

Exergy Cost Analysis of New Method for Efficiency Improvement in Small Gas Turbines Using LNG Cold Exergy

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Abstract

The gas turbine systems are some equipment; that have extensive industrial and military applications due to their unique characteristics. One of the major problems in association with utilization of gas turbines is the output power and thermal efficiency. One of the major methods that are used for increasing the power and thermal efficiency in this type of power systems is compressor's inlet air cooling. In this paper, the effect of fuel replacement for methane gas-fired gas turbines with LNG and cooling the compressor's air inlet with the cold exergy of LNG fuel evaporation will be examined. In the first step, the energy balance equation is applied to the gas turbine with Methane and LNG fuel separately. Then the problem of fuel replacement in some mini and micro gas turbine has been examined and the appropriate engine with the most power and thermal efficiency is selected. In the next step, the exergetic analysis of considered systems is performed and exergetic parameters of the two systems are compared together. Finally, the advance exergy analysis is performed and effects of fuel replacement on exergetic cost parameter and environmental characteristics are evaluated.

Keywords: Gas turbine; LNG; cooling; power; efficiency; exergy cost.

1. Introduction

Gas turbines are the subset of power generation systems that have many major applications due to their specific capabilities from low power range to high power. One of the problems related to gas turbines is the low thermal efficiency in these kinds of turbines. Thus, many extensive types of research to increase the thermal efficiency of these turbines have been performed. The methods that used for thermal efficiency increment can be divided into three different categories.

- a) Cooling the compressor air inlet
- b) Heat recovery from exhaust
- c) Using energy storage

The most proposed methods in the second and third categories required the change in the gas turbine cycle, combustion mechanism, design parameters and thermodynamic conditions of the cycle. Therefore, the use of these methods is only possible for companies that have access to turbine design specifications. In this section, all of the methods used for improvement in the gas turbine engine are reviewed.

Foster [1] proposed a method and apparatus for increasing output power and the thermal efficiency of a gas turbine power plant. In this research, the inlet air is super-chilled before it enters the compressor. Enhancing gas turbine performance by intake air cooling using an absorption chiller was studied by Mohanty and Paloso Jr [2]. Authors found that air temperature reduction from the ambient condition to 15°C increase the instantaneous output power between 8 to 13% and 11% additional electricity could be generated from the same gas turbine power plant. De Lucia et al. [3] investigated the benefit of compressor

inlet air cooling for gas turbine cogeneration plants with the comparative analysis of different cooling methods. These analyses have performed for the LM6000 gas turbine in a cogeneration plant operated in base load. In this research, the absorption and evaporative air cooling systems are considered and their performance and economic benefits compared for the dry low-NOx LM6000 version. Najjar [4] compared the effects an inlet air pre-cooler on performance enhancement of gas turbine engines. In this paper, the cooling system is connected to the evaporator of an ammonia absorption chiller that is driven by the heat recovered from the engine exhaust gases. Palsson et al. [5] studied the combined solid oxide fuel cell and gas turbine systems for efficient power and heat generation. In this paper, the performance of the gas turbine system and the effect of operational parameters are evaluated. Results show that the pressure ratio has an effective impact on the performance and the electrical efficiencies of more than 65% are possible at low-pressure ratios. The effects of the evaporative air inlet and outlet cooling on the recuperated gas turbine cycle were analyzed by Bassily [6]. Four different layouts of the recuperated gas turbine cycle were presented. The results showed that evaporative cooling of the inlet air could boost the efficiency up to 3.2% and the evaporative outlet cooling could increase the power up to 110% and cycle efficiency up to 16%. Integration of steam injection and inlet air cooling for a gas turbine system was investigated by Wanga and Chiou [7]. In this study, the computer code was developed to simulate a Taipower's Frame 7B simple cycle gas turbine generator sets. Under the condition of local summer weather, the benefits obtained from the system implementing both steam injection gas turbine and inlet air cooling system

features are more than a 70% in power and 20.4% improvement in heat rate. A comparison between two different types of air coolers, namely water spraying system, and the cooling coil system was carried out by Alhazmya and Najjar [8]. Yang et al. [9] conducted an analytical method for evaluation of the inlet air cooling for gas-steam combined cycle power plant with absorption chiller and saturated evaporative cooler. It was shown that inlet fogging is superior in power efficiency at T_a 15-20°C though it gains a smaller profit margin than inlet chilling. GTCC inlet chilling with absorption chiller is preferable in the zones with $T_a > 25^\circ\text{C}$ and $\text{RH} > 0.4$. Farzaneh-Gord et al. [10] carried out a comparison between two commons and one novel inlet air cooling method using turbo-expanders to improve the performance of a gas turbine located at the Khangiran refinery in Iran. Olivenza et al. [11] presented the new analytical model for a hybrid solar driven gas-turbine plant. The model validation by comparing with real prototype plants is very satisfactory. Dabwan and et al. [12] analysis the new integrated solar gas turbine cogeneration plant (ISGCPP). In this paper, results show that optimal integration of LFR with gas turbine cogeneration system can reduce CO₂ emission by 18%. The new combined-cycle of gas turbine and Stirling engine is proposed for power generation by Entezari et al [13]. In this paper, the exergoeconomic optimization is performed on the gas turbine cycle as well as combined-cycle. Genetic algorithm and TOPSIS method implemented to find the optimal operating point. Santos and et al. [14] modeled Brayton hybrid thermo-solar plants and considered Multi-stage recuperative configurations and different working fluids. Results show the good improvement over a reference plant.

In this paper, the effect of using LNG as the fuel in gas turbines and using its cold exergy to cool the compressor's inlet air is investigated. First, the energy balance rules are applied to each component and the operational parameters are evaluated. Then the effect of the selected cooling method on gas turbine systems efficiency is performed. For this object, gas turbines with different capacities (30 – 2000 KW) have been examined and finally, suitable capacity for the suggested cooling procedure is chosen. After choosing suitable power, the effect of different parameters on the power and efficiency of the gas turbine is examined. To accurate evaluation of the proposed method, the advanced exergy cost analysis is examined and exergy cost factors and environmental impacts of the proposed system are evaluated.

2. Thermal Simulation

A schematic view of the gas turbine cycle with methane and LNG fuel is shown in Figures 1. The LNG contains 90.28 percent methane, 6.33 percent ethane, 2.49 percent propane, 0.49 percent butane, and 0.41 percent nitrogen. In the gas turbine cycle with LNG fuel, first, the available LNG in the tank enters to the LNG vaporizer then in the warming up process, heat is transferred from the air at compressor inlet to LNG and the LNG converted to steam. The LNG vaporizer is uncomplicated heat exchanger which vaporizes liquefied gas by using heat absorbed from the ambient air. In this heat exchanger, the LNG passes through a number of interconnected tubes in various series and parallel paths. Initially, energy analysis accordance with the first thermodynamic law is carried out to investigate the effects of replacement methane fuel with LNG and use cold exergy from LNG evaporation, and according to specified

parameters and exergy analysis, the amount of inefficiency in the various components of the cycle is determined.

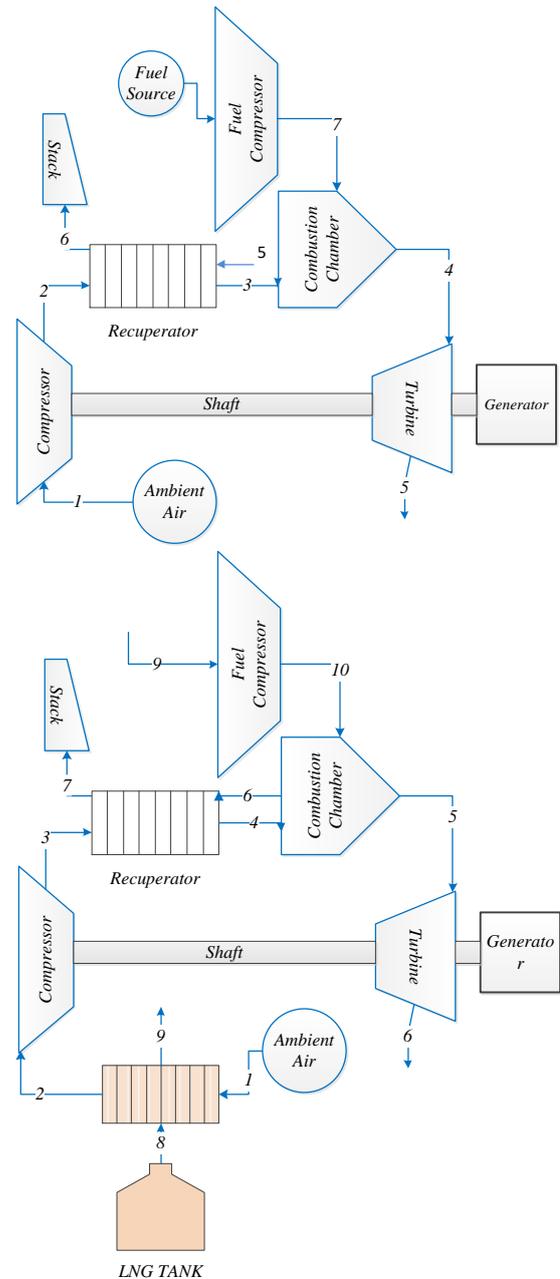


Figure 1. Schematic of gas turbine cycle with methane (CH₄) and LNG fuel.

2.1 Energy analysis

To simulate the gas turbine system, the energy balance equation in steady state conditions for each component is applied:

$$\sum_{in} \dot{m}_{in} h_{in} - \sum_{out} \dot{m}_{out} h_{out} - \dot{W} + \dot{Q} = 0 \quad (1)$$

Energy balance equations for each component based on the above equation and decision and input variables is evaluated and are mentioned in Appendix A.

2.2 Exergy analysis

The exergy equation for a selected control volume is generally expressed as below [15]:

$$\sum \dot{Q} \left(1 - \frac{T_0}{T}\right) - \dot{W} + \sum_{in} \dot{m}_{in} e_{in} - \sum_{out} \dot{m}_{out} e_{out} = \dot{E}_D \quad (2)$$

Assuming the adiabatic behavior for all equipment, the term $\dot{E}_Q (\dot{Q}(1 - T_0/T))$ is ignored in the exergy equations in steady state conditions. \dot{E}_D is the exergy destruction of the irreversible system. The specific exergy value for each input or output flow is determined from the following equation [15]:

$$e = e_{ph} + e_k + e_p + e_{ch} \quad (3)$$

where e_{ph} , e_k , e_p and e_{ch} , are physical, kinetic, potential and chemical exergy respectively.

In this study, the amount of kinetic and potential exergy has not been considered and the chemical exergy is calculated based on exergy table. So special exergy in each case is equal to total physical and chemical exergy that is calculated from the equation (4).

$$e = e_{ph} + e_{ch} = [(h - h_0) - T_0(s - s_0)] + \left[\sum x_k \bar{e}_{CH,k} + \bar{R}T_0 \sum x_k \ln x_k \right] \quad (4)$$

where, s and h are specific entropy and specific enthalpy, T_0 is the ISO temperature, s_0 , and h_0 are entropy and enthalpy in T_0 and P_0 , \bar{R} is gas constant and x_k is the molar fraction of each element in the mixture (reference point ISO condition: $T_0 = 15^{0c}$ and $P_0=1.013$ bar).

2.3 Exergy cost analysis

Exergy cost of a flow is the number of exergy sources that consumed by the system for producing the flow. There are various applications that not only the exergy content is the critical parameter, but the required exergy for flow production is important. If the required resources are calculated in terms of exergy, the embodied exergy can be evaluated. The detail of this concept so-called exergy cost analysis is presented in [16] and in this section, the basic concept of this method is presented.

The exergy cost (Ex_i^*) of specific flow (e_i) is the amount of exergy needed to produce this flow. Based on this definition, the unit exergy cost of a flow is the exergy cost per unit

exergy $c_i = Ex_i^*/Ex_i$. The exergy cost of a flow can be with a set of interrelated cost values:

- The exergy cost is a conservative property and, therefore, the cost of the inlets is equal to that of the outlets ($A.Ex^* = 0$)
- In the absence of external assessment, the exergy cost of each flow entering the plant is equal to its exergy ($Ex_i^* = Ex_i$)
- If a fuel stream of a component has an output flow (non-exhausted fuel), its unit exergy cost is equal to input flow of this fuel stream.
- If the product stream of a component has several outputs flows, all of the flows have the same unit exergy cost.

The production of component (i) is used as fuel of other components or as a part of the total production of the plant:

$$P_i = Ex_{i0} + \sum_{j=1}^n Ex_{ij}, \quad i = 0, 1, \dots, n \quad (5)$$

In the above expression, if the component 0 is considered as the system environment and j present the other component. The resources consumed by component (i) can be written as:

$$F_i = Ex_{0i} + \sum_{j=1}^n Ex_{ji}, \quad i = 0, 1, \dots, n \quad (6)$$

In accordance with the proposed model, the cost of the resources and product in component (i) can be calculated with Eq. (7) and (8):

$$C_{F,i} = Ex_{i0}^* + \sum_{j=1}^n Ex_{ij}^* \quad (7)$$

$$C_{P,i} = Ex_{0i}^* + \sum_{j=1}^n Ex_{ij}^* \quad (8)$$

The unit exergy consumption (k) is defined as the ratio of unit exergy cost of the product to unit exergy cost of fuel. Thus the component with minimum k defines as the ideal component in each system.

Table 1. The operational parameters of selected gas turbines.

Manufacturer	Engine's name	Output Power (KW)	rotational speed (RPM)	Pressure ratio	Stack mass flow rate (kg/s)	Stack temperature ($^{\circ}C$)
Capstone	C30	30	96000	4	0.31	275 $^{\circ}C$
Capstone	C200	200	61000	4	1.3	280 $^{\circ}C$
Capstone	C1000	1000	61000	4	6.7	280 $^{\circ}C$
OPRA	OP16	1.88×10^3	26000	6.7	8.9	573 $^{\circ}C$

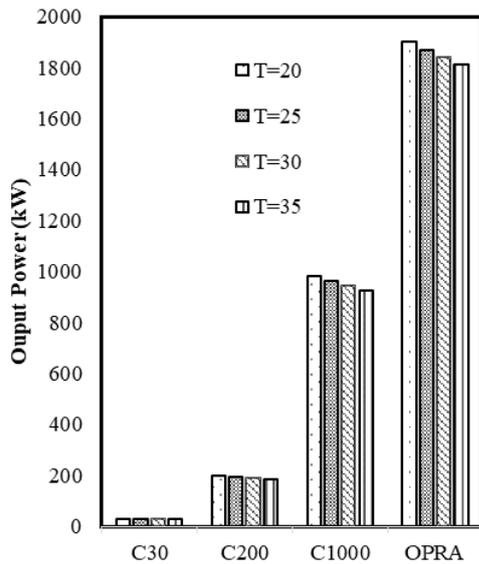
Table 2. Comparison between power and efficiency of different gas turbines (methane and LNG) in ISO conditions.

Company's name and engine's name	Output power (KW)			Thermal efficiency		
	CH4	LNG	Improvement (%)	CH4	LNG	Improvement (%)
C30	30	31.23	4.1	24.35	24.71	1.4784
C200	200	206.2	3.1	33.42	33.54	0.359
C1000	1000	1031.8	3.18	32.95	33.1	0.4552
OPRA	1933.3	2041.1	5.5759	24.71	25.36	2.6305

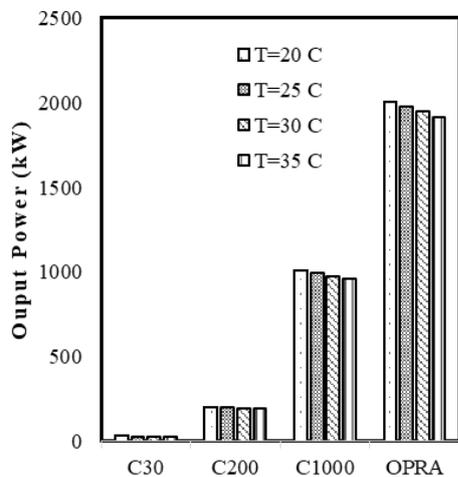
3. Results and Discussion

Based on the thermodynamic analysis, the various energy and exergy parameters are calculated and listed in Appendix C. In order to analyze the effect of using cold exergy of LNG evaporation and cooling the inlet air to the compressor, four engines with different output power (20 – 2000 KW) are modeled and examined. The operational characteristics and useful variables of the selected engines are shown in Table 1.

The comparison between thermal efficiency and output power for selected gas turbines are shown in Table 2. The results show that using LNG instead of methane in the gas turbine cycle will always be accompanied by increased power and efficiency. This increase in the output power of the cycle is varied from 3 to 5 percent and for power varied from 0.3 to 2.6. Also, the results show that the highest increase in power and efficiency is related to the 2 MW OP16. So this engine is select for further analysis.

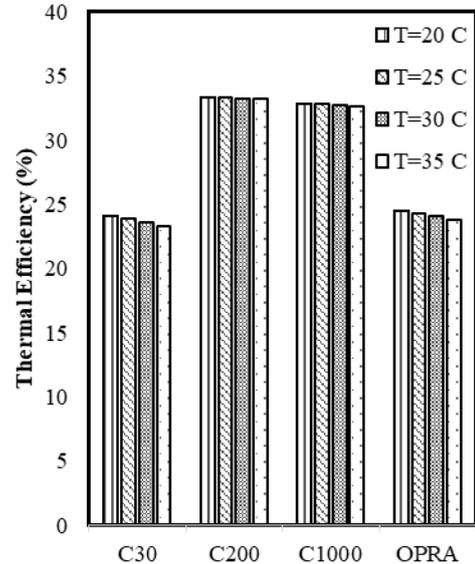


A) methane fuel

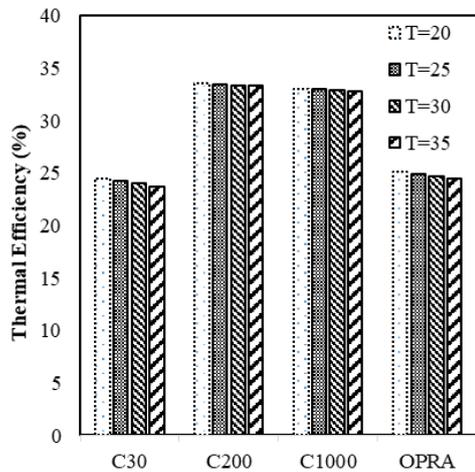


B) LNG fuel

Figure 2. Amount of output power for different motor engines.



A) methane fuel



B) LNG fuel

Figure 3. Amount of thermal efficiency for different motor engines. A) methane fuel. B) LNG fuel.

Comparison between output power and thermal for selected gas turbines with methane and LNG fuel at various ambient temperatures are shown in Figure 2 and Figure 3. Results show that the output power and thermal efficiency are decreased with increase in ambient air temperature. Also, for at all ambient temperature, the OP16 2 MW gas turbine has the highest output power and efficiency.

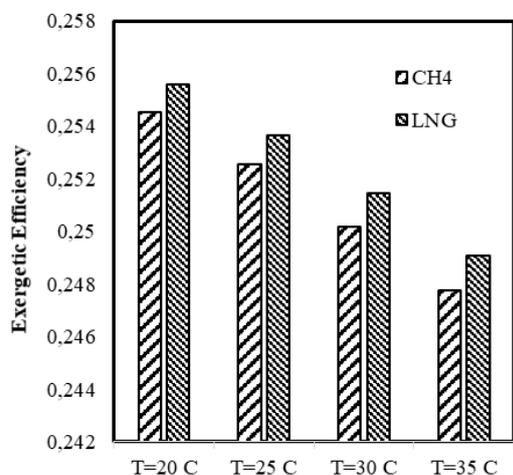


Figure 4. Comparison of exergetic efficiency for gas turbine with LNG and Methane fuel.

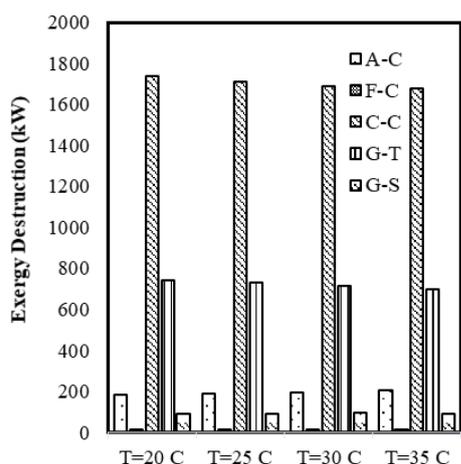


Figure 5. Comparison of the exergy destruction for different equipment for 2MW gas turbine with methane fuel. (A-C: air compressor, F-C: fuel compressor, C-C: combustion chamber, G-T: gas turbine, G-S: stack).

The magnitude of exergetic efficiency at different ambient temperature for a 2MW engine for two modes of methane and LNG fuel is shown in Figure 4. The results show that, for both cases, the exergetic efficiency is decreased with increase in ambient air temperature. This decrement can be analyzed according to the power reduction in gas turbine cycle due to the increase in ambient temperature. The results show that the exergy efficiencies increase slightly (less than 1%) by fuel replacement. This improvement in exergetic efficiency can be analyzed with the cooling of the air entering the compressor in this cycle as well as the reduction of fuel consumption. Although the improvement in exergetic efficiency is low, however, due to the suitable increase in the thermal efficiency (despite the increase of a new heat exchanger to the system), this is a good improvement for the proposed system. The amount of exergy destruction in the different equipment of 2MW engine cycle is also shown in Figure 5. As the results show, the greatest amount of exergy destruction occurs in the combustion chamber. Therefore, it can be concluded that reducing fuel consumption can have an effective contribution to the rate of destruction of exergy in the combustion chamber and exergy destruction of the gas turbine system.

The exergy cost analysis of 2MW gas turbine cycle with LNG fuel at different temperatures (15 and 25 °C) is shown in the following figures and tables. Table 3 shows a summary of fuel exergy, product exergy, irreversibility, the unit exergy cost of the product and its decomposition for each component. In this table, the results show that the combustion chamber and air compressor have the maximum unit exergy cost consumption, respectively. It is shown that these two components have the lowest efficiency in gas turbine cycle. Also, the maximum amount of product exergy cost is related to the stack. This cost shows the waste cost and thus the greatest effect of environmental destruction is due to this equipment. Furthermore, the comparison between Tables 3 and 4 show that, with increasing ambient temperature, cost of unit exergy in the compressor, heat exchanger, fuel compressor which is before the combustion process is increased and in the remaining equipment is decreasing. Also, the results show that the fuel compressor has the lowest unit exergy and thus this component can be defined as an ideal device.

Table 3. The parameters related to the exergy cost in the various equipment for 2-MW gas turbine cycle at $T_a=15^\circ\text{C}$.

Component	Fuel Exergy [kW]	Product Exergy [kW]	Irreversibility I [kW]	k (kJ/kj)	cp (kJ/kj)
Compressor	2019.6	1816.8	202.8	1.1116	1.1116
Combustion	9919.5	7824.4	2095.2	1.2678	1.3147
Gas Turbine	7824.4	7081.5	742.9	1.1049	2.4739
Heat Exchanger	8200.5	8041.6	158.9	1.0198	1.0198
Fuel Compressor	8108.1	8102.7	5.4	1.0007	1.0203
Stack	2923.5	2848.4	75.0	1.0263	2.5391

Table 4. The parameters related to the exergy cost in the various equipment for 2-MW gas turbine cycle at $T_a=25^\circ\text{C}$.

Component	Fuel Exergy [kW]	Product Exergy [kW]	Irreversibility I [kW]	k (kJ/kJ)	cp (kJ/kJ)
Compressor	2098.6	1884.2	214.4	1.1138	1.1138
Combustion	9843.5	7822.5	2021.0	1.2584	1.3067
Gas Turbine	7822.5	7092.0	730.5	1.1030	2.4513
Heat Exchanger	8055.6	7897.6	158.0	1.0200	1.0200
Fuel Compressor	7965.2	7959.3	5.9	1.0007	1.0206
Stack	2922.0	2861.6	60.4	1.0211	2.5030

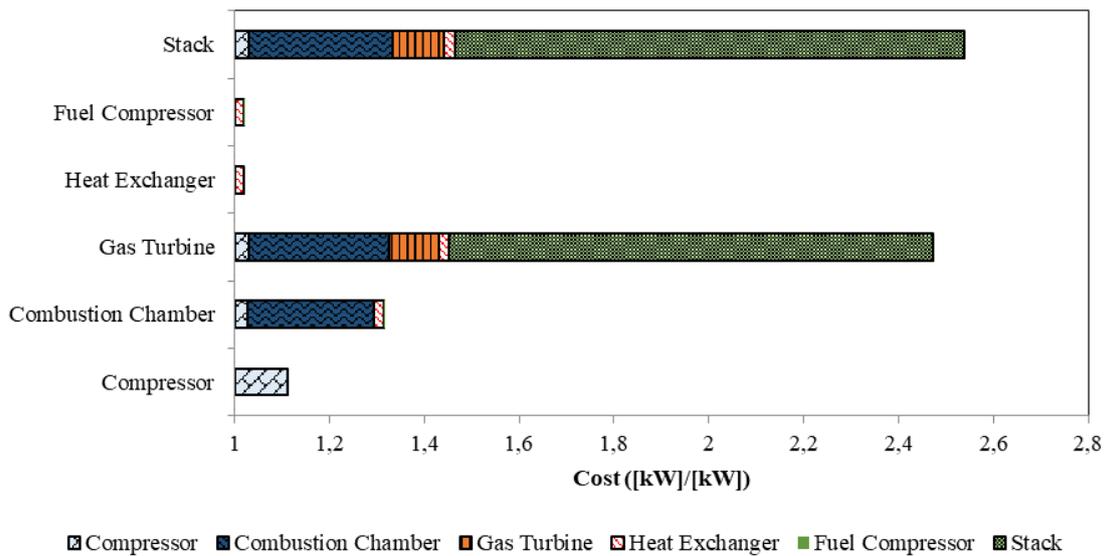


Figure 6. Exergy cost ratio in the different equipment of 2-MW gas turbine cycle at the temperature of 15°C .

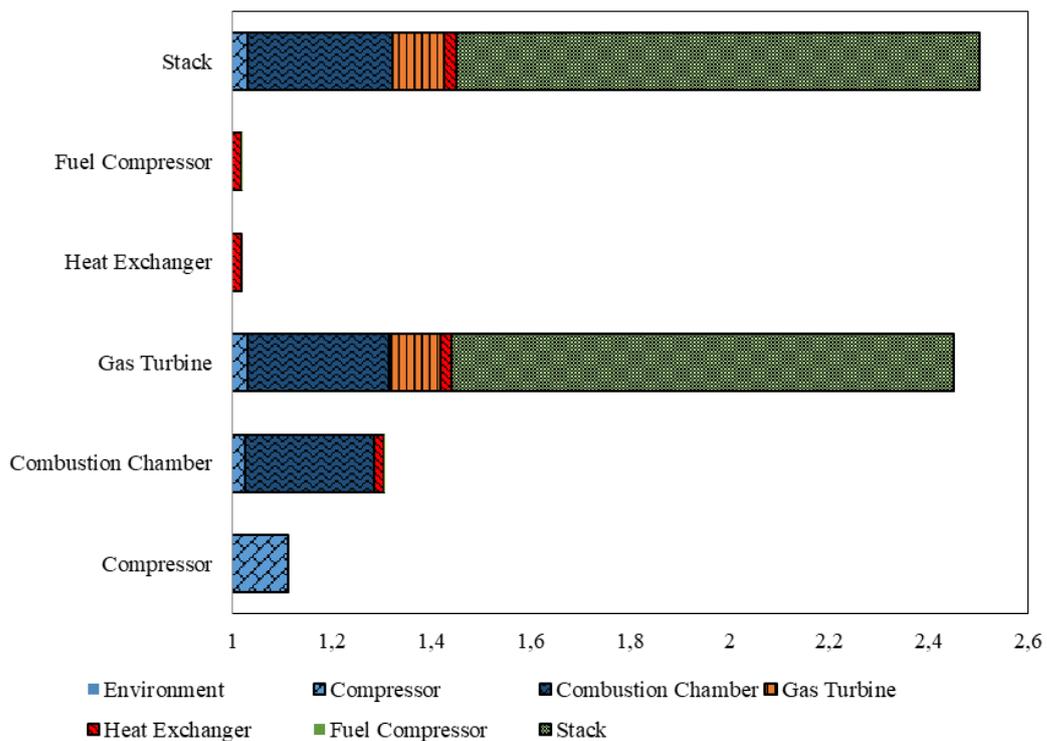


Figure 7. Exergy cost ratio in the different equipment of 2-MW gas turbine cycle at the temperature of 25°C .

The graph of the unit exergy cost of the products of different component in the gas turbine cycle is shown in Figure 6 and 7. In these figures, the effect of the irreversibilities taking place in the production process and the residues are considered. Figures show that highest influence of the cost is stack and combustion chamber. In these devices have major effects on the unit cost of all components. But LNG heat exchanger only affects its own cost. Compression of Figure 6 and 7 show that increasing in ambient air temperature does not affect unit exergy cost of components.

4. Conclusions

In this paper, the effect of using LNG fuel instead of methane on thermal efficiency, exergetic parameters and environmental effect in small gas turbines is considered. The results if the first and second law of thermodynamics show that the power and efficiency of gas turbine engines in the range of micro and mini power is improved using the cold exergy of LNG. But the highest improvement has been seen in a 2-MW gas turbine engine. The results of exergy analysis show that, in the gas turbine with LNG fuel, the exergetic efficiency is higher than gas turbine with methane fuel. Results of exergy cost analysis show that the combustion chamber and stack have the maximum unit exergy cost in gas turbine cycle and these components have major effects on the unit cost of all components. Thus improvement in these components can decrease the exergy destruction and environmental disadvantage.

Nomenclature

Efficiency	η
Enthalpy	h
Entropy	s
Exergy	e
Exergy Cost	c
Exergy Destruction	\dot{E}
Gas Constant	\bar{R}
Heat	\dot{Q}
Mass Flow Rate	\dot{m}
Molar Fraction	x_k
Output Power	\dot{W}
Pressure Drop	ΔP
Pressure	P
Specific Heat	C_p
Temperature	T
Thermal Fuel Value	LHV

Subscripts

Potential	p
Kinetic	k
ISO	0
Combustion Chamber	cc
Chemical	ch
Destruction	D
Physical	ph

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Appendix A

The heat exchanger (LNG vaporizer): in LNG vaporizer the inlet air into the compressor is dried by the cold exergy of LNG evaporation and its temperature is reduced.

$$\sum (\dot{m} h)_{in} = H_{fg} + \sum (\dot{m} h)_{out} \quad A-1$$

$$\begin{aligned} \dot{m}_{air} h_1 + \dot{m}_{fuel} h_8 \\ = H_{fg} + \dot{m}_{air} h_2 + \dot{m}_{fuel} h_9 \end{aligned} \quad A-2$$

Air compressor: The compressor with 86% isentropic efficiency increases the inlet air pressure. The required compressor work is part of the output power of the gas turbine.

$$\eta_{is,comp} = \frac{h_{3s} - h_2}{h_3 - h_2} \quad A-3$$

$$h_3 = \frac{h_{3s} - h_2}{\eta_{is,comp}} + h_2 \quad A-4$$

$$\dot{W}_C = \dot{m}_{air}(h_3 - h_2) \quad A-5$$

$$h = c_{p,air} \Delta T \quad A-6$$

Combustion chamber: the amount of thermal fuel value (LHV) is equal to 50000 KJ/Kg. The heat dissipation from the combustion chamber is considered 2% of the combustion heat. It is also assumed that the pressure decreases by 2% of the input pressure.

$$\dot{m}_a h_4 + \dot{m}_f LHV = \dot{m}_g h_5 + \dot{Q}_{l,CC} \quad A-7$$

$$\dot{Q}_{l,CC} = \dot{m}_f LHV(1 - \eta_{CC}) \quad A-8$$

$$\dot{m}_g = \dot{m}_a + \dot{m}_f \quad A-9$$

$$P_5 = P_4(1 - \Delta P_{CC}) \quad A-10$$

The heat exchanger (Recuperator): in the recuperator, the exhaust air from the compressor is preheated by exhaust gases of the turbine. This method avoids heat dissipation and increases cycle efficiency.

$$\sum (\dot{m} h)_{in} = \sum (\dot{m} h)_{out} \quad A-11$$

$$\dot{m}_{air} h_3 + \dot{m}_g h_6 = \dot{m}_{air} h_4 + \dot{m}_g h_7 \quad A-12$$

Gas turbine: power is made by combining the exhaust gases from the combustion chamber to the gas turbine's blades with an isentropic efficiency of 90%,

$$\eta_{is,Tur} = \frac{h_6 - h_5}{h_{6s} - h_5} \quad A-13$$

$$h_6 = \eta_{is,Tur}(h_{6s} - h_5) + h_5 \quad A-14$$

$$\dot{W}_{Tur} = \dot{m}_g(h_6 - h_5) \quad A-15$$

$$h = c_{p,air} \Delta T \quad A-16$$

Output profit: the output profit of a cycle is equal to the difference between the turbine production work and the compressor's work.

$$\dot{W}_{net} = \dot{W}_{Tur} - \dot{W}_C \quad A-17$$

Appendix B

The heat exchanger (LNG vaporizer): in the heat exchanger, the exergy values of work are equal to zero. The heat dissipation from heat exchanger is considered a part of the external irreversibility, and is integrated in the exergy destruction term, and the term of exergy destruction in equations includes the amount of thermal exergy loss.

$$\sum_{in} \dot{m}_{in} e_{in} - \sum_{out} \dot{m}_{out} e_{out} = \dot{E}_D \quad B-1$$

$$(\dot{m}_1 e_1 + \dot{m}_8 e_8) - (\dot{m}_2 e_2 + \dot{m}_9 e_9) = \dot{E}_D \quad B-2$$

$$e_1 = e_{ph} \quad B-3$$

$$e_2 = e_{ph} \quad B-4$$

$$e_8 = e_{ph} + e_{ch} \quad B-5$$

$$e_9 = e_{ph} + e_{ch} \quad B-6$$

Air compressor: assuming that, it is adiabatic, the air compressor has no dissipation, so the exergy temperature value is zero and the exergy analysis is defined as follows.

$$-\dot{W} + \sum_{in} \dot{m}_{in} e_{in} - \sum_{out} \dot{m}_{out} e_{out} = \dot{E}_D \quad B-7$$

$$\dot{W} + \dot{m}_2 e_2 - \dot{m}_3 e_3 = \dot{E}_D \quad B-8$$

$$e_3 = e_{ph} \quad B-9$$

The heat exchanger (recuperator): in the recuperator, similar to the heat exchanger (LNG vaporizer), the heat exergy and work are equal to zero, so the exergy analysis is obtained as follows.

$$\sum_{in} \dot{m}_{in} e_{in} - \sum_{out} \dot{m}_{out} e_{out} = \dot{E}_D \quad B-10$$

$$(\dot{m}_3 e_3 + \dot{m}_6 e_6) - (\dot{m}_4 e_4 + \dot{m}_7 e_7) = \dot{E}_D \quad \text{B-11}$$

$$e_4 = e_{ph} \quad \text{B-12}$$

$$e_6 = e_{ph} + e_{ch} \quad \text{B-13}$$

$$e_7 = e_{ph} + e_{ch} \quad \text{B-14}$$

Combustion chamber: by removing the heat and work exergy from Eq. 18, the following equation is obtained for the combustion chamber.

$$\sum_{in} \dot{m}_{in} e_{in} - \sum_{out} \dot{m}_{out} e_{out} = \dot{E}_D \quad \text{B-15}$$

$$(\dot{m}_4 e_4 + \dot{m}_{10} e_{10}) - (\dot{m}_5 e_5) = \dot{E}_D \quad \text{B-16}$$

$$e_{10} = e_{ph} + e_{ch} \quad \text{B-17}$$

$$e_5 = e_{ph} + e_{ch} \quad \text{B-18}$$

Gas turbine: by removing heat exergy from Eq. 18, the exergy destruction of the gas turbine is obtained from the equation.

$$-\dot{W} + \sum_{in} \dot{m}_{in} e_{in} - \sum_{out} \dot{m}_{out} e_{out} = \dot{E}_D \quad \text{B-19}$$

$$-\dot{W} + \dot{m}_5 e_5 - \dot{m}_6 e_6 = \dot{E}_D \quad \text{B-20}$$

Exergetic efficiency is one of the important parameters for irreversibility evaluation that defined based on fuel-product definition as bellow:

$$\varepsilon = \text{product} / \text{fuel} \quad \text{B-21}$$

Appendix C

Table C.1: The calculated thermodynamic parameters (Capston C30).

Flow	Mass flow (kg/s)		Temperature (C)		Pressure (bar)	
	CH4	LNG	CH4	LNG	CH4	LNG
1	0.307	0.307	15	15	1.013	1.013
2	0.307	0.307	196.9	8.312	4.052	0.974
3	0.3077	0.307	454.6	186.2	3.896	3.897
4	0.31	0.31	776.7	444.7	3.82	3.747
5	0.31	0.31	524.9	776.7	1.005	3.674
6	0.31	0.31	275	525	0.985	0.966
7	0.0023	0.31	143.2	275	5.455	0.947
8	---	0.002	---	-162	---	1.033
9	---	0.002	---	9.995	---	1.013
10	---	0.002	---	116	---	5.246

Table C.2: The Exergy destruction of CH4 in the various equipment for 2-MW gas turbine cycle at Ta=15°C

Component	Exergy destruction (KW)
Compressor	179.04
Fuel Compressor	11.98266
Combustion	1767.3816
Gas Turbine	748.31613
Stack	91.9438

Table C.3: The Exergy destruction of LNG in the various equipment for 2-MW gas turbine cycle at Ta=15°C

Component	Exergy destruction (KW)
LNG Vap	185.5871
Compressor	176.089
Fuel Compressor	5.3863
Combustion	2095.1817
Gas Turbine	742.8945
Stack	65.0481