\eta_{II} = \frac{\dot{Q}_{evap} \cdot \left[1 - \frac{T_o}{T_{s,evap}}\right]}{\dot{Q}_{gen} \cdot \left[1 - \frac{T_o}{T_{s,gen}}\right] + \dot{W}_{pump}} \quad (9)$$

The COP is also expressed in the form of the following equation:

$$COP = \frac{\dot{Q}_{evap}}{(\dot{Q}_{gen} + \dot{W}_{pump})} \quad (10)$$

The exergy balance equations for the components of the novel absorption refrigeration cycle are developed in the following forms [1]:

Pump:

$$\dot{E}_D = \dot{W}_{pump} - \dot{E}_{t11} + \dot{E}_{t10} \quad (11)$$

Compressor:

$$\dot{E}_D = \dot{W}_{com} - \dot{E}_{t19} + \dot{E}_{t8} \quad (12)$$

Expander:

$$\dot{E}_D = \dot{E}_{t21} - \dot{E}_{t22} - \dot{W}_{exp} \quad (13)$$

Generation chamber:

$$\begin{aligned} \dot{E}_D = & \dot{Q}_{gen} \cdot \left[1 - \frac{T_o}{T_{s,gen}}\right] - \dot{E}_{t15} - \dot{E}_{t11} + \dot{E}_{t12} + \\ & \dot{E}_{t1f} - \dot{Q}_{rec} \cdot \left[1 - \frac{T_o}{T_{s,rec}}\right] - \dot{Q}_{con} \cdot \left[1 - \frac{T_o}{T_{s,con}}\right] \\ & - \dot{Q}_{abs} \cdot \left[1 - \frac{T_o}{T_{s,abs}}\right] \end{aligned} \quad (14)$$

Evaporation chamber:

$$\begin{aligned} \dot{E}_D = & \dot{E}_{t1} + \dot{E}_{t9} + \dot{E}_{t17} + \dot{E}_{t22} - \dot{E}_{t8} - \dot{E}_{t20} + \\ & \dot{E}_{t10} - \dot{E}_{t1f} - \dot{Q}_{evap} \cdot \left[1 - \frac{T_o}{T_{s,evap}}\right] \end{aligned} \quad (15)$$

SHX:

$$\dot{E}_D = \dot{E}_{t15} - \dot{E}_{t17} - \dot{E}_{t12} + \dot{E}_{t11} \quad (16)$$

Exergy balance equations for components of the conventional absorption refrigeration cycle are inferred as follows:

Evaporator:

$$\begin{aligned} \dot{E}_D = & \dot{E}_1 + \dot{E}_9 - \dot{E}_5 - \dot{Q}_{evap} \cdot \left(1 - \frac{T_o}{T_{s,evap}}\right) \\ & - \dot{Q}_{con} \cdot \left(1 - \frac{T_o}{T_{s,con}}\right) - \dot{Q}_{abs} \cdot \left(1 - \frac{T_o}{T_{s,abs}}\right) \end{aligned} \quad (17)$$

Generator:

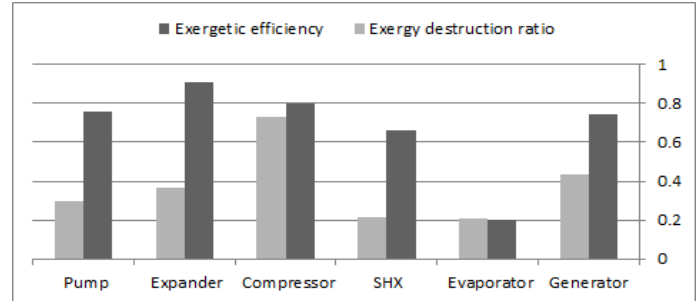
$$\dot{E}_D = \dot{Q}_{gen} \cdot \left(1 - \frac{T_o}{T_{s,gen}}\right) + \dot{E}_7 - \dot{E}_8 - \dot{E}_1 \quad (18)$$

Pump:

$$\dot{E}_D = \dot{W}_{pump} + \dot{E}_5 - \dot{E}_6 \quad (19)$$

SHX:

$$\dot{E}_D = \dot{E}_8 - \dot{E}_9 + \dot{E}_6 - \dot{E}_7 \quad (20)$$



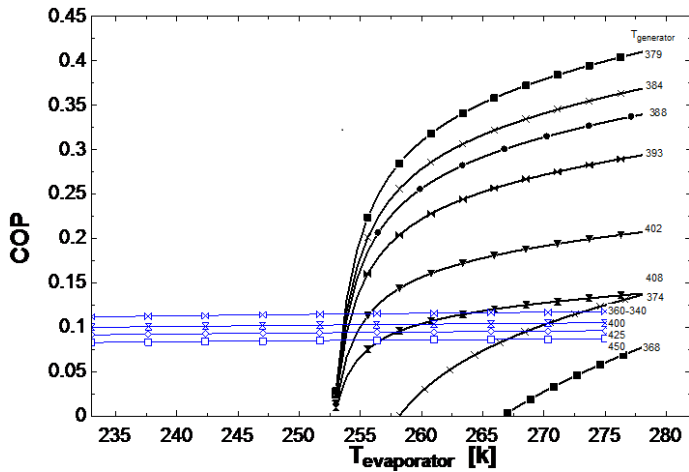
**Figure 3** Exergetic efficiency ( $\epsilon$ ) and exergy destruction ratio ( $\gamma$ ) for components of the novel absorption refrigeration cycle.

## TRENDS AND RESULTS

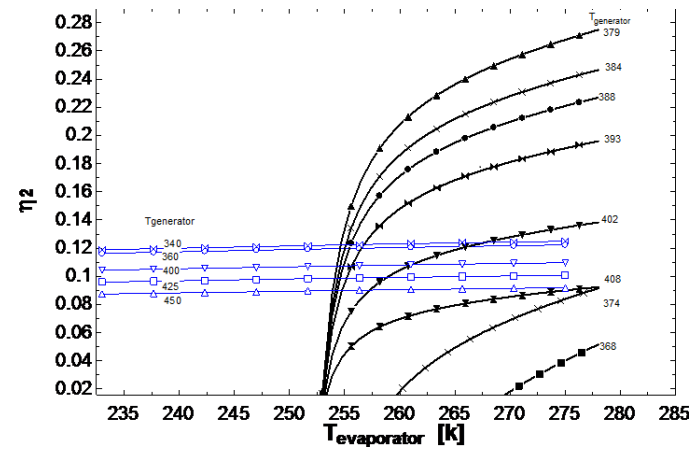
Figure 3 demonstrates exergetic efficiencies and exergy destruction rates (i.e. the component exergy destruction rate per total exergy destruction rate) for each component of the novel absorption refrigeration cycle.

According to the Figure 3, the evaporation chamber has the least exergetic efficiency among the rest of components and this parameter reaches its maximum value for expander.

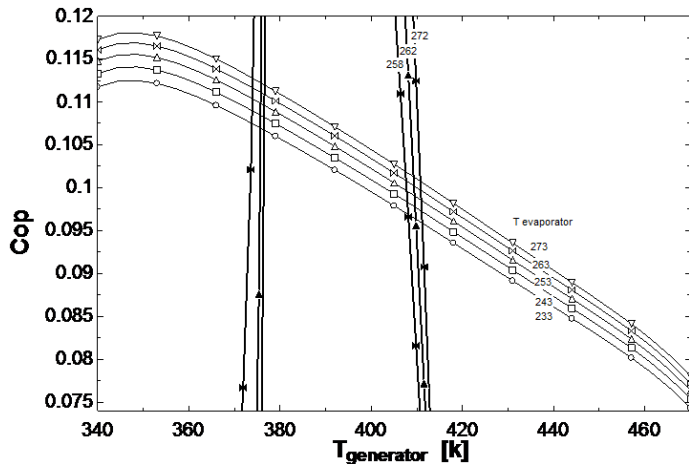
As Figure 4 and Figure 5 demonstrate, a conventional absorption refrigeration cycle could not provide refrigeration temperatures below than 252 K. For the evaporation temperatures of 233 K, the novel absorption refrigeration cycle showed a COP of 0.1109 and an exergetic efficiency of 0.1156 for generation temperature of 360 K. Schematics also demonstrate that for the evaporation temperatures more than 252 K the conventional system operated with a higher COP and  $\eta_2$  until the generation temperature was less than 402 K.



**Figure 4** The effect of evaporation and generation temperatures on COP versus evaporation temperature (thick lines belong to conventional cycle and thin lines belong to the novel system)

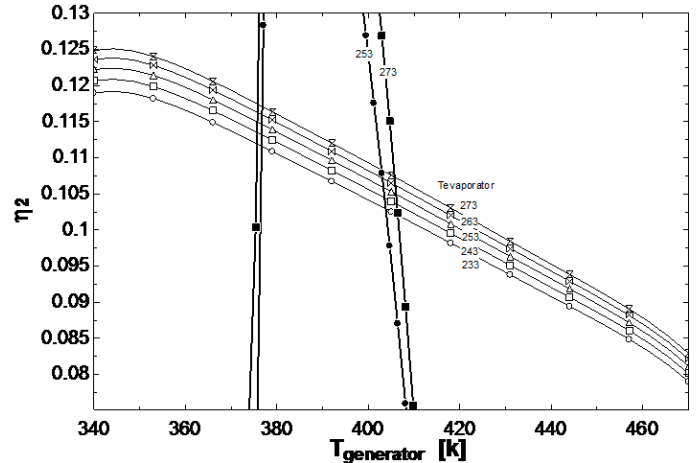


**Figure 5** The effect of evaporation temperature on exergetic efficiency with regard to evaporation temperature (thick lines belong to conventional system and thin lines belong to the novel system profile)

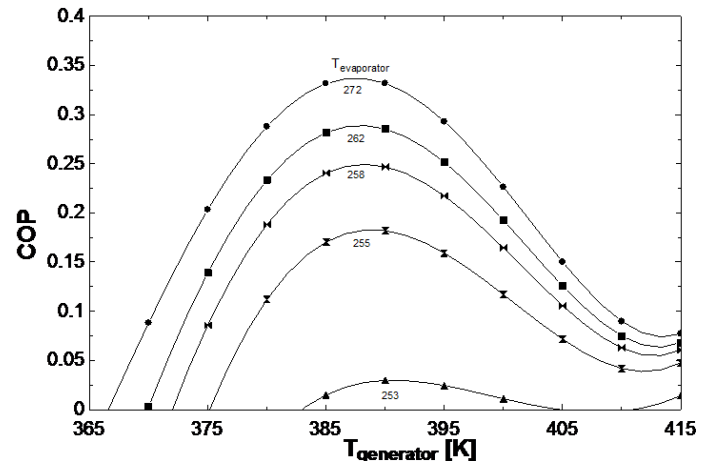


**Figure 6** The influence of generation temperature on COP with regard to evaporation temperature

Figure 6 and Figure 7 provide information about various values for COP and second law efficiencies at different generation temperatures with respect to evaporation temperature. The thin parabolic lines show the coefficient of performance and exergetic efficiency for the novel absorption refrigeration cycle. It can be observed from the figures that the COP values experienced a pick at approximately 350 K for generation temperature and after this point by increasing the temperature for the generator COP and  $\eta_2$  decreased in value simultaneously. Furthermore, figures illustrate that when the evaporator operates in a higher temperature, the COP and exergetic efficiency rise accordingly.



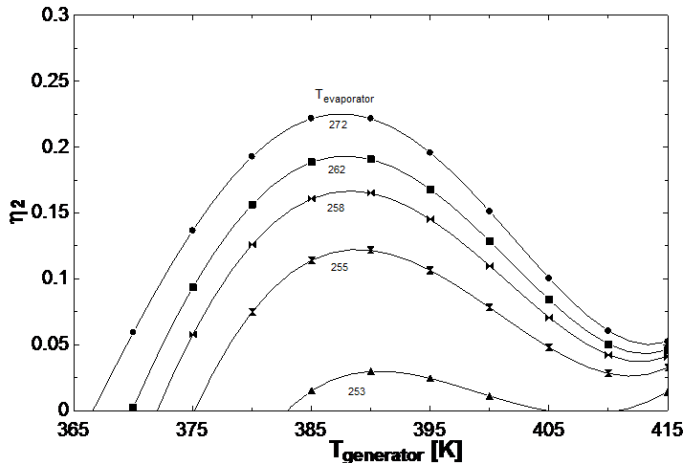
**Figure 7** The influence of generation temperature on exergetic efficiency with respect to evaporation temperature



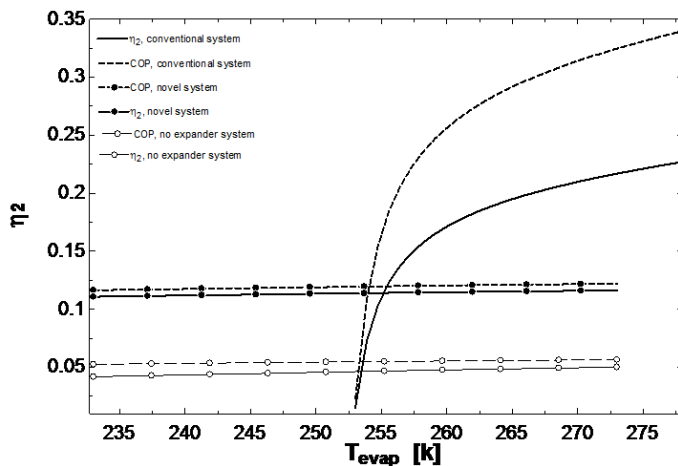
**Figure 8** The influence of generation temperature on COP for the conventional absorption refrigeration cycle with regard to evaporation temperature

Moreover, the thicker lines demonstrate parabolic profiles of COP and  $\eta_2$  for the conventional absorption refrigeration cycle. The figure demonstrates that the novel system operated at a far wider temperature range compared with the conventional cycle for generator and evaporator. In other words, the conventional system operated only between the temperatures approximately 366 and 410 K for the generator and 252 and 278 K for the evaporator. Nevertheless, at this small temperature range the conventional system showed a

higher COP and exergetic efficiency relatively. The frames of the Figure 4 and 5 do not include the whole parabolas because the COP and exergetic efficiency values of the conventional system were much more spread than those of the novel system; therefore, the Figure 6 and Figure 7 are presented to focus on the conventional absorption refrigeration cycle performance singularly.



**Figure 9** The influence of generation temperature on exergetic efficiency for the conventional absorption refrigeration cycle with respect to evaporation temperature



**Figure 10** The comparison of the novel and the conventional absorption refrigeration cycle with the novel system without expander

Figure 8 and Figure 9 illustrate that for the conventional system by enhancing the temperature for generator between the related values, at which the quantities for COP and  $\eta_2$  were sensible, at the temperature of nearly 388 K both COP and  $\eta_2$  reached their maximum values and after that as the temperature rose more the both parameters went down. The figures also confirm that an increase in the refrigeration temperature caused an improvement in system performance as the Figure 4 and 5 brings to light.

Figure 10 represents that if the expander, the super heater, and the vapour-liquid separators are eliminated from the

configuration of the novel system, an external energy resource has to be used to supply the power needed to drive the compressor. Thus, as it is obvious in the Figure 10 the COP and exergetic efficiency for the conventional system drop to less than half of these values for the novel system. Furthermore, our findings show that the performance of a simple absorption refrigeration system over almost all of its operating temperature range is much higher than the other systems.

## CONCLUSION

Our results indicated that the novel absorption refrigeration cycle has a higher COP and exergetic efficiency at the temperature range of value one-value two compared with the conventional absorption refrigeration cycle but not for all the temperature range. In other words, there is a region for both evaporation and generation temperatures that the conventional system is far more efficient than the novel absorption refrigeration cycle. Indeed, the purpose of the present was to identify these regions and make a comparison between the performances of two systems at each situation. These findings suggest that if the  $T_{gen}$  is more than 366 K and less than 415 K ( $366\text{ K} < T_{gen} < 415\text{ K}$ ) and if the  $T_{evap}$  is more than 252 K and less than 278 K ( $252\text{ K} < T_{evap} < 278\text{ K}$ ), the conventional absorption refrigeration cycle is the better choice; however, if  $T_{gen}$  is less than 366 K ( $T_{gen} < 366\text{ K}$ ) or  $T_{evap}$  is less than 252 K ( $T_{evap} < 252$ ), the novel absorption refrigeration cycle is the better choice.

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