Exergy Analysis of a Trigeneration System Driven by an Internal Combustion Engine with a Steam Ejector Refrigeration System

M. Najafi*
Department of Mechanical and Aerospace Engineering, Science and Research Branch, Islamic Azad University, Tehran, Iran
E-mail: M.njafi36@gmail.com
*Corresponding author

K. Javaherdeh
Department of Mechanical Engineering, University of Gilan, Iran
E-mail: Javaherdeh@guilan.ac.ir

B. Liravinia
Department of Mechanical and Aerospace Engineering, Science and Research Branch, Islamic Azad University, Tehran, Iran
E-mail: Behrouz_Liravi@yahoo.com

Received: 11 December 2011, Revised: 27 June 2012, Accepted: 30 June 2012

Abstract: The present study analyzes a trigeneration system based on an internal combustion engine and a steam ejector refrigeration system from an exergy point of view. The present system is to produce cooling energy, heating energy and to supply power simultaneously. The analyses include a thorough implementation of the first law of thermodynamics and exergy formulations on each and every component of the designed cycle. Having computed the system exergy, the system exergy destruction sources are determined for both the summer and winter seasons. Based on the exergy analyses, about 360 kw by the internal combustion engine and about 70 kw by the steam generator are found to be the main exergy destruction sources in the cycle. Furthermore the heat generator and heat exchanger within the proposed cycle show to produce the maximum and minimum amount of exergy, respectively.

Keywords: Exergy Analysis, Gas Fired Engine, Steam Ejector Refrigeration System, Trigeneration


Biographical notes: M. Najafi received his PhD in mechanical engineering from the University of Alabama in 1989. His current research interests include energy storage systems, cooling systems, heating systems, mesh less numerical works in heat transfer and fluid flow in thermodynamic analysis. K. Javaherdeh received his PhD in mechanical engineering from University of Nancy in 1996. He is currently an assistant professor in the department of mechanical engineering, in Gilan University, Gilan, Iran. His current research interests include energy storage systems, cooling systems, design of heat exchangers, and optimization in thermodynamic analysis. B. Liravinia has received his master in mechanical engineering, from Science and Research Branch, Azad University, Tehran, Iran. His current research focuses on, thermodynamics and heat transfer.

© 2012 IAU, Majlesi Branch
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 point of views. 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 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 cogeneration systems for a building in which they investigated the effects of certain operating parameters such as steam pressure, and water temperature on the 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 performed an exergy analysis to examine the system efficiency. In this research a parameters study on the power and refrigeration output of the steam turbine and the ejector refrigeration system was carried out [6].

Cardona and Piacentino investigated for an optimal design of CCHP system using thermodynamic analysis to be implemented in buildings [7].

Cogeneration system is not a new concept, but coupling it with a gas-fired engine beside a steam ejector refrigeration system has not been investigated from exergy point of view for both summer and winter seasons. This study presents an optimized cogeneration cycle in order to meet the needs for heating, cooling and power in summer and winter. In addition, the effects of some parameters affecting system performance are discussed.

2 CYCLE DESCRIPTIONS

A schematic view of the designed cogeneration heating, cooling, and power 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 by the working engine. It should be mentioned that in summer, a small amount of power 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 seasons, is divided into two parts. Path 7 (Fig. 1) is considered for the winter and path 8 (Fig. 1) for the summer.

In summer, steam enters the steam ejector refrigeration system and the required cooling is generated. In winter, the produced steam in the heat recovery steam generator enters the heat exchanger (heat exchanger No. 1 shown in Fig. 1) and the required heating is generated. To provide 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

Fig. 1  Simplified schematic view of a simultaneous energy generation system for cooling, heating and power, based on an internal combustion engine and a steam ejector refrigeration system
pressure is decreased at the outlet of the nozzle and this pressure drop forces 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 a diffuser 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 portion of the condensed water, after passing through an expansion valve enters the evaporator to absorb the heat from the environment, and the cooling cycle is repeated again.

3 THERMODYNAMIC CALCULATIONS OF CYCLE COMPONENTS

Assumptions used in this study are presented in Table 1.

<table>
<thead>
<tr>
<th>Table 1 Required assumptions for investigating performance of the cycle</th>
</tr>
</thead>
<tbody>
<tr>
<td>Heat transfer from the engine to the environment</td>
</tr>
<tr>
<td>environment air conditions in summer</td>
</tr>
<tr>
<td>environment air conditions in winter</td>
</tr>
<tr>
<td>Output power required from the engine</td>
</tr>
<tr>
<td>Input energy to the engine</td>
</tr>
<tr>
<td>Outlet gas Temperature from the engine</td>
</tr>
<tr>
<td>Compression ratio of the engine</td>
</tr>
<tr>
<td>Heat transfer to the engine casing</td>
</tr>
<tr>
<td>Evaporator temperature</td>
</tr>
<tr>
<td>Heat generator pressure</td>
</tr>
<tr>
<td>Condenser temperature</td>
</tr>
<tr>
<td>HRSG pressure</td>
</tr>
<tr>
<td>Efficiency of the ejector mixing part</td>
</tr>
<tr>
<td>Efficiency of the ejector nozzle</td>
</tr>
<tr>
<td>Efficiency of the ejector diffuser</td>
</tr>
</tbody>
</table>

3.1 GAS FIRED ENGINE – GENERATOR

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

\[
CH_4 + a(0.2059 \ O_2 + 0.7748 \ N_2 + \\
+ 0.0003 \ CO_2 + 0.019 \ H_2 O(g)) \rightarrow \ (1 + 0.0003 \ a)CO_2 + (2 + 0.019 \ a)H_2 O(g) \\
+ (a - 2)O_2 + 0.7748a \ N_2.
\]

Variable \( a \) determines the amount of excess air in Eq. (1). It is necessary to write the first law for the engine to determine variable \( a \):

\[
Q_{c,v} - W_{c,v} = nCO_2(\bar{h}_f + \Delta \bar{h})CO_2 + nH_2 O(g) \\
(\bar{h}_f + \Delta \bar{h})H_2 O(g) + nO_2(\bar{h}_f + \Delta \bar{h})O_2 \\
+ nN_2(\bar{h}_f + \Delta \bar{h})N_2 - nCH_4(\bar{h}_f + \Delta \bar{h})CH_4 \\
- nO_2(\bar{h}_f + \Delta \bar{h})O_2 - nN_2(\bar{h}_f + \Delta \bar{h})N_2 \\
- nCO_2(\bar{h}_f + \Delta \bar{h})CO_2 \\
- nH_2 O(\bar{h}_f + \Delta \bar{h})H_2 O.
\]

In Eq. (2) \( \bar{h}_f \) and \( \Delta \bar{h} \) are the formation enthalpy and the enthalpy difference between each mode assessed at the base conditions. Also, \( n \) is the number of moles. By finding the formation enthalpy \( (\bar{h}_f) \) and enthalpy difference \( (\Delta \bar{h}) \) using tables of thermodynamics, the unknown coefficients of reaction \( (a) \) will be determined.

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

\[
m_{tot} \frac{h}{tot} = m_A \frac{h}{A} + m_B \frac{h}{B} \rightarrow \frac{h}{tot} = \frac{m_A}{m_{tot}} \frac{h}{A} + \frac{m_B}{m_{tot}} \frac{h}{B}.
\]

Thus, for these two points, it can be written that,

\[
h_2 = \frac{1}{m_2}(m_{CO_2} \ \frac{h}{CO_2} + m_{H_2 O} \ \frac{h}{H_2 O} + m_{N_2} \ \frac{h}{N_2} + m_{o_2} \ \frac{h}{o_2}).
\]
where \( m \) is the mass. In Eq. (4), enthalpy of point 2 is calculated at the engine exhaust temperature. Now to calculate entropy, the gas mixture equation is needed [11]:

\[
\overline{S}_{\text{mix}} = \sum_i y_i \overline{S}_i^* \tag{5}
\]

\[
\overline{S}_i^* = \overline{S}_i^o - R \ln \frac{y_i P}{P_o} \tag{6}
\]

In the above equation, \( \overline{S}_i^* \) is the specific entropy of component \( i \) in the mixture, \( y_i \) is the component mole fraction, \( P \) is total pressure, and \( \overline{S}_i^o \) is the specific entropy at the given temperature and base pressure. Using Eq. (5) and (6), the following equation can be stated:

\[
\overline{S}_2 = y_{CO_2} \left( \overline{S}_2^o - R \ln \frac{y_{CO_2} P_2}{P_o} \right) + y_{H_2O} \left( \overline{S}_2^o - R \ln \frac{y_{H_2O} P_2}{P_o} \right)
\]

\[
+ y_{O_2} \left( \overline{S}_2^o - R \ln \frac{y_{O_2} P_2}{P_o} \right) + y_{N_2} \left( \overline{S}_2^o - R \ln \frac{y_{N_2} P_2}{P_o} \right). \tag{7}
\]

In the above equation, \( S \) is the specific entropy, \( R \) is the gas constant, subscripts \( o \) show environment conditions, and subscripts 1, 2, … show points of cycle.

### 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 super-heater. Water enters the economizer and its temperature is increased to saturation temperature; water then enters an evaporator and is converted into saturated vapor, and then, to enter a super-heater it is converted into a superheated steam. Having the pinch point, (temperature difference of the inlet gas and outlet water from the economizer) the following equation can be written:

\[
T_{2^*} = T_{23^*} + T_{\Delta \text{Pinch}}, \tag{8}
\]

where the subscripts indicate temperatures measured for various parts of Fig. 2. To calculate \( \dot{m}_{23} \), the first law should be written for the economizer:

\[
\dot{m}_{23}(h_{2^*} - h_{3}) \times \eta_{\text{HRSG}} = \dot{m}_{23}(h_{23^*} - h_{23}) \tag{9}
\]

\[
\dot{m}_{23} \left( h_{2^*} - h_{3} \right) \times \eta_{\text{HRSG}} = \dot{m}_{23} \left( h_{23^*} - h_{23} \right)
\]

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

Where \( \eta_{\text{HRSG}} \) is the efficiency of the heat recovery steam generator and \( \dot{m} \) is the mass flow rate of different components in the cycle. In Eq. (9), \( h_{2^*} \) can be found based on the previous equation providing its temperature is known. To find \( h_6^* \) and its corresponding temperature, the first law should be utilized for the super-heater.

Temperature of point \( 2^* \) is found based on trial and error. Such that an initial temperature is guessed for point \( 2^* \) (a temperature between \( 2^\prime \) and \( 2^\prime\prime \)), then like the previous method (Eq. (4).), \( h_{2^*} \) is calculated at this temperature. Now, the calculated results should be compared with \( h_{2^*} \) which was obtained from the general equation for the evaporator (Eq. (10).)

\[
h_{2^*} = \dot{m}_{23} \left( h_{23^*} - h_{23} \right) \tag{10}
\]

\[
\dot{m}_{23} \left( h_{23^*} - h_{23} \right) + h_{2^*}
\]

If the difference is reasonable, then:

\[
h_{2^*} \eta_{\text{HRSG}} = h_{2^*} \tag{11}
\]
3.3 HEAT EXCHANGER NO. 1 (E.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]:

\[ m_7 (h_7 - h_{10}) \eta_{E.1} = m_{21} (h_{22} - h_{21}) \]  \hspace{1cm} (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 of point 8 is quite similar to those of point 6, conditions of point 8 are defined and because 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]:

\[ \dot{Q} - \dot{W} = m (P_{\text{Enter}} - P_{\text{Exit}}) v \rightarrow h_17 = h_{15} + \left( P_{\text{Exit}} - P_{\text{Enter}} \right) v \]  \hspace{1cm} (13)

It should be noted that, \( P_{\text{Enter}} \) is the pressure of point 15 \( (P_{15}) \) which is equal to the saturation pressure at the condenser outlet temperature, \( \dot{W} \) is rate of work, and \( \dot{Q} \) is rate of heat transfer. Now, by writing the first law for the heat generator (H.G), \( m_{17} \) can be found [11]:

\[ m_8 (h_8 - h_9) = m_{17} (h_{11} - h_{17}) \]  \hspace{1cm} (14)

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

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

\[ x_{1E} = \frac{S_{11} - S_f @ P = \text{Pevap}}{S_f g @ P = \text{Pevap}} \]  \hspace{1cm} (15)

\[ h_{1ES} = h_f @ P = \text{Pevap} + x_{1E} h_f g @ P = \text{Pevap} \]  \hspace{1cm} (16)

\[ \frac{V_{1E}^2}{2} = \eta_n (h_{11} - h_{1ES}) \]  \hspace{1cm} (17)

\[ h_{1E} = h_{11} - \eta_n (h_{11} - h_{1ES}) \]  \hspace{1cm} (18)

where the subscript s shows isentropic states, \( V \) stands for speed, \( x \) stands for quality, subscript \( f \) shows saturated liquid, subscript \( g \) shows saturated gas, subscript \( fg \) shows the difference between saturated liquid and vapor, subscript \( n \) shows nozzle condition, subscript \( evap \) shows evaporator condition, and subscript E stands for ejector. Having found \( h_{1ES} \), \( h_{1E} \) and \( V_{1E} \), the ratio of the mass flow rate belonging to point 12 to the mass flow rate belonging to point 11, denoted by ‘W’, follows[11]:

\[ W = \frac{m_{12}}{m_{11}} \]  \hspace{1cm} (19)

Fig. 3. Schematic view of an ejector
It should be mentioned here that, $m_{11}$ is obtained from the present analysis for the heat generator. Using the following equation, it is concluded that, for $V_{2E}$ [11]:

$$ (m_{11} + m_{12})V_{2E} = \eta m_{11}V_{1E}^2, \quad (20) $$

where subscript $m$ indicates mixture. Using the following equation $h_{2E}$ is obtained [11]:

$$ (m_{11} + m_{12}) \left[ h_{2E} + \frac{V_{2E}^2}{2} \right] = m_{12} h_{12} + m_{11} h_{1E} + \frac{V_{1E}^2}{2}, \quad (21) $$

Now, the speed of sound should be calculated [11]:

$$ C_{2E} = \sqrt{K_{2E}R_{2E}T_{2E}}, \quad (22) $$

where $C$ is the velocity of sound. If $V_{2E} > C_{2E}$, shock will occur (this generally develops in the ejector). Therefore, from [12]:

$$ x_{2E} = \frac{h_{2E} - h_{fg} @ P = P_{evap}}{h_{fg} @ P = P_{evap}}, \quad (23) $$

$$ Ma_{2E} = \frac{V_{2E}}{C_{2E}}, \quad (24) $$

$$ Ma_{2E}^2 = \frac{2}{k_{3E} - 1} - \frac{2 + \frac{2}{Ma_{2E}^2} - 1}{2Ma_{2E}^2 + k_{3E} - 1}, \quad (25) $$

where $Ma$ is the Mach number.

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

$$ P_{3E} = \frac{1 + K_{2E}Ma_{2E}^2}{1 + K_{3E}Ma_{3E}^2}, \quad (26) $$

$$ P_{2E} = \frac{1}{1 + K_{3E}Ma_{3E}^2}, \quad (27) $$

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

$$ T_{3E} = \frac{P_{3E}V_{3E}^2}{P_{2E}V_{2E}^2}, \quad (28) $$

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

$$ T_{3E} = \left( \frac{P_{3E}}{P_{2E}} \right)^{\frac{2}{Ma_{3E}}} \cdot \frac{Ma_{3E}^2}{Ma_{2E}^2}, \quad (29) $$

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

$$ Ma_{3E} = \frac{V_{3E}}{C_{3E}}, \quad (29) $$

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

$$ h_{13} = h_{3E} + \frac{V_{3E}^2}{2\eta_d}, \quad (30) $$

where subscript $d$ stands for the diffuser. 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 ($\frac{m_{12}}{m_{11}}$) should be changed and the procedure should be repeated.

Thus, enthalpy, entropy and mass flow rate for all components of the cycle are hereby found. A program has been written in MATLAB software for the above process and for all iteration steps, using trial and error method.

### 4 Exergy Analyses

Equation (31) shows the total exergy for each component of the cycle, where $e_k$, $e_p$, $e_{ch}$ and $e_{ph}$ represent the kinetic, potential, chemical, and physical exergies, respectively.
where \( e \) stands for specific exergy. For our purposes, in the above equation both the kinetic and potential exergies can be neglected, and physical and chemical exergies can be found as follows [10, 16]:

\[
e = e_{ph} + e_{ch} + e_p + e_k,
\]

(31)

where \( e \) stands for specific exergy. For our purposes, in the above equation both the kinetic and potential exergies can be neglected, and physical and chemical exergies can be found as follows [10, 16]:

\[
e_{ph} = (h - h_o) - T_o (S - S_o),
\]

(32)

\[
\bar{\tau}_{ch, mix} = \sum y_k \tau_{ch,k} + \partial T_o \sum y_k \ln y_k.
\]

(33a)

\[
\tau_{ch,k} = -R T_o \sum y_k \ln y_k.
\]

(33b)

\[
e_{ch, fuelGas} = LHV \left[ 1.033 + 0.0169 \frac{b}{a} - 0.0698 \right],
\]

(34)

where LHV stands for low heating value. Based on the extended Matlab code, developed for the present study, exergy for all the states in Fig. 1 can now be determined as follows:

Equation (34) can be used to find chemical exergy for point 26 (Fig. 1). Should fuel enter the engine at the surrounding conditions, as Eq. (32) shows, the physical exergy becomes zero. Knowing that fuel enters the engine at the environment temperature, the only remaining exergy to be calculated for this point, would be chemical exergy.

For state 1, it is obvious that the physical and chemical exergies are zero. For state 2, to find the reference enthalpy and entropy for the exiting gases from the engine [16], first, the amount of water vapor in the product being liquefied to reach the surrounding temperature should be determined. To do so, the water vapor partial pressure and its corresponding dew point temperature can be determined from the appropriate tables. Now, should the dew point temperature exceed the reference temperature, some of the available water vapor in the product will be condensed. For the remaining water vapor the following procedure [10] determines the amount of physical exergies for points 2 and 3 (Fig. 1):

\[
p_v = \frac{n_v}{n_{CO_2} + n_O_2 + n_{N_2} + n_v} p_2,
\]

(35)

where \( n_v, p \) and \( n_v \) are the number of moles, water vapor partial pressure at the reference temperature, and the Knol of the vapor which is not condensed. Therefore, the number of moles for point 2 at the reference conditions and in vapor form is:

\[
n_{2,new} = n_{CO_2} + n_{O_2} + n_{N_2} + n_v.
\]

(36)

Now, to find \( \bar{h} \) and \( \bar{S} \), the following equations are used:

\[
\bar{h}_o = n_{N_2} h_{N_2, o} + n_{O_2} h_{O_2, o} + n_{CO_2} h_{CO_2, o} + n_{H_2O} h_{H_2O, o}.
\]

(37)

where \( \bar{h}_o \) indicates liquid, and \( n_v \) is the number of condensed moles. Also,

\[
\bar{S}_o = n_{N_2} S_{N_2, o} + n_{O_2} S_{O_2, o} + n_{CO_2} S_{CO_2, o} + n_{H_2O} S_{H_2O, o}.
\]

(38)

\[
S_{CO_2, o} = \frac{y_{N_2, new} P(2)}{P_o} \ln \frac{y_{CO_2, new} P(2)}{y_{O_2, new} P(2)} - n_{CO_2} \bar{R}
\]

\[
\ln \frac{y_{O_2, new} P(2)}{P_o} - n_{CO_2} \bar{R} \ln \frac{y_{CO_2, new} P(2)}{P_o} - n
\]

\[
\bar{R} \ln \frac{y_{v, new} P(2)}{P_o}.
\]

(39)

where \( y_{v, new} \) is the mole portion at the reference condition. Using the above equation, the physical exergy for both points 2 and 3 (Fig. 1) can be determined. To find the chemical exergy for states 2 and 3, the following equations can be employed [10]:

\[
e_{ch,2} = [-\bar{R} T_o] \begin{bmatrix}
0.0003 \\
0.2059 \\
0.7748 \\
0.019 \\
0.0019 \\
-0.2059 \\
-0.0019 \\
-0.2059
\end{bmatrix} \begin{bmatrix}
y_{CO_2, new} Ln \\
y_{CO_2, new} Ln \\
y_{O_2, new} Ln \\
y_{H_2O(g), new} Ln \\
y_{H_2O(g), new} Ln \\
y_{H_2O(g), new} Ln \\
y_{H_2O(g), new} Ln \\
y_{H_2O(g), new} Ln
\end{bmatrix}.
\]

(40)

where \( M \) is the molecular weight. Also,
Now, using Eqs. (32) and (33), the physical and chemical exergies for the rest of the states in Fig. 1, can be found. Tables (2) and (3) show the physical, chemical, and total exergies for all the states indicated in Fig. 1, these tables are the results of the developed computer code for the present study. The standards used to evaluate the performance of the cogeneration systems are [13-15]:

\[
\varepsilon_{HRSG} = \frac{\dot{E}_6 - \dot{E}_{23}}{\dot{E}_2 - \dot{E}_3}
\]

(42)

\[
\varepsilon_{Ej} = \frac{\dot{E}_{13}}{\dot{E}_{11} - \dot{E}_{12}}
\]

(43)

\[
\varepsilon_{H.G} = \frac{\dot{E}_{11} - \dot{E}_{17}}{\dot{E}_8 - \dot{E}_9}
\]

(44)

\[
\varepsilon_{Evap} = \frac{\dot{E}_{20} - \dot{E}_{19}}{\dot{E}_{18} - \dot{E}_{12}}
\]

(45)

\[
\varepsilon_{E.1} = \frac{\dot{E}_{22} - \dot{E}_{21}}{\dot{E}_7 - \dot{E}_{10}}
\]

(46)

\[
\varepsilon_{motor} = \frac{\dot{W}_{net} + \dot{E}_5 - \dot{E}_4}{\dot{E}_{26} + \dot{E}_4 + \dot{E}_1}
\]

(47)

where \( \varepsilon \) is exergy efficiency, Evap. stands for the evaporator, Ej stands for the ejector, and \( \dot{E} \) is the total rate of exergy. Exergy destruction rate for each component in Fig. 1, can be listed as,

\[
\dot{I}_{HRSG} = \dot{E}_2 + \dot{E}_{23} - \dot{E}_3 - \dot{E}_6
\]

(48)

\[
\dot{I}_{H.G} = \dot{E}_8 + \dot{E}_{17} - \dot{E}_9 - \dot{E}_{11}
\]

(49)

\[
\dot{I}_{cond} = \dot{E}_{13} + \dot{E}_{24} - \dot{E}_{14} - \dot{E}_{25}
\]

(50)

\[
\dot{I}_{Evap} = \dot{E}_{19} + \dot{E}_{18} - \dot{E}_{12} - \dot{E}_{20}
\]

(51)

\[
\dot{I}_{Ej} = \dot{E}_{11} + \dot{E}_{12} - \dot{E}_{13}
\]

(52)

\[
\dot{I}_{E.1} = \dot{E}_7 + \dot{E}_{21} - \dot{E}_{22} - \dot{E}_{10}
\]

(53)

\[
\dot{I}_{motor} = \dot{E}_{26}(1 - \frac{\dot{W}_{net} + \dot{E}_5 - \dot{E}_4}{\dot{E}_{26} + \dot{E}_4 + \dot{E}_1})
\]

(54)

where \( \dot{I} \) is the rate of exergy destruction, and Cond. stands for the condenser.

### 5 RESULTS AND VALIDATIONS

After the above calculations, the following results are achieved through the developed computer code:

As Fig. (4a) and (4b) show, the internal combustion engine (Fig. 1) causes the maximum exergy destruction (irreversibility) in both summer and winter seasons, respectively.

<table>
<thead>
<tr>
<th>STATE</th>
<th>( \dot{m} ) (Kg/s)</th>
<th>T (°C)</th>
<th>P (bar)</th>
<th>( h ) (KJ/Kg)</th>
<th>S (KJ/Kg °K)</th>
<th>( \varepsilon_{pp} ) (KJ/Kg)</th>
<th>( \varepsilon_{eh} ) (KJ/Kg)</th>
<th>( E_{tot} ) (KJ/Kg)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>0.2263</td>
<td>10</td>
<td>1</td>
<td>276.5</td>
<td>6.8192</td>
<td>0</td>
<td>0</td>
<td>0</td>
</tr>
<tr>
<td>2</td>
<td>0.2365</td>
<td>600</td>
<td>10</td>
<td>1050.1</td>
<td>7.8313</td>
<td>3015.7</td>
<td>3015.7</td>
<td>724.8016</td>
</tr>
<tr>
<td>3</td>
<td>0.2365</td>
<td>167</td>
<td>9.5</td>
<td>507.6</td>
<td>7.6982</td>
<td>2510.9</td>
<td>2510.9</td>
<td>605.4163</td>
</tr>
<tr>
<td>4</td>
<td>0.7093</td>
<td>10</td>
<td>1</td>
<td>42.1</td>
<td>0.1511</td>
<td>0</td>
<td>0</td>
<td>2.128</td>
</tr>
<tr>
<td>5</td>
<td>0.7093</td>
<td>70</td>
<td>1</td>
<td>293.1</td>
<td>0.955</td>
<td>23.3</td>
<td>23.3</td>
<td>18.655</td>
</tr>
<tr>
<td>6</td>
<td>0.0484</td>
<td>297.85</td>
<td>15</td>
<td>3033</td>
<td>6.9115</td>
<td>1077.1</td>
<td>1077.1</td>
<td>52.2768</td>
</tr>
<tr>
<td>7</td>
<td>0.0484</td>
<td>297.85</td>
<td>15</td>
<td>3033</td>
<td>6.9115</td>
<td>1077.1</td>
<td>1077.1</td>
<td>52.2768</td>
</tr>
<tr>
<td>10</td>
<td>0.0484</td>
<td>100</td>
<td>15</td>
<td>420.1</td>
<td>1.3059</td>
<td>51</td>
<td>51</td>
<td>2.6136</td>
</tr>
<tr>
<td>21</td>
<td>0.5045</td>
<td>10</td>
<td>1</td>
<td>42.1</td>
<td>0.1511</td>
<td>0</td>
<td>0</td>
<td>1.5135</td>
</tr>
<tr>
<td>22</td>
<td>0.5045</td>
<td>70</td>
<td>1</td>
<td>293.1</td>
<td>0.955</td>
<td>23.3</td>
<td>23.3</td>
<td>13.268</td>
</tr>
<tr>
<td>23</td>
<td>0.0484</td>
<td>100</td>
<td>15</td>
<td>420.1</td>
<td>1.3059</td>
<td>51</td>
<td>51</td>
<td>2.614</td>
</tr>
<tr>
<td>26</td>
<td>0.0103</td>
<td>10</td>
<td>1</td>
<td>590.8</td>
<td>11.5146</td>
<td>0</td>
<td>0</td>
<td>532.593</td>
</tr>
</tbody>
</table>

© 2012 IAU, Majlesi Branch
Figures (5a) and (5b) show the exergy efficiency for different components of Fig. 1, for summer and winter seasons, respectively. As Fig. (5a) indicates, for the summer season, the heat generators and the engine have the highest and lowest exergies, respectively. Also as Fig. (5b) shows, for the winter season, the heat recovery steam generator and number 1 heat exchanger (Fig. 1) have the highest and lowest exergies, respectively. As Fig. 6 from reference [17] shows, in their study like the present work in Figs. (4a) and (4b), the maximum exergy destruction belongs to the internal combustion engine and heat recovery steam generators. Again, Figs (4a) and (4b) in this study like Fig. 6 by [17] agree that the least exergy destruction belongs to the condenser and evaporator. These comparisons prove the validity of the present work.

Table 3 Summer season exergy results (Fig. 1)

<table>
<thead>
<tr>
<th>STATE</th>
<th>m (Kg/s)</th>
<th>T (°C)</th>
<th>P (bar)</th>
<th>h (KJ/Kg)</th>
<th>S (KJ/Kg °K)</th>
<th>e_ph (KJ/Kg)</th>
<th>e_ch (KJ/Kg)</th>
<th>E_tot (KJ/Kg)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>0.2342</td>
<td>30</td>
<td>1</td>
<td>276.5</td>
<td>6.8662</td>
<td>0</td>
<td>0</td>
<td>0</td>
</tr>
<tr>
<td>2</td>
<td>0.2445</td>
<td>600</td>
<td>10</td>
<td>1050.1</td>
<td>7.8223</td>
<td>2904.1</td>
<td>49</td>
<td>722.033</td>
</tr>
<tr>
<td>3</td>
<td>0.2445</td>
<td>167</td>
<td>9.5</td>
<td>507.6</td>
<td>7.6911</td>
<td>2403.6</td>
<td>49</td>
<td>599.6607</td>
</tr>
<tr>
<td>4</td>
<td>1.0643</td>
<td>30</td>
<td>1</td>
<td>42.1</td>
<td>0.4368</td>
<td>0</td>
<td>3</td>
<td>3.1929</td>
</tr>
<tr>
<td>5</td>
<td>1.0643</td>
<td>70</td>
<td>1</td>
<td>293.1</td>
<td>0.955</td>
<td>10.2</td>
<td>3</td>
<td>14.0488</td>
</tr>
<tr>
<td>6</td>
<td>0.0499</td>
<td>297.63</td>
<td>15</td>
<td>3033</td>
<td>6.9107</td>
<td>944.6</td>
<td>3</td>
<td>47.2852</td>
</tr>
<tr>
<td>7</td>
<td>0.0499</td>
<td>297.63</td>
<td>15</td>
<td>3033</td>
<td>6.9107</td>
<td>944.6</td>
<td>3</td>
<td>47.2852</td>
</tr>
<tr>
<td>8</td>
<td>0.0499</td>
<td>100</td>
<td>15</td>
<td>420.1</td>
<td>1.3059</td>
<td>30.8</td>
<td>0</td>
<td>1.53692</td>
</tr>
<tr>
<td>11</td>
<td>0.0514</td>
<td>133.5254</td>
<td>3</td>
<td>2724.9</td>
<td>6.9916</td>
<td>612</td>
<td>3</td>
<td>31.611</td>
</tr>
<tr>
<td>12</td>
<td>0.0128</td>
<td>12</td>
<td>0.014</td>
<td>2522.9</td>
<td>8.8514</td>
<td>153.8</td>
<td>3</td>
<td>-1.93024</td>
</tr>
<tr>
<td>13</td>
<td>0.0642</td>
<td>237.1558</td>
<td>0.0963</td>
<td>2952.2</td>
<td>9.0702</td>
<td>209.2</td>
<td>3</td>
<td>13.62324</td>
</tr>
<tr>
<td>14</td>
<td>0.0642</td>
<td>45</td>
<td>0.0959</td>
<td>188.4</td>
<td>0.6386</td>
<td>1.4</td>
<td>3</td>
<td>0.28248</td>
</tr>
<tr>
<td>15</td>
<td>0.0514</td>
<td>45</td>
<td>0.0959</td>
<td>188.4</td>
<td>0.6386</td>
<td>1.4</td>
<td>3</td>
<td>0.22616</td>
</tr>
<tr>
<td>16</td>
<td>0.0128</td>
<td>45</td>
<td>0.0959</td>
<td>188.4</td>
<td>0.6386</td>
<td>1.4</td>
<td>3</td>
<td>0.05632</td>
</tr>
<tr>
<td>17</td>
<td>0.0514</td>
<td>45</td>
<td>3</td>
<td>188.4</td>
<td>0.6386</td>
<td>1.4</td>
<td>3</td>
<td>0.22616</td>
</tr>
<tr>
<td>18</td>
<td>0.0128</td>
<td>12</td>
<td>0.014</td>
<td>188.4</td>
<td>0.6647</td>
<td>-6.5</td>
<td>3</td>
<td>-0.0448</td>
</tr>
<tr>
<td>19</td>
<td>0.448</td>
<td>30</td>
<td>1</td>
<td>125.8</td>
<td>0.4368</td>
<td>0</td>
<td>3</td>
<td>1.344</td>
</tr>
<tr>
<td>20</td>
<td>0.448</td>
<td>14</td>
<td>1</td>
<td>58.9</td>
<td>0.2099</td>
<td>1.8</td>
<td>3</td>
<td>2.1504</td>
</tr>
<tr>
<td>23</td>
<td>0.0499</td>
<td>100</td>
<td>15</td>
<td>420.1</td>
<td>1.3059</td>
<td>30.8</td>
<td>3</td>
<td>1.68662</td>
</tr>
<tr>
<td>24</td>
<td>0.8488</td>
<td>30</td>
<td>1</td>
<td>125.8</td>
<td>0.4368</td>
<td>0</td>
<td>3</td>
<td>2.5464</td>
</tr>
<tr>
<td>25</td>
<td>0.8488</td>
<td>80</td>
<td>1</td>
<td>335</td>
<td>1.0754</td>
<td>15.6</td>
<td>3</td>
<td>15.78768</td>
</tr>
<tr>
<td>26</td>
<td>0.0103</td>
<td>30</td>
<td>1</td>
<td>636</td>
<td>11.6775</td>
<td>0</td>
<td>51708</td>
<td>532.593</td>
</tr>
</tbody>
</table>

Fig. 4a Exergy destruction rate for different components of Figure 1 during summer

Fig. 4b Exergy destruction rate for different components of Figure 1 during winter
6 CONCLUSIONS AND DISCUSSIONS

In the present study, a simultaneous energy generation cycle for cooling, heating and power supplying is designed and each component of the system is analyzed from thermodynamic and exergy aspects. The maximum and minimum exergy generation as well as the maximum and minimum exergy destruction by the components for both the summer and winter cycles are distinguished, and are validated through comparisons made with the appropriate results of another investigator. Based on the exergy analyses, the internal combustion engine and the heat recovery steam generator prove to be the most exergy destroying sources by about 360 KW and 70 KW, respectively. The highest and lowest exergy production sources prove to be the heat generator and the heat exchanger, respectively.

In general, utilizing a simultaneous energy generation system definitely invites economic advantage. However, selection of the appropriate capacity for each cycle component plays an important role in achieving this advantage, and needs much experimental as well as analytical study as future work.

ACKNOWLEDGEMENTS

The authors would like to thank the Iranian fuel conservation company (IFCC) for sponsoring this research project.

REFERENCES