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Thermodynamic analysis of three combined power and refrigeration Systems based on a demand

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Abstract: Three combined power and refrigeration system are introduced to compare and analyze for a defined demand and same fuel consumption based on thermodynamic parameters in a 24 hours period. Gas turbine and/or steam turbine are used for power generation and also ejector refrigeration cycle is used to produce cooling. These three systems are named as GER, SER and GSER. The results of three systems are compared and it's shown that SER has so much lower cogeneration efficiency than two other systems. GSER produces 284 MWhr per year more power than GER with the same fuel consumption that can easily cover the additional capital cost of the system. Moreover, exergy efficiency of GSER is 12% higher than GER and its cogeneration efficiency is 8% higher than GER too. So, between these three systems we would recommend GSER because of better performance like higher exergy and energy efficiency and also better conformity with our demand.

Keywords: Combined power and refrigeration; Ejector refrigeration system; Exergy; Cogeneration

1. Introduction

Many researchers have been done on combined power and refrigeration cycle to improve energy efficiency and performance of systems. A combined cycle was proposed by Xu, Goswami and Bhagwat in 2000 (Xu, Goswami, & Bhagwat, 2000). Goswami worked on combined power and cooling system with some other researchers both theoretically and experimentally (Goswami, Vijayaraghavan, Lu, & Tamm, 2004; Hasan, Goswami, & Vijayaraghavan, 2002; Martin & Goswami, 2006; Sadrameli & Goswami, 2007; Tamm, Goswami, Lu, & Hasan, 2004; Vidal, Best, Rivero, & Cervantes, 2006; Vijayaraghavan & Goswami, 2006). A combined power and cooling cycle was introduced by Zheng et al. based on the Kalina cycle (Zheng, Chen, Qi, & Jin, 2006). They used a rectifier which could obtain a higher concentration ammonia-water vapor for refrigeration. Also, a condenser and an evaporator were used between the rectifier and the second absorber. Liu and Zhang presented an ammonia-water cycle for a cogeneration system (Liu & Zhang, 2007). They used a splitting/absorption unit into the cogeneration system. An ammonia–water system for the power and refrigeration system was presented by Zhang and Lior (N. Zhang & N. J. I. J. o. R. Lior, 2007). The plant operated in a parallel combined cycle mode with an ammonia–water Rankine cycle and an ammonia refrigeration cycle, interconnected by absorption, separation and heat transfer processes. Zhang and Lior also presented several novel combined refrigeration and power systems using ammonia–water as working fluid and summarized some guidelines for integration of refrigeration and power systems to produce higher energy and exergy efficiencies (N. Zhang & N. J. J. o. E. R. T. Lior, 2007). Wang et al. also proposed a combined refrigeration and power system which combined Rankine cycle and absorption refrigeration cycle (Wang, Dai, & Gao, 2008). Although much research has been carried out on the combined power and refrigeration cycle, most of them have combined the Rankine cycle or Kalina cycle with the absorption refrigeration cycle, and little attention has been paid to the combination of Rankine cycle and the ejector refrigeration cycle. Alexis in 2007 studied a combined power and refrigeration cycle with ejector refrigeration cycle (Alexis, 2007). This cycle used extraction steam from steam turbine in conventional

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Rankine cycle to heat the working fluid in an independent steam ejector refrigeration cycle. The ejector refrigeration cycle which many studies have been devoted to, has many advantages such as less movable parts and low operating, installation and maintenance cost except for the relatively low performance (Li & Groll, 2005; Pianthong et al., 2007; Sankarlal & Mani, 2007; R. Yapıcı, Ersoy, & management, 2005; R. Yapıcı, Yetişen, & Management, 2007; R. J. E. C. Yapıcı & Management, 2008). In addition, the ejector refrigeration cycle has the possibility of using a wide range of refrigerants with the system. Wang et al. in 2009 presented a combined power and ejector refrigeration cycle which used steam extraction turbine between HRVG and ejector (Wang, Dai, & Sun, 2009). Ameri et al. in 2010 presented a tri-generation system based on a gas turbine and ejector refrigeration which power and refrigeration cycles were independent (Ameri, Behbahaninia, & Tanha, 2010). Many other studies are done about different cogeneration and tri-generation systems with different approaches to upgrade these system performances by using thermodynamic and economic analysis and also optimization methods (Ghorbani, Mafi, Amidpour, MOUSAVI, & Salehi, 2013; Ghorbani, Shirmohammadi, & Mehrpooya, 2018; Ghorbani, Shirmohammadi, Mehrpooya, & Mafi, 2018; Mohammadi, Ahmadi, et al., 2017; Mohammadi, Kasaeian, Pourfayaz, & Ahmadi, 2017; Niasar et al., 2017).

In the present study, time dependent analysis is done for three systems. Three combined power and refrigeration systems which use a micro-gas turbine, an extraction steam turbine and ejector refrigeration is presented. Also, R141 is used as working fluid in the cycle. These three cogeneration systems are analyzed for specified power and cooling demand and results are compared with each other to select the best one.

2. System Description

The schematic of GSER is shown in Fig. 1. Air at atmospheric pressure is compressed in compressor and it passes through the recuperator. The air temperature increases in this stage. The fuel, i.e. natural gas, is injected into the combustion chamber and ignition occurs. The hot flue gas passes through turbine blades and produces power. The flue gases pass through the recuperator and enter generator.

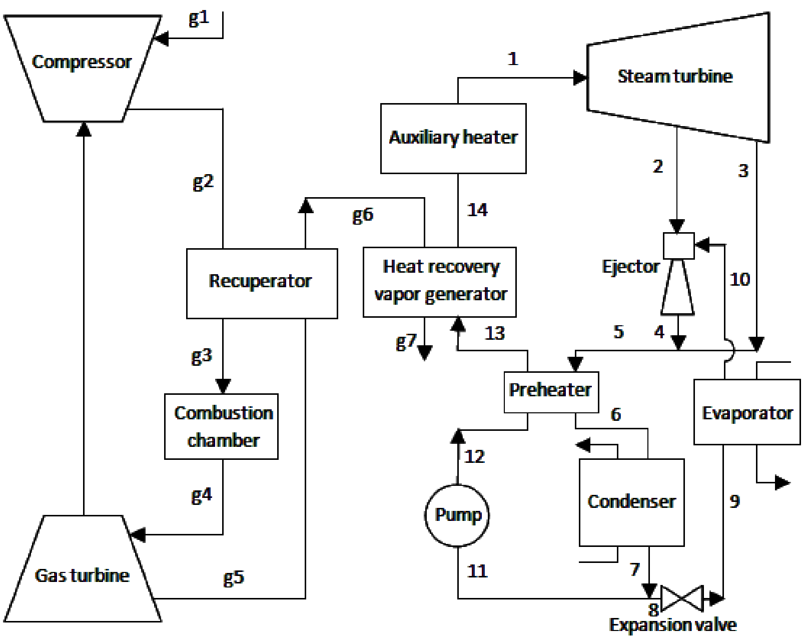

Figure 1. Schematic diagram of GSER

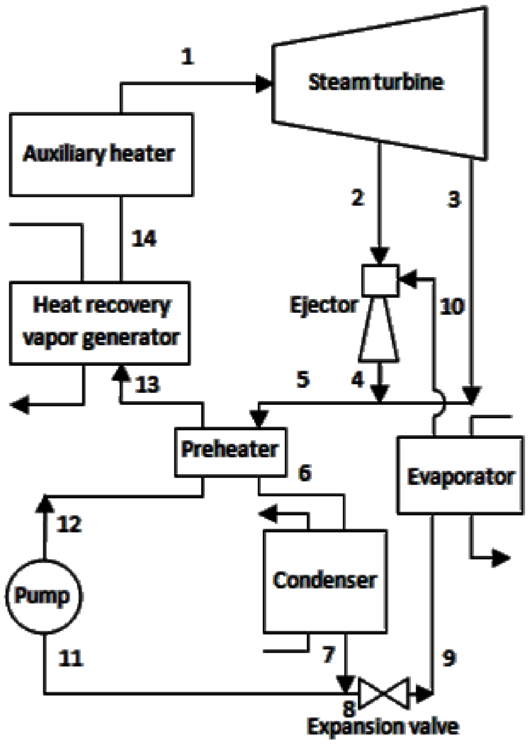

Figure 2. Schematic diagram of SER

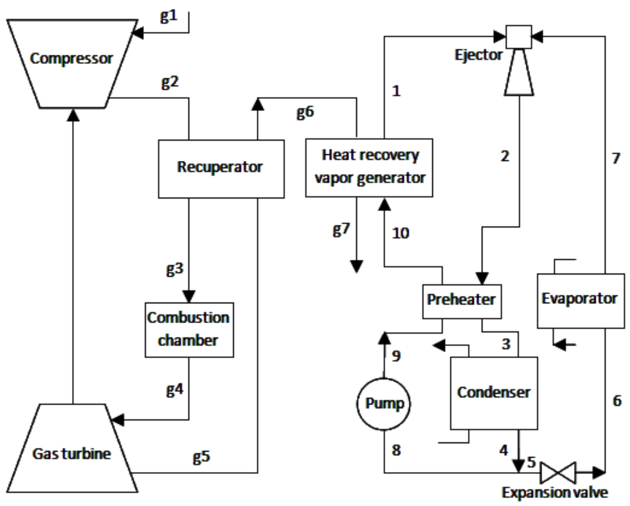

Figure 3. Schematic diagram of GER

These hot gases increase the temperature of R141b as the working fluid of the next cycle. The extraction turbine and ejector play important roles in this combined cycle. The HRVG is a device in which high pressure and temperature vapor is generated by absorbing heat from flue gases of gas turbine. The high pressure and temperature vapor is expanded through the turbine to generate power. The extracted vapor from the turbine enters the ejector as the primary vapor. The very high velocity vapor at the exit of the nozzle produces a high vacuum at the inlet of the mixing chamber and entrains secondary vapor into the chamber from the evaporator. The stream out of the ejector mixed with turbine exhaust is cooled at the preheater and enters the condenser where it condenses from vapor to liquid by rejecting heat to the surroundings. Some of the working fluid leaving the condenser enters the evaporator after passing through the throttle valve, and the other part flows to the pump. The high pressure working fluid is heated at the preheater before entering heat recovery vapor generator to be vaporized again. The working fluid with low pressure and temperature from the valve is vaporized at the evaporator, providing a cooling effect for cooling user such as a cold-storage room. SER and GER are two other cogeneration systems that we want to compare them with GSER. These two systems are presented in Fig. 2 and Fig. 3. R141b is selected as the working fluid as a low-pressure, nontoxic, nonflammable and non-corrosive refrigerant.

Table 1 contains the main assumptions made for the simulation of the system. Moreover, there is a cooling demand that is presented for a typical day in Fig. 4.

3. Mathematical Model

In the present work, there are some assumptions for exergy calculation, ejector and the whole cycle performance simulation that are the same as J. Wang's work in 2009 (Wang et al., 2009). A computer program has been developed to calculate all necessary parameters. The parameters for the gas turbine cycle are presented as it follows:

$$W_{comp} = \dot{m}_{g1}\left(h_{g2} - h_{g1}\right) \tag{1}$$

$$\dot{Q}_{CC} = \dot{m}_{CC} LHV_{CH_4} . \eta_{CC} \tag{2}$$

$$W_g = \dot{m}_{g4}\left(h_{g4} - h_{g5}\right) - W_{comp} \tag{3}$$

$$\dot{Q}_g = \dot{m}_{g6}\left(h_{g6} - h_{g7}\right) \tag{4}$$

The energy balance at the mixing point of the ejector is presented as it follows:

$$\left(m_2 + m_{10}\right) h_4 = m_{10} h_{10} + m_2 h_2 \tag{5}$$

At the mixing section of the ejector yields, mass conservation is written as follows:

$$\left(m_2 + m_{10}\right) c_m = m_{10} c_e + m_2 c_2 \tag{6}$$

The entrainment ratio is written as:

$$\omega = \frac{m_{10}}{m_2} = \sqrt{\eta_n \eta_m \eta_d \frac{h_2 - h_{n,s}}{h_{d,s} - h_m}} - 1 \tag{7}$$

From the conservation law for energy, the basic equations in the components are presented for evaporator, turbine and pump.

$$Q_e = m_{10}\left(h_{10} - h_9\right) \tag{8}$$

$$W_{st} = m_1\left(h_1 - h_2\right) + (m_1 - m_2)\left(h_2 - h_3\right) \tag{9}$$

$$W_p = m_p\left(h_{12} - h_{11}\right) \tag{10}$$

The system performance is evaluated as equation (11) and (13). The thermal efficiency is presented as the useful energy output divided by the total energy input, given by:

$$\eta_{th} = \frac{W_{net} + Q_e}{Q_{in}} \tag{11}$$

Where W_{net} is the turbine power output, Q_e is the refrigeration output and Q_{in} is the total heat input from the boiler.

Table 1. Main assumptions for the combined cycle

Environment temperature (˚C)	31
Environment pressure (MPa)	0.10135
Steam turbine inlet pressure (MPa)	1.251
Steam turbine inlet temperature (˚C)	130
Steam turbine extraction pressure (MPa)	0.283
Isentropic efficiency of steam turbine (%)	85
Combustion chamber efficiency (%)	100
Pinch point temperature difference (˚C)	40
Approach point temperature difference (˚C)	5
Isentropic efficiency of gas turbine (%)	85
Isentropic efficiency of compressor (%)	85
Pressure ratio of compressor	10
Generator efficiency (%)	98
Condenser temperature (˚C)	36
Evaporator temperature (˚C)	8

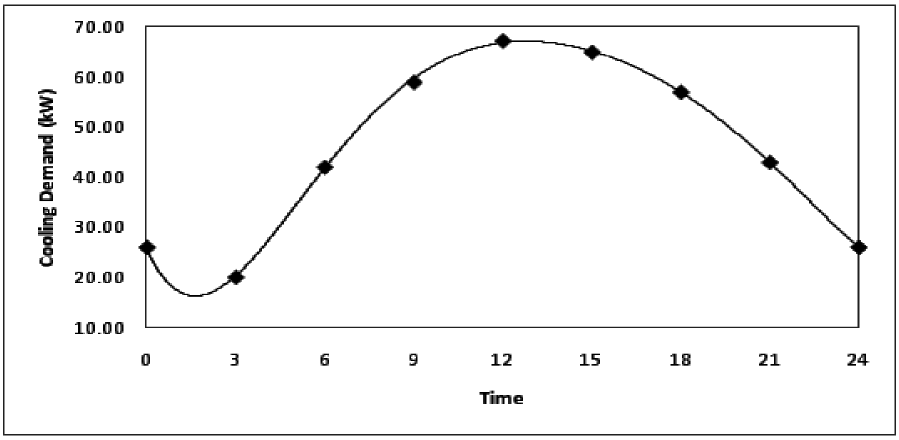

Figure 4. Hourly cooling demand variations in a day

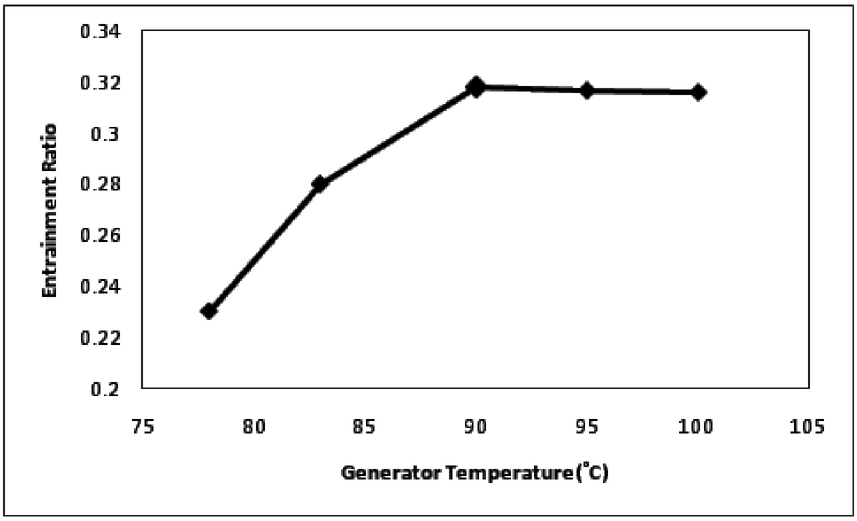

Figure 5. Effect of motive fluid temperature on entrainment ratio

The physical exergy can be calculated as:

$$E = m\left(\left(h_i - h_o\right) - T_0\left(s_i - s_9\right)\right) \tag{12}$$

Exergy efficiency is defined as the summation of exergy of the network and the refrigeration divided by the exergy of the heat source.

$$\eta_{ex} = \frac{W_{net} + E_e}{E_{in}} \tag{13}$$

E_{in} is the exergy of the heat source fluid.

$$E_{in} = Q_b\left(1 - \frac{T_0}{T_b}\right) \tag{14}$$

Since the heat source fluid is finally exhausted into the environment, the calculation of the exergy input is based on the difference between its initial state and the environment state. E_e is the exergy associated with the refrigeration output, which is calculated as the working fluid exergy difference across the evaporator.

$$E_e = m_e\left(\left(h_9 - h_{10}\right) - T_0\left(s_9 - s_{10}\right)\right) \tag{15}$$

4. Results and Discussion

The simulation of the combined GSER, GER and SER is carried out using a simulation program written in Matlab by present authors. Moreover, thermodynamics properties of R141b were calculated by using EES database. The thermodynamic analysis of GSER is done and the results are brought in comparison with SER and GER with the same fuel consumption based on the assumptions.

Table 2 shows the thermodynamic properties of the cycle states for GSER. The results of thermodynamic analysis of GER and SER for their states are shown in table 3 and 4. In Table 5 the analysis results of three systems are brought together.

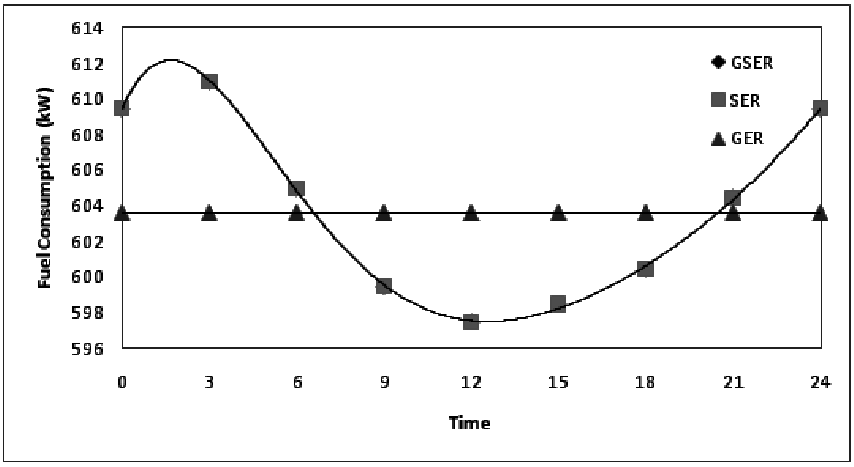

Figure 6. Hourly fuel consumption variations of the systems

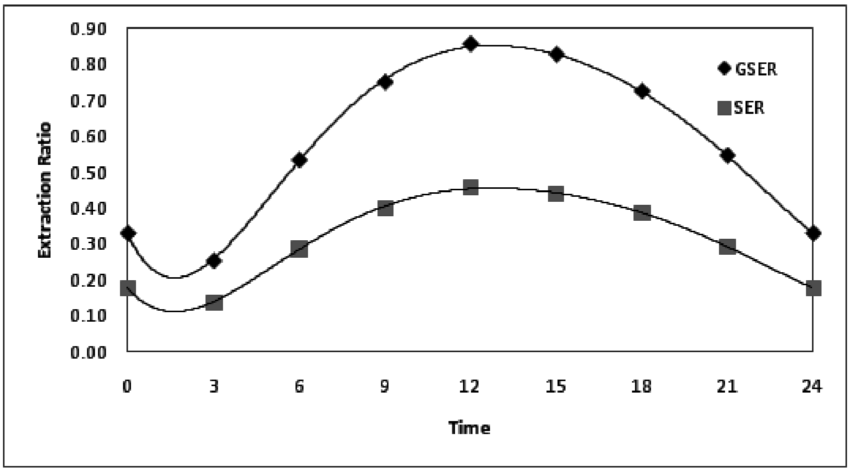

Figure 7. Hourly extraction ratio variations of the systems

Fig. 5 shows the effect of generator temperature on the entrainment ratio. It's clear that entrainment ratio increases with generator temperature until 90˚C. For the generator temperature more than it, entrainment ratio doesn't increase anymore. So 90˚C is the optimum temperature for steam turbine extraction. Fig. 6 shows the fuel consumption of three systems. It's constant for GER, but the others are time dependent. The minimum fuel consumption for GER and GSER is at noon when the cooling demand is high. The three systems have the same average of fuel consumption.

Fig. 7 shows the extraction ratio of GSER and SER. The extraction ratio is calculated based on our time dependent demand. As shown in the figure, the maximum extraction is at noon, because of the maximum cooling demand. GSER has higher extraction ratio, because its steam turbine size is smaller and in order to achieve the cooling demand, more extraction ratio is needed.

Fig. 8 shows the produced cooling by GSER and the two other systems in comparison. SER produced the same cooling as GSER that is equal to the demand. But the produced cooling by GER is constant and produces the maximum of demand. So, there is useless produced cooling at the most hours of the day.

Fig. 9 shows the produced power by GSER and the two other systems in comparison. As it's clear in the figure, the maximum produced power is for GSER and the minimum is for SER that is less than our demand. We should consider that SER is just recommended for using low temperature heat sources, because of low boiling temperature of the working fluid. So, Sources like geothermal and solar energy and also low temperature waste heat, can produce power in this cycle.

Fig. 10 shows the cogeneration efficiency of the GSER and two other systems in comparison. GSER and GER cogeneration efficiency is more than SER.

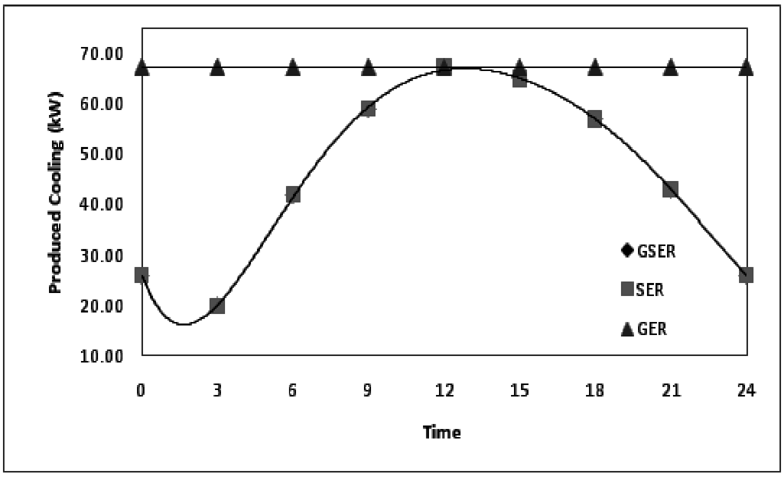

Figure 8. Hourly produced cooling variations of the systems

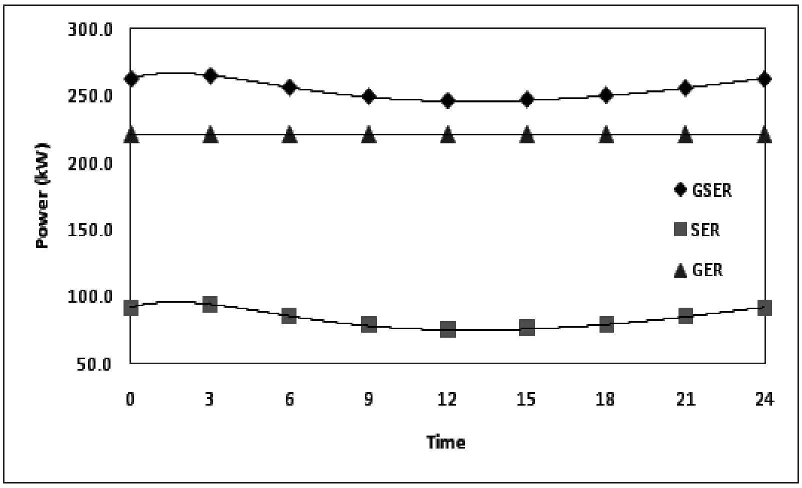

Figure 9. Hourly produced power variations of the systems

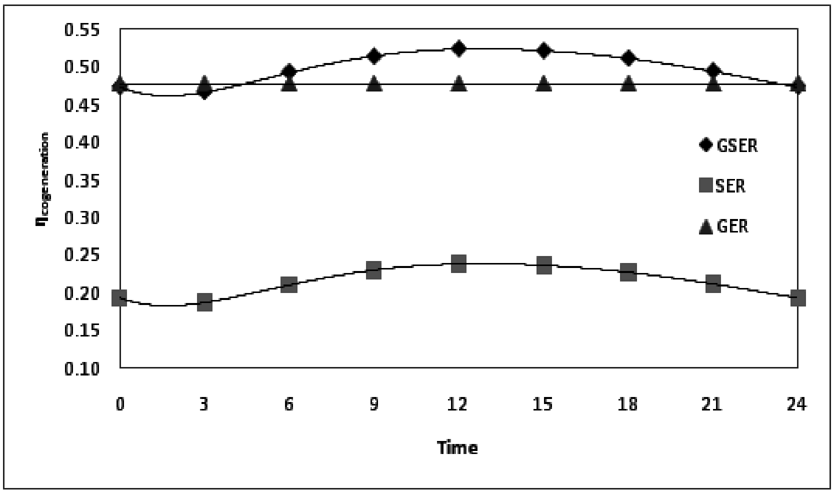

Figure 10. Hourly cogeneration efficiency variations of the systems

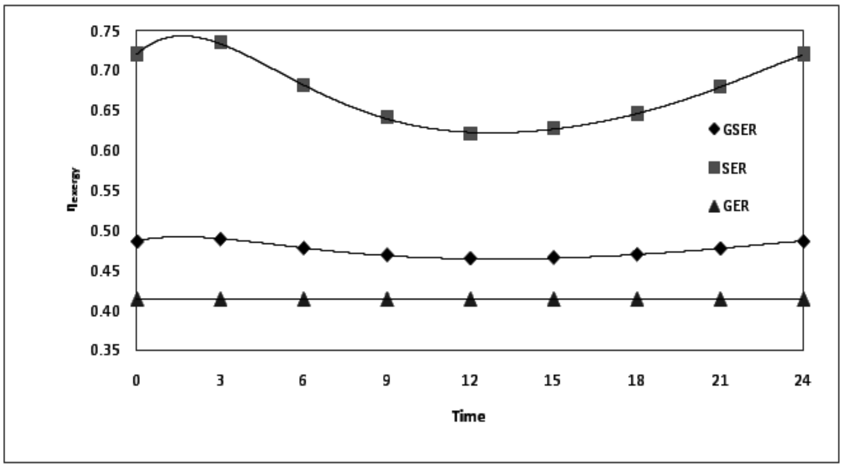

Figure 11. Hourly exergy efficiency variations of the systems

Table 2. Thermodynamic properties of GER states

State	s (kJ kg-1 K-1)	h (kJ kg-1)	P (kPa)	T (°C)	$\dot{E}$ (kW)
g1	5.715	304.6	101.35	31	0.000
g2	5.808	644.7	1014	362.5	279.5
g3	6.225	958.3	978.8	649.4	446.9
g4	6.791	1603	912.1	1200	886.1
g5	6.904	1014	112.6	698.4	317.3
g6	6.544	705.5	107.6	419.6	135.7
g7	6.072	437	103.4	162.1	21.73
1	1.02	338.1	538.5	90	38.91
2	1.083	324.7	116.1	64.15	7.009
3	1.026	305.9	116.1	41	5.016
4	0.2962	80.31	116.1	36	0.08274
5	0.2962	80.31	116.1	36	0.02006
6	0.2962	78.75	39.93	8	-0.4951
7	1.021	282.6	39.93	8	-5.976
8	0.2962	80.31	116.1	36	0.06268
9	0.2962	80.7	538.5	36	0.4652
10	0.3748	105.5	538.5	56.78	1.388

The maximum efficiency is for GSER and the other two systems work with lower cogeneration efficiency.

Fig. 11 shows the exergy efficiency of GSER and two other systems. GSER has less efficiency than SER but more than GER. The maximum exergy efficiency is at 2:00 a.m. that cooling demand is minimized. It's because the exergy efficiency of cooling cycle is low.

5. Conclusion

Three cogeneration systems are introduced to cover our power and cooling demand and also analyzed to select the best one. The results show that GSER is the best choice for our demand. Because GER produces enough power and cooling with the same fuel consumption based on our demand and it has lower capital cost, this system is a good choice too. But production of 284 MWhr per year more power than GER can easily cover the higher capital cost and can produce income after the additional capital cost payback date. SER has much lower efficiency, but because of working in low temperature, it's a good choice for low temperature heat sources, such as low temperature waste heat. Moreover, exergy and cogeneration performances of GSER are more than GER. These reasons can approve the great performance of GSER for our demand.

Table 3. Thermodynamic properties of GSER states

State	s (kJ kg-1 K-1)	h (kJ kg-1)	P (kPa)	T (°C)	$\dot{E}$ (kW)
g1	5.715	304.6	101.4	31.0	0
g2	5.808	644.7	1014	362.5	276.5
g3	6.225	958.3	978.8	649.4	442.2
g4	6.791	1603.0	912.1	1200	876.7
g5	6.904	1014.0	112.6	698.4	314.0
g6	6.544	705.5	107.6	419.6	134.3
g7	5.852	349.9	102	76.0	3.278
s1	1.033	362.0	1251	130	70.05
s2	1.076	343.4	283.1	90	17.99
s3	1.061	317.3	116.1	55.1	2.318
s4	1.095	328.7	116.1	69.0	5.024
s5	1.083	324.5	116.1	64.0	7.105
s6	1.026	305.9	116.1	41.0	5.290
s7	0.296	80.3	116.1	36.0	0.08725
s8	0.296	80.3	116.1	36.0	0.01346
s9	0.302	80.3	39.9	8.0	-0.3573
s10	1.021	282.6	39.9	8.0	-4.011
s11	0.296	80.3	116.1	36.0	0.07379
s12	0.296	80.7	1251	36.0	0.5476
s13	0.367	102.8	1251	54.6	1.420
s14	1.020	356.6	1251	130.	68.29

Table 4. Thermodynamic properties of SER states

State	**s (kJ kg-1 K-1)**	**h (kJ kg-1)**	**P (kPa)**	**T (°C)**	**$\dot{E}$ (kW)**
1	1.033	362.0	1251.00	130.0	131.9
2	1.076	343.4	283.10	90.0	18.06
3	1.061	317.3	116.10	55.1	7.066
4	1.095	328.7	116.10	69.0	5.045
5	1.074	321.5	116.10	60.2	11.76
6	1.026	305.9	116.10	41.0	9.244
7	0.296	80.3	116.10	36.0	0.1520
8	0.296	80.3	116.10	36.0	0.01400
9	0.302	80.3	39.93	8.0	-0.3590
10	1.021	282.6	39.93	8.0	-4.027
11	0.296	80.3	116.10	36.0	0.1390
12	0.296	80.7	1251.00	36.0	1.031
13	0.351	97.8	1251.00	50.5	1.789
14	1.025	358.7	1251.00	130.0	129.9

Table 5. Results of simulation for three cycles

6:00	12:00	18:00	24:00	Time	
42	67	57	26	Cooling demand (kW)	
0.53	0.86	0.72	0.33	GSER	Extraction Ratio
0.28	0.45	0.38	0.18	SER	
X	X	X	X	GER	
42.00	67.33	57.00	26.00	GSER	Produced cooling (kW)
42.00	67.33	57.00	26.00	SER	
67.33	67.33	67.33	67.33	GER	
255.8	245.6	249.7	262.2	GSER	Power (kW)
85.3	75.1	79.3	91.8	SER	
221.2	221.2	221.2	221.2	GER	
605	598	601	610	GSER	Fuel Consumption (kW)
605	598	601	610	SER	
604	604	604	604	GER	
0.49	0.52	0.51	0.47	GSER	Cogeneration Efficiency
0.21	0.24	0.23	0.19	SER	
0.48	0.48	0.48	0.48	GER	
0.48	0.46	0.47	0.49	GSER	Exergy Efficiency
0.68	0.62	0.65	0.72	SER	
0.41	0.41	0.41	0.41	GER	

Nomenclature

C	Velocity (m s^{-1})
E	Exergy (kW)
h	Enthalpy (kJ kg^{-1})
LHV	Low Heating Value (kJ kg^{-1})
m	Mass Rate (kg s^{-1})
Q	Heat Transfer Rate (kW)
s	Entropy (kJ kg^{-1} K^{-1})
T	Temperature (K)
W	Power (kW)
Greek letters	
η	Efficiency
ω	Entrainment Ratio
Subscripts	
CC	Combustion Chamber
d	Diffuser
e	Evaporator
ex	Exergy
g	Heat Recovery Steam Generator
gt	Gas Turbine
m	Mixture
n	Nozzle
p	Pump
s	Isentropic
st	Steam Turbine
th	Thermal

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