Karaelmas Science and Engineering Journal
Journal home page: http://fbd.beun.edu.tr
DOI: 10.7212%2Fzkufbd.v8i1.999

Research Article
Received / Geliş tarihi : 18.05.2017
Accepted / Kabul tarihi : 08.08.2017

Thermodynamic Analysis of Vapour Compression- Absorption Two Stage Refrigeration Cycle

Buharlı Sıkıştırmalı-Absorbsiyonlu Çift Kademeli Soğutma Çevriminin Termodinamik Analizi

Canan Cimsit*

Golcuk Vocational School of Higher Education, Kocaeli University, Golcuk, Kocaeli, Turkey

Abstract

At low temperature applications and large cooling capacities, ammonium refrigerant used widely and these type refrigeration applications have electrical energy consuming by using vapour compression cycle. This causes increasing cooling cost, consuming more fossil fuel and to create environment problem. The vapour compression-absorption two stage refrigeration cycle is alternative to solve this overcoming problems. In this study, a thermodynamic analysis of the vapour compression-absorption two stage refrigeration (novel cycle) was performed. While NH3-H2O was used as fluid pair in the absorption section, NH3 was used in the vapour compression section. It was presented that electrical energy consumption in the novel cycle is 60% lower than classical one stage vapour compression refrigeration cycle and 24% lower than single effect vapour compression-absorption cascade refrigeration cycle. The thermodynamics analysis was performed for different condenser and generator temperatures. The result shows that COP of the novel cycle increases by increasing the generator temperature, while it decreases by increasing the condenser temperatures. The exergetic efficiency of the system decreases on increasing the generator and condenser temperatures. The novel cycle provide by using inexpensive alternative heat energy sources like solar energy, geothermal energy and waste heat at low temperatures.

Keywords: Absorption, Refrigeration, Thermodynamic analysis, Two stage refrigeration

Öz

Düşük sıcaklık uygulamalarında ve büyük soğutma kapasitelerinde, yaygın olarak kullanılan amonyak soğutucu ve bu tür soğutma uygulamaları, buhar sıkıştırma çevrimi kullanılarak elektrik enerjisi tüketmektedir. Bu durum soğutma maliyetinin artmasına, daha fazla fosil yakıt tüketmesine ve çevre sorunlarının oluşmasına neden olmaktadır. Buharlı sıkıştırmalı-absorbsiyonlu çift kademeli soğutma çevrimi, bu problemleri çözmek için bir alternatiftir. Bu çalışmada buharlı sıkıştırmalı-absorbsiyonlu çift kademeli soğutma çevriminin bir termodinamik analizi yapılmıştır. Akışkan olarak absorbsiyonlu kısımda NH3-H2O akışkan çiftinin ve buhar sıkıştırmalı kısmında da NH3 soğutucu akışkanının kullanıldığı kabul edilmiştir. Analiz sonuçlarına göre buharlı sıkıştırmalı-absorbsiyonlu çift kademeli soğutma çevriminin, tek kademeli buhar sıkıştırmalı soğutma çevrimine göre de % 60 ve tek kademeli buhar sıkıştırmalı-absorbsiyonlu kaskad soğutma çevrimine göre de % 24 daha az elektrik enerjisine ihtiyaç olduğu görülmüştür. Termodinamik analiz, farklı kondenser ve kaynatıcı sıcaklıklarında gerçekleştirildi. Analiz sonuçları, yeni soğutma çevriminin kondenser sıcaklığının artmasıyla çevrimin soğutma tesir katsayısının azaldığını buna karşın artan kaynatıcı sıcaklıklarında ise artmakta olduğunu göstermiştir. Kaynatıcı ve kondenser sıcaklıklarının artmasıyla sistemin ekserji verimi azalmaktadır. Bu yeni çevrim, güneş enerjisi, jeotermal enerji ve atık ısı gibi ucuz alternatif enerji kaynaklarını kullanarak düşük sıcaklıklarda soğutma yapmayı sağlamaktadır.

Anahtar Kelimeler: Absorbsiyonlu, Soğutma, Termodinamik analiz, Çift kademeli soğutma

1. Introduction

Absorption cycles have have zero global warming potential and do not contribute to stratospheric ozone layer depletion. Absorption systems use different energy sources such as fossil fuels, renewable energies and waste heat from other thermal systems.

There are many studies on the absorption refrigeration systems in the literature. Adewusi ve Zubair (2004) made thermodynamic analysis of absorption refrigeration cycle with NH_3-H_2O. They calculated the entropy generation of each component and the total entropy generation of all the system. Aman et al. (2014) discussed and analyzed the

*Corresponding author: ccimsit@kocaeli.edu.tr

Canan Cimsit orcid.org/0000-0003-1569-3180

potential of a solar driven ammonia-water absorption chiller for residential air conditioning application. They developed a thermodynamic model based on a 10 kW air cooled ammonia-water absorption chiller driven by solar thermal energy. Both energy and exergy analyses have been conducted to evaluate the performance of this residential scale cooling system. The analyses uncovered that the absorber is where the most exergy loss occurs (63%) followed by the generator (13%) and the condenser (11%). Furthermore, the exergy loss of the condenser and absorber greatly increase with temperature, the generator less so, and the exergy loss in the evaporator is the least sensitive to increasing temperature. Sencan et al. (2005) carried out an exergy analysis of a single-effect lithium bromide-water absorption system for cooling and heating applications. They presented that the coefficient of performance (COP) of the system increase slightly when increasing the heat source temperature and the exergetic efficiency of the system decreases for both cooling and heating. Le Lostec et al. (2012) made the performance of an ammonia-water absorption chiller. This single-stage 10 kW absorption chiller is cooled with a water-ethylene glycol solution. The required heat source is hot water between 75°C and 85°C. Different operating conditions can be imposed by varying temperatures and flow rates of secondary circuits and the flow of the rich solution. This equipment, designed for solar air conditioning applications, was tested under various operating conditions to assess its performance. This study shows that the performance of the absorption chiller decreases significantly with the evaporator temperature. Sarbu and Sebarchievici (2013) presented a detailed review of different solar refrigeration and cooling methods. There are presented theoretical basis and practical applications for cooling systems within various working fluids assisted by solar energy and their recent advances. Thermally powered refrigeration technologies are classified into two categories: sorption technology (open systems or closed systems) and thermo-mechanical technology (ejector system). Solid and liquid desiccant cycles represent the open system. The liquid desiccant system has a higher thermal coefficient of performance (COP) than the solid desiccant system. Bouaziz and Lounissi (2015) present an energy and exergy analysis of a novel configuration of absorption cooling system operating at low enthalpy sources. The double stage cycle is operating with water-ammonia. In this investigation, modeling and simulation of the proposed configuration is attempted. Also, a thermodynamic model based on the energy and exergy balances is developed. They compared the obtained numerical results obtained with those corresponding to the conventional machine. Mohamed and Brahim (2016) made a modeling in dynamic state of the refrigeration phase of a solar absorption machine using NH_3-H_2O as a fluid of work. As a matter of fact, a simulation program is used to calculate the solution densities and the dynamic systemstorage in an absorption cycle phase. In times of discharge, the evaporator and the absorber are the only devices in the cycle to operate either in energy or in the upgrading refrigeration. Such a study allows us to select the cooling phase with three storage tanks.

There are a lot of studies on the absorption-vapour compression combined and absorption-vapour compression cascade systems that are formed by combining absorption and vapour compression cycles in the literature. Ayala et al. (1997) made analysis of absorption-vapour compression combined cycle with NH_3-$LiNO_3$. They presented that the performance of the absorption-vapour compression combined cycle is higher than of the vapor-compression or absorption refrigeration cycle. Kairouani and Nehdi (2006) made analysis of compression-absorption refrigeration system assisted by geothermal energy. They used NH_3-H_2O fluid pair is used at the absorption section of the compression-absorption cascade refrigeration cycle, and three different fluids (R717, R22 and R134a) in the vapour compression section. They compared compression-absorption refrigeration system with the conventional cycles for the same operating conditions. They determined that the coefficient of performance (*COP*) can be improved by 37-54% than the conventional cycles for the same operating conditions. Cimsit and Ozturk (2012) carried out theoretically by using LiBr-H_2O fluid couple in the absorption section of cascade refrigeration systems as an alternative to fluid NH_3-H_2O and using different refrigerants (R-134a, R410A and NH_3) in the vapour compression section. Also, in a sample application in which cascade refrigeration systems have been compared with mechanical vapour compression refrigeration systems, it is found that cascade systems are required less electrical energy ranged between 48% and 52% compared with the other system to obtain the same amount refrigeration in the same conditions. Cimsit et al. (2015) carried out the thermoeconomic analysis of compression-absorption cascade refrigeration cycles as the first time in the literature, and the thermoeconomic optimization of the LiBr/H_2O-R134a compression-absorption cascade refrigeration cycle has been performed in this study. The analysis has included detailed exergy analysis of all cascade cycle and components.

It is well known that solar energy is the source of life on earth. This energy affects living organisms, such as plants, in addition to the planning and design of building heating and cooling systems. Turkey has high solar energy potential because of its location in the northern hemisphere with latitudes 36–42^{0}N and longitudes 26–4^{0}E (Sayla et al 2002). Turkey has very rich geothermal energy resources and It is ranking fifth in the world after China, Japan, USA and Iceland in geothermal heat and thermal spring applications (Kilic and Kilic 2013). In this context, the thermodynamic analysis of the vapour compression-alternative energy (solar, geothermal and waste heat) aided absorption two stage refrigeration cycle (VCATSRC) was carried out in this study. The novel cycle has been compared with alternative cycles (single effect vapour compression-absorption cascade cycle (SEVCACC) and one stage vapour compression refrigeration (VCR) cycle) for the same operating conditions. Also, the thermodynamic analysis was performed for different condenser and generator temperatures of the novel cycle.

2. Material and Methods

2.1. The Single Effect Vapour Compression-Absorption Cascade Cycle

The single effect vapour compression-absorption cascade cycle is shown in Figure 1. NH_3-H_2O is used in the absorption section, and NH_3 is used in the vapour compression section of this cycle. In this cycle the strong solution of NH_3-H_2O leaves the absorber, and it is circulated through the heat exchanger by a pump. Then, the high-pressure warm mixture enters the generator, where the heat is added to drive away the ammonia from the solution. The weak solution leaves the generator, and its pressure is reduced to absorption pressure by using an expansion valve (Exv-2). Then, it enters the absorber. In order to increase temperature in stream (6) and decrease the temperature in stream (9), a counter flow heat exchanger is located between strong and weak solutions. After the refrigerant condenses in the condenser 2 and leaves it as a saturated liquid, then it passes through an expansion valve (Exv-1). At the vapour

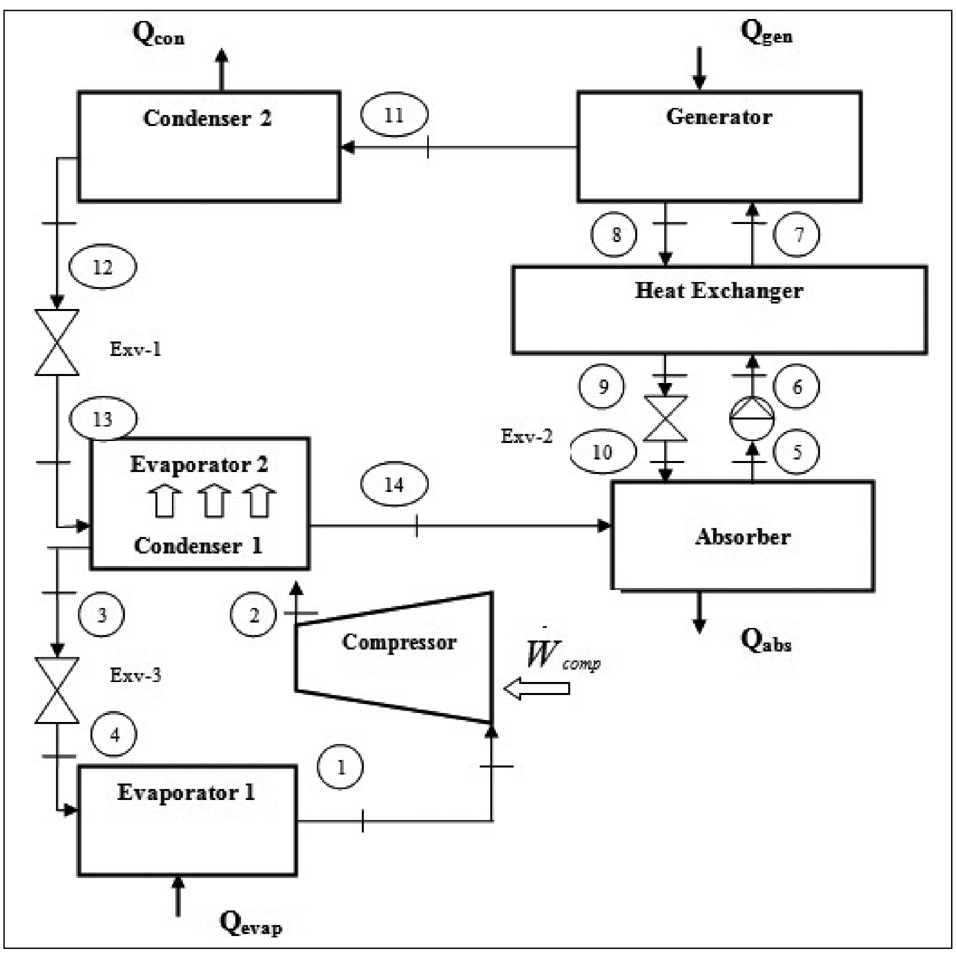

Figure 1. Schematic illustration of the single effect vapour compression-absorption cascade refrigeration cycle.

compression section, the fluid is compressed to the high pressure in the compressor, and then enters to the evaporator 2. After condensing, the pressure of refrigerant is reduced in the expansion valve (Exv-3), and then enters into evaporator

2.2 The Vapour Compression-Absorption Two Stage Refrigeration Cycle (Novel Cycle)

The vapour compression-absorption two stage refrigeration cycle is shown in Figure 2. The vapour compression and absorption refrigeration cycles have been combined as two stage refrigeration cycle that is first stage as vapour compression cycle at low temperature level and second stage as absorption cycle at high temperature level. NH_3-H_2O is used in the absorption section, and NH_3 is used in the vapour compression section of this cycle. In this cycle the strong solution of NH_3-H_2O leaves the absorber, and it is circulated through the heat exchanger by a pump. Then, the high-pressure warm mixture enters the generator. In the generator, the refrigerant is separated from the absorbent by the heat provided. The weak solution leaves the generator, and its pressure is reduced to absorption pressure by using an expansion valve (Exv-2). Then, it enters the absorber. In order to increase temperature in stream (6) and decrease the temperature in stream (9), a counter flow heat exchanger is located between strong and weak solutions. After the refrigerant condenses in the condenser and leaves it as a saturated liquid, then it passes through an expansion valve (Exv-1). At the vapour compression section, the fluid is compressed to the high pressure in the compressor, and then enters to intercooler. After condensing, the pressure of refrigerant is reduced in the expansion valve (Exv-3), and then enters into evaporator.

2.3. Thermodynamic Model of the Novel Cycle

Following basic assumptions have been made for the thermodynamic analysis of the cycle:

- The analyses are performed under steady conditions.
- Refrigerants are in saturated states (at condenser and evaporator).
- Pressure losses in the system are neglected.
- Isentropic efficiency of the compressor is taken as 0.8.
- The refrigerant vapour concentration at the generator exit is 100% ammonia.
- Reference temperature and pressure are taken as 298.15 K and 101.325 kPa, respectively.

In this cycle, working pressure of the intercooler should be between condenser and evaporator presures:

$$P_{\text{int}} = \sqrt{P_{evap} P_{con}} + 35 \tag{1}$$

For steady state conditions the mass and energy balance equations for each system component can be written as follows:

$$\sum \dot{m}_i = \sum \dot{m}_e = 0 \tag{2}$$

$$\sum \dot{m}_i x_i = \sum \dot{m}_e x_e = 0 \tag{3}$$

$$\sum \dot{m}_i h_i = \sum \dot{m}_e h_e - \sum \dot{Q} - \dot{W} = 0 \tag{4}$$

$$\dot{Q}_{gen} = \dot{m}_{11}.h_{11} + \dot{m}_s.h_s - \dot{m}_7.h_7 \tag{5}$$

$$\dot{Q}_{abs} = \dot{m}_{10}.h_{10} + \dot{m}_{14} h_{14} - \dot{m}_5.h_5 \tag{6}$$

$$\dot{Q}_{con} = \dot{m}_{11}(h_{11} - h_{12}) \tag{7}$$

$$\dot{Q}_{HEX} = \dot{m}_8(h_8 - h_9) \tag{8}$$

$$\dot{W}_{comp} = \dot{m}_1.(h_2 - h_1) \tag{9}$$

$$\dot{Q}_{evap} = \dot{m}_1(h_1 - h_4) \tag{10}$$

$$\dot{Q}_{\text{int}} = \dot{m}_1(h_2 - h_3) = \dot{m}_{14}(h_{14} - h_{13}) \tag{11}$$

Circulation ratio

$$f = \frac{\dot{m}_7}{\dot{m}_{11}} = \frac{1 - x_8}{x_7 - x_8} \tag{12}$$

The coefficients of the performance for the vapour compression section of the novel is calculated as

$$COP_{vapor-comp} = \dot{Q}_{evap} / \dot{W}_{comp}) \tag{13}$$

The coefficient of performance for the novel cycle is defined as

$$COP_{cyclegen} = \dot{Q}_{evap} / (\dot{Q}_{gen} + \dot{W}_{comp} + \dot{W}_p) \tag{14}$$

Exergy is defined as maximum theoretical available (useful) work obtainable when the system interacts to equilibrium with the environment. The exergy balance applied to a fixed control volume is given as generally by the following equation (Bejan et al. 1996):

$$\sum \dot{m}_i e_i - \sum \dot{m}_e e_e + \sum \dot{Q}(1 - \frac{T_0}{T}) - \sum \dot{W} - \dot{E}_D = 0 \tag{15}$$

The first two terms represent the sum of the time rates of exergy input and output of the system, respectively. The third term is the exergy transfer rate accompanying heat transfer. $\dot{W}$ accounts for time rate of energy transfer by mechanical work, and $\dot{E}_D$ is the time rate of exergy destruction due to irreversibilities within the control volume. Neglecting the kinetic and potential energies the stream exergy flow rate is:

$$e = (h - h_0) - T_0(s - s_0) \tag{16}$$

where T_o is the reference temperature.

The rate of exergy destruction in each component of a compression-absorption cascade refrigeration cycle can be determined according to Eq.15.

The exergetic efficiency (*ECOP*), also called the second law efficiency, can be used to measure the performance of the cascade refrigeration cycle. *ECOP* is the ratio between the useful exergy obtained from a system and the useful exergy supplied to the system. Therefore, the exergetic efficiency can be written as:

$$ECOP = -\frac{\dot{Q}_{evap}\,1 - \frac{T_0}{T_{evap}}}{\dot{Q}_{gen}(1 - \frac{T_0}{T_{gen}}) + \dot{W}_{comp} + \dot{W}_p} \tag{17}$$

Table 1. Operating conditions of the novel cycle.

Point	T (K)	m(kg/s)	x(NH%)	m.e (kW)
1	243.15	0.0816		-23.035
2	345.34	0.0816		-8.606
3	273.15	0.0816		0.350
4	243.15	0.0816		0.350
5	307.15	0.701	50	0.557
6	307.55	0.701	50	0.916
7	335.95	0.701	50	7.537
8	363.15	0.594	41	18.194
9	329.79	0.594	41	5.541
10	329.79	0.594	41	5.541
11	363.15	0.107		6.066
12	307.15	0.107		0.222
13	273.15	0.107		0.222
14	273.15	0.107		-11.250

Detailed explanation of the calculations of the solution enthalpies can be found in the literature for NH_3-H_2O solution (Ziegler and Trepp 1984).

3. Results and Discussion

While NH_3-H_2O was used as fluid pair in the absorption section, NH_3 was used in the vapour compression section. At this analysis, operating conditions are assumed as T_1=243.15 K, T_{12} =307.15 K and cooling load Q_{evap}=100 kW (Table 1). The cycle has been compared with alternative cycles for the same operating conditions. It was presented that electrical energy consumption in the novel cycle is 60% lower than the classical one stage vapour compression refrigeration cycle and 24% lower than the vapour compression-absorption cascade refrigeration cycle. The thermal energy should be loaded up to 227.812 kW to the vapour compression-absorption two stage refrigeration cycle (Table 2). Also, the thermal energy consumption in the novel cycle is 6% lower than the single effect vapour compression-absorption cascade refrigeration cycle.

Figure 3 shows that the $COP_{cyclegen}$ increases on increasing the generator temperature of the novel cycle (for T_1=243.15 K, T_{12} =307.15 K and cooling load Q_{evap}=100 kW). The maximum $COP_{cyclegen}$ is obtained at 378.15 K generator temperature. The $COP_{cyclegen}$ initially exhibits significant increase with an increasing generator temperature, and then the values of the $COP_{cyclegen}$ become almost flat.

As generator temperature increases, ECOP falls (Figure 4). Because the generator needs more external thermal energy input to the system at higher generator temperatures, the irreversibility in the generator increases with rising generator temperatures. The other irreversibility source of

Table 2. Heat loads, compressor work and coefficients of performance of the novel cycle and alternative cycles for base case.

	The vapour compression-absorption two stage refrigeration cycle	The single effect vapour compression-absorption cascade refrigeration cycle	The classical one stage vapour compression refrigeration cycle
	NH_3-H_2O NH_3	NH_3-H_2O NH_3	NH_3
$\dot{Q}_{gen}$ (kW)	227.812	241.639	-
$\dot{Q}_{evap}$ (kW)	100	100	100
$\dot{W}_{comp}$ (kW)	17.373	22.941	43.104
$\dot{W}_p$ (kW)	1.823	1.893	
$COP_{vapour-comp}$	5.756	4.359	2.320
$COP_{cyclegen}$	0.405	0.375	-

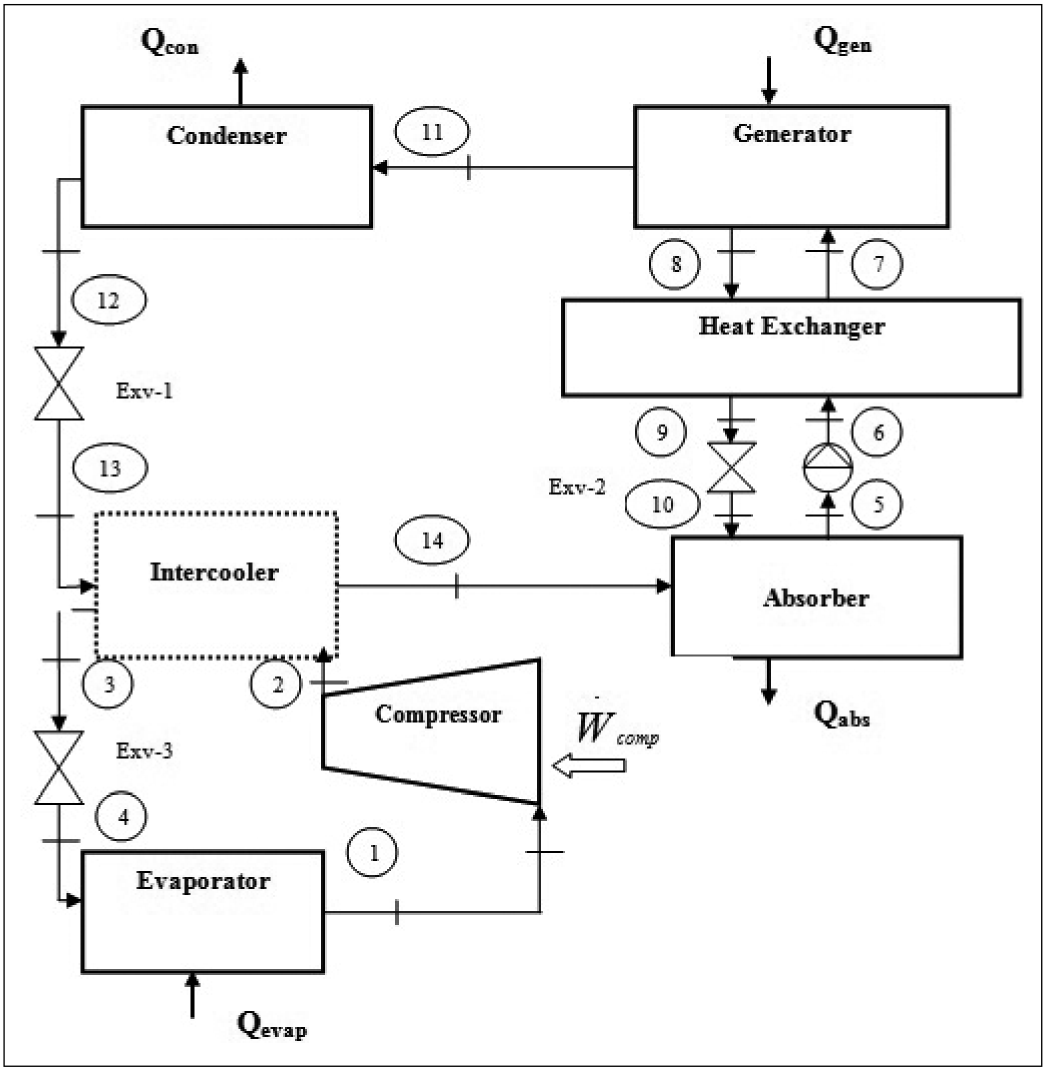

Figure 2. The schematic illustration of the novel cycle.

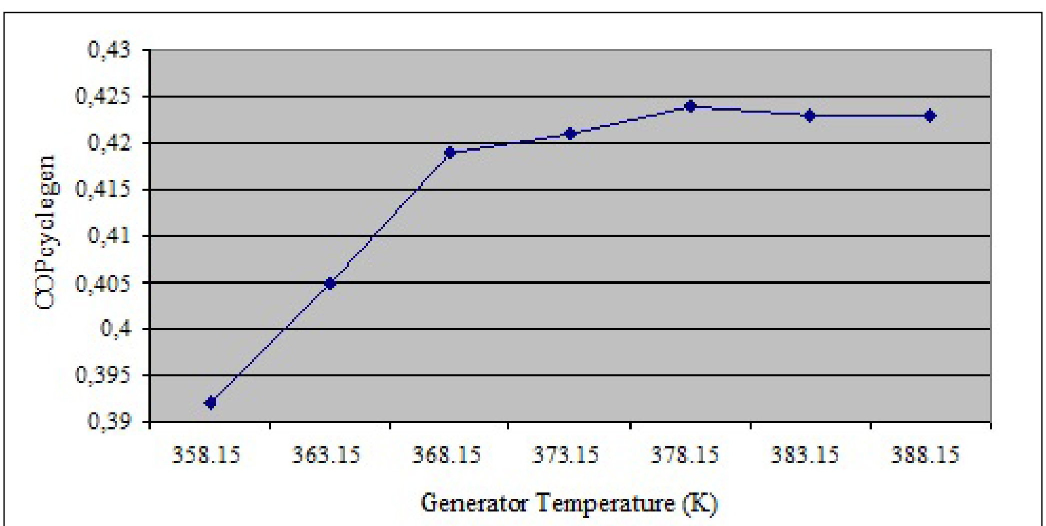

Figure 3. Variation of $COP_{cyclegen}$ with Generator Temperature (K) of the novel cycle.

the generator is the mixing process. As a result, the rising generator temperature generally increases the COP, in spite of the decline due to the resulting decrease in the exergetic efficiency.

Figure 5 and 6 show the the $COP_{cyclegen}$ and *ECOP* decrease on increasing the condenser temperature, as expectedly (for T_1=243.15 K, T_8 =363.15 K and cooling load Q_{evap}=100 kW). The thermal energy of the generator, the electrical energy consumption of the compressor and the pump increase with increasing condenser temperature, thereby lowering $COP_{cyclegen}$. Moreover, as the temperature difference between the condenser and the surrounding air increase due to increasing condenser temperature, the heat transfer in the condenser causes higher irreversibility, thus lowering exergetic efficiency of the novel cycle.

4. Conclusion

The vapour compression-absorption two stage refrigeration cycle, combined with absorption and vapour compression cycles, is analysed in this study. The thermodynamic analysis results show that the vapour compression-absorption two stage refrigeration cycle has the advantage less electric energy consumption than classical vapour compression cycle for low temperature cooling applications.

The thermodynamic analyses have been performed for different operating temperatures of the novel cycle. Increasing the generator temperatures promotes the *COP* of the novel cycle, while decreasing the condenser it. The exergetic efficiency of the system decreases on increasing the generator and condenser temperatures.

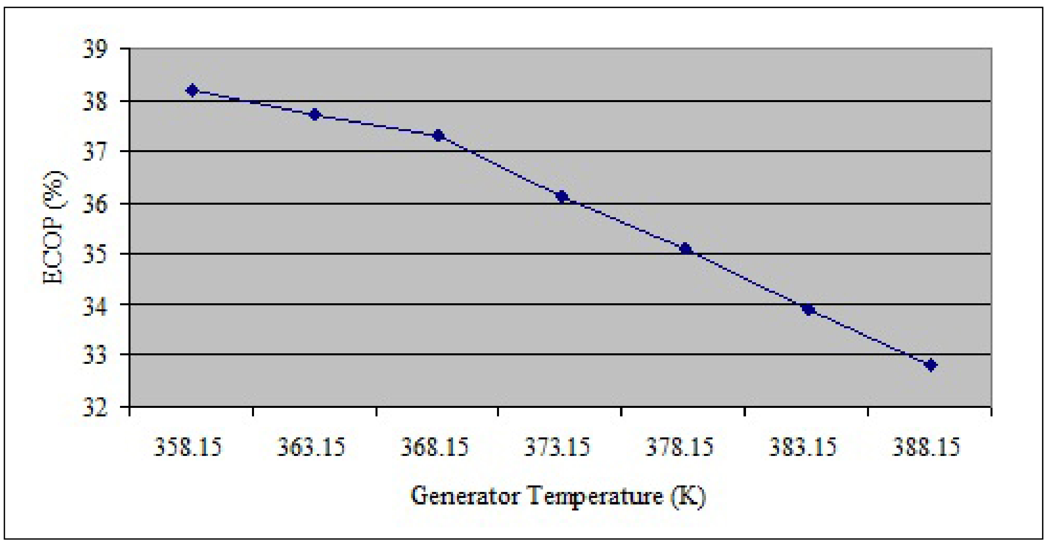

Figure 4. Variation of ECOP with Generator Temperature (K) of the novel cycle.

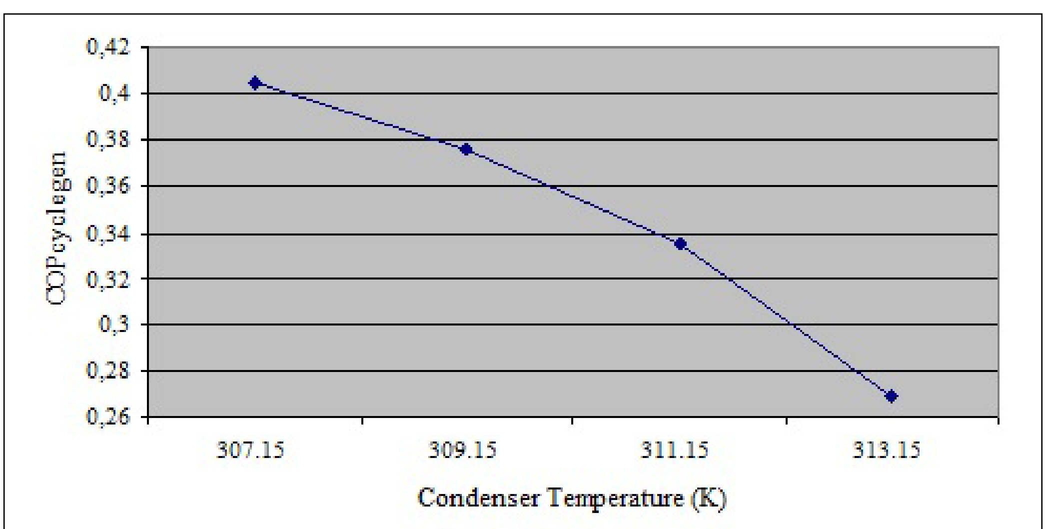

Figure 5. Variation of $COP_{cyclegen}$ with Condenser Temperature (K) of the novel cycle.

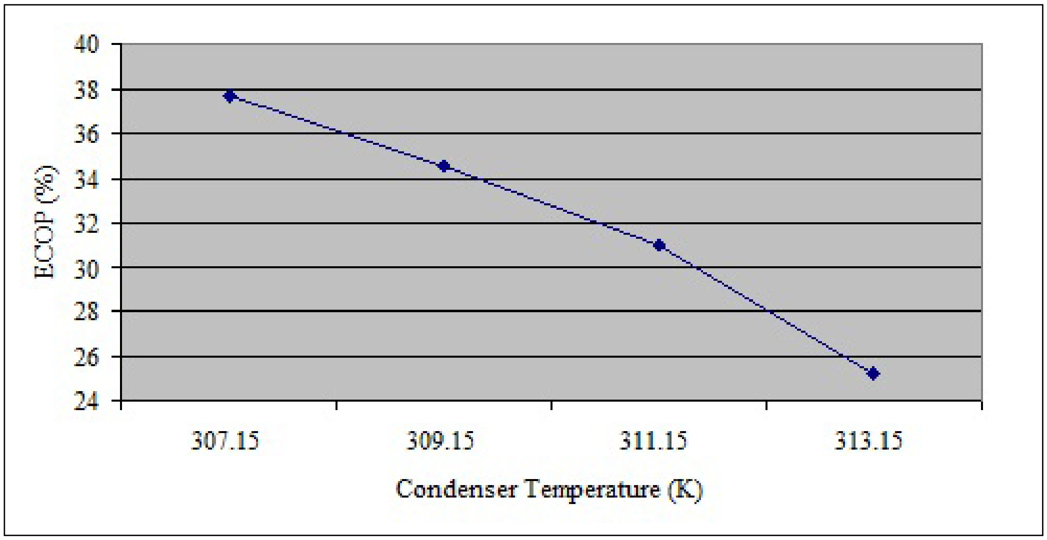

Figure 6. Variation of ECOP with Condenser Temperature (K) of the novel cycle.

The novel cycle provide economical use of the alternative energy sources. The electrical energy consumption the novel cycle has lower than the mechanical vapour compression systems. Thus, the novel cycle helps in reducing peak summer electric demands. This the novel cycle is a good choice to make effective use of the alternative energy source in Turkey. Therefore, the novel cycle is both economic and do not contribute ozone layer depletion due the environmentally friendly refrigerants.

5. References

Adewusi, SA., Zubair, S. 2004. Second law based thermodynamic analysis of ammonia-water absorption systems. *En. Con. Man.*, 45: 2355-2369.

Aman, J., Ting, DSK., Henshaw, P. 2014. Residential solar air conditioning: Energy and exergy analyses of an ammoniaewater absorption cooling system. *Appl. Therm. Eng.*, 62: 424-432.

Ayala, R., Heard, CL., Hollabd, FA. 1997. Ammonia-lithium nitrate absorption/ compression refrigeration cycle. Part I. Simulation. *Appl. Therm. Eng.*, 17 (3): 223-233.

Bejan, A., Tsatsaronis, G., Moran, M. 1996. *Thermal Design & Optimization*, Wiley, New York.

Cimsit, C., Ozturk, I. 2012. Analysis of compression-absorption cascade refrigeration cycles. *Appl. Therm. Eng.*, 40: 311-317, 2012.

Cimsit, C., Ozturk, I., Kıncay, O. 2015. Thermoeconomic optimization of LiBr/H_2O-R134a compression-absorption cascade refrigeration cycle. *Appl. Therm. Eng.*, 76: 105-115.

Kairouani, L., Nehdi, E. 2006. Cooling performance and energy saving of a compression- absorption refrigeration system assisted by geothermal energy. *Appl. Therm. Eng.*, 26: 288-294.

Kilic, FÇ., Kiliç, MK. 2013. Geothermal Energy and Turkey. *Eng. Machn.*, 54(639): 45-56.

Le Lostec, B., Galanis, N., Millette, J. 2012. Experimental study of an ammonia-water absorption chiller. *Int. J. Refrigeration*, 35: 2275-2286.

Mohamed, H., Brahim, AB. 2016. Modeling of the absorption phase of a cycle with solar absorption using the couple NH_3-H_2O for in-sight energy storage. *In. J. Hydrogen En.*, 1-7.

Nahla Bouaziz, N., Lounissi, D. 2015. Energy and exergy investigation of a novel double effect hybrid absorption refrigeration system for solar cooling. *Int. J. Hydrogen En.*, 40: 13849-13856.

Sarbu, I., Sebarchievici, C. 2013. Review of solar refrigeration and cooling systems. *En. Build.*, 67: 286–297.

Saylan, L., Sen, O., Toros, H., Arısoy, A. 2002. Solar energy potential for heating and cooling systems in big cities of Turkey. *En. Con. Man.*, 43: 1829–1837.

Sencan, A., Yakut, KA., Kalogirou, SA. 2005. Exergy analysis of lithium bromide/water absorption systems. *Ren. En.*, 30: 645-657.

Ziegler, B., Trepp, C. 1984. Equation of state for ammonia-water mixtures. *In. J. Refrigeration*, 7(2):101-6.

6. Symbols

COP	coefficient of performance
e	specific exergy (kJ kg^{-1})
$\dot{E}$	exergy flow rate (kW)
ECOP	exergetic efficiency
h	enthalpy (kJ kg^{-1})
$\dot{m}$	mass flow rate (kg s^{-1})

P	pressure (kPa)
s	specific entropy (kJ kg^{-1} K^{-1})
$\dot{Q}$	heat flow rate (kW)
T	temperature (K or ^{o}C)
$\dot{W}$	work flow rate or power of compressor (kW)

Subscripts

abs	absorber
comp	compressor
con	condenser
cyclegen	cycle general
e	streams leaving a component or system
evap	evaporator
gen	generator
HEX	*heat e*xchanger
i	inlet
int	intercooler
o	ambient
p	pump
total	total system
vapor-comp	vapour compression