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Thermodynamic Analysis of an Integrated System for LNG Regasification and Power Production

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ABSTRACT— Today, natural gas is used in domestic as well as for various industrial purposes. Natural gas being found in remote and specific locations, it has to be transported for long distance before supplied to customers around the globe. Producing liquefied natural gas (LNG) is a highly energy intensive process and consumes about 10 – 15% of total energy spent for LNG production. However, eventually for the end use, natural gas need to be supplied in its gaseous form and the process is known as regasification. Energy spent for the liquefaction of natural gas is wasted unless it is recovered during this regasification process. Cold exergy of LNG can be utilized for improving the performance of Rankine cycle based power plants. This paper has proposed a power system in which low temperature waste heat can be effectively recovered and LNG can be vaporized to atmospheric conditions. The system consists of propane Rankine cycle and LNG power generation system using direct expansion. It is modeled by considering mass and energy balance in each component. The result shows that proposed cycle has good in performance and gives an exergy efficiency of 37.25 %. The effect of key parameters on system performance was also investigated.

Keywords— LNG, Regasification, Exergy, Rankine cycle, Power generation.

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1. INTRODUCTION

LNG is produced by liquefaction of natural gas after removal of impurities, water and acid components. The liquefaction process requires a significant amount of energy to lower the temperature (about 110K) [1].LNG should be regasified at the regasification terminals. Large amount of cold energy is released during this process and usually this transferred to sea water (Open rack vaporizers) or Atmospheric air (ambient air vaporizers). Typically sea water is used as the heat source to vaporize LNG [2]. This process not only consumes a large amount of power for driving the sea water pump but also wastes plenty of cold energy. With the increasing demand for cleaner fuels, LNG is now playing an even significant role as energy resource [3]. The type of the regasification technique used depends on many factors such as location of the plant, conditions of surroundings, local rules and regulations etc. From the Carnot cycle, we know that the efficiency of a thermal cycle is proportional to the temperature difference between heat source and heat sink. The power cycle based on the cold energy of LNG and low-grade heat source enlarges the temperature difference between source and sink, which can not only efficiently recover and utilize the energy consumed during the liquefying process of natural gas but also greatly increase the utilizing efficiency of low-grade energy. It is significant for and environment protection [4].

In previous studies, the temperature of the heat source of Rankine cycle is usually higher [5, 6]. The temperature of gas turbine exhaust or heat from process industries are about several hundreds. By utilizing the cold exergy of LNG this waste heat can be recovered effectively since very low temperature heat sink is available for rejection [7, 8].

2. DESCRIPTION OF THE PROPOSED PLANT

The proposed system consists of a Rankine cycle with propane as the working fluid and natural gas expanding cycle as shown in figure. In Rankine power cycle, low temperature waste heat such as exhaust gas from IC engines, Heat from process industries or from incinerators used as heat source and LNG as heat sink. The LNG cycle with Natural gas direct expansion utilizes physical exergy of LNG. The gaseous natural gas is continuously warmed and supplied to distribution networks.

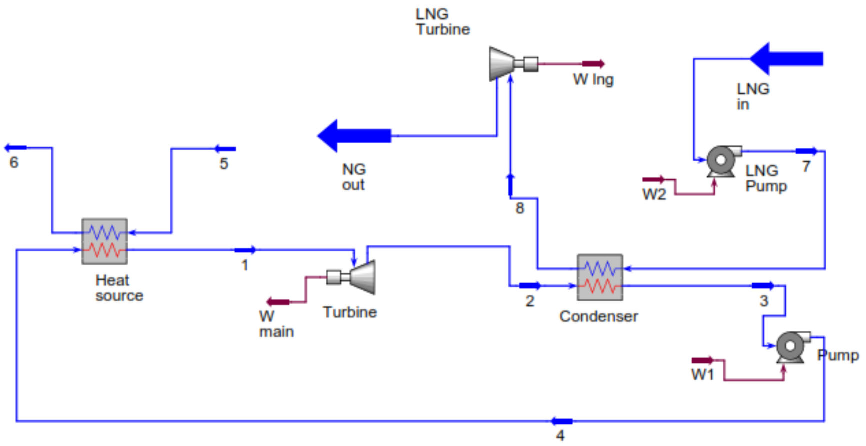

Figure 1: Proposed Combined system

The main advantages of proposed system with low temperature Rankine cycles are: Low temperature Heat can be converted to work effectively. Physical exergy of LNG can be utilized by direct expansion process. The proposed combined system can not only efficiently recover low-temperature waste heat but also fully utilize the cold energy of LNG. Because Propane is a working fluid with very low boiling point and the cold energy generated during the LNG vaporization is used to condense the propane turbine exhaust, the propane vapor can expand to a much lower temperature. The composition of LNG adopted here is summarized in Table 1 [14].

Table 1: Composition of LNG

Component	**Mole fraction**
N_2	0.0007
CH_4	0.8877
C_2H_6	0.0754
C_3H_8	0.0259
n-C_4H_8	0.0056
i-C_4H_8	0.0045
n-C_5H_{12}	0.0001
i-C_5H_{12}	0.0001

3. THERMODYNAMIC MODELING AND ANALYSIS

3.1 Energy analysis

The thermal analysis of each facility in the Rankine power cycle described above can be done with following equations:

Pump, $W_{in} = m\times (h_4 - h_3)$

Turbine, $W_{out} = m\times (h_1 - h_2)$

Net work output $W_{net} = W_{out} - W_{in}$

Thermal efficiency of the cycle $= W_{net} / Q_{in}$ (1)

Heat input, $Q_{in} = m\times Cp\times \Delta T$

where m is mass flow rate in kg/s, h is enthalpy in kJ/kg, ΔT is difference in temperature in K, Cp is specific heat in kJ/kg-K, W is work in kW and Q_{in} is heat in kW

3.2 Exergy analysis

Exergy is defined as the maximum theoretical work obtainable by bringing the fluid into equilibrium with the environment. In the comparison, exergy analysis rather than energy analysis is adopted as the former provides a more accurate representation in terms of conversion. Exergy analysis combines the first and second laws of thermodynamics, and is a powerful tool for analyzing both the quantity and the quality of energy utilization.

$$\Sigma j\ Qj\ (1-T_0/T_j) - W + \Sigma in\ Ex_{mass} - \Sigma out\ Ex_{mass} - Ex_{dest} = 0 \qquad (2)$$

In the above equation, $\Sigma j\ Qj\ (1-T_0/Tj)$ represents the exergy transfer associated with heat and T0 is the reference state temperature (K). Exergy transfer accompanying the work interaction equals the electrical or mechanical work (kW) itself. Terms $\Sigma in\ Ex_{mass}$, $\Sigma out\ Ex_{mass}$ of equation respectively correspond to the exergy transfer accompanying mass crossing the boundaries at the inlet and outlet of the control volume. This is expressed as:

$$Exmass = m \times Ex = m\ [(h-ho)-To(s-so)] \qquad (3)$$

Where, m represents the mass flow rate, (kg/s); Ex represents the exergy per unit mass, (kJ/kg), h and s correspond to the enthalpy and entropy of the fluid respectively. h_0 and s_0 represent the enthalpy and entropy at the reference state (T_0, P_0) respectively. For this analysis, the atmospheric condition is taken as the reference state (T0 = 300 K and P0 = 1.01325 bar).

In the case of heat exchangers employed in a refrigeration/liquefaction system, the objective is to increase the exergy of the high pressure hot stream such that finally it is capable enough to absorb heat from the given external load. So the rational exergetic efficiency of the heat exchanger [6] is defined as

$$\eta_{EX\ hx} = m_{HP}\ (Ex_{HOUT} - Ex_{HIN})\ /\ m_{LP}\ (Ex_{CIN} - Ex_{COUT}) \qquad (4)$$

4. RESULTS AND DISCUSSIONS

The following assumptions are made for the proposed system analysis:

- The flow is steady and the state of the working fluid at each specific location within the system does not change with the time.
- All components are well insulated.
- Pressure drop and heat loss in pipe lines are neglected.

In order to find out the effect of parameters on exergy efficiency of the cycle, analysis has been carried out at different conditions, such as turbine inlet temperature and turbine inlet pressure. The simulation conditions are summarized in Table 2 and results are shown in the form of graphs

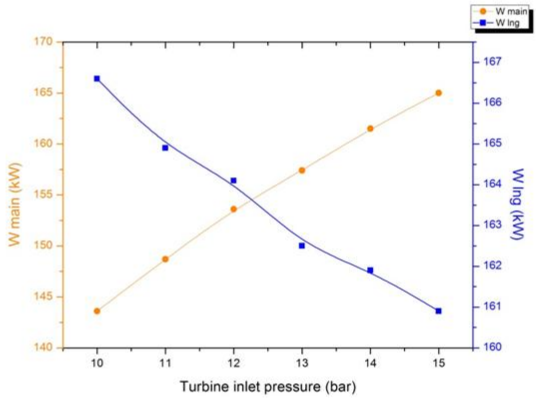

Figure 2: Variations of main and LNG turbine output with Turbine inlet pressure

Table 2: Simulation Conditions

	Simulation Conditions	
	Parameters	**Proposed cycle**
Rankine cycle	Turbine isentropic efficiency	75%
	Pumps isentropic efficiency	74%
	High pressure	1.5MPa
	Low pressure	0.1 MPa
LNG cycle	Storage pressure	200 KPa
	Storage temperature	120.4 K
	Pump isentropic efficiency	74%

Fig 2 and 3 shows the work output from each turbine and net work output from the plant respectively. As turbine inlet pressure increases, the propane turbine output increases and LNG turbine output decreases,

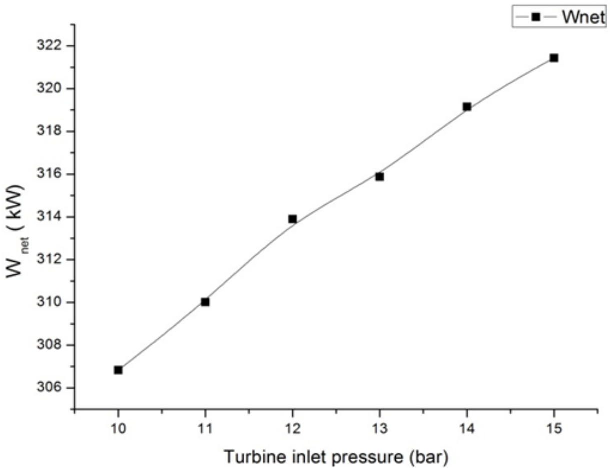

Figure 3: Variations of net work output with Turbine inlet pressure

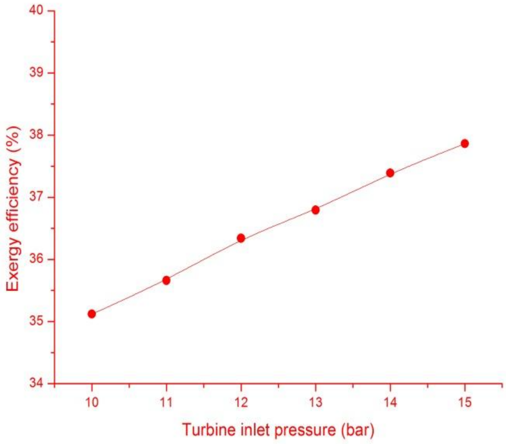

Figure 4: Variations of exergy efficiency with Turbine inlet pressure

The performance of the proposed combined system is evaluated for LNG turbine inlet pressures from 10 to 15 bars with the thermal conditions summarized in Table 2 are kept unchanged. The maximum net work output obtained from both turbines was 321.90 kW at 15bar as shown in graph. The exergy efficiency increases with turbine inlet pressure as shown in figure 4. Maximum exergy efficiency obtained was 37.8 % at 15 bars. Increase in exergy efficiency is due to reduction in irreversibility due to finite temperature differences in condenser.

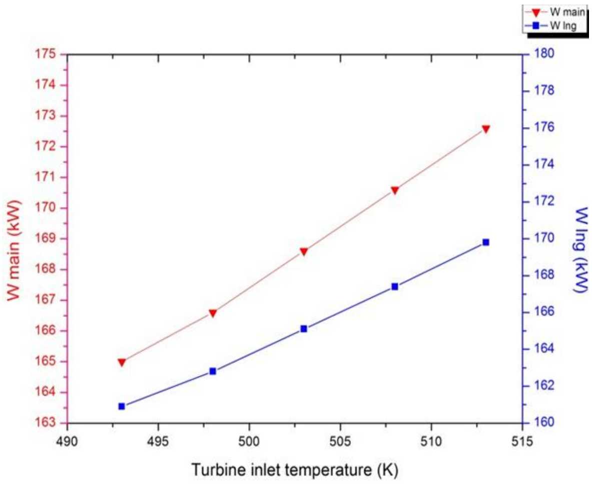

Figure 5: Variations of main and LNG turbine output with Turbine inlet Temperature

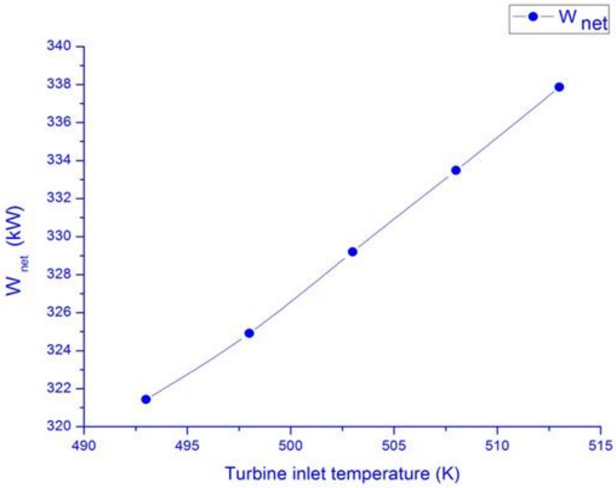

Figure 6: Variations of net work output with Turbine inlet temperature

Figure 5 shows changes in work output of LNG and main (propane) turbine with turbine inlet temperature. As the turbine inlet temperature increases, the work output of main turbine and turbine outlet temperature of working fluid (propane) increases. Hence the heat load on condenser increases and work output of LNG turbine also increases. The maximum work output was found to be172.8 kW for main turbine and 169.8 for LNG turbine. Figure 6 shows the variation of net work output from both turbine, which increases with turbine inlet temperature because the work output from both turbines are increased with increase in Turbine inlet temperature. The maximum work output obtained is 338.1 kW at 512.15 K as shown in figure. Figure 7 shows the exergy efficiency changes with turbine inlet temperature. As the turbine inlet temperature increases, exit temperature also will increase and this can't be used effectively, hence reduces the exergy efficiency of the system.

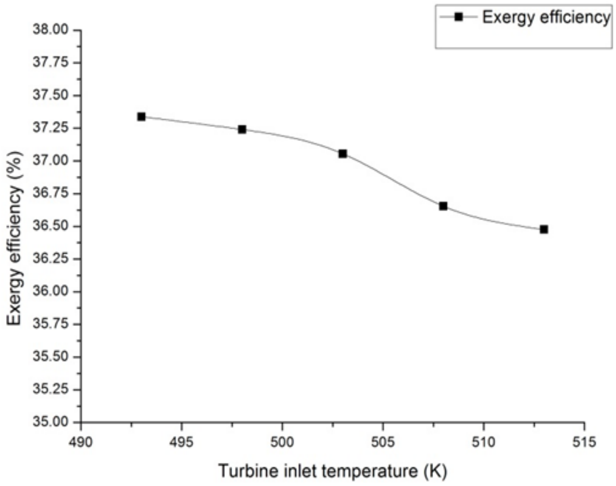

Figure 7: Variations of exergy efficiency with Turbine inlet temperature

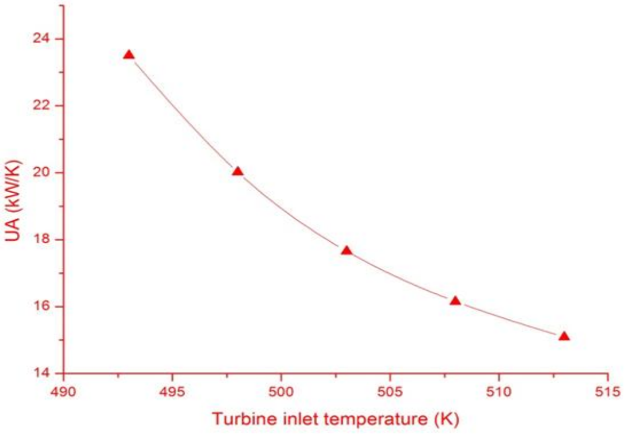

Figure 8: Variations of UA with Turbine inlet temperature

Figure 8 explains the variations of UA with Turbine inlet temperature. The UA indicates the size of heat exchanger/ condenser. As the turbine inlet temperature increases, the outlet temperature also increases. Which improves the ΔT in condenser, since heat transfer rate is constant, condenser requires less amount of working fluid for vaporizing given amount of LNG.

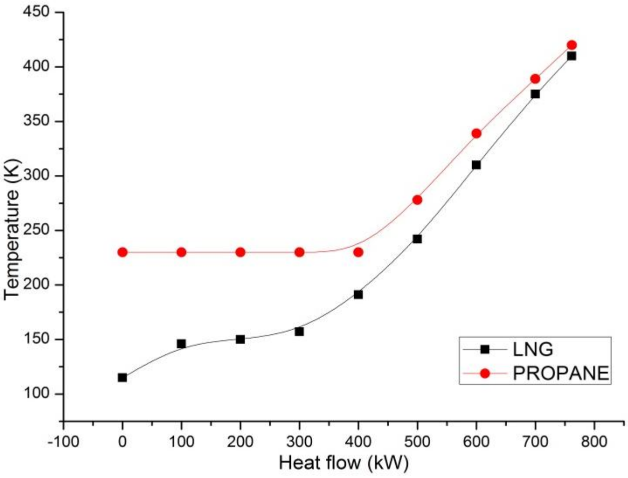

Figure 9: Heat transfer profile of Condenser

Figure 9 shows the heat transfer profile of condenser where LNG get vaporized to warm Natural gas and propane (working fluid) condensed to liquid state. The reason for lower exergy efficiency of the system is irreversibility development due to heat transfer between finite temperature differences.

5. CONCLUSIONS

This study uses power system with propane as working fluid and LNG as its heat sink to recover the low grade waste heat. The steady-state component models are developed in consideration of mass, energy and exergy balances. Based on the thermodynamic analysis, the effects of key parameters such as turbine inlet temperature and turbine inlet pressure on the system performance were investigated. The result shows:

- The exergy efficiency of the proposed system increases with increase in Turbine inlet pressure and decreases with increase in Turbine inlet temperature.
- The maximum exergy efficiency of the system is 37.25 %, which is higher than existing Rankine cycles.
- An increase in Turbine inlet temperature reduces the size of condenser due to the reduction in mass flow rate of working fluid.

- Heat transfer profile of cold and hot fluid in Condenser shows the irreversibility development and which is accounted for reduction in exergy efficiency.

In order to evaluate the advantages of proposed power system a more detailed techno-economic study based on above results are needed.

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