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Modeling of Combined Heat and Power Cycle based on Micro Gas Turbines using Biogas Fuel: an Investigation of Key Parameters

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Abstract
Distributed electricity generation has been a long-standing focus for researchers and policy-makers. With the global rise in electricity demand, various generation methods such as solar, wind, fuel cells, and internal combustion engines—are being implemented, each with distinct advantages and drawbacks. Micro gas turbines have emerged as a viable candidate for a reliable, cost-effective, and accessible energy production system. To enhance overall system efficiency, the heat produced from fuel combustion in these turbines can also be used to generate hot water. This study investigates micro gas turbines fueled by biogas, analyzing the effects of several critical parameters: Turbine Inlet Temperature (TIT), Compressor Pressure Ratio (CPR), and recuperator effectiveness within the cycle. The thermodynamic modeling uses the thermally perfect gas model and was conducted in EES (Engineering Equation Solver), with a selected commercial gas microturbine used for validation. Variable fluid thermodynamic properties are accounted for based on temperature, providing accuracy under diverse operational scenarios. It is found that to achieve the maximum overall efficiency; there is an optimal value for the CPR, while it increases with increment in the TIT and recuperator effectiveness.

Keywords: *Microturbine, Electricity, Heat, Pressure drop, Cogeneration, Biogas, Renewable Energy, Heat recovery unit.*

1. Introduction

Energy is one of humanity's most strategic needs, and ensuring its supply has become a significant concern for nations. With fossil fuel resources gradually depleting and their adverse environmental impacts becoming more apparent [1, 2], the necessity of shifting towards clean energy technologies and performance improvement of current energy systems optimizing energy production methods is increasingly clear. The renewable energy sources such as wind, solar, etc. are applicable for several purposes like thermal comfort, cooking, and electrification [3–5]. Electricity consumption in the world has significantly increased in the recent decades due to a variety of reasons like population growth, urbanization, and industrialization. Thermal power plants consuming fossil fuels have a key role in world power generation. According to the data presented by the International Energy Agency (IEA), coal had the highest contribution to electricity generation among different sources that was followed by natural gas [6]. Aside from the development of renewable energy systems for electricity generation to make this sector more environmentally friendly, performance improvement of the technologies that consume fossil fuels and their modifications would be helpful in shifting towards sustainability.

Efficiency improvement in thermal power plants has been focused by different scholars in the recent years. Shajahan et al. [7] performed a study on the efficiency improvement of thermal power plants by use of robust controller and optimization. The data collected in their work helped them to propose a precise model of fuel, air and feedwater flow rate. By employment of evolution and optimization methods, the controllers for all the considered loops were regulated to perfection. In another study by Mandi et al. [8], auxiliary components of a thermal power plant were improved via operations optimization. Aside from carrying out optimization on the cycle or components of the systems, two primary strategies are being adopted globally in the power industry:

1. Combined Heat and Power (CHP): Reusing waste thermal energy for simultaneous electricity and heat production [9]. It should be noted that the concept of heat recovery from the waste of thermal power plants can be applied for providing more products such as cooling, desalination, etc. [10].
2. Distributed Generation (DG): Placing power generation facilities closer to consumption points to minimize transmission losses [11].

Traditionally, electricity for industrial, commercial, agricultural, and residential consumers is produced in large, centralized power plants, and then transmitted through distribution networks. DG systems, however, utilize smaller power stations near consumption sites, significantly reducing transmission losses [12]. Micro gas turbines have shown significant promise in small-scale and decentralized power generation. With output capacities ranging from 25 to 500 kW, microturbines are a relatively recent technology attracting considerable attention in the DG market [13].

One of the renewable energy sources with applicability for different energy-related purposes is biofuel. Scholars have evaluated the use of biofuels for a variety of applications such as heating, desalination, power generation, and polygeneration systems. Depending on the type of biofuel, they can be used for different applications. For instance, biodiesels, as sulfur-free fuels, are applicable as fuels of diesel engines [14]. In a study by Wierzbicka et al. [15], use of biofuels in district heating system was assessed. In this work, particle emission related to district heating systems by use of three biofuels namely pellets, forest residues and sawdust were compared. They reported that overall concentration of emitted particles with aerodynamic diameter of less than 5 micrometer (PM5) at medium working load was in range of 6.3-7.7 $\times 10^7$ particles/cm_n^3, with a bit higher value in case of forest residues combustion. In another study [16], performance of a desalination system based on the heat recovery from combustion engine utilizing biofuel blends was investigated. It was reported that despite the disadvantage of biofuel in terms of low heat recovery potential, use of these fuels leads to more reasonable emissions compared with gasoline. Use of biofuels in the gas turbines is a suitable alternative for the power generation systems using fossil fuels. Use of these fuels can result in significant decrease in the emission of CO_2 by utilizing existing technologies applicable for combustion, while remarkable changes appear to be needed, and more development related to the technological aspect is vital [17]. It is possible to apply biofuels in microturbines to benefit from its advantages. In a study by Devi et al. [18], impact of high oxygenated biofuels on the micro gas turbine engine was evaluated. They reported that the blend with incremented oxygenated compounds in the fuel blends shows a better performance. Microturbines fueled by biogas offer additional environmental benefits, and align with renewable energy goals, making them particularly suitable for distributed generation. Recent studies have demonstrated that biogas-fueled microturbines not only reduce greenhouse gas emissions but also utilize otherwise wasted organic materials [19,20]. Aside from the systems with one type of outputs, biofuels are usable in polygeneration technologies. Fong and Lee [21] implemented a study on trigeneration systems driven by biofuel for non-residential buildings applications. They reported that the systems considered have high potential for primary energy saving, up to 15%, and a decrement of carbon emissions by at least 86% in comparison with the conventional technologies.

As mentioned, it is advantageous in terms of environment and technical aspects to utilize biofuels in microturbines. In this regard, getting deeper insight into these technologies and influential factors on their performance would be helpful for designers. This study simulates a CHP cycle using a Micro-Gas Turbine (MGT) and biogas fuel, examining critical variables such as Turbine Inlet Temperature (TIT), Compressor Pressure Ratio (CPR), and recuperator effectiveness within the cycle. To evaluate performance under these conditions, thermodynamic modeling with the Engineering Equation Solver (EES) software is applied. Impact of these factors on the performance of a combined heating and power cycle based on MGT using cow manure biogas as the fuel is evaluated for the first time, which is the main novelty of this study. The key points of the present work, which represent the importance of the study are consideration of a renewable energy source, biogas, as alternative for natural gas for use in a MGT. The proposed system is a cleaner alternative compared with the MGTs using fossil fuels. Moreover, the proposed system is not only applicable for power generation in small-scale, but also it is usable for heating applications due to the consideration of heat recovery unit. Furthermore, sensitivity analysis is applied to compare the effect of mentioned factors on the overall efficiency of the designed system.

2. Methodology

The simulation approach for this study is adapted from a model originally developed for an industrial gas turbine, as presented in Ref. [22]. Since this research focuses on an MGT with a recuperator, significant adjustments are made to the original model to account for the unique characteristics of MGT systems. These modifications involve revising key thermodynamic equations to match the specific operational parameters of the microturbine, particularly in the presence of a recuperator. The revised thermodynamic equations are implemented and executed within the Engineering Equation Solver (EES) software. EES enabled precise modeling of the MGT's performance across a range of operating conditions, accounting for the effects of recuperation, TIT and CPR on both overall cycle efficiency and fuel consumption. The schematic of the designed cycle for power generation, and heating is represented in figure 1. The main benefits of the proposed system are as follows:

- Use of biogas as a renewable energy source.
- Simultaneous production of heating and power by using heat recovery system.
- Applicability for DG.

According to the methodology represented in the study by Banjac et al. [22], each operational state within the cycle can be formulated, as represented in the following parts of the article.

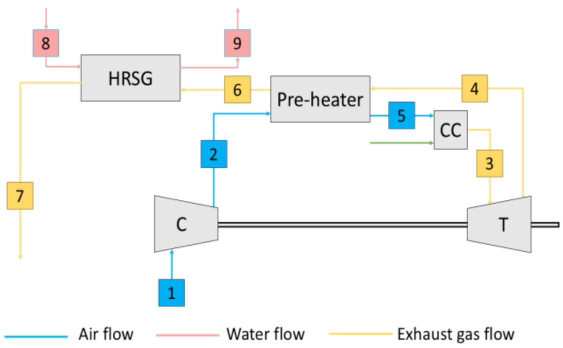


Figure 1. Schematic representation of the designed cycle.

State 1: The inlet conditions for the compressor, with pressure p_{01} and temperature T_{01}, serve as the starting point. Using these parameters, enthalpy h_{01}, specific entropy s_1, and integral of entropy change during an isobaric process I_1 at state 1 are calculated, as follows (Equations (1) and (2)):

$$\mathbf{h_{01} = h(T_{01}.x_1).\ I_1 = I(T_{01}\ .\ x_1)\ , x_1 = 0} \quad (1)$$

$$\mathbf{s_1 = s_R + I_1 - R\ln\ (p_{01}/p_{ref})} \quad (2)$$

State 2s (Ideal compression): For an idealized isentropic process, entropy remains constant, as presented in equation (3).

$$\mathbf{s_{2s} = s_1} \quad (3)$$

Isentropic enthalpy (I) and temperature are calculated according to equations (4) and (5).

$$\mathbf{I_{2s} = s_{2s} - s_R + R\ln\ (p_{02s}/p_{ref})} \quad (4)$$

$$\mathbf{T_{02s} = T(I_{2s}\ .\ x_1) \rightarrow h_{02s} = h(T_{02s}\ .\ x_1)} \quad (5)$$

State 2: From here, the intermediate temperature, enthalpy, and entropy (s) are derived for ideal compression followed by actual values using compressor efficiency η_{iC}, as represented in equations (6) to (8), respectively.

$$\mathbf{h_{02} = h_{01} + (h_{02_s} - h_{01})/\eta_{iC}.\ p_{02} = p_{02s}{=}CPR \times\ p_{01}} \quad (6)$$

$$\mathbf{T_{02} = T(h_{02}.x_1).\ I_2 = I(\ T_{02}.x_1)} \quad (7)$$

$$\mathbf{s_2 = s_R + I_2 - R\ln\ (p_{02s}/p_{ref})} \quad (8)$$

State 5 (Cold recuperation side): The recuperator transfers heat between the exhaust and the compressed air. The effectiveness (ε) is calculated by using equation (9). Equations (10) to (12) are used for determination of enthalpy, isentropic enthalpy and entropy in point (5).

$$\mathbf{\varepsilon = \frac{T_{05} - T_{02}}{T_{04} - T_{02}}.\ p_{05} = p_{02}} \quad (9)$$

$$\mathbf{h_{05} = h(T_{05}\ .\ x_1)} \quad (10)$$

$$\mathbf{I_5 = I(\ T_{05}.x_1)} \quad (11)$$

$$\mathbf{s_5 = s_R + I_5 - R\ln\ (p_{05}/p_{ref})} \quad (12)$$

State 3 (Combustion calculations): The enthalpy (h_{03}) is obtained given (T_{03}), but an iterative approach is required to determine the stoichiometric mass part x, according to equation (13).

$$\mathbf{Q_{cc} = (1+\beta) \times h(T_{03}.x) - h_{02}\ ,\ \beta_{new} = Q_{cc}/(\eta_{CC} \times LHV)} \quad (13)$$

The equation is iterated until convergence, using $= [\beta(1+b)]/[b(1+\beta)]$.

Once the iterative loop converges and the mass part $x = x_3$ is determined, it is necessary to update the reference entropy (s_R) and the specific gas constant (R) based on this new composition (x_3). These values are calculated by the use of equation (14).

$$\mathbf{s_R = s_R(x_3).\ R = R(x_3)} \quad (14)$$

The new mass fraction $x = x_3$ is now applied in subsequent calculations. The pressure (P_{03}) in the combustion chamber is adjusted according to the pressure drop specified, either as a relative or absolute value. In this case, however, we assume no pressure loss in the chamber. The entropy at state s_3 is calculated using the enthalpy $I_3 = I(T_{03}.x_3)$, and the expression provided in Equation (15).

$$s_3 = s_R + I_3 - R\ln(p_{03}/p_{ref}) \quad (15)$$

States 4s and 4 (Expansion process): In the turbine expansion process, if no cooling air is applied, gas composition remains constant. The calculations proceed similarly to compression stages, with constant properties through states 4s and 4.

State 6 (Hot recuperation side): The energy balance in the recuperator is defined according to equation (16). Temperature, isentropic enthalpy, and entropy of the stream in point (6) are calculated by use of equation (17) to (19), respectively.

$$\dot{m}_{air} \times (h_{05} - h_{02}) = \dot{m}_{exh} \times (h_{04} - h_{06}) \quad (16)$$

$$T_{06} = T(h_{06}.x_3).\, p_{06} = p_{04} \quad (17)$$

$$I_6 = I(T_{06}.x_3) \quad (18)$$

$$s_6 = s_R + I_6 - R\ln(p_{06}/p_{ref}) \quad (19)$$

Considered gas mixture

The model assumes specific compositions for both the air and fuel involved in the combustion process.

- **Dry air composition**

The composition of dry air, expressed in mass fractions, is detailed in table 1. These values are used in the model to calculate combustion reactions and properties of the exhaust gases.

Table 1. Dry air composition (mass fraction)

component	Mass Fraction
Ar	0.01288
N_2	0.7552179
O_2	0.2314212
CO_2	0.0004809
Other	0

Biogas fuel properties and air-fuel ratio

The Lower Heating Value (LHV) of the biogas fuel mixture, derived from cow manure, is calculated based on its methane (CH_4) and carbon dioxide (CO_2) composition that given in table 2. Given that methane has an LHV of approximately 50,000 kJ/kg and CO_2 does not contribute to the heating value, the LHV of biogas with 60.2% CH_4 content is calculated by use of equation (20).

$$LHV_{biogas} = \left(\frac{CH_4\%}{100}\right) \times LHV_{CH_4} + \left(\frac{CO_2\%}{100}\right) \times LHV_{CO_2} \quad (20)$$

Table 2. Compositions of chosen biogas produced from biological wastes [23, 24].

Samples		1
Resource		Cow manure
Composition (%)	CH_4	60.2
	CO_2	39.8
	H_2	Not detected

The stoichiometricAir-Fuel Ratio (AFR) for methane combustion is determined using the reaction according to Equation (21).

$$CH_4 + 2(O_2 + 3.76\, N_2) \rightarrow CO_2 + 2H_2O + 7.52\, N_2 \quad (21)$$

For a methane-based fuel with 60.2% CH_4 content, the effective AFR is calculated by scaling the pure methane AFR based on CH_4 percentage.

Heat recovery unit

The Heat Recovery Unit (HRU) in the system is responsible for transferring heat from the exhaust gases to a cold-water stream entering at approximately 20°C. The mass flow rate of the water is adjusted to achieve an outlet temperature of 60°C [10]. The heat transfer and mass flow rates are governed by equations (22) and (23)

3. Model validation

For validation, performance data from the Turbec T100 microturbine—a commercial micro gas turbine commonly used in combined heat and power (CHP) applications—was utilized. Key operational characteristics of the Turbec T100, used as benchmarks, are presented in table 3.

The assumptions considered for all models are as follows:

- The effectiveness of the recuperator is 0.8.
- Pressure loss is assumed to be neglected.
- The power output of the turbine kept constant at the value of 500 kW.
- The temperature of water at the outlet of HRSG is assumed to be 60˚C.
- AFR is equal to the stoichiometric value.

Table 3. T100 Specifications [13, 25].

Variable	T100 Microturbine	Unit
T_{01}	288.15	[K]
P_{01}	101325	[Pa]
Relative Humidity	60	[%]
CPR	4.5	[-]
T_{02}	487.15	[K]
η_C	0.768	[-]
$\dot{m}_{air}$	0.7833	[kg/s]

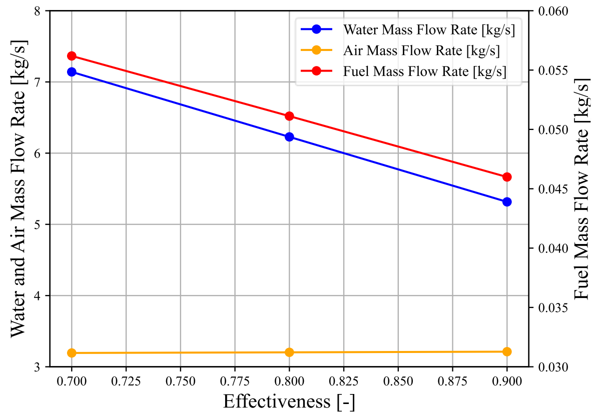
Rotational Speed	70000	[RPM]
Power$_C$	159	[kW]
Effectiveness	90	[%]
T_{06}	543.15	[K]
$\dot{m}_{fuel}$	0.0067	[kg/s]
P_{05}	450000	[Pa]
Fuel type	Natural Gas	[-]
$\dot{Q}_{Input}$	333	[kW]
$\dot{m}_{exh}$	0.79	[kg/s]
T_{03}	1223	[K]
T_{04}	923.15	[K]
η_T	0.826	[-]
Power$_T$	282	[kW]
T_{08}	50	[°C]
T_{09}	70	[°C]
$\dot{m}_{water}$	2	[Lit/s]
T_{07}	80	[°C]
Net electric output	100	[kW]
η_{ele}	30	[%]
η_{CHP}	80	[%]

Table 4. Comparison between numerical results and experimental data.

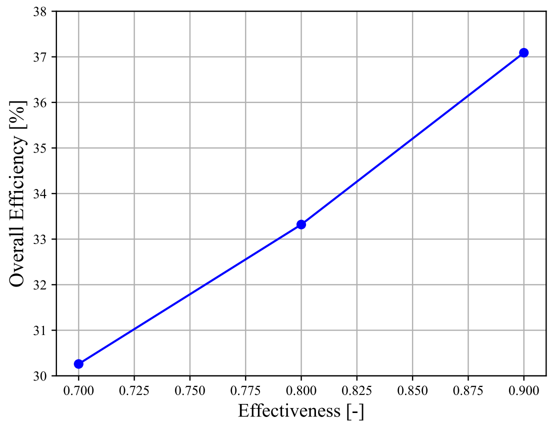
Variable	Numerical results	Relative error (%)	Unit
CPR	4.61	2.542	[-]
T02	486.04	0.227	[K]
η_C	0.79	2.737	[-]
$\dot{m}_{air}$	0.8	1.954	[kg/s]
Rotational speed	70416.59	0.595	[RPM]
Power$_{Compressor}$	161	1.26	[kW]
T_{06}	544.77	0.289	[K]
$\dot{m}_{fuel}$	0.0067	0.030	[kg/s]
P_{05}	441000	2	[Pa]
Fuel type	Pure Methane	-	[-]
Thermal power$_{Input}$	330.85	0.644	[kW]
$\dot{m}_{exhaust}$	0.81	1.938	[kg/s]
T_{03}	1213.55	0.773	[K]
T_{04}	923.15	0	[K]
η_T	0.83	0.309	[-]
Power$_{Turbine}$	277.08	1.743	[kW]
T_{07}	75.7	1.22	[°C]
Power electric$_{Output}$	99.65	0.346	[kW]
$\eta_{electricity}$	0.3	0.401	[%]
η_{CHP}	0.78	2.5	[%]

4. Result and discussion

In the present study, the effects of different factors, namely the recuperator effectiveness, TIT, and CPR, on the outputs of the cycle is evaluated. In all the represented results, the output power of the system is kept constant, and equal to 500 kW. In figure 2, effect of effectiveness of the heat exchanger on the outputs of the system for TIT = 1173.15 K, CPR = 4.5 and ambient temperature of 283.15 K is represented. As shown in this figure, the fuel mass flow rate decreases significantly due to improved compressed air preheating, which reduces the energy demand in the combustion chamber to achieve the desired turbine inlet temperature (TIT). Since the system operates with a fixed Air-to-Fuel Ratio (AFR), the air mass flow rate also decreases proportionally with the reduction in fuel consumption. Additionally, the water mass flow rate decreases slightly because the improved recuperator effectiveness transfers more heat to the compressed air, leaving less waste heat in the exhaust gases for the Heat Recovery Unit (HRU). Despite this, the overall cycle efficiency increases linearly with the recuperator effectiveness, driven by reduced fuel consumption.



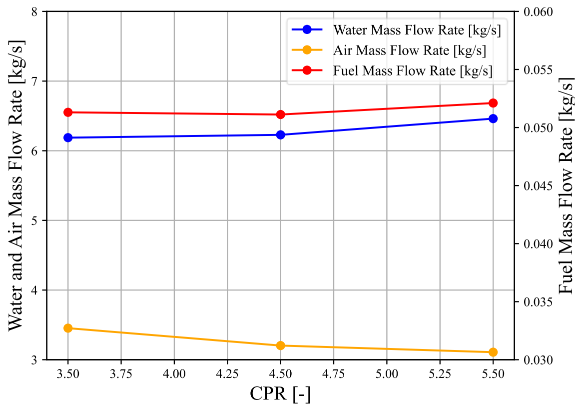
(a)



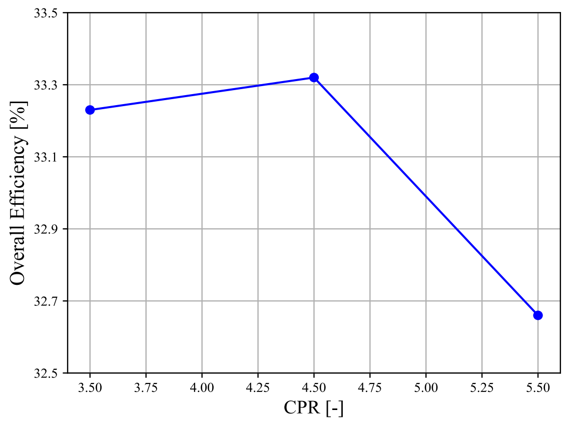
(b)

Figure 2. Effect of heat exchanger effectiveness on a) mass flow rates of the streams and b) overall efficiency.

In figure 3 effect of CPR on the outputs of the system for TIT = 1173.15 K, ε = 0.8, and ambient temperature of 283.15 K is represented. By increasing the CPR, the density of the compressed air rises due to higher pressure, which may reduce the required mass flow rate to maintain a specific power output. Due to a constant AFR, it means the fuel mass flow rate should be decreased. As shown in this figure, the water mass flow rate increases with increasing CPR. This behavior can be explained by the changes in the exhaust gas temperature leaving the recuperator. As CPR increases, the pressure ratio across the turbine grows, leading to a lower turbine outlet temperature due to the larger enthalpy drop during expansion. However, with the fixed effectiveness of the recuperator, the outlet temperature of the exhaust gas rises. The higher exhaust gas temperature entering the Heat Recovery Unit (HRU) provides more energy for heating water, requiring a greater water mass flow rate to maintain the desired energy transfer and temperature rise in the HRU. The relationship between CPR and overall efficiency is typically a parabolic curve. At low CPR, the compression process is inefficient, and at a very high CPR, the increased compression work and associated losses outweigh the benefits of better combustion, leading to a decrease in overall efficiency. There is indeed an optimal CPR for maximum efficiency, depending on the turbine design and operating conditions.



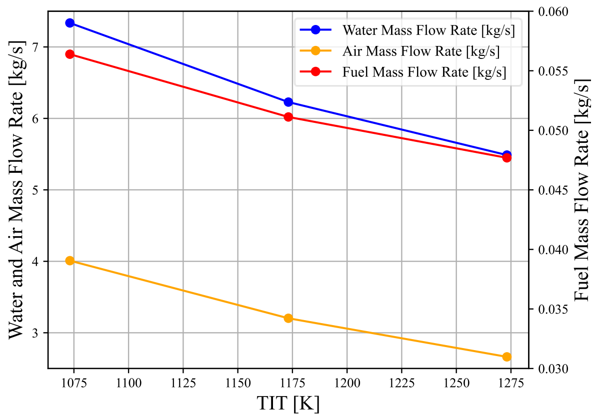
(a)



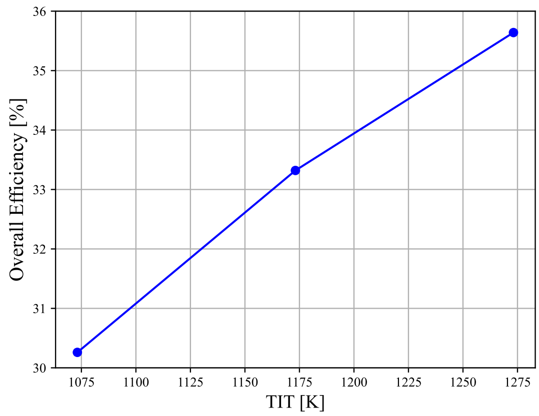
(b)

Figure 3. Effect of CPR on a) mass flow rates of the streams and b) overall efficiency.

Finally, effect of TIT on the characteristics of the cycle for CPR = 4.5 K, $\varepsilon = 0.8$ and ambient temperature of 283.15 K is represented in figure 4. With increase in the TIT, mass flow rates of both air and fuel reduce for the same power output (500 kW); this reduction occurs because the higher TIT improves the thermodynamic efficiency of the cycle, allowing for the same power to be generated with less input mass flow. However, the mass flow rate of water decreases since the available energy for recovery in the corresponding heat exchanger is decreased due to reduction in the mass flow rate of the exhaust gases. In addition, with increase in the TIT, the overall efficiency of the cycle is increased, which can be attributed to the reduction in the input energy to the cycle related to the decrement in the fuel. Despite the benefits of higher TITs, material constraints and thermal stresses limit how far this parameter can be pushed. Scientists are working on improving turbine materials to withstand higher temperatures, facilitating greater power generation and allowing for smaller Micro Gas Turbines (MGTs) to be used as the cycle's primary mover.



(a)



(b)

Figure 4. Effect of TIT on a) mass flow rates of the streams and b) overall efficiency.

5. Sensitivity analysis

For evaluation of importance of different considered parameters on the overall efficiency of the systems, sensitivity analysis is carried out. The considered variables in the previous section were TIT, CPR, and recuperator effectiveness. For sensitivity analysis, it is required to calculate the relevancy factor, a parameter in range of -1 to 1.

Comparison between the absolute values of the relevancy factors reveals their impact level on the considered output which is overall efficiency here. The relevancy factor for each parameter is obtained by the use of Equation (24), as follows [26, 27]:

$$\mathbf{r} = \frac{\sum_{i=1}^{N}(\mathbf{X}_{k,i} - \bar{\mathbf{X}}_k)(\mathbf{y}_i - \bar{\mathbf{y}})}{\sqrt{\sum_{i=1}^{N}(\mathbf{X}_{k,i} - \bar{\mathbf{X}}_k)^2 \sum_{i=1}^{N}(\mathbf{y}_i - \bar{\mathbf{y}})^2}} \quad (24)$$

In equation (24), $\bar{y}$, y_i, $\bar{X}_k$, and $X_{k,i}$ are the output average value, ith input value, average value of kth input parameter, and ith input value of kth input parameter, respectively. In figure 5, calculated values of relevancy factors for each input parameter are presented. These values reveal that recuperator effectiveness has the highest impact on the overall efficiency of the designed cycle under the mentioned assumptions.

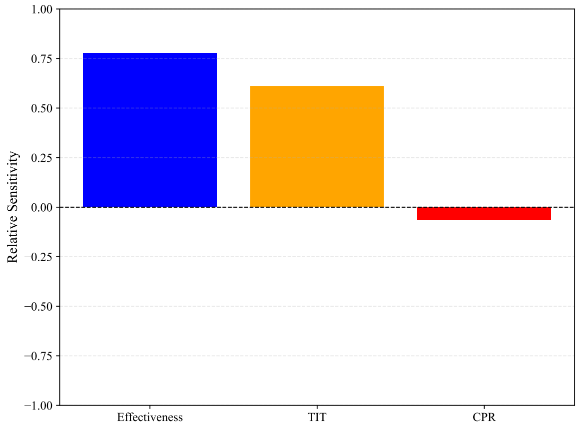


Figure 5. Relevancy factor values for three inputs of the cycle.

6. Recommendations for future studies

In the previous sections, applied method for modeling and the governing equations, results of the modeling by consideration of different factors variation and sensitivity analysis were represented. There are some ideas that could be considered in future works, which would be worthwhile for designers and scholars. For instance, in addition to the energy efficiency of the systems, some other criteria and indices such as exergy efficiency or environmental factors can be applied for analysis of the present system. Furthermore, it would be useful to consider economic factors for evaluation of the present system and the similar configurations. Moreover, it is recommended to use other biofuels in the combustion chamber of the present system, and evaluate the impact of fuel on different facets of the proposed systems. Besides the factors considered in this work, it would be useful to consider other influential parameters such as the ambient temperature and characteristics of different components on the output of the proposed system. The generated data by the thermodynamics models can be used to develop models based on machine learning methods such as artificial neural network.

Aside from the suggestions mentioned that are applicable for the current configurations, there are some other suggestions on similar systems with different structures. For example, the recovered heat from the exhaust gases can be used for other purposes such as desalination and cooling. These productions require some additional components such as thermal desalination systems and absorption chillers that would increase the complexity of the system.

7. Conclusion

Microturbines could be attractive options for distributed generation applications. In order to make these systems more environmentally friendly, it is possible to use biofuels as renewable energy sources and apply HRU. In the present study, effects of different factors namely TIT, recuperator effectiveness and CPR on the overall efficiency of the system and characteristics of the cycle with constant output electrical power is investigated. According to the findings of the models, solved by use of EES software; it can be concluded that increase in TIT leads to elevation in the overall efficiency of the cycle. In addition, it is found that there is an optimal value for the CPR of the applied compressor and deviation from the design point would lead to reduction in the overall efficiency. Furthermore, it is observed that with elevation in the recuperator effectiveness, cycle overall efficiency improves. Findings of the present article would be beneficial for the designers and engineers working on the cycle efficiency improvement and greenhouse gases emissions reduction of power generation systems. In future studies, different types of biofuels can be used, and the cycle characteristics can be evaluated for deeper insight into these alternative fuels.

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