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Cascade organic Rankine cycle using LNG cold energy: Energetic and exergetic assessments

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A B S T R A C T

This study deals with a cascade organic Rankine cycle driven by linear Fresnel solar collector as a source and cold energy of LNG to produce power. An exergy concept is used to model the system and the performance of the desired system is assessed by varying the major design parameters namely solar subsystem feed mass flow rate, LNG mass flow rate and turbines inlet and back pressures. A Parametric study shows that among the design parameters solar subsystem feed mass flow rate and turbine 1 back pressure have positive effects on the performance of the system and maximum efficiencies increments are achieved within 5.24% as turbine 1 back pressure increases.

1. Introduction

Solar energy is one of the cleanest energy resources that does not compromise or add to the global warming. The sun radiates more energy in one second than people have used since beginning of time. Solar energy is often called "alternative energy" to fossil fuel energy sources such as oil and coal. Availability of cheap and abundant energy with minimum environmental and ecological hazards associated with its production and use is one of the important factors for desired improvement in the quality of life of the people. The growing scarcity of fossil fuels has raised global interest in the harnessing of solar energy.[1] Recently, many investigations focus on using the liquefied natural gas (LNG) as low temperature source for organic Rankine cycle (ORC).

Karellas et al. [2] proposed an ORC driven by heat waste in cement industry. The simulation was performed using the exergy and economic concepts. A parametric evaluation was carried out to assess the performance of the system. Shi and Che [3] proposed a combined power system, in which low-temperature waste heat can be efficiently recovered and cold energy of LNG can be fully utilized as well. This system consisted of an ammonia–water mixture Rankine cycle and an LNG power generation cycle. The proposed system was modelled by considering energy and species balances for each component.

Wang et al. [4] investigated a transcritical CO_2 cycle using geothermal resources to generate electricity. Liquefied natural gas (LNG) was employed as heat sink to drop the CO_2 turbine back pressure sharply. The mathematical model of the transcritical CO_2 geothermal power generation system was established for system simulations under steady-state conditions. A parametric analysis was conducted to evaluate the effect of several key thermodynamic parameters on system performance. Additionally, a multi-objective optimization using NSGA-II method was carried out to find the optimum performance of system from both thermodynamic and economic aspects. Sun et al. [5] studied a solar transcritical CO2 power cycle for hydrogen production. Liquefied Natural Gas (LNG) was utilized to condense the CO_2. An exergy analysis of the whole process was performed to evaluate the effects of the key parameters, including the boiler inlet temperature, the turbine inlet temperature, the turbine inlet pressure and the condensation temperature, on the system power outputs and to guide the exergy efficiency improvement. In addition, parameter optimization was conducted via Particle Swarm Optimization to maximize the exergy efficiency of hydrogen production. Song et al. [6] proposed a transcritical CO_2 power cycle driven by solar energy while utilizing the cold heat rejection to a liquified natural gas (LNG) evaporation system. In order to ensure a continuous and stable operation for the system, a thermal storage system was introduced to store the collected solar energy and to provide stable power output when solar radiation was insufficient. A mathematical model was developed to simulate the solar-driven transcritical CO_2 power cycle under steady-state conditions, and a modified system efficiency was defined to better evaluate the cycle performance over a period of time. The thermodynamic analysis focuses on the effects of some key parameters, including the turbine inlet pressure, the turbine inlet temperature and the condensation temperature, on the system performance. Liu and Guo [7] proposed a novel cryogenic cycle by using a binary mixture as working fluids and combined with a vapor absorption process to improve the energy recovery efficiency of an LNG (liquefied natural gas) cold power generation. The cycle was simulated with seawater as the heat source and LNG as the heat sink, and the optimization of the power generated per unit LNG was performed.

The objective of this study is the simulation of a cascade ORC between solar and LNG cold energies as source and sink, respectively. The simulation of the proposed system was performed using the exergy concept. The effects of major parameter was evaluated on the performances of the system.

Nomenclature	
h	enthalpy (kJ/kg)
$\dot{E}x$	Exergy flow rate (kW)
$\dot{m}$	Mass flow rate (kg/s)
$\dot{Q}$	Heat flow rate (kW)
$\dot{W}$	Power (kW)
η	Efficiency
Subscripts	
Ch	Chemical component
D	destruction
En	Energy
Ex	Exergy
k	the kth component of the system
L	Loss
P	Product
Ph	physical component
PMP	Pump
sw	sea water
Tor	turbine
wg	waste gas

2. Materials and Methods

2.1. Thermodynamic modeling

Figure.1 illustrates the schematic diagram of the proposed system. The desired system consists of two cycle cascaded via H-1C and LNG vaporization process. R227ea is used in the upper cycle and R116 is used in the bottom cycle. In the upper cycle, the low temperature liquid R227ea is pumped to the required pressure and routed to H-1B and 1A to receive the require heat from the solar subsystem. The high pressure R227ea enters turbine T-1 and produce the power. The low pressure vapor passes through H-1C, and finally is led to the condenser C-1 to be liquefied by natural gas. The bottom cycle is driven by the heat loss of H-1C and solar heat via H-2A. The LNG is heated and evaporated in condenser C-1 and C-2, respectively.

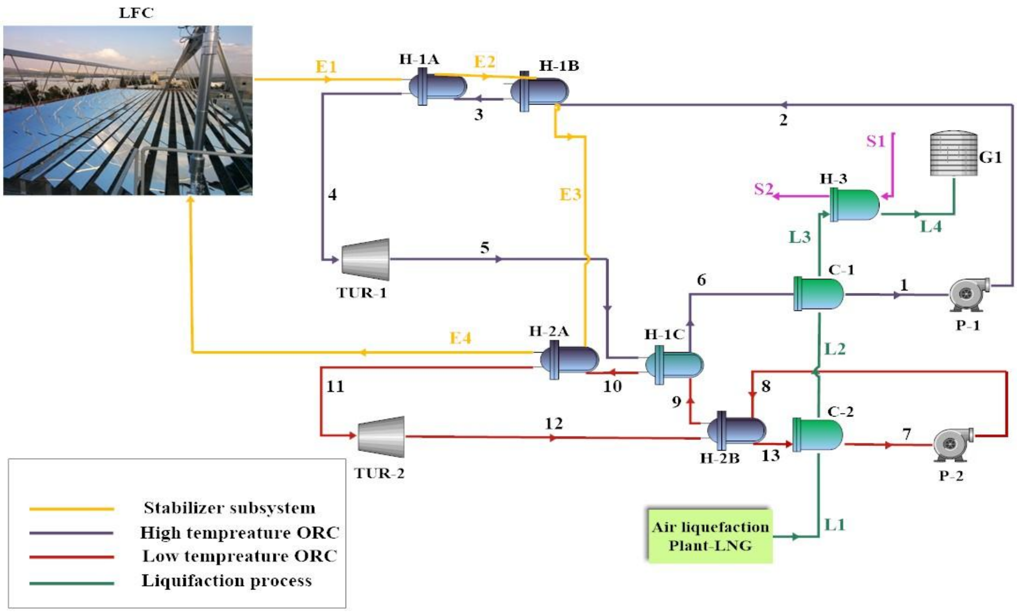

Figure 1. Schematic diagram of desired system

2.2. Energy analysis

The thermodynamic properties of cascade is calculated by Engineering Equation solver (EES). By simplifying the model, some thermodynamics assumptions are made below:

1. The system operates under steady state condition.
2. Pressure drops along the pipelines and heat loss to the surroundings are neglected.
3. The isentropic efficiencies of the pumps and the turbine are set to 0.8 and 0.85, respectively.
4. In order to improve the heat transfer efficiency and reduce the heat exchanger size, counter flow heat exchangers are adopted in the system. Each heat exchanger has its own pinch temperature difference, which is not less than 5K.

The equations of each balance is expressed as follows [8]:

Mass balance:

$$\sum \dot{m}_{in} = \sum \dot{m}_{out} \tag{1}$$

Energy balance:

$$\dot{Q} - \dot{W} = \sum \dot{m}_{out} h_{out} - \sum \dot{m}_{in} h_{in} \tag{2}$$

Thermal efficiency (X.Xue et al 2015 [9]):

$$\eta_{En} = \frac{\dot{W}_{net}}{\dot{Q}} = \frac{\dot{W}_{Tor1} + \dot{W}_{Tor2} - \dot{W}_{P1} - \dot{W}_{P2}}{\dot{Q}_{wg} + \dot{Q}_{sw}} \tag{3}$$

2.3. Exergy analysis

The total exergy of the system is divided into physical exergy and chemical exergy parameters:

$$\dot{E}x = \dot{E}x_{Ph} + \dot{E}x_{Ch} \tag{4}$$

When the fluid experiences negligible changes in its kinetic and potential exergies and there is no chemical exergy, the physical exergy component is calculated using the following relation:

$$\dot{E}x_{Ph} = \dot{m}[(h - h_0) - T_0(s - s_0)] \tag{5}$$

In order to evaluate the performance of a system for the second law point of view, it is necessary to identify both the product and the fuel for each component of system. The product represents the desired result produced by the component or the system and the fuel represents the source that is consumed in generating the product. The exergy destruction rate in a component is calculated from the exergy rate balance for the component as follows [10]:

$$\dot{E}x_{D,k} = \dot{E}x_{F,k} - \dot{E}x_{P,k} - \dot{E}x_{L,k} \tag{6}$$

Where $\dot{Ex}_F$, $\dot{Ex}_P$ and $\dot{Ex}_L$ are the exergy flow rates of the fuel, the product and the losses for the component, respectively. Table 1 shows the fuel and product exergies of each component.

Exergy efficiency [9]:

$$\eta_{Ex} = \frac{\dot{W}_{net}}{\dot{E}_{in}} = \frac{\dot{W}_{Tor1} + \dot{W}_{Tor2} - \dot{W}_{P1} - \dot{W}_{P2}}{\dot{E}_{wg,in} + \dot{E}_{LNG,in} + \dot{E}_{sw,in}} \tag{7}$$

Table 1. The fuel-product-lost exergy for each component of the system

Comp.	Fuel	Product	Lost
ORC Subsys.			
H-1A	$\dot{E}x_{E1} - \dot{E}x_{E3}$	$\dot{E}x_4 - \dot{E}x_3$	-
H-1B	$\dot{E}x_{E2} - \dot{E}x_{E3}$	$\dot{E}x_3 - \dot{E}x_2$	-
P-1	$\dot{W}_{P1}$	$\dot{E}x_2 - \dot{E}x_1$	-
C-1	$\dot{E}x_{L2} - \dot{E}x_{L3}$	-	$\dot{E}x_1 - \dot{E}x_6$
TUR-1	$\dot{E}x_4 - \dot{E}x_5$	$\dot{W}_{T1}$	-
P-2	$\dot{W}_{P2}$	$\dot{E}x_8 - \dot{E}x_7$	-
C-2	$\dot{E}x_{L1} - \dot{E}x_{L2}$	-	$\dot{E}x_7 - \dot{E}x_{13}$
H-2B	$\dot{E}x_8 - \dot{E}x_9$	$\dot{E}x_{13} - \dot{E}x_{12}$	-
H-1C	$\dot{E}x_5 - \dot{E}x_9$	$\dot{E}x_6 - \dot{E}x_{10}$	-
H-2A	$\dot{E}x_{E3} - \dot{E}x_{E4}$	$\dot{E}x_{11} - \dot{E}x_{10}$	-
TUR-2	$\dot{E}x_{11} - \dot{E}x_{12}$	$\dot{W}_{T2}$	-
H-3	$\dot{E}x_{L3} - \dot{E}x_{L4}$	$\dot{E}x_{S3} - \dot{E}x_{S1}$	-

3. Results & Discussion

By considering the input data listed in Table 2. The simulation results are obtained according to Table 3.

Table 2. Input Data

Term	Type of fluids	Value
Turbine inlet pressure, P_4	R227ea	2400
Mass flow rate of top cycle, $\dot{m}_1$	R227ea	27
Turbine inlet pressure, P_{11}	R116	2800
Mass flow rate of bottom cycle, $\dot{m}_7$	R116	25.2
Turbine outlet pressure, P_5	R227ea	100
Turbine outlet pressure, P_{12}	R116	100
LNG mass flow rate, $\dot{m}_{L1}$	methane	8.3
Pressure of LNG, P_{L1}	methane	2500

As results show, the exergy efficiency of the system is obtained within 24.54% which is lower than the energy efficiency and the net power output is calculated by about 19.16 kW.

Table 3. Performance

Item	Value
Energy efficiency, η_{En} (%)	27.41
Exergy efficiency, η_{Ex} (%)	24.54
Cooling effect, $\dot{Q}_{Cooling}$ (kW)	5251
Power produced, $\dot{W}$ (kW)	1916

3.1. Parametric Assessments

Figure. 2 shows the effects of solar subsystem feed mas flow rate ($\dot{m}_E$) on the efficiencies and net power output of the system. When $\dot{m}_E$ varies from 144 kg/s to 151 kg/s, the energy efficiency of the system increases within 1.43% (from 27.19% to 27.58%) due to the increment of power produced by about 1.42%. In this case the turbine 1 inlet temperature increase. As clearly observed, the exergy efficiency of the system is also grows from 24.39% to 25.67% by the increment of $\dot{m}_E$. This growth is due to the decrement of the total exergy destruction of the system.

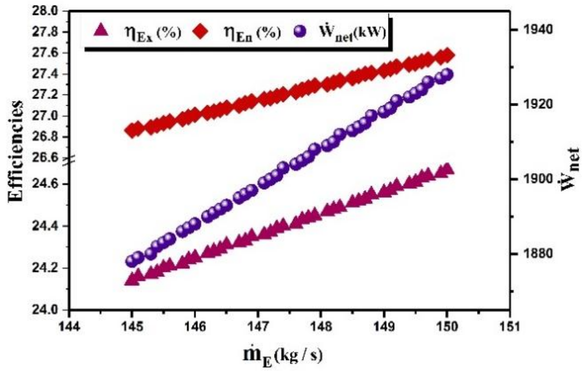

Figure 2. The effects of solar subsystem feed mas flow rate on the performances

Figure. 3 shows the variation of efficiencies and net power output when turbine 1 back pressure is supposed to change from 80 kPa to 110 kPa. As clearly observed, both energy and exergy efficiencies increase. The reason of these increments is related to the increases of power output within 2.65% as can be observed in Fig. 3. The growth of energy efficiency is by about 2.5% (from 26.96% to 27.64%). Moreover, the exergy efficiency increases within 2.5%.

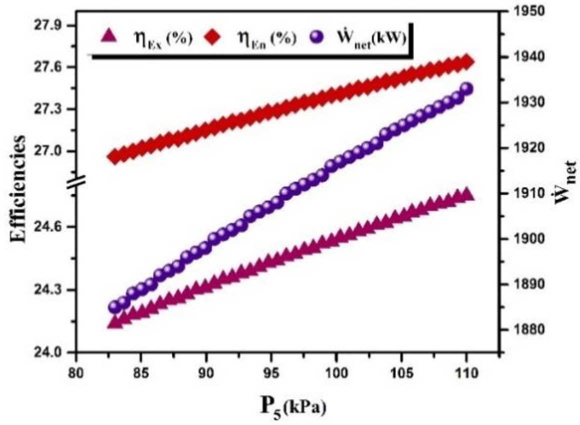

Figure 3. The effects of turbine 1 back pressure on the performances

Figure 4 illustrates the effects of turbine 1 inlet pressure on the performances of the system. The increment in turbine 1 inlet pressure the thermodynamic efficiencies reduce due to the increase of power output. When pressure increases from 2090 kPa to 2800 kPa, the energy efficacy decreases within 2.3% and exergy efficiency also lessens slightly from 24.62% to 24.06%.

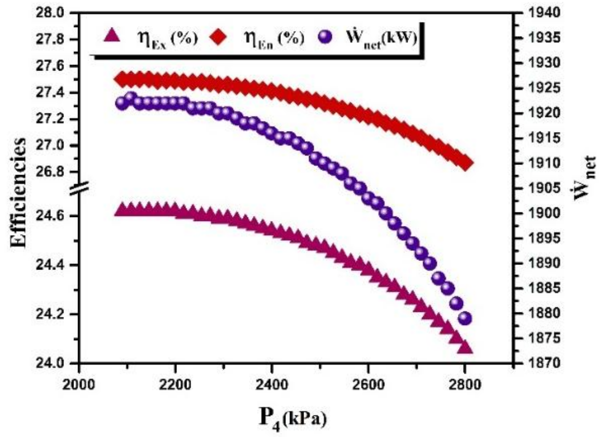

Figure 4. The effects of turbine 1 inlet pressure on the performances

Figure 5 shows the effects of turbine 2 back pressure on the energy and exergy efficiencies and net power output. When P12 increases from 40 kPa to 110 kPa, the energy efficiency reduces within 12.99%. The decrement of energy efficiency is due to the power drop by about 12.97% (from 2158 kW to 1878 kW). Further results indicates the 12.99% reduction in exergy efficiency as back pressure increases.

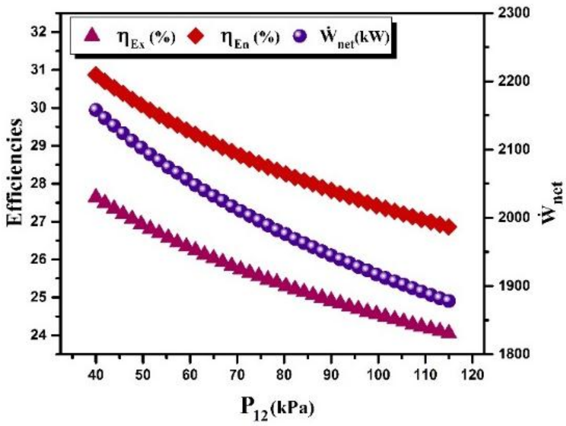

Figure 5. The effects of turbine 2 back pressure on the performances

The effects of turbine 1 inlet pressure (P11) from 2590 kPa to 3200 kPa on the performances of the desired system is illustrated in Fig. 6. As clearly observed the P11 increment has a slight negative effect on the energy exergy due to the reduction of power output within 1.58%. This trend is observed for the exergy efficiency so that 1.6% decrement may be obtained as pressure increases.

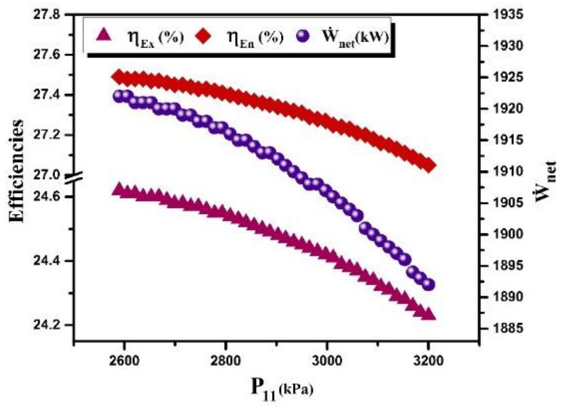

Figure 6. The effects of turbine 2 inlet pressure on the performances

Figure 7 shows the effect of LNG mass flow rate on the efficiencies and power output of the system. The variation of the LNG mas flow rate does not affect the energy efficiency and power produced while it affects the exergy efficiency within 55.88%, negatively.

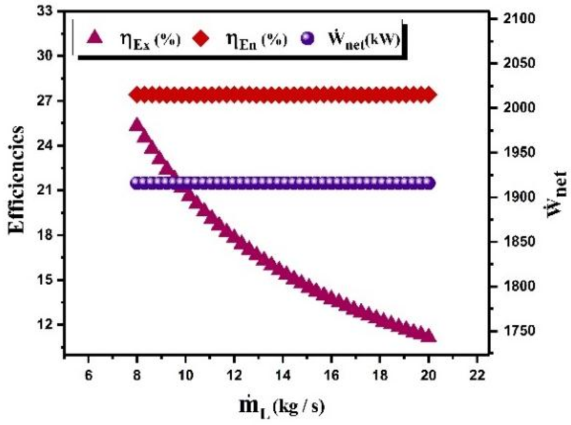

Figure 7. The effects of LNG mass flow rate on the performances

4. Conclusions

A solar driven cascade ORC is presented and modeled using the exergy method. The desired system produces power and vaporized LNG. The effect of major parameters are evaluated on the performances of the system. The main conclusions are listed as follow:

1. The increases of $\dot{m}_E$ has a positive effects on the energy and exergy efficiencies within 1.43% and 5.24%, respectively.
2. The increase of P_5 affects the efficiencies positively by about 5.24%.
3. The increments of turbines pressures have negative impacts on the performances of the system so that P12 reduces efficiencies within 12.99%.

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