THERMODYNAMIC ANALYSIS OF ORGANIC RANKINE CYCLE USING LNG COLD ENERGY DEPENDING ON SOURCE TEMPERATURES

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Abstract- This paper presents a thermodynamic analysis for a combined cycle consisting of an organic Rankine cycle (ORC) and a liquefied natural gas (LNG) Rankine cycle for the recovery of low-grade heat source energy and LNG cold energy. Based on the thermodynamic models of the combined cycle, the effects of the turbine inlet pressure of the working fluid and the source temperature on the system performance are parametrically investigated. Results show that the specific net work has the peak value and the net heat input rate of system decreases with increasing turbine inlet pressure. The thermal efficiency increases with increasing turbine inlet pressure and the source temperature, however, the effect of the source temperature on the thermal efficiency is insignificant.

Keywords- Organic RankineCycle (ORC), Liquefied Natural Gas (LNG), Low-Grade Heat Source, Power Generation.

I. INTRODUCTION

Since the liquefiednatural gas (LNG) is stored at a cryogenic temperature of approximately −165°C, it contains a great amount of cold exergy. Recently, the most widely-studied application to exploit LNG exergy is that of improving power cycle efficiency using LNG as a heat sink and as a contributor of additional exergy. When a low-grade heat source is used for the power cycle in the form of sensible energy, the Rankine cycle is considered most suitable and the LNG cold exergy is used for cooling the condenser. During the process of LNG regasification, the thermal exergy released during vaporization and the heating of natural gas is used to condense the working fluid. Low temperature condensation can improve the cycle efficiency due to the decrease in turbine backpressure [1],[2].

The organic Rankine cycle (ORC) applies the principle of the steam Rankine cycle, but uses organic working fluids with low boiling points to recover heat from lower temperature heat sources [3]. During the past decades the ORC have attracted much attention as they are proven to be the most feasible methods to achieve high efficiency in converting the low-grade thermal energy to more useful forms of energy [4]-[6]. Heberle and Brueggemann [7] investigated the CHP generation for geothermal resources with series and parallel circuits of the ORC. Tchanche et al. [8] comparatively investigated the performances as well as thermodynamic and environmental properties of the fluids for use in low-grade solar ORC systems.Gao et al. [9] carried out the analysis of a supercritical organic Rankine cycle system driven by exhaust heat using 18 organic working fluids. Li et al. [10] performed an exergo-economic analysis and optimization of a condenser for a binary mixture in ORC system. Walraven et al. [11] investigated comparatively the performance of ORC and Kalina cycle. Kim and Perez-Blanco [12] reported a thermodynamic analysis of cogeneration of power and refrigeration activated by low-grade sensible energy.

Miyazaki et al. [13] compared the conventional refuse incineration power cycle with the combined power cycle consisting of ammonia-water and LNG Rankine cycles. Shi and Che [14] studied a combined cycle of ammonia-water and LNG Rankine cycles using a separator of the ammonia-water mixture. Wang et al. [15] analyzed the optimal performance of an ammonia-water Rankine cycle with LNG as a heat sink. Kim et al. [16],[17] carried out the energy and exergy analysis for a combined cycle of ammonia-water and LNG Rankine cycles to examine the effects of key system parameters on the system performance.Choi et al. [18] proposed and investigated a cascade Rankine cycle that consisted of multiple stages of ORC and recovered LNG cold energy for power generation. They presented a review of previous studies on power generation cycles using cold energy sources. Rao et al. [19] studied a combined cycle that consisted of the ORC and the LNG Rankine cycles using solar energy as a low-temperature heat source.

Soffiato et al. [20] investigated ORC to exploit the low-grade waste heat rejected by a ship power generation plant-driven LNG carrier. Lee and Kim [2] presented energy and exergy analyses for a combined cycleconsisting of an organic Rankine cycle (ORC) and a liquefied natural gas (LNG) Rankinecycle for the recovery of low-grade heat sources and LNG cold energy. Xue et al. [21] conducted an analysis of a two-stage ORC with the low-grade heat of the exhaust flue gas of a gas-steam combined cycle power-generating unit, as well as the cryogenic energy of LNG. R227ea and R116 were selected as working fluids for the system. Lee et al. [22] studied

an ORC that used an R601-R23-R14 ternary mixture as its working fluid and was integrated with a steam cycle as a bottoming cycle and LNG as the cold sink of the working fluid.

This study performsthe thermodynamic analysis of the combined power cycle consisting of the ORC and the LNG Rankine cycle, where LNG is used to produce power output as well as to condense the working fluid of ORC as a heat sink. This work focuses on the effects of key system parameters, such as the turbine inlet pressure and the source temperature including the mass flow rate of working fluid, the heat input rate, the net power production, and the thermal efficiency of the system.

II. SYSTEM ANALYSIS

A schematic diagram of the combined cycle, consisting of the ORC and the LNG power generation cycle by direct expansion is shown in Fig 1.

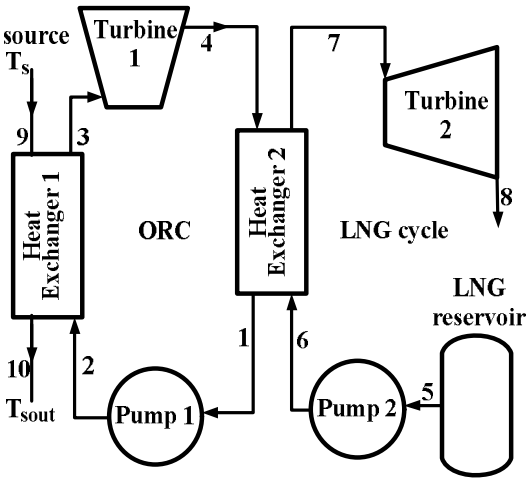


Fig.1. Schematic diagram of the system.

In the ORC, the working fluid is compressed in Pump 1 from a saturated liquid state (State 1) to a compressed liquid state (State 2). The fluid is then heated in the evaporator with the source fluid to a saturated or superheated vapor state (State 3). After the mechanical energy is obtained owing to the expansion process in Turbine 1, the working fluid (State 4) enters the condenser and exchanges heat with the LNG, then returns to State 1. In the LNG cycle, the LNG is supplied from a reservoir as a saturated liquid (State 5) and is pressurized in Feed Pump 2 to a compressed liquid (State 6). It is then heated and vaporized in the condenser to the NG vapor state (State 7), whereby mechanical energy is also extracted in Turbine 2 by the expansion process (State 8). On the other hand, the source fluid enters the evaporator at temperature T_s (State 9) and leaves at temperature T_{sout}(State 10)after heating the working fluid.

The important assumptions made for the proposed system analysis are as follows: 1) the flow is steady 2) all components are well insulated. 3) the LNG is assumed to be pure methane. 4) the fluid is in a pure vapor form at the turbine inlet. 5) the performances of the pumps or turbines are characterized by constant isentropic efficiencies. 6) the turbine inlet pressure is lower than the critical pressure of the working fluid, so the cycle is limited to a subcritical one. Additionally, each of the heat exchangers is assumed to be operated with a pinch point condition, which means that the minimum temperature difference between the hot and cold streams in the heat exchanger reaches the prescribed value of the pinch temperature difference [2].

For the specified mass flow rate of the source fluid m_s, the mass flow rate of the working fluid m_w in the ORC and the LNG mass flow rate m_c in the LNG cycle can be determined from the energy balances at the evaporator and the condenser as follows:

$$m_w = \frac{m_s c_s (T_s - T_{sout})}{h_3 - h_2} \quad (1)$$

$$m_c = \frac{m_w (h_4 - h_1)}{h_7 - h_6} \quad (2)$$

wherec_s is the specific heat of source fluid and h is the specific enthalpy of the working fluid. The heat addition rate to the system (Q_{in}), the net power productions of the ORC (W_{n1}), the LNG cycle (W_{n2}) and the combined system (W_{net}), can then be evaluated in accordance to the following equations:

$$Q_{in} = m_s c_s (T_9 - T_{10}) = m_w (h_3 - h_2) \quad (3)$$

$$W_{n1} = m_w (h_3 - h_4) - m_w (h_2 - h_1) \quad (4)$$

$$W_{n2} = m_c (h_7 - h_8) - m_c (h_6 - h_5) \quad (5)$$

$$W_{net} = W_{n1} + W_{n2} \quad (6)$$

The heat transfer capability can reflect to a certain degree of the heat transfer area of the heat exchanger. The total heat transfer capability of the combined system, UA_{tot}, can be described as follows [2]:

$$UA_{tot} = \frac{Q_{HX1}}{\Delta T_{m,HX1}} + \frac{Q_{HX2}}{\Delta T_{m,HX2}} \quad (7)$$

where Q is the heat transfer rate and ΔT_m is the logarithmic mean temperature difference in the heat exchanger, which is expressed as:

$$\Delta T_m = \frac{\Delta T_{max} - \Delta T_{min}}{\ln(\Delta T_{max} / \Delta T_{min})} \quad (8)$$

The thermal efficiency of the system η_{th}, based on the first law of thermodynamics is defined as the ratio of the net power production to the heat input rate to the system, as follows:

$$\eta_{th} = W_{net} / Q_{in} \quad (9)$$

In this paper, thermodynamic properties of working fluid and LNG are evaluated by using the Patel-Teja equation [23],[24].

III. RESULTS AND DISCUSSION

In this study, it is assumed that the inlet pressure of Turbine 1 is lower than critical pressure. Therefore, the reduced turbine inlet pressure (RTIP), which is defined as the ratio of the turbine inlet pressure to the critical pressure of the working fluid, is limited to the value lower than unity. It is also assumed that the source fluid is the standard air with a mass flow rate of 1 kg/s and the working fluid is R123. The basic data used for the cycle simulation are as follows: inlet temperature of turbine 1 T_{H1} = T_s - 10°C, condensation temperature T_c = -20°C, inlet pressure of turbine 2 P_{H2} = 15 bar, exit pressure of turbine 2 P_{L2} = 1atm, isentropic pump efficiency η_p= 0.7, isentropic turbine efficiency η_p=0.7, pinch temperature difference ΔT_{pp} = 8°C.

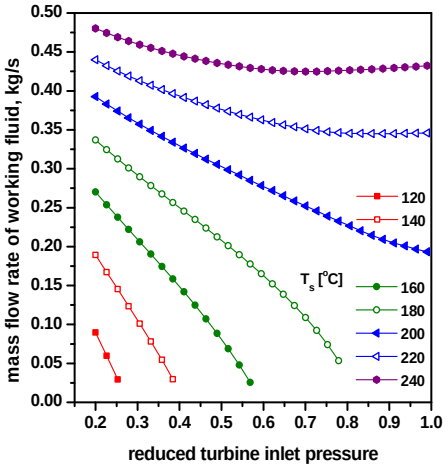


Fig.2 Effects of turbine inlet pressure on the mass flow rate for various source temperatures.

Fig. 2 displays the effects of turbine inlet pressure on the mass flow rate of the working fluid in the ORC for various source temperatures. The mass flow rate decreases with increasing RTIP, since the evaporating temperature of working fluid increases as RTIP, which leads to higher exhaust temperature of source and subsequently low mass flow rate of working fluid in the ORC. For a given RTIP, the mass flow rate increases with increasing source temperature, since the specific heat input of working fluid increases as the source temperature increases. It can be seen from the figure that the decreasing rate of mass flow rate with respect to RTIP grows as the source temperature decreases. Therefore, for low source temperatures, there exists an upper limit of RTIP for positive mass flow rate of working fluid.

Fig. 3 shows the effects of turbine inlet pressure on the specific net work in the ORC for various source temperatures. As RTIP increases, the specific workincreases at first, reaches a maximum value, and then decreases again, so it has a peak value with respect to RTIP. It is because as follows. As RTIP increases, the pressure ratio across the turbine increases but the enthalpy at turbine inlet decreases, so it has both the increasing and decreasing factors with respect to RTIP. For a specified RTIP, the specific net work increases with increasing source temperature, due to increased enthalpy of working fluid at the turbine inlet.

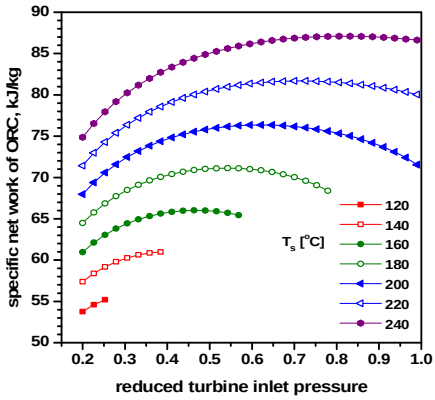


Fig.3 Effects of turbine inlet pressure on the specific net work for various source temperatures.

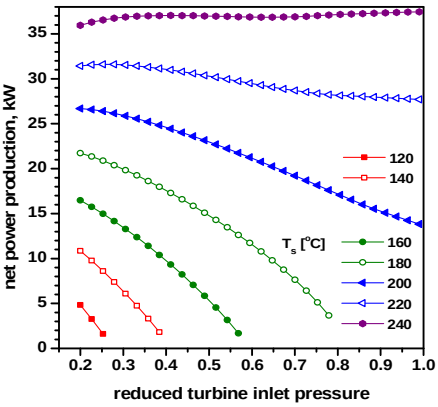


Fig.4 Effects of turbine inlet pressure on the net power production for various source temperatures.

Fig. 4 illustrates the effects of turbine inlet pressure on the net power production of the system in the ORC for various source temperatures. The net power production decreases with increasing RTIP for low source temperatures. The decreasing rate with respect to RTIP decreases with increasing source temperature. So it can be seen from the figure that the net power production increases with increasing RTIP for high source temperatures such as T_s = 240°C. It can be explained as follows. At each cycle of the system, the net power production is obtained as the product of specific net work and the mass flow rate. As previous stated, the specific net work at the ORC has a peak with respect to RTIP, but the mass flow rate of working fluid decreases with increasing RTIP and it plays a dominant role on the result. For a given RTIP,

the net power production increases with increasing source temperature, due to the increase in the mass flow rate.

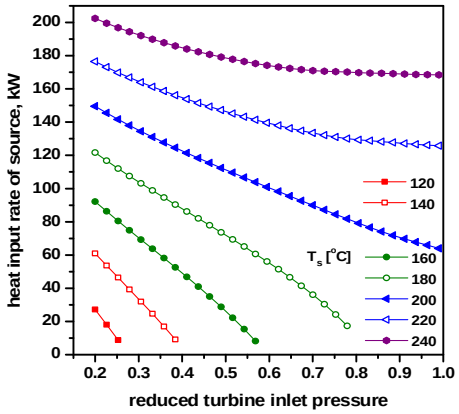


Fig.5 Effects of turbine inlet pressure on the heat input rate for various source temperatures.

Fig. 5 displays the effects of turbine inlet pressure on the heat input rate by source fluid for various source temperatures. The heat input rate is obtained from the product of mass flow rate of working fluid and the specific heat input, and it is proportional the temperature difference between source inlet and exit temperatures. The heat input rate decreases with increasing RTIP, since as RTIP increases in the ORC, the evaporation temperature of working fluid in the heat exchanger increases, which leads to an increased source exhaust temperature. For a given RTIP, the heat input rate increases with increasing source temperature, due to the increased temperature difference between source inlet and exit temperatures. Fig. 6 shows the effects of turbine inlet pressure on the total heat transfer capability of the system for various source temperatures. The total heat transfer capability decreases with increasing RTIP, since as RTIP increases in ORC, the heat input rate decreases and the logarithmic mean temperature difference increases due to the increased source exhaust temperature. For a given RTIP, the total heat transfer capability increases with increasing source temperature, due to the increased mass flow rate of working and the heat input rate.

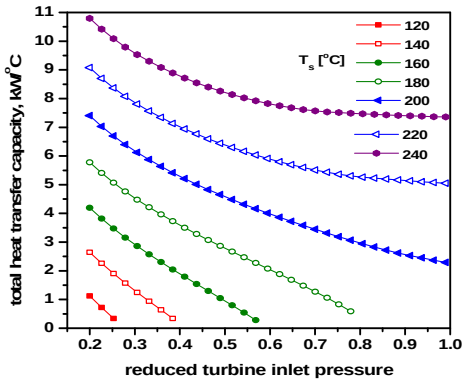


Fig.6 Effects of turbine inlet pressure on the total heat transfer capability for various source temperatures.

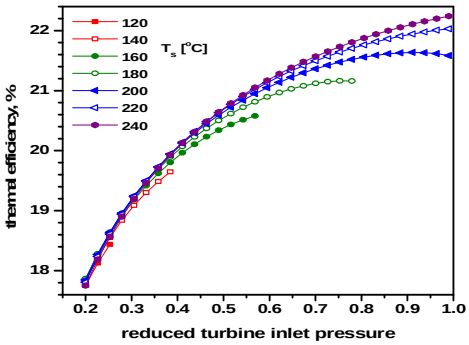


Fig.7 Effects of turbine inlet pressure on the thermal efficiency for various source temperatures.

Fig. 7illustrates the effects of turbine inlet pressure on the thermal efficiency of the system for various source temperatures. The thermal efficiency increases with increasing RTIP. It is because the thermal efficiency is defined as the ratio of net power production to the heat input rate to the system, as is seen in Eq. (9).As RTIP increases, both the heat input rate and the net power production decrease, but the effect ofdecreasing heat input rate is dominant compared the net power production. For a given RTIP, the thermal efficiency increases with increasing source temperature. It is because as the source temperature increases, both the heat input rate and the net power production increase, but the effect of increasing net power production is dominant compared the heat input rate. However, the effect of source temperature on the thermal efficiency is insignificant compared to the RTIP.

CONCLUSIONS

This paper presents the thermodynamic analysis of the combined cycle consisting of the organic Rankine cycle and the LNG Rankine cycle for the recovery of low-grade heat source energy and LNG cold energy. Air with 1 kg/s is assumed to be the heat source, and R123 is considered as the working fluid. The parametric study is carried out with respect to the reduced turbine inlet pressure (RTIP) and the source temperature.The results show that the specific net work of ORC has a peak but the net heat input rate of system decreases and the thermal efficiency increases as RTIP increases. The net power production and the thermal efficiency increase with increasing source temperature but the effect of source temperature on the thermal efficiency is insignificant.

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