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Thermodynamic Analysis of a Trigeneration System Driven by an Internal Combustion Engine and a Steam Ejector Refrigeration System

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Abstract: This paper aims at studying a trigeneration system, based on an internal combustion engine and a steam ejector refrigeration system. The designed cycle is to generate cooling and heating energies, and supply power simultaneously. The cycle is studied from thermodynamics point of view and for this purpose, the first law of thermodynamics is applied to all the components of the cycle. The efficiency of the cycle is studied by changing a number of parameters such as the pinch point temperature, evaporator temperature, heat generator temperature, and condenser temperature to simulate summer and winter seasons. The results show an increase in the fuel efficiency up to 88% for the winter and 71% for the summer, and also a 28% fuel savings for the winter and 18% for the summer seasons.

Keywords: Combined Generation, Gas Fired Engine, Steam Ejector Refrigeration System, Thermodynamic Analysis

1. Introduction

In recent years numerous and extensive researches have been conducted to evaluate cogeneration of cooling, heating, and power systems (CCHP) from both thermodynamic and exergy points of view.

In 2009, exergy analysis of combined generation of heating, cooling, and power system was conducted by Abdul Khaliq and in this research the effects of different parameters on the efficiencies of the first and second laws were studied [1, 2]. In 2006, Wu and Wang made a detailed review on the types of cogeneration systems and stated specifications of those systems, briefly [3]. In 2005, an experimental investigation on a combined generation of cooling, heating, and power system driven by a gas engine and a micro absorption chiller was conducted by Kong and Wang [4].

In 2008, Mehmet Kanoglu and Ibrahim Dincer conducted a performance assessment of various ogeneration systems for a building in which they investigated the effects of certain operating parameters such as steam pressure and water temperature on energy and exergy efficiencies [5]. In 2008, Yiping Dai and his colleagues presented a cogeneration system based on a steam turbine and an ejector refrigeration system. They did an exergy analysis to examine the system efficiency. In this study a parametric study on the power and refrigeration output of the steam turbine and the ejector refrigeration system was made [6]. Cardona and Piacentino did a research on optimal design of CCHP system using thermodynamic analyses to use their system in buildings [7].

Investigating the trigeneration simultaneous production systems have been conducted from determining cooling and heating loads point of view. However, combining such systems with internal combustion engines and determining the effect of different thermodynamic properties to optimize the efficiency of the system for both summer and winter seasons have not been focused on thoroughly. The present work does just that for when the cycle

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goes through producing cooling and heating energies together with providing the required power simultaneously.

2. Cycle descriptions

A schematic view of the designed heating, cooling, and power cogenerating system coupled with a gas-fired engine and a steam ejector refrigeration system is shown in Fig. 1.

As the first stage, air and fuel at the environment temperature (depending on the season) and atmospheric pressure enter the engine. Combustion occurs with excess air in the engine. Next, combustion products enter a heat recovery steam generator (HRSG) and the required amount of heating and cooling energies are produced. After exchanging heat in the heat recovery steam generator, combustion products are discharged to the atmosphere.

To supply electricity, an internal combustion engine is coupled with a generator and the required electricity is produced here by the working engine. It should be mentioned that, in summer, a small amount of work produced by the engine is utilized to provide enough energy for the pump in the cooling cycle.

Moreover, water jacket around the engine which is used to cool the engine, supplies hot water for all seasons. Steam exiting the heat recovery steam generator, depending on the the season, is divided into two parts. Path 7 (Fig. 1) is considered for the winter and path 8 (Fig. 1) for the summer. In the summer, steam enters the steam ejector refrigeration system and the required cooling is generated. In the winter, the produced steam in the heat recovery steam generator enters the heat exchanger (heat exchanger No. 1 shown as E.1 in Fig. 1) and the required heating is generated. To provide the cooling energy, the produced steam in the heat recovery steam generator enters the heat generator of the refrigeration system and the required steam (primary flow) is produced. Next, the steam in the heat generator enters the nozzle. Since the steam entering the nozzle has a high pressure, this pressure is decreased at the outlet of the nozzle and this pressure drop makes the generated steam to be sucked through the evaporator (secondary flow). The primary and secondary flows are mixed together inside the ejector and the mixture enters the diffuser shown in Fig. 1 in order to provide the necessary pressure for exiting from the ejector and entering the condenser. After this stage, the flow in the condenser is condensed and a part of it is pumped into the heat generator by a pump and the process is repeated. The other part of the condensed water, after passing through an expansion valve enters the evaporator to capture the heat from the environment, and the cooling cycle is repeated again.

3. Thermodynamic calculations of cycle components

The assumptions used in this study are presented in Table 1.

3.1. Gas fired engine– generator

First, the reaction carried out in the engine is considered [9,10],

$$\begin{aligned} & CH_4 + a(0.2059\ O_2 + 0.7748\ N_2 + \\ & \quad + 0.0003\ CO_2 + 0.019\ H_2O(g)) \rightarrow \\ & \quad (1+0.0003a)CO_2 + (2+0.019a)H_2O(g) \\ & \quad + (a-2)O_2 + 0.7748\,a\,N_2 \end{aligned} \tag{1}$$

where a determines the amount of excess air. It is necessary to write the first law for the engine to obtain variable a:

$$\begin{aligned} Q_{c.v} - W_{c.v} = {} & n_{CO_2}(\bar{h}_f^\circ + \Delta\bar{h})_{CO_2} + n_{H_2O(g)} \\ & (\bar{h}_f^\circ + \Delta\bar{h})_{H_2O(g)} + n_{O_2}(\bar{h}_f^\circ + \Delta\bar{h})_{O_2} \\ & + n_{N_2}(\bar{h}_f^\circ + \Delta\bar{h})_{N_2} - n_{CH_4} \\ & (\bar{h}_f^\circ + \Delta\bar{h})_{CH_4} - n_{O_2}(\bar{h}_f^\circ + \Delta\bar{h})_{O_2} \\ & - n_{N_2}(\bar{h}_f^\circ + \Delta\bar{h})_{N_2} - n_{CO_2} \\ & (\bar{h}_f^\circ + \Delta\bar{h})_{CO_2} - n_{H_2O}(\bar{h}_f^\circ + \Delta\bar{h})_{H_2O} \end{aligned} \tag{2}$$

In Eq. (2) $\bar{h}_f^\circ$ and $\Delta\bar{h}$ are formation enthalpy, and enthalpy difference between each mode and enthalpy at the base conditions, respectively. Also, n is the number of moles. By finding the formation

Table 1. Required assumptions for investigating the cycle performance

Heat transfer from the engine to the environment	21 kW
The environment air conditions in summer	T= 30 °C, P = 1bar
The environment air conditions in winter	T= 10 °C, P = 1bar
Output required potency from the engine	150 kW
Input energy into engine	515 kW
Outlet gas temperature from engine	627 °C
Engine compression ratio	10
Generated heat load by the engine casing	178 kW
Evaporator temperature	12, 14, 16, 18 °C
Heat generator pressure	3, 4, 5, 6, 7 bar
Condenser temperature	42.5, 45, 50, 52.5 °C
HRSG pressure	15 bar
Efficiency of the ejector's mixing part	1
Efficiency of the ejector's nozzle	1
Efficiency of the ejector's diffuser	0.5

enthalpy $(\bar{h}_f^\circ)$ and the enthalpy difference $(\Delta\bar{h})$ using tables of thermodynamics, the unknown coefficients of the reaction (a) will be determined.

If a gas is composed of a mixture of two gases A and B [9],

$$m_{tot}h_{tot} = m_A h_A + m_B h_B \rightarrow h_{tot} = \frac{m_A}{m_{tot}} h_A + \frac{m_B}{m_{tot}} h_B \quad (3)$$

Thus, for these two points (points 1 and 2), for example, for point 2, h_2 becomes [9],

$$h_2 = \frac{1}{m_2}(m_{CO_2}h_{CO_2} + m_{H_2O}h_{H_2O} + m_{N_2}h_{N_2} + m_{o_2}h_{o_2}) \quad (4)$$

In Eq. (4) enthalpy of point 2 (Fig. 1) is calculated at the engine exhaust temperature. Now, to calculate entropy, the equation for mixture of gases is needed [11]:

$$\bar{S}_{mix} = \sum_i y_i \bar{S}_i^* \quad (5)$$

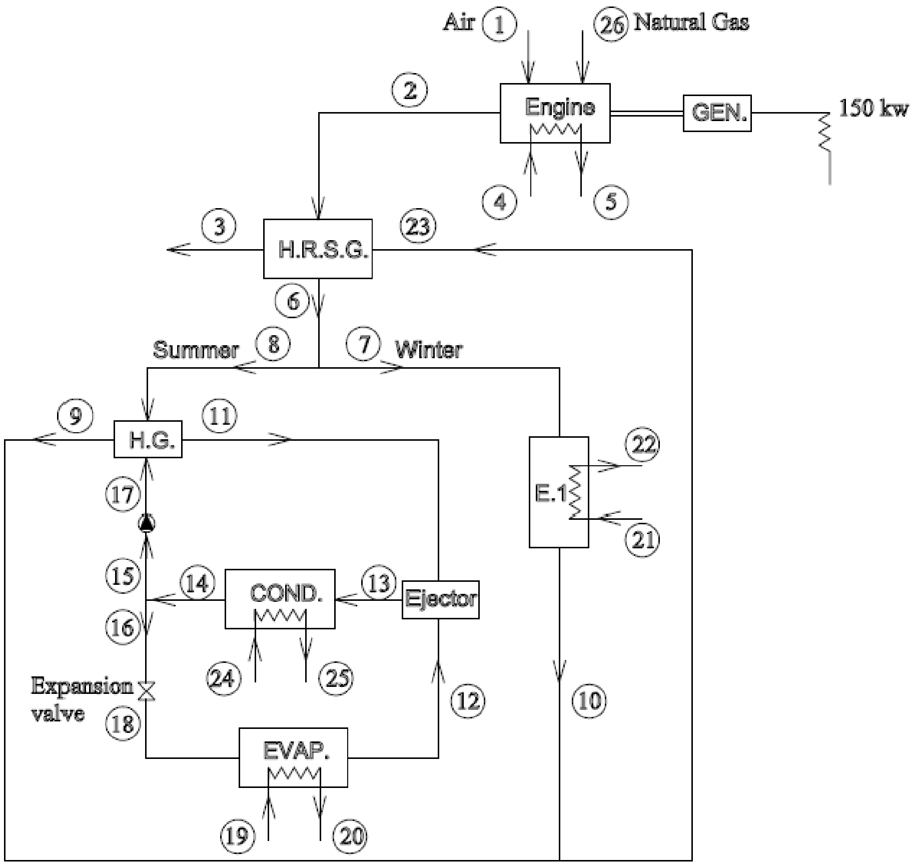

Fig. 1. Simplified schematic view of a simultaneous power, cooling and heating energy generating system driven by an internal combustion engine and a steam ejector refrigeration system.

where,

$$\bar{S}_i^* = \bar{S}_{T_i}^{\circ} - \bar{R}\, In \frac{y_i P}{P_\circ} \tag{6}$$

In the above equations, $\bar{S}_i^*$ is the specific entropy of component i in the mix, y_i is the component mole fraction, P is total pressure, and $\bar{S}_{T_i}^{\circ}$ is the specific entropy at the given temperature and base pressure. Using Eqs. (5) and (6), the following equation can be obtained [11]:

$$\begin{aligned}\bar{S}_2 = {} & y_{CO_2}(\bar{S}_{T_2}^{\circ} - \bar{R}\, In \frac{y_{CO_2} P_2}{P_o}) + y_{H_2O}(\bar{S}_{T_2}^{\circ} \\ & - \bar{R}\, In \frac{y_{H_2O} P_2}{P_o}) + y_{O_2}(\bar{S}_{T_2}^{\circ} - \bar{R}\, In \frac{y_{O_2} P_2}{P_o}) \\ & + y_{N_2}(\bar{S}_{T_2}^{\circ} - \bar{R}\, In \frac{y_{N_2} P_2}{P_o})\end{aligned} \tag{7}$$

3.2. Heat recovery steam generator

As shown in Fig. 2, a heat recovery steam generator consists of three major parts: an economizer, an evaporator and a superheater. Water enters the economizer where its temperature is increased to the saturation point. The water then enters the evaporator, where it is converted into saturated vapor.

To find the pinch point, (temperature difference of the inlet gas and outlet water from the economizer) the following equation can be written:

$$T_{2''} = T_{23'} + \Delta Pinch \tag{8}$$

where the subscript indicates various parts of Fig. 2. To calculate $\dot{m}_{23}$, the first law is written for the economizer [9],

$$\dot{m}_2(h_{2''} - h_3) \times \eta_{HRSG} = \dot{m}_{23}(h_{23'} - h_{23}) \tag{9}$$

In Eq. (9), $h_{2''}$ can be readily found based on the previous equation with having T_2. To find h_6 and the temperature at that point, the first law should be written for the superheater.

Temperature of point $2'$ is found based on a trial and error. In this way, first a temperature is guessed for point $2'$ (a temperature between 2 and $2'$) then like the previous method (Eq. (4)), $h_{2'N}$ is calculated at this temperature. Now, the calculated result should be compared with $h_{2'}$ which was obtained from the general equation for the evaporator (Eq. 10).

$$h_{2'} = \dot{m}_{23} \frac{(h_{23''} - h_{23'})}{\dot{m}_2 \eta_{HRSG}} + h_{2''} \tag{10}$$

If the difference is reasonable, then,

$$h_{2'N} = h_{2'} \tag{11}$$

3.3. Heat exchanger No. 1

This heat exchanger is used only in winter. For analysis, writing the first law for heat exchanger No. 1 (E.1) is required [9],

$$\dot{m}_7(h_7 - h_{10})\eta_{E.1} = \dot{m}_{21}(h_{22} - h_{21}) \tag{12}$$

3.4. Analysis of steam ejector refrigeration system

Entrainment Ratio is the most important factor which is used to evaluate a steam ejector system (Fig. 3).

To analyze this system, which works only in the summer, first, the heat generator is examined. To find the mass flow rate, the enthalpy of the input and output to / from the heat generator are required. Knowing that the given conditions for point 8 are quite similar to those of point 6, conditions of point 8 are defined and, because the the conditions of points 9 and 23 are quite similar, the conditions of point 9 must be known. Assuming that pressure is 3bar for the heat generator, the enthalpy of point 11 which is the enthalpy of the saturated steam is computed. To find enthalpy of point 17, the pump is analyzed. By writing the first law for the pump, enthalpy of point 17 is calculated as follows [9]:

$$\left.\begin{aligned}\dot{Q} - \dot{W} &= \dot{m}(h_{17} - h_{15}) \\ \dot{W} &= \dot{m}(P_{Enter} - P_{Exit})v\end{aligned}\right\} \rightarrow h_{17} = h_{15} + (P_{Exit} - P_{Enter})v \tag{13}$$

It should be noted that, P_{Enter} is the pressure of point 15 (P_{15}), which is equal to the saturation pressure at the condenser outlet temperature. Also,

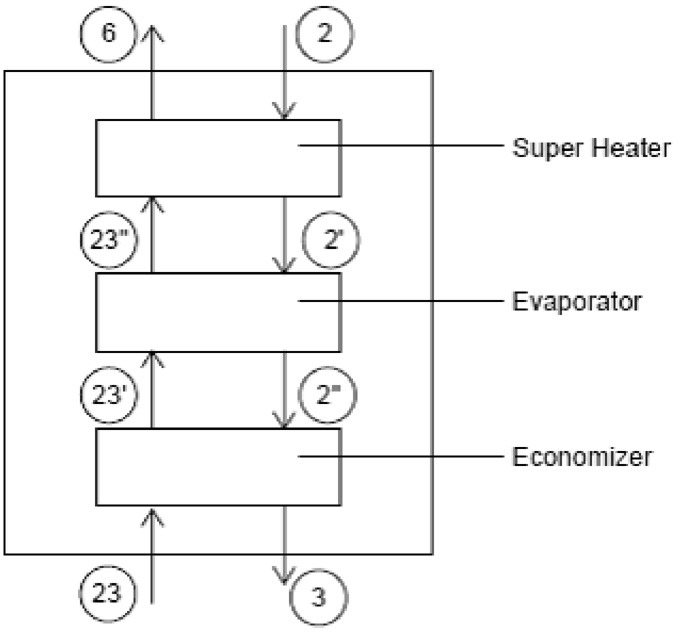

Fig. 2. Schematic view of various parts of heat recovery steam generator.

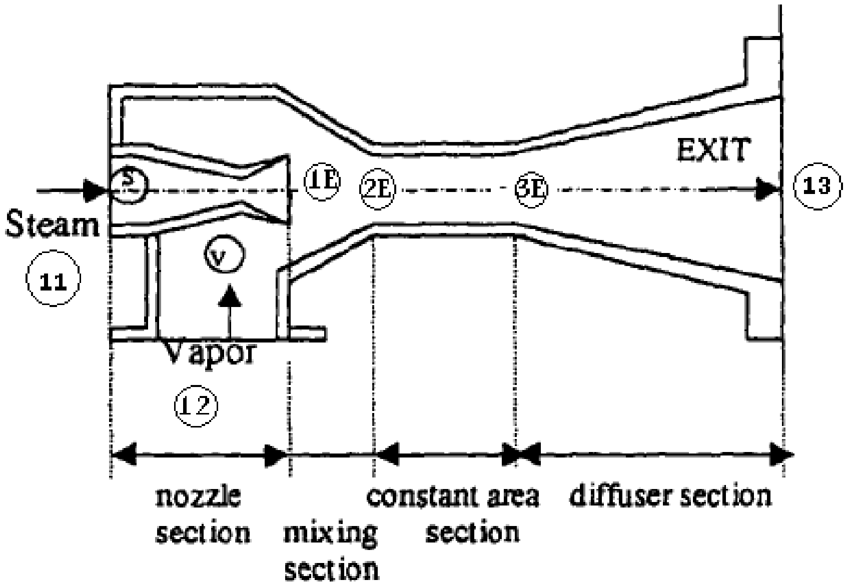

Fig. 3. Schematic view of an ejector.

$\dot{W}$ is the rate of work, and $\dot{Q}$ is the rate of heat transfer. Now, by writing the fist law for the heat generator (H.G), $\dot{m}_{17}$ can be found [11]:

$$\dot{m}_8(h_8 - h_9) = \dot{m}_{17}(h_{11} - h_{17}) \tag{14}$$

The conditions for points 16 and 14 are also quite similar. Enthalpy is constant in the expansion valve, and the temperature of the evaporator is known. So the pressure of point 18 can be found by knowing the saturation pressure at point 18.

Considering these information, the following equations are used to analyze the ejector [9,11,12]:

$$x_{1E} = \frac{S_{11} - S_{f\ @\ P=Pevap}}{S_{fg\ @\ P=Pevap}} \tag{15}$$

$$h_{1ES} = h_{f\ @\ P=Pevap} + x_{1E} h_{fg\ @\ P=Pevap} \tag{16}$$

$$\frac{V_{1E}^{2}}{2}=\eta_{n}(h_{11}-h_{1ES}) \tag{17}$$

$$h_{1E}=h_{11}-\eta_{n}(h_{11}-h_{1ES}) \tag{18}$$

Considering that h_{1ES}, h_{1E}, and V_{1E} are found, the ratio of the mass flow rate for point 12 to that for point 11 can be estimated, and is denoted W [11],

$$W=\frac{\dot{m}_{12}}{\dot{m}_{11}} \tag{19}$$

It should be mentioned here that, $\dot{m}_{11}$ is obtained from the present analysis for the heat generator. Using the following equation, it is concluded that, for V_{2E} [11],

$$(\dot{m}_{11}+\dot{m}_{12})V_{2E}=\eta_{m}\dot{m}_{11}V_{1E} \tag{20}$$

Using the following equation, h_{2E} can be obtained [11]:

$$(\dot{m}_{11}+\dot{m}_{12})\left[h_{2E}+\frac{V_{2E}^{2}}{2}\right]=\dot{m}_{12}h_{12}+\dot{m}_{11}\left[h_{1E}+\frac{V_{1E}^{2}}{2}\right] \tag{21}$$

Now, the speed of the sound is calculated [11]:

$$C_{2E}=\sqrt{K_{2E}R_{2E}T_{2E}} \tag{22}$$

If $V_{2E}>C_{2E}$, the shock will occur (this generally develops in the ejector). Therefore, from [12],

$$x_{2E}=\frac{h_{2E}-h_{f\,@\,P=Pevap}}{h_{fg\,@\,P=Pevap}} \tag{23}$$

$$Ma_{2E}=\frac{V_{2E}}{C_{2E}} \tag{24}$$

$$Ma_{3E}^{2}=\frac{Ma_{2E}^{2}+\frac{2}{k_{3E}-1}}{\frac{2Ma_{2E}^{2}\times K_{3E}}{K_{3E}-1}-1} \tag{25}$$

Now, P_{3E} is obtained through the following equation [12]:

$$\frac{P_{3E}}{P_{2E}}=\frac{1+K_{2E}Ma_{2E}^{2}}{1+K_{3E}Ma_{3E}^{2}} \tag{26}$$

where T_{3E} is also obtained through Eq. (27) [12],

$$\frac{T_{3E}}{T_{2E}}=\frac{P_{3E}V_{3E}}{P_{2E}V_{2E}} \tag{27}$$

and V_{3E} is obtained through Eq. (28) [12],

$$\frac{T_{3E}}{T_{2E}}=\left(\frac{P_{3E}}{P_{2E}}\right)^{2}\left(\frac{Ma_{3E}}{Ma_{2E}}\right)^{2} \tag{28}$$

C_{3E} can be obtained by Eq. (29) [11, 12],

$$Ma_{3E}=\frac{V_{3E}}{C_{3E}} \tag{29}$$

Finally, after using the above equations, enthalpy of point 13 is determined as [11],

$$h_{13}=h_{3E}+\frac{V^{2}_{3E}}{2\eta_{d}} \tag{30}$$

Considering the assumptions of most references, entropy of point 13 is equal to the entropy of point C_{3E}. Now, having two properties for point 13 (entropy and enthalpy), pressure for point 13 is obtained. Since the pressure drop in the condenser is assumed to be zero, the pressure for point 13 is equal to that of point 14. Now, the obtained pressure in the ejector is compared with the pressure of point 14. If there are obvious differences, the initial value for ($\dot{m}_{12}/\dot{m}_{11}$) should be changed and the procedure should be repeated. Therefore, enthalpy, entropy, and the mass flow rate of all components of the cycle are now found. A program has been written for the above processes and for all the steps of the repetitions, using the trial and error method.

4. Thermodynamic parameters performance evaluations

The standards used to evaluate the performance of the cogeneration systems were studied by X.

Feng in 1998. The most important of them which are usually used to evaluate performance of co-generation systems are [13,14,15]:

1. Energy utilization factor (EUF)
2. Fuel energy saving ratio (FESR)
3. Coefficient of performance (COP)
4. Power to heat ratio (PHR)

These standards are expressed as the following [13,14,15]:

Energy utilization factor for winter is,

$$EUF_{chp} = \frac{\dot{W}_{net} + \dot{Q}_{E.1} + \dot{Q}_{Jacket}}{\dot{Q}_{input}} \quad (31)$$

Energy utilization factor for summer is,

$$EUF_{ccp} = \frac{\dot{W}_{net} + \dot{Q}_{evaperator} + \dot{Q}_{Jackat}}{\dot{Q}_{input}} \quad (32)$$

Fuel energy saving ratio in winter is,

$$FESR_{chp} = 1 - \frac{\dot{Q}_{input}}{\frac{\dot{Q}_{E.1}}{\eta_{bo}} + \frac{\dot{Q}_{Jackat}}{\eta_{Jackat}} + \frac{\dot{W}_{net}}{\eta_{otto}}} \quad (33)$$

Fuel energy saving ratio in summer is,

$$FESR_{ccp} = 1 - \frac{\dot{Q}_{input}}{\frac{\dot{Q}_{Evaperator}}{\eta_{bo}} + \frac{\dot{Q}_{Jackat}}{\eta_{Jackat}} + \frac{\dot{W}_{net}}{\eta_{otto}}} \quad (34)$$

Coefficient of performance is,

$$CO\,Pr\,ef = \frac{\dot{Q}_{Evaperator}}{\dot{Q}_{Generator}} \quad (35)$$

Power to heat ratio for summer is,

$$PHR_{ccp} = \frac{\dot{W}_{net}}{\dot{Q}_{Jacket} + \dot{Q}_{evaporator}} \quad (36)$$

Power to heat ratio for winter is,

$$PHR_{chp} = \frac{\dot{W}_{net}}{\dot{Q}_{E.1} + \dot{Q}_{Jacket}} \quad (37)$$

5. Results and discussion

Using the above equations, the following results are achieved through the mentioned computer code:

Fig. 4 shows the effects of the pinch point temperature variations on energy utilization factor in summer, at temperature 12 °C for the evaporator and 45 °C for the condenser. As it can be seen, by decreasing the pinch point temperature, the energy utilization factor is reduced.

Fig. 5 shows the effects of the pinch point temperature variations on energy utilization factor in winter at temperature 12 °C for the evaporator and 45 °C for the condenser. As it can be seen, by decreasing the pinch point temperature, the energy utilization factor also decreases in winter.

Fig. 6 shows the effects of the pinch point temperature variations on energy utilization factor in summer, at condenser temperatures of 42.5, 45, 50, and 52.5 °C for an evaporator temperature of 12 °C.

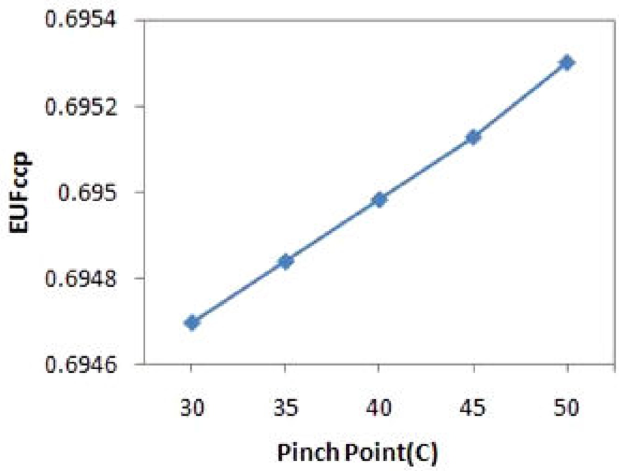

Fig. 4. Effects of pinch point temperature variations on EUFccp for summer.

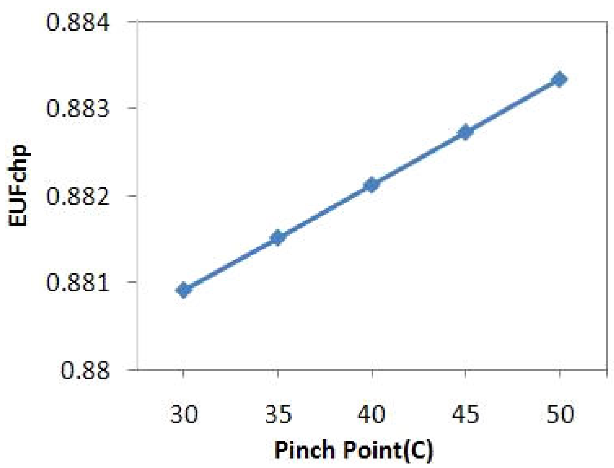

Fig. 5. Effects of pinch point temperature variations on EUFchp for winter.

As it can be seen, by decreasing the condenser temperature, the energy utilization factor increases. Also, by decreasing the pinch point temperature, (For a fixed condenser temperature), the energy utilization factor decreases**.**

Fig. 7 shows the effects of the pinch point temperature changes on fuel energy saving factor in winter. As it can be seen, by decreasing the pinch point temperature, the fuel energy saving factor decreases.

Fig. 8 shows the effects of the pinch point temperature changes on fuel energy saving factor in summer at evaporator temperatures of 12, 14, 16 and 18 °C for a condenser temperature of 45 °C. As it can be seen, decreasing the evaporator temperature causes a decrease in the fuel energy saving factor. Also, decreasing the pinch point temperature, (for a fixed evaporator temperature), causes some small decrease in the fuel energy saving factor.

Fig. 9 shows the effects of heat generator pressure changes on the COPref for different evaporator temperatures in summer. As it can be seen, reducing the evaporator temperature, causes the COP, for a fixed heat generator pressure, to decrease.

Additionally, for a specific evaporator temperature, an increase in the pressure of the heat generator, causes an increase in the COP.

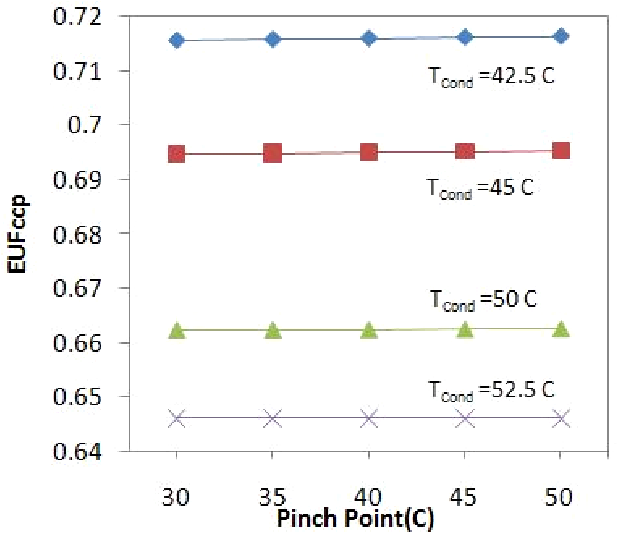

Fig. 6. Effects of pinch point temperature on EUFccp for different condenser temperatures for summer.

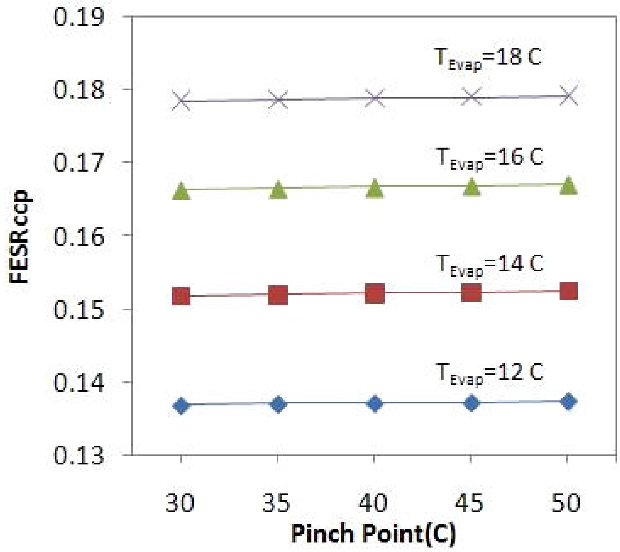

Fig. 8. Effects of pinch point temperature on FESRccp for different evaporator temperatures for winter.

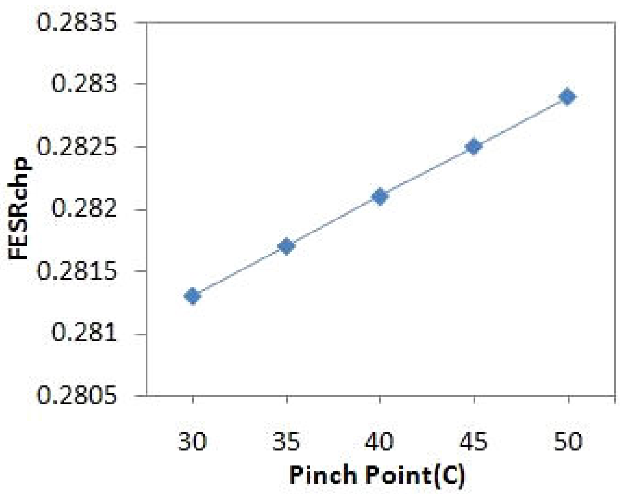

Fig. 7. Effects of pinch point temperature variations on FESRchp for winter.

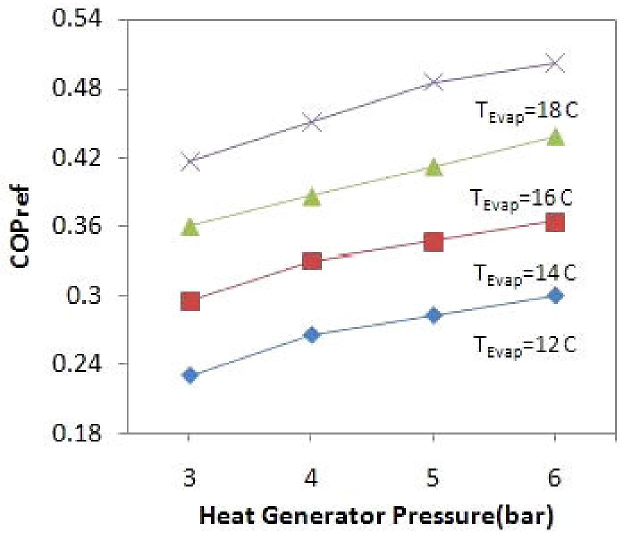

Fig. 9. Effects of generator pressure variations on COPref for different evaporator temperatures.

In Figs. 10 and 11 the influence of the pinch point temperature on power to heat ratio in winter and summer, respectively are examined. Both figures show that with increasing the pinch point temperature, the ratio is reduced.

Fig. 12 shows the influence of heat generator pressure changes on energy utilization factor at the condenser temperature of 45 °C in summer. As it can be seen, decreasing the pressure of the heat generator, causes the energy utilization factor to decrease.
Fig. 13, which is taken from reference 15, shows that an increase in the heat generator pressure causes an increase in the energy utilization factor and the coefficient of performance.

By comparing the results of the present study shown in Figs. 9 and 12 with those of Fig. 13 [15], the same trends of the appropriate curves show the validity of the obtained results in this work.

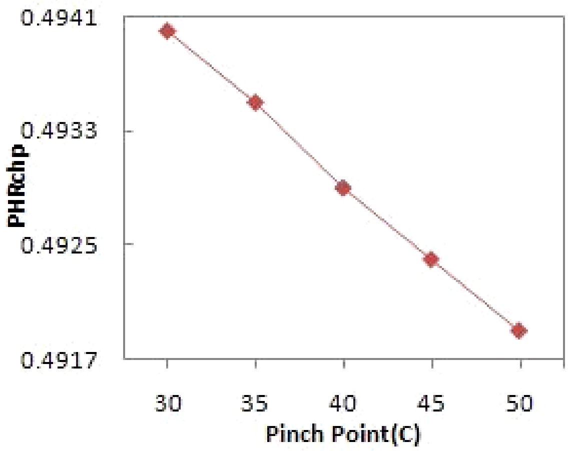

Fig. 10. Effects of pinch point temperature variations on PHRchp for winter.

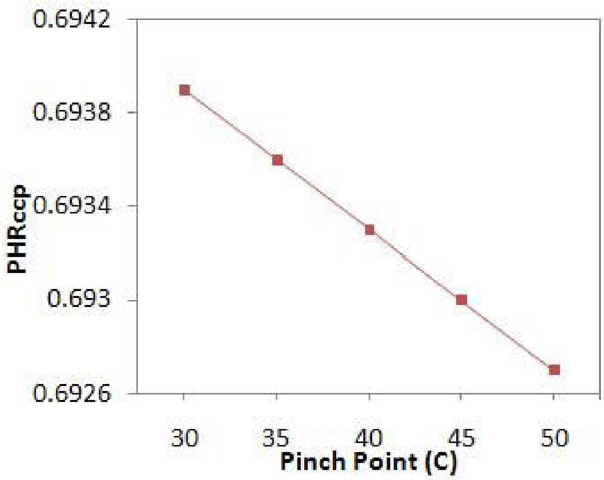

Fig. 11. Effects of pinch point temperature variations on PHRccp for summer.

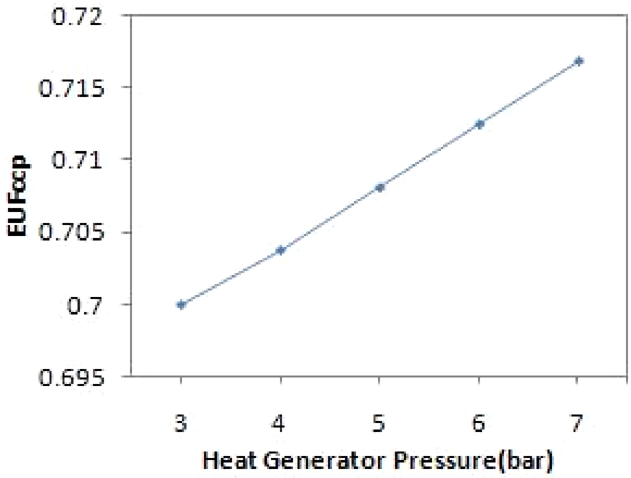

Fig. 12. Effects of generator pressure variations on EUFccp.

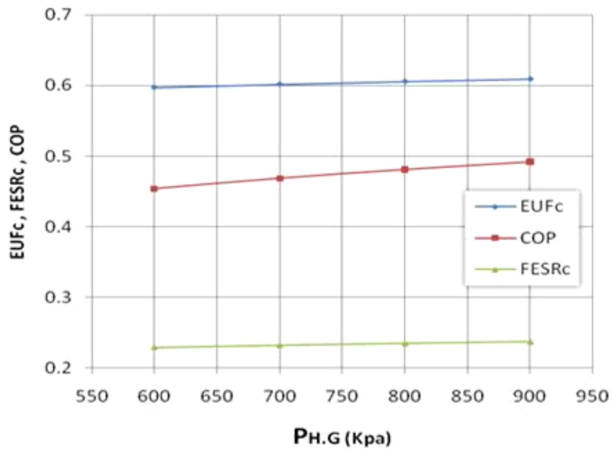

Fig. 13. Effects of heat generator pressure of ejector on total efficiency of cycle, fuel energy saving ratio (in generating power and cooling modes), and coefficient of steam ejector refrigeration system performance [15].

6. Conclusions

In this study a thermodynamic cycle capable of simultaneously generating cooling and heating energies and supplying power is designed. The first law of thermodynamics was implemented to analyze each component of the cycle thoroughly. A parametric study was executed to obtain the cycle efficiency through changing the pinch point, evaporator, heat generator, and condenser temperatures for both the winter and summer seasons.

Based on the obtained results, optimizing the efficiency of the cycle would result in an increase in the energy utilization efficiency up to 88% in winter (Fig. 5) and 71% in summer (Fig. 6) seasons. It should be mentioned that if the cooling or heating system works individually (not simultaneously), this energy utilization efficiency would be reduced to about 30%. The results of the analyses also indicate 28% fuel efficiency in winter (Fig. 7) and an 18% in summer (Fig. 8) seasons.

7. Acknowledgements

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Nomenclature

A	Condition A
B	Condition B
c	Velocity of sound (m/s)
CCHP	Combined cooling, heating and power generation
E.1	Heat exchanger number 1
E.V	Expansion valve
h	Specific enthalpy (kJ/kg)
$\Delta\bar{h}$	Difference between reference and specified enthalpy (kJ/kg)
$\bar{h}_f^\circ$	Saturated liquid enthalpy (kJ/kg)
H.G	Heat generator
HRSG	Heat recovery steam generator
h_g	Saturated vapor enthalpy (kJ/kg)
h_m	Enthalpy of mixture (kJ/kg)
m	Mass (kg)
M	Molecular weight (kg/Kmol)
$\dot{m}$	Mass flow rate (kg/s)
M_a	Mach number
n	Number of mole
n_v	Number of condensate mole (Kmol)
NG	Natural gas
p	Pressure (bar)
Q	Heat transfer (kJ)
$\bar{R}$	Gas constant (kJ/kg k)
s	Specific entropy (kJ/kg k)
$\bar{s}^*$	Specific entropy of a mixture constituents (KJ/kg k)
$\bar{s}_{T_i}^\circ$	Specific entropy at a given temperature and base pressure (kJ/kg k)
T	Temperature (C)
V	Velocity (m/s)
W	Work (kJ)
x	Quality
y	Mole fraction
η	Efficiency (%)

Subscripts

ccp	Combined cooling & power generation
chp	Combined heating & power generation
Cond.	condenser
d	diffuser
E	Ejector location
E.1	Heat exchanger number 1
Evap.	Evaporator
f	Saturated liquid
fg	Difference between Saturated liquid and vapor
g	Saturated gas
H.G	Heat generator
HRSG	Heat recovery steam generator
i	i component
m	Mixture
n	Nozzle
P	Product
q	Heat transfer
$\dot{Q}$	Rate of heat transfer
Ref	Refrigeration
s/is	Isentropic
T	Temperature
w	Output work
1,2,3,…	Cycle location
2′,...	Cycle location
2″,...	Cycle location

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