J Biochem Tech (2020) Special Issue (1): 111-120
ISSN: 0974-2328

The Effects of Evaporator Temperature and Pressure Ratio on Exergy Efficiency and Exergy Destruction of Cooling Production Cycle

Peyman Asemani Kalejahi

Received: 12 August 2019 / Received in revised form: 07 December 2019, Accepted: 15 December 2019, Published online: 25 January 2020

Abstract

Much thermal energy is produced by combustion of fossil fuels to produce steam and heat for various applications. By utilization of an absorption refrigeration cycle, cooling could be generated from this wasted heat. The purpose of this study was to investigate the effect of temperature and evaporator pressure ratio on exergy efficiency and exergy destruction in cooling production process. The results showed that the refrigeration capacity increases by increasing the evaporator temperature. The net output of the cycle decreases as the evaporator temperature increases. Therefore, it could be concluded that the produced cycle power is inversely related to the evaporator temperature. By increasing the compressor pressure ratio, at first, the refrigeration capacity increases and then decreases. As the pressure ratio of compressor No. 1 increases, the exergy efficiency of the cycle decreases. This could be explained by the fact that the output work of the cycle decreases by increasing the compressor pressure ratio.

Key words: Evaporator Temperature and Pressure, Cooling Production, Exergy Destruction.

Introduction

Nowadays, the international community is moving towards the fighting with environmental degrading agents. This subject is observed in the international treaties like the Kyoto protocol, which implies on the environmental protection and not using its pollutants. Chloro-fluorocarbon (CFC) refrigerants that are the ozone layer destroyer and are used in the compression air conditioning machines are recognized as one of the main pollutants. Also, the compression systems use electrical energy as the input. This electricity is usually generated in thermal power plants and by combustion of fossil fuels that are one of the most important sources of earth pollution. In comparison with the compression systems, the absorption systems use chillers that are harmless to the environment. Also, the electricity consumption of the absorption systems is lower.

On the other hand, the accessible energy sources and global warming are the two concerning issues for the stability of the production amount of various types of the world energy in the future. For this reason, more attention has been paid to design of refrigeration systems in small scales with prime movers like micro-turbines, spark ignition engines and other types of internal combustion engines. These systems could be widely used for the production of cooling in the residential and commercial area using the wasted heat of the prime mover systems.

In most of the industries, very much thermal energy is produced by combustion of fossil fuels for the production of vapor and heat in various applications. A considerable amount of the produced energy is always entering the environment as wasted heat, using an absorption refrigeration cycle, cooling could be generated from this wasted heat.

Some researches' have been performed in this area. The date of the first use of the absorption cycle returns to AD 1700. On that time, they recognized that ice could be produced by the evaporation of pure water near a sulfuric acid in a tank that is placed in a vacuum tank. (Alexis, 2007; Kotas, 1985) Rabah Gomri performed a study about the single effect and double effect absorption systems with working fluid of lithium bromide –water in 2009. (Gomri, 2009) In the studies of Morosuk et al., two gas turbines were combined for power generation according to the Brayton cycle with a cooling cycle based on the LNG. The total cost of the studied cycle was relatively low and the efficiency was high. Also, they studied the cycle according to the thermodynamics second law and the advanced exergy analysis was performed on the cycle members. (Morosuk and Tsatsaronis, 2011; Tsatsaronis and Morosuk, 2010)

Liu et al., (2011) proposed a novel cooling system using the dual mixtures as the working fluid. The intended cycle was combined with the vapor absorption process for improving and increasing the recycling efficiency of cooling exergy of LNG. Herold et al., (1991) by designing the branched-GAX showed that in this cycle, with the increase of concentrated fluid flow rate in the high-temperature section, better heat transfer could be established between the hot and cold sections. Ramesh Kumar and Ramesh Kumar (2008) in the continuation of their studies about the HGAX cycle, investigated the effect of compressor pressure ratio. The results showed that the optimum efficiency,

Peyman Asemani Kalejahi
MA, Department of mechanical engineering, mechanical engineering Faculty, University of Tabriz,Iran.
Email: peyman.a.k@gmail.com

which is related to the optimum pressure ratio, is independent of the evaporator temperature. Yari et al., (2011) investigated the effect of generator temperature on the GAX and HGAX cycle by the thermodynamic first and second laws and reported that the generator temperature has a more significant effect on the thermodynamic second law compared to the first law.

Kang and Kashiwagi (2000) compared the performance of PGAX and single effect absorption cooling cycles for the application of heating panels (PSE) and reported that the effect of UA on the efficiency of PGAX cycle is more than the common single effect absorption cooling cycles. Kang et al., (2004) have performed the parametric study on four different types of HGAX. These cycles were planned for various purposes like high efficiency, having a low evaporator and generator temperatures and capability of production of hot water. Ramesh Kumar and Udaya Kumar (2008) modeled the GAXAC cycle for the water-ammonia solution and studied the effect of concentration and absorbent pressure differences on the cycle efficiency. Zheng et al., (2007), by investigating the GAX cycle showed that the first and second law efficiency of the GAX cycle is 31% and 78% higher than the single effect absorption cooling cycles, respectively. According to this, the aim of this study is to investigate the effect of temperature and evaporator pressure ratio on the exergy efficiency and exergy destruction in the cooling production process.

Research method

Exergy calculation:

When the system only exchanges heat with the environment and the final state of the system is dead state, the reversible work is simplified as follows:

$$w_{rev,sf} = (h_0 - T_0 s_0) - \left(h + \frac{V^2}{2} + gz - T_0 s\right) \qquad (1)$$

For the steady flow process with the input state of 1 and output state of 2, the flow exergy changes are calculated as follows:

$$\psi_2 - \psi_1 = (h_2 - h_1) - T_0(s_2 - s_1) + (Ke_2 - Ke_1) + (Pe_2 - Pe_1) \qquad (2)$$

Contrary to the non –flow exergy, Φ, the flow exergy, Ψ, can have negative values. This happens when the pressure is lower than the environment pressure (P_0).

$$\frac{d\phi_{CV}}{dt} = \dot{\phi}_Q + \sum_{in} \dot{m}b - \sum_{out} \dot{m}b + \left(\dot{W}_{act} + P_0 \frac{dV_{CV}}{dt}\right) - \dot{I}_{CV} \qquad (3)$$

AS we have:

$$\dot{\phi}_Q = \sum \dot{Q}\left(1 - \frac{T_0}{T_i}\right) \qquad (3)$$

This equation explains that the exergy change rate in a control volume depends on the exergy transfer accompanied by the heat transfer, mass transfer and work transfer in the control surface and decreases due to the internal irreversibility. A schematic of the exergy calculation of an open system (volume control) shown in Figure (1).

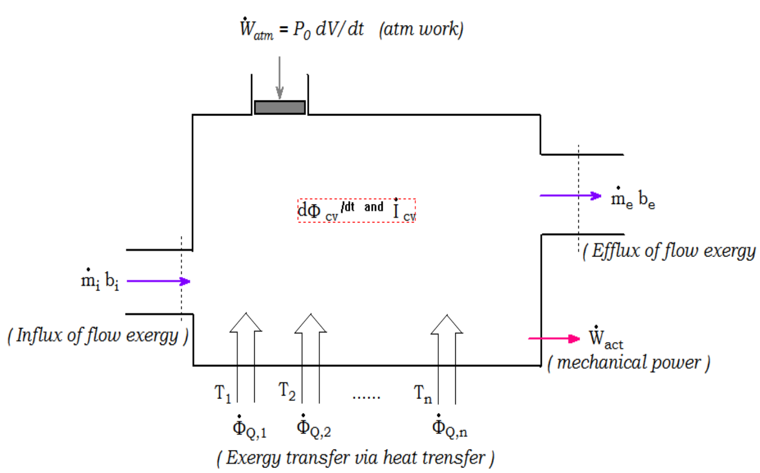

Figure 1: A schematics of exergy calculation of an open system.

Also, if one input and one output are considered, the equation will be simplified as follows:

$$0 = \dot{\phi}_Q - \dot{m}\Delta\psi + \dot{W}_{act,sf} - \dot{I}_{CV} \tag{4}$$

Irreversibility:

One of the fundamental concepts in engineering topics is the waste in work production capability due to irreversibility. Many factors cause the irreversibility of the process. Four main factors are friction, free expansion, and heat transfer due to the limited temperature difference and mixing of two different materials.

The relationship between the actual work and reversible work for the system is as follows:

$$\dot{W}_{act,u} - \dot{W}_{rev,u} = \dot{W}_{act} - \dot{W}_{rev} = T_0\dot{\sigma}_{cv} \geq 0 \tag{5}$$

The amount of work that is wasted in a process due to irreversibility is defined as its irreversibility.

$$\dot{I}_{cv} = \dot{W}_{act,u} - \dot{W}_{rev,u} = \dot{W}_{act} - \dot{W}_{rev} = T_0\dot{\sigma}_{cv} \geq 0 \tag{6}$$

Calculation of irreversibility:

The amount of irreversibility of the volume control per unit in the steady state condition is as follows:

$$i_{CV} = T_0\frac{\dot{\sigma}_{CV}}{\dot{m}} = q_i\left(1 - \frac{T_0}{T_I}\right) - (b_2 - b_1) + w_{act} \tag{7}$$

Also, the following equation could be presented for the total irreversibility of volume control system:

$$i_{sf} = T_0(\Delta s_0 + s_2 - s_1 + \Delta s_R) = T_0\Delta s_{tot} \tag{8}$$

The system exergy analysis leads to the determination of maximum unit efficiency and also the determination of exergy wasting locations. The equations related to the exergy wastes for the various equipment according to Figure (1) are as follows:

$$I_{gen} = T_{ref}(\mathrm{m_g}(\mathrm{S_1} - \mathrm{S_6})) \tag{9}$$

$$I_{eje} = T_{ref}((\mathrm{m_g} + \mathrm{m_e}).\mathrm{S_3} - \mathrm{m_g}.\mathrm{S_1} - \mathrm{m_e}.\mathrm{S_2}) \tag{10}$$

$$I_{Con} = T_{ref}((\mathrm{m_g} + \mathrm{m_e}).(\mathrm{S_4} - \mathrm{S_3}) + (^{\mathrm{Q_c}}/_{\mathrm{T_{ref}}})) \tag{11}$$

$$I_{pump} = W_{pump} + m_g.\{(\mathrm{h_6} - \mathrm{h_4}) - T_{ref}.(\mathrm{S_6} - \mathrm{S_4})\} \tag{12}$$

$$I_{exp} = m_e.\{T_{ref}.(\mathrm{S_5} - \mathrm{S_4})\} \tag{13}$$

$$I_{eva} = T_{ref}(\mathrm{m_e}.(\mathrm{S_2} - \mathrm{S_5}) + (^{\mathrm{Q_{eva}}}/_{\mathrm{T_{ref}}})) \tag{14}$$

$$I_{tot} = I_{gen} + I_{eje} + I_{con} + I_{pump} + I_{exp} + I_{eva} \tag{15}$$

Second law efficiency:

The efficiency of a thermodynamic system in energy conversion from one form to other form evaluates the effectiveness of conversion of one input to the final use. The first law efficiency shows this kind of transformation according to the input energy and not considering the utilization of this energy. The first law efficiency (for a thermal motor) is defined as the ratio of used energy to the input energy:

$$\eta_{\mathrm{I}} = \frac{\mathrm{W}}{Q_{\mathrm{H}}} \tag{16}$$

That in the above equation, Q_H is the transferred heat from the high-temperature source. The weakness of the first law efficiency is that practically, only a portion of the input energy is used for the effective performance of the work. The first law specifies that energy cannot be generated or destroyed. But the thermodynamic wastes cannot be revealed only by energy balance. The processes like heat transfer by throttling, without any energy loss, resulting in the energy quality degradation. The exergy concept (serviceability) is used for the determination of the efficiency of a system for doing the work. The performance of a system is the ratio of the produced real useful work to the reversible useful work or useable work. Please note that in this definition, the input useful energy is used; because the input useful energy shows the potential of maximum work in a system for performing the work. Therefore, the performance that is also called the second law efficiency could be considered as the proper criterion for the potential of performing the work which is defined as follows:

$$E = \frac{W}{\phi_Q} \qquad (17)$$

Because the principle and origin of exergy are derived from the thermodynamics second law, the performance parameter of a process based on the exergy with the second law efficiency or exergy efficiency is defined as the ratio of the output effective exergy to the input exergy:

Thermomechanical exergy:

When a system gets from a definite thermodynamic state (initial temperature of T and pressure of P) to the limited dead state (temperature of T_0 and pressure of P_0), the maximum work obtained in this process is defined as the thermomechanical exergy. In the limited dead state, the system is in the thermal and mechanical equilibrium with its surroundings but it has not reached the chemical equilibrium. The thermomechanical exergy equation could be presented as follows:

$$\dot{E}^{th} = (h_i - h_0) - T_0 (s_i - s_0) \qquad (18)$$

In order to evaluate the system according to the thermodynamics second law, it is necessary to determine the fuel and product for each of the cycle elements. Each of the cycle elements is the results that each component is designed or bought for that reason and the fuel is the source of energy that is consumed in each component for producing the product.

Chemical exergy:

The chemical exergy is defined as the maximum produced work in a condition that the chemical species have the possibility of chemical mixing or reaction with the present species in the environment. This reaction results in the production of excessive work that is called the chemical exergy. It is worth noting that in the present work, the chemical exergy of the output gases of the engine is considered as the diffusion exergy that this kind of exergy includes only the types that are present both in the system and environment. In this research, these species include the O_2, CO_2, N_2, and H_2O. For this condition, we need to determine the temperature and pressure of the real dead state (environment). In the present work, the environment temperature is considered to be about T_0 = 298K and the pressure is P_0 = 1 bar. Also, the combination of these species in the environment condition is according to Table (1).

Table 1: The mixture of the species in the environment condition

	By volume%			
component	N_2	O_2	H_2O	CO_2
	77.48	20.59	1.9	0.03

The chemical exergy for the mixture of the gases is calculated with the following equation:

$$E_i^{CH} = -\bar{R} T_0 \sum y_k \ln \frac{y_{0,k}}{y_k} \qquad (19)$$

In the above equation, y_k, and $y_{0,k}$ are a molar fraction of the k species in the limited dead state and real dead state (environment), respectively.

Exergy destruction:

The exergy destruction in each of the cycles' elements could be calculated by balancing the exergy:

$$\dot{E}_{D,k} = \dot{E}_{F,k} - \dot{E}_{P,k} - \dot{E}_{L,k} \qquad (20)$$

The second law has been applied to each of the cycles' element and is shown in Table (2).

Table 2: The exergy balance according to the thermodynamics second law for different elements of simultaneous production combined cycle according to LNG and GAX cycle

Components	Exergy balance
Generator	$\dot{I}_{des\&abs} = T_0 (\dot{m}_{34}s_{34} + \dot{m}_{36}s_{36} + \dot{m}_{31}s_{31} - \dot{m}_{35}s_{35} - \dot{m}_{37}s_{37} - \dot{m}_{43}s_{43}$ $-\dot{m}_{32}s_{32} + \dot{m}_g (s_{out,gen} - s_{in,gen}) + \dot{m}_{abs} (s_{out,abs} - s_{in,abs}))$

GAXD	$\dot{I}_{des\&abs}=T_0(\dot{m}_{34}s_{34}+\dot{m}_{36}s_{36}+\dot{m}_{31}s_{31}-\dot{m}_{35}s_{35}-\dot{m}_{37}s_{37}-\dot{m}_{43}s_{43}$ $-\dot{m}_{32}s_{32}+\dot{m}_g(s_{out,gen}-s_{in,gen})+\dot{m}_{abs}(s_{out,abs}-s_{in,abs}))$
GAXA	$\dot{I}_{des\&abs}=T_0(\dot{m}_{34}s_{34}+\dot{m}_{36}s_{36}+\dot{m}_{31}s_{31}-\dot{m}_{35}s_{35}-\dot{m}_{37}s_{37}-\dot{m}_{43}s_{43}$ $-\dot{m}_{32}s_{32}+\dot{m}_g(s_{out,gen}-s_{in,gen})+\dot{m}_{abs}(s_{out,abs}-s_{in,abs}))$
Absorber	$\dot{I}_{des\&abs}=T_0(\dot{m}_{34}s_{34}+\dot{m}_{36}s_{36}+\dot{m}_{31}s_{31}-\dot{m}_{35}s_{35}-\dot{m}_{37}s_{37}-\dot{m}_{43}s_{43}$ $-\dot{m}_{32}s_{32}+\dot{m}_g(s_{out,gen}-s_{in,gen})+\dot{m}_{abs}(s_{out,abs}-s_{in,abs}))$
Evaporator	$\dot{I}_{eva}=T_0(\dot{m}_{42}s_{42}-\dot{m}_{41}s_{41}+\dot{m}_e(s_{out,eva}-s_{in,eva}))$
Condenser	$\dot{I}_{cond}=T_0(\dot{m}_{39}s_{39}-\dot{m}_{38}s_{38}+\dot{m}_{cond}(s_{out,cond}-s_{in,cond}))$
Rectifier	$\dot{I}_{rec}=T_0(\dot{m}_{37}s_{37}+\dot{m}_{38}s_{38}-\dot{m}_{36}s_{36}+\dot{m}_{rec}(s_{out,rec}-s_{in,rec}))$
Pump 1	$\dot{I}_p=T_0(\dot{m}_{32}s_{32}-\dot{m}_{31}s_{31})$
Expansion Valve 1	$\dot{I}_{EV\,1}=T_0(\dot{m}_{35}s_{35}-\dot{m}_{34}s_{34})$
Expansion Valve 2	$\dot{I}_{EV\,1}=T_0(\dot{m}_{40}s_{40}-\dot{m}_{41}s_{41})$
Pre- Cooler	$\dot{I}_{RHX}=T_0(\dot{m}_{40}s_{40}+\dot{m}_{43}s_{43}-\dot{m}_{39}s_{39}-\dot{m}_{42}s_{42})$
Compressor 1	$\dot{E}_{D,comp1}=\dot{E}_{21}+\dot{W}_{comp1}-\dot{E}_{22}$
Compressor 2	$\dot{E}_{D,comp2}=\dot{E}_{23}+\dot{W}_{comp2}-\dot{E}_{24}$
Compressor 3	$\dot{E}_{D,comp3}=\dot{E}_{11}+\dot{W}_{comp3}-\dot{E}_{12}$
Turbine 1	$\dot{E}_{D,Turb1}=\dot{E}_{26}-\dot{W}_{Turb1}-\dot{E}_{27}$
Turbine 2	$\dot{E}_{D,Turb2}=\dot{E}_{13}-\dot{W}_{Turb2}-\dot{E}_{14}$
Turbine 3	$\dot{E}_{D,Turb3}=\dot{E}_{3}-\dot{W}_{Turb3}-\dot{E}_{4}$
Heat exchanger 1	$\dot{E}_{D,HX\,2}=\dot{E}_{27}+\dot{E}_{12}-\dot{E}_{28}-\dot{E}_{13}$
Heat exchanger 2	$\dot{E}_{D,HX\,3}=\dot{E}_{14}+\dot{E}_{2}-\dot{E}_{11}-\dot{E}_{3}$
Pump 2	$\dot{E}_{D,Pump}=\dot{E}_{1}+\dot{W}_{Pump}-\dot{E}_{2}$
Combustion chamber	$\dot{E}_{D,C.C.}=\dot{E}_{24}+\dot{E}_{25}-\dot{E}_{26}$

Findings:

The effect of evaporator temperature on the cooling capacity, cycle net produced power, energy efficiency, exergy efficiency, and cycles' total exergy destruction is investigated in Figures (2) to (6), respectively and the related graphs are plotted.

According to Figure (2), it could be understood that with the increase of evaporator temperature, the cooling capacity increases. This could be explained as more the evaporator temperature increases the output enthalpy of the evaporator increases, however, the input enthalpy of the evaporator is constant, therefore, it could be claimed that in fact, the difference between the input and output enthalpy of the evaporator increases, that this is followed by the increase of the cooling capacity of the cycle.

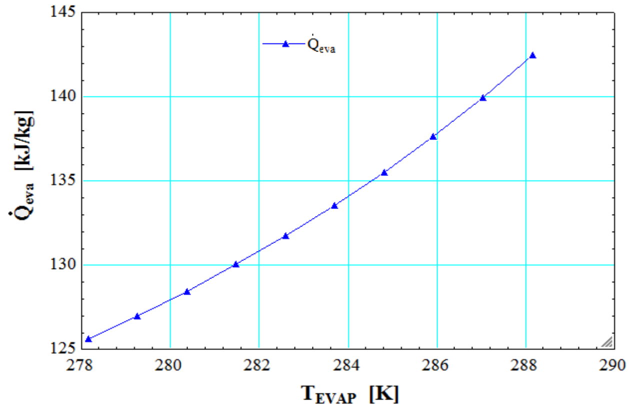

Figure 2: The variation of cooling capacity versus the evaporator temperature.

The variation of the cycle net produced power versus evaporator temperature is shown in Figure (3). As could be seen in the figure, the cycle net produced power decreases as the evaporator temperature increases. Therefore, it could be concluded that the cycle produced power has an inverse relationship with the evaporator temperature.

Form the studied graphs, it could be deduced that with the increase of the evaporator temperature, the cooling capacity increases but the cycle net produced power decreases. But the amount of the cooling capacity increase overcomes the decrease in the produced power and with the increase of the temperature the thermodynamics first law, efficiency (energy efficiency) increases that is shown in Figure (4).

Also, the graph of the second law efficiency (exergy efficiency) and total exergy destruction of the cycle is shown in Figures (5) and (^), respectively.

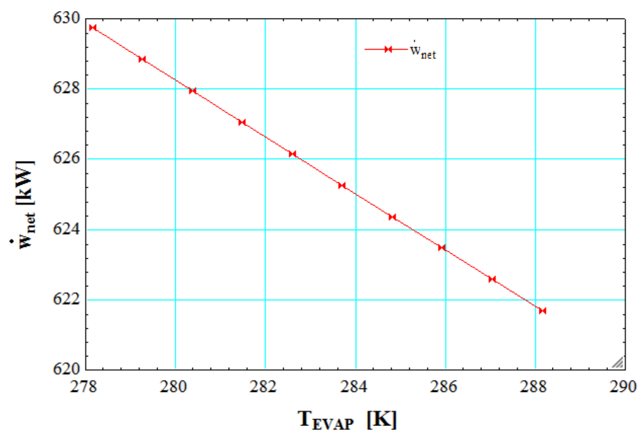

Figure 3: The graph of cycles' net power produced versus the evaporator temperature.

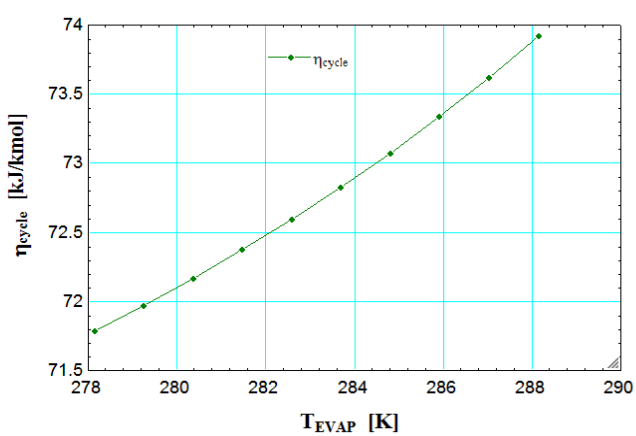

Figure 4: The variation of cycles' energy efficiency versus the evaporator temperature.

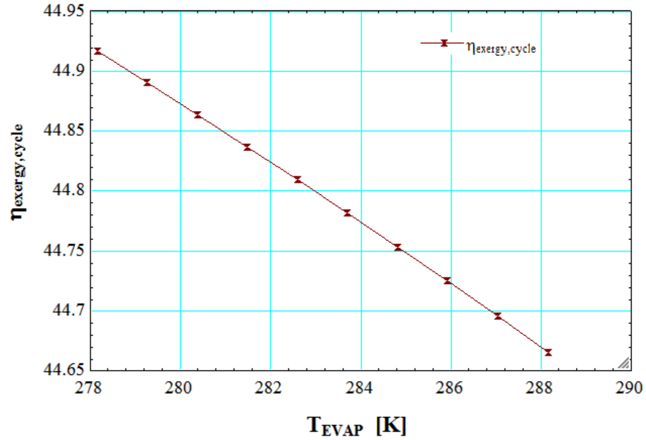

Figure 5: The variation of cycles' exergy efficiency versus the evaporator temperature.

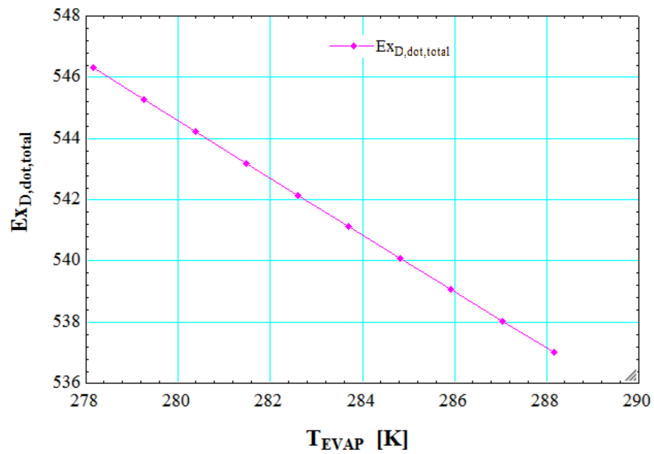

Figure 6: The variation of cycles' total exergy destruction versus the evaporator temperature.

The effects of the compressor number one pressure ratio on the cooling capacity, cycles' net produced power, energy efficiency, exergy efficiency, and cycles' total exergy destruction is investigated in Figures (7) to (11), respectively and the related graphs are plotted.

According to Figure (7), it could be observed that with the increase of compressor number one pressure ratio (P_{Rc1}) at first, the cooling capacity increases and then decreases. According to the figure, the cooling capacity is maximum in the pressure ration equal to 6.5.

The variation in the cycles' net power versus the compressor number one pressure ratio is shown in Figure (8).

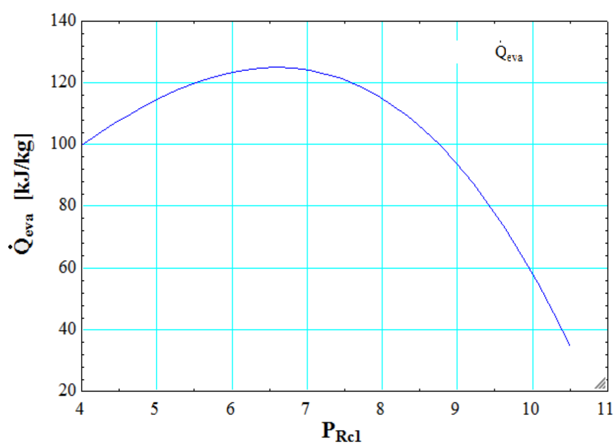

Figure 7: The variation of the cooling capacity versus the compressor number one pressure ratio.

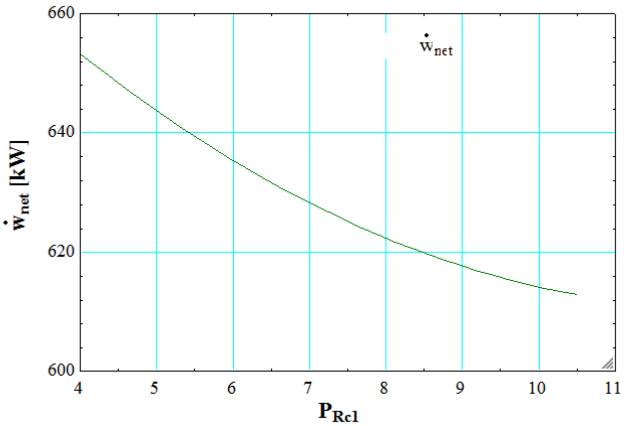

Figure 8: The variation of the cycles' net produced power versus the compressor number one pressure ratio.

According to Figure (8), with the increase of compressor number one pressure ratio, the cycles' net produced power decreases. Because as the compressor pressure ratio increases, the consuming work of the compressor increases and this results in the decrease of the cycles' net produced power decrease.

Considering that the cycles' energy efficiency is affected by the cycles' cooling capacity and net power variations, it is expected that the energy efficiency shows ascending behavior in a range and descending behavior in another range with the increase of the compressor number one pressure ratio. And have the extremum like the cooling capacity graph. Figure (9) shows the variation of energy efficiency versus the compressor number one pressure ratio.

The variation of cycles' exergy efficiency and cycles' total exergy destruction versus the compressor number one pressure ratio are shown in Figures (10) and (11).

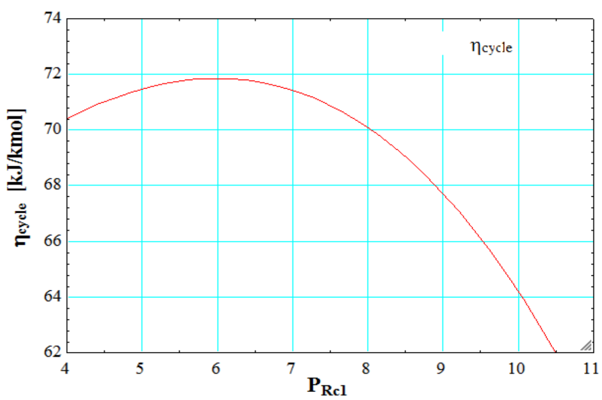

Figure 9: Variation of the cycles' energy efficiency versus the compressor number one pressure ratio.

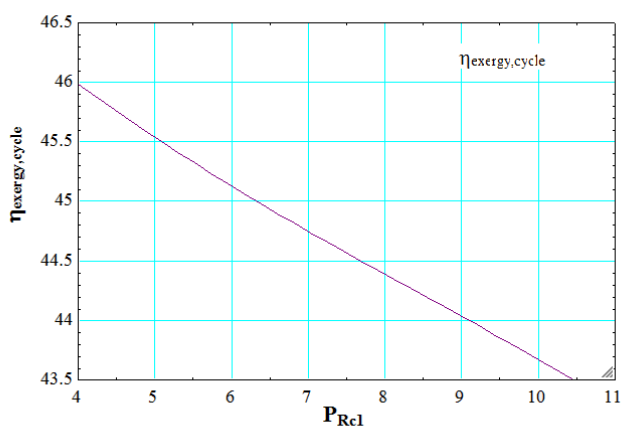

Figure 10: Variation of the cycles' exergy efficiency versus the compressor number one pressure ratio.

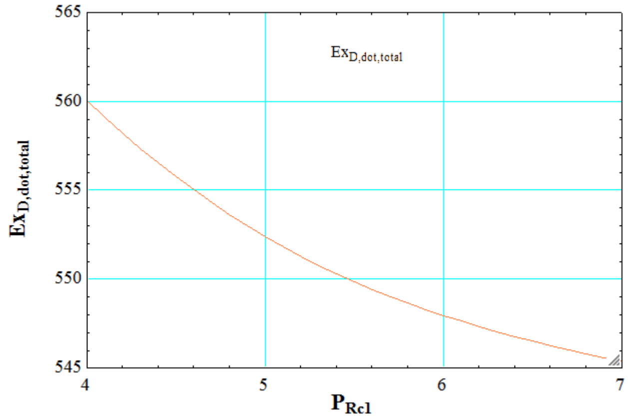

Figure 11: Variation of the cycles' exergy destruction versus the compressor number one pressure ratio.

According to Figure (10), the cycles' exergy efficiency decreases with the increase of compressor number one pressure ratio. This could be explained that with the increase of compressor number one pressure ratio, as discussed in Figure (9), the output work of the cycle decreases, knowing that the effect of cycles' net output power variation that is identical to exergy, is very higher than the cooling capacity variation that is different from the cooling exergy, it is expected that the exergy efficiency to have similar behavior like the power variation behavior.

Also, it could be observed in Figure (11) that with the increase of the compressor number one pressure ratio, the total exergy destruction will also decrease.

Conclusion:

In this study, the effect of temperature and evaporator pressure ratio was studied on the exergy efficiency and cycles' total exergy destruction. The results showed that with the increase of evaporator temperature the cooling capacity increases. With the increase of the evaporator temperature, the cycles' net produced power decreases. Therefore, it could be concluded that the cycles' produced power has inversely related to the evaporator temperature. With the increase of the pressure ratio, at first, the cooling capacity increases and then decreases. With the increase of the compressor number one pressure ratio, the exergy efficiency of the cycle decreases. This could be explained in this way that with the increase of the compressor number one pressure ratio, the output work of the cycle decreases.

References

Alexis, G. K. (2007). Performance parameters for the design of a combined refrigeration and electrical power cogeneration system. *International journal of refrigeration*, *30*(6), 1097-1103.

Gomri, R. (2009). Second law comparison of single effect and double effect vapour absorption refrigeration systems. *Energy Conversion and Management*, *50*(5), 1279-1287.

Herold, K. E. (1991). The branched GAX absorption heat pump cycle. In *Tokyo: Proceedings of International Absorption Heat Pump Conference, Sep30.-Oct. 2, 1991*.

Kang, Y. T., & Kashiwagi, T. (2000). An environmentally friendly GAX cycle for panel heating: PGAX cycle. *International Journal of Refrigeration*, *23*(5), 378-387.

Kang, Y. T., Hong, H., & Park, K. S. (2004). Performance analysis of advanced hybrid GAX cycles: HGAX. *International Journal of Refrigeration*, *27*(4), 442-448.

Kotas, T. J. (1985). *The Exergy Method of Thermal Plant Analysis*. Butterworths,

Liu, Y., & Guo, K. (2011). A novel cryogenic power cycle for LNG cold energy recovery. *Energy*, *36*(5), 2828-2833.

Morosuk, T., & Tsatsaronis, G. (2011). Comparative evaluation of LNG–based cogeneration systems using advanced exergetic analysis. *Energy*, *36*(6), 3771-3778.

Tsatsaronis, G., & Morosuk, T. (2010). Advanced exergetic analysis of a novel system for generating electricity and vaporizing liquefied natural gas. *Energy*, *35*(2), 820-829.

Udayakumar, M. (2008). Studies of compressor pressure ratio effect on GAXAC (generator–absorber–exchange absorption compression) cooler. *Applied Energy*, *85*(12), 1163-1172.

Yari, M., Zarin, A., & Mahmoudi, S. M. S. (2011). Energy and exergy analyses of GAX and GAX hybrid absorption refrigeration cycles. *Renewable energy*, *36*(7).

Zheng, D., Deng, W., Jin, H., & Ji, J. (2007). α–h diagram and principle of exergy coupling of GAX cycle. *Applied thermal engineering*, *27*(11-12), 1771-1778.