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Thermodynamic analysis of gas turbine power plant of PT PLN Belawan generation implementation unit

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Abstract

The low quality of the thermodynamic process in a gas turbine power plant results in the waste of potential energy and impacts the power plant's efficiency. Analysing the thermodynamic performance of a gas turbine power plant is crucial to evaluating its efficiency in converting fuel energy into useful work. This analysis helps identify opportunities for improvement and optimise the plant's design for better performance by examining the components (e.g., the compressor, combustion chamber, and turbine). This study aims to evaluate the performance of a Gas Turbine Power Plant (GTPP) through thermodynamic analysis considering the variation of cycle loads. The study was conducted based on the field survey data obtained from the GTPP PT PLN Belawan generation implementation unit. The collected operation data was used to perform a thermodynamic analysis by applying the principles of conservation of mass and energy, along with the laws of thermodynamics. The study examined five cycle load variations: 31.7 MW, 34.3 MW, 48.1 MW, 60.7 MW, and 71.7 MW. Results showed a consistent reduction in the gas turbine heat rate as the load increased, with a significant 53.3% drop in heat rate from 34.3 MW to 71.7 MW. Higher cycle loads also correlated with increased turbine and compressor work, with the turbine producing 55.8% more work than the compressor at 71.7 MW. The turbine's thermal efficiency ranged from 40% to 44%, with potential for a 5% efficiency increase.

Keywords:
Thermodynamic analysis, energy, exergy, heat rate, cycle load, efficiency.

1 Introduction

Electrical energy plays an important role in everyday human life. To be able to produce electricity, a power generation unit that can convert a form of energy is needed [1]. Currently, the generation units that we most often encounter, especially in the Sumatra region, are Steam Turbine Power Plants (STPP) and Gas Turbine Power Plants (GTPP) [2]. Both generation units basically have the same working principle, namely utilising kinetic energy from heating the fluid that is used to operate the turbine. The turbine's rotational power is used to drive the generator to produce electricity. In a STPP, the fluid's kinetic energy is acquired by heating water in the boiler, while in a GTPP, energy is obtained by combusting a mixture of fuel and compressed air [3][4].

GTPP frequently encounters load variations to satisfy electrical power needs, which can alter at any time based on customer demand [5]. Adjusting GTPP loads will affect the performance of each component, including the turbine, combustion chamber, and generator. Changes in the load trigger automatic adjustments to the fuel supply and combustion air [6]. The highest efficiency of the GTPP can be determined by identifying the load at which it operates most efficiently.

In order to identify the locations and types of waste and losses in these systems, the power system analysis shall normally be carried out with heat optimisation instruments. Energy and exergy analyses are the investigation and evaluation tools used to optimise the thermal energy system [7][3]. Energy analysis, as is known, does not give precise information about the energy degradation in this process [8]–[11].

Khaliq [12] suggested a notional trigeneration system based on the ordinary gas turbine cycle. He used first and second law techniques, as well as computational analysis, to study how governing factors affected exergy destruction in each component.

An exergy analysis of the combined Brayton and Rankine power cycles for the different components of the power plant has been presented by Tiwari et al. [13]. Maximum exergy losses have been found in the combustion chamber of the gas turbines. Changes in the pressure ratio and inlet temperature of the turbines alter these effects on exergy loss.

Ghazikhani et al. [14] conducted an exergy comparison of a standard gas turbine and a gas turbine with an air-bottoming cycle. The results show that at a low pressure ratio, the air-bottoming cycle can provide more specified work with less specific fuel consumption than a simple gas turbine.

Nondy and Gogoi [15] suggest a Combined Cycle Power Plant (CCPP) that combines a recuperative gas turbine cycle and a reheat-regenerative steam turbine cycle. The primary goal of this study is to evaluate the performance of the proposed CCPP using energy and exergy assessments. The results demonstrate that the CCPP generates 63.59 MW of net power, with energy and exergy efficiency of 49.08% and 47.42%, respectively. It also shows that the combustion chamber has the most exergy destruction, accounting for 63.30% of the total system irreversibilities, while the gas turbine is the most efficient component, with an exergy efficiency of 94.91%.

Energy and energy-related numerical analysis have been carried out by Ahmadi et al. [16] for a gas turbine cycle that is connected with an Organic Rankine Cycle (ORC) cycle. In order to identify the ideal working pressure and working fluid, the optimisation of the ORC cycle's working pressure has been studied using both energy and exergy analysis. It was found that dimethyl carbonate and o-xylene demonstrated the most and least desirable criteria in both energy and exergy analysis out of six working fluids that were examined for study.

Current research primarily focusses on the operational conditions of gas turbines at peak efficiency, often referred to as full-load conditions. However, in real-world scenarios, gas turbines frequently operate at partial loads due to fluctuating power demand. This variability in operation generally results in notable reductions in efficiency and increased fuel consumption. Consequently, there is an urgent need for comprehensive research into the thermodynamic performance of gas turbines during partial-load conditions.

This study aims to evaluate the performance of GTPP through thermodynamic analysis considering the variation of cycle loads. The performance of the GTPP that is investigated includes the heat rate, actual work of the compressor and turbine, and both thermal and exergy efficiency of the GTPP.

2 Research Method

This section discusses the collection of operational data during the field survey as well as energy and exergy analysis. The steps for evaluating GTPP performance, such as heat rate, compressor and turbine actual work, and thermal and exergy efficiency, are extensively covered in the energy and exergy analysis sections. Investigating the thermodynamic analysis of the GTPP involved examining five different operational loads, which were 31.75 MW, 34.26 MW, 48.1 MW, 60.7 MW, and 71.7 MW.

2.1 Operational Data

Data for this study was collected during a field survey at PT PLN Belawan generation implementation unit. Tables 1-5 present the collected data on the operational parameters of the GTPP used for the computational and thermodynamic analysis.

Table 1. Operational parameters with a cycle load of 31.75 MW

Parameter	Value
Load (MW)	31.75
Ambient air temperature (°C)	32
Compressed air temperature (°C)	309
Exhaust gas temperature (°C)	370
Environmental air pressure (atm)	1
Absolute pressure of compressed air (bar)	8.3
Fuel mass flow rate (kg/s)	3.7
Fuel calorific value (LHV) (kJ/kg)	46144
Specific heat of air (kJ/kgK)	1.006

Table 2. Operational parameters with a cycle load of 34.26 MW

Parameter	Value
Load (MW)	34.26
Ambient air temperature (°C)	28.8
Compressed air temperature (°C)	307
Exhaust gas temperature (°C)	369
Environmental air pressure (atm)	1
Absolute pressure of compressed air (bar)	8.5
Fuel mass flow rate (kg/s)	3.9
Fuel calorific value (LHV) (kJ/kg)	46144
Specific heat of air (kJ/kgK)	1.006

Table 3. Operational parameters with a cycle load of 48.1 MW

Parameter	Value
Load (MW)	48.1
Ambient air temperature (°C)	30.1
Compressed air temperature (°C)	329
Exhaust gas temperature (°C)	370
Environmental air pressure (atm)	1
Absolute pressure of compressed air (bar)	9.84
Fuel mass flow rate (kg/s)	4.7
Fuel calorific value (LHV) (kJ/kg)	46144
Specific heat of air (kJ/kgK)	1.006

Table 4. Operational parameters with a cycle load of 60.7 MW

Parameter	Value
Load (MW)	60.7
Ambient air temperature (°C)	31
Compressed air temperature (°C)	342
Exhaust gas temperature (°C)	394
Environmental air pressure (atm)	1
Absolute pressure of compressed air (bar)	10.5
Fuel mass flow rate (kg/s)	5.3
Fuel calorific value (LHV) (kJ/kg)	46144
Specific heat of air (kJ/kgK)	1.006

Table 5. Operational parameters with a cycle load of 71.7 MW

Parameter	Value
Load (MW)	71.7
Ambient air temperature (°C)	31.4
Compressed air temperature (°C)	348
Exhaust gas temperature (°C)	492
Environmental air pressure (atm)	1
Absolute pressure of compressed air (bar)	10.82
Fuel mass flow rate (kg/s)	6
Fuel calorific value (LHV) (kJ/kg)	46144
Specific heat of air (kJ/kgK)	1.006

2.2 Thermodynamic of Gas Turbine

The electricity generation cycle, known as the Brayton cycle, makes use of combustion gas as a working fluid. A gas turbine is employed in the Brayton cycle, converting the energy from hot gas into mechanical energy. The mechanical energy produced by the gas turbine drives the generator to generate electrical energy. Power plants that utilise the Brayton cycle are referred to as gas power plants [4].

2.2.1 Energy Analysis

The energy analysis of the gas turbine power plant system adheres to the principles of the Brayton cycle. Within the Brayton cycle, there are multiple continuous processes occurring. The Brayton cycle commonly employed in GTPP is typically an open cycle. An open Brayton cycle is a cycle that acquires working fluid from the ambient air (atmosphere) and releases it back into the atmosphere at the completion of the cycle. Fig. 1 presents the P-v and T-s diagrams of the air-standard ideal Brayton cycle.

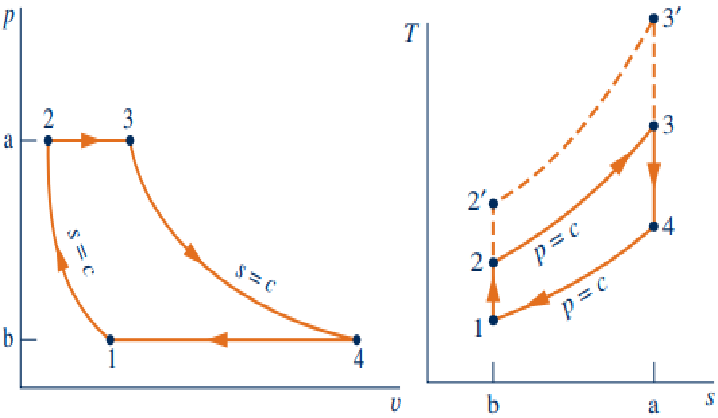

Fig. 1. Air-standard ideal Brayton cycle [3].

The gas turbine's actual state, depicted more realistically in Fig. 2, is influenced by friction in the compressor and turbine. This causes an increase in specific entropy throughout the components, leading to a decrease in pressure as the working fluid moves through the heat exchanger. With irreversibility effects becoming more noticeable in the turbine and compressor, the turbine's work output decreases and the compressor's work input increases, ultimately leading to a significant reduction in the power plant's network. Hence, achieving the desired network in the power generation system necessitates relatively high turbine and compressor efficiency. Through decades of development efforts, gas turbine power generation systems have achieved component efficiencies ranging from 80% to 90% [5].

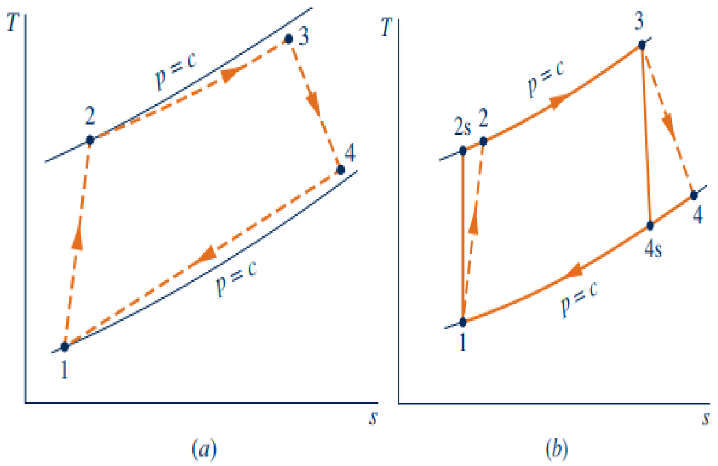

Fig. 2. Irreversibility in a gas turbine [3].

The energy conservation law equation can be used to analyse the work and heat transfer calculations in each part of the system. If we assume that there is no heat transfer from the turbine and compressor to the surroundings, the system is in a steady state, and any changes in kinetic energy and potential energy are not considered, then we can determine the work and heat involved in each component.

Work required by the compressor (Eq. 1):

$$\frac{\dot{W}_c}{\dot{m}} = h_2 - h_1 \tag{1}$$

where $\dot{W}_c$ is the compressor work (kJ/s), $\dot{m}$ is the air mass flow rate (kg/s), h_2 is the enthalpy at state 2 (kJ/kg), h_1 is the enthalpy at state 1 (kJ/kg).

Considering irreversibility occurs in the compressor, the enthalpy at state 2 h_2 is obtained from the relationship of the equation of isentropic efficiency as Eq. 2:

$$\eta_c = \frac{(\dot{W}_c/\dot{m})_s}{(\dot{W}_c/\dot{m})} = \frac{h_{2s}-h_1}{h_2-h_1} \tag{2}$$

where η_c is the isentropic efficiency of the compressor, $(\dot{W}_c/\dot{m})_s$ is the ideal work developed by the compressor per unit of air mass flowing (kJ/kg), $(\dot{W}_c/\dot{m})$ is the actual work developed by the compressor per unit of air mass flowing (kJ/kg), h_{2s} and h_2 are the enthalpy at state 2s and 2 respectively (kJ/kg), and h_1 is the enthalpy at state 1 (kJ/kg).

Using the isentropic relation (Eq. 3), the specific enthalpy at state 2s (h_{2s}) is found from the thermodynamic table of the ideal gas properties of air.

$$p_{r2} = \frac{P_2}{P_1} p_{r1} \tag{3}$$

where p_{r2} and p_{r1} are the reduced pressure at states 2 and 1 respectively, and $\frac{P_2}{P_1}$ is the ratio of compressor pressure at states 2 and 1.

Heat entered the combustion chamber (Eq. 4):

$$\frac{\dot{Q}_{in}}{\dot{m}} = h_3 - h_2 \tag{4}$$

where $\frac{\dot{Q}_{in}}{\dot{m}}$ is the heat added to the cycle per unit of mass (kJ/kg), h_2 and h_3 are the enthalpy at states 2 and 3 respectively (kJ/kg).

Work produced by the turbine (Eq. 5):

$$\frac{\dot{W}_t}{\dot{m}} = h_3 - h_4 \tag{5}$$

where $\dot{W}_t$ is the turbine work (kJ/kg), $\dot{m}$ is the sum of the mass flow rates of air and fuel (kg/s), h_3 and h_4 are the enthalpy at states 3 and 4 respectively (kJ/kg).

Due to irreversibility, the enthalpy at state 4 h_4 is determined based on the relationship of isentropic efficiency of the turbine.

$$\eta_t = \frac{(\dot{W}_t/\dot{m})}{(\dot{W}_t/\dot{m})_s} = \frac{h_3-h_4}{h_3-h_{4s}} \tag{6}$$

where η_t is the turbine isentropic efficiency, $(\dot{W}_t/\dot{m})$ is the actual work developed by the turbine per unit of mass flowing (kJ/kg), $(\dot{W}_t/\dot{m})_s$ is the ideal work developed by the turbine per unit of mass flowing (kJ/kg), h_3, h_4, h_{4s} are the enthalpy at states 3, 4, 4s.

The specific enthalpy at state 4s (h_{4s}) is found from the table of ideal gas properties of air by using the isentropic relation (Eq.7)

$$p_{r4} = \frac{P_4}{P_3} p_{r3} \tag{7}$$

where p_{r4} and p_{r3} are the reduced pressure at states 4 and 3, respectively, and $\frac{P_4}{P_3}$ is the ratio of turbine pressure at states 4 and 3.

Heat dissipated (Eq. 8):

$$\frac{\dot{Q}_{out}}{\dot{m}} = h_4 - h_1 \tag{8}$$

where $\frac{\dot{Q}_{out}}{\dot{m}}$ is the heat rejected per unit of mass (kJ/kg), h_1 and h_4 are the enthalpy at states 1 and 4, respectively (kJ/kg).

The efficiency of the Bryton cycle can be evaluated by comparing the power generated by the generator to the fuel power needed in the combustion process. The generator's power is calculated as the difference between the turbine's power output and the compressor's power input. Efficiency values are typically stated as percentages (Eq. 9):

$$\eta = \frac{\frac{\dot{W}_t}{\dot{m}} - \frac{\dot{W}_c}{\dot{m}}}{\frac{\dot{Q}_{in}}{\dot{m}}} = \frac{(h_3-h_4)-(h_2-h_1)}{h_3-h_2} \tag{9}$$

where η is the thermal efficiency of the Bryton cycle, $\frac{\dot{W}_t}{\dot{m}}$ is the work developed by the turbine per unit of mass flowing (kJ/kg), $\frac{\dot{W}_c}{\dot{m}}$ is the work developed by the compressor per unit of mass flowing (kJ/kg), and $\frac{\dot{Q}_{in}}{\dot{m}}$ is heat added to the cycle per unit of mass (kJ/kg).

In addition to efficiency, another factor to examine when evaluating generator performance is heat rate, which is the proportion of the heat required by the generator to the electrical power generated.

In the Brayton cycle, the combustor acquires heat by burning a specific amount of fuel. Improved generator performance is indicated by a lower heat rate value, typically expressed in units of kCal/kWh [5].

$$Heat\ Rate = \frac{\dot{W}_{Generator}}{\dot{Q}_{input}} \tag{10}$$

2.2.2 Exergy Analysis

The low thermal efficiency of gas turbine producing systems motivates academics to conduct more thermodynamic analysis to improve the efficiency of power plants rather than exploring for new alternative energy sources [6].

The three primary parts of GTPP are the compressor, combustion chamber, and turbine. According to the first law of thermodynamics, energy cannot be created or destroyed but can change forms, which is fundamental to energy analysis. The second law of thermodynamics indicates that entropy within a system tends to be unstable and increase with time, which is a key concept in exergy analysis.

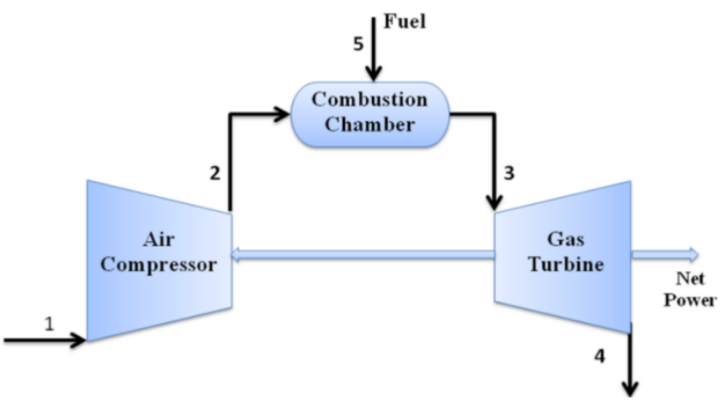

Fig. 3. Exergy rate in the Bryton cycle [17].

Based on the two statements provided, exergy refers to the maximum potential work attainable when the system interacts at equilibrium. Analysis of the isentropic process during air compression begins in the compressor. Subsequently, the air undergoes additional heating as it enters the combustion chamber and is burned in an isobaric state. The combustion gas products proceed to the turbine, where they expand and generate work on the turbine [8].

Upon understanding the exergy flow rate for every part of the component, it is also possible to compute the exergy destruction of each component. Essentially, there will be a decrease in exergy flow during each process. The general exergy destruction equation can be expressed as Eq. 11:

$$\dot{E}_{x\,in} - \dot{E}_{x\,out} = \dot{E}_{dest} \quad (11)$$

where $\dot{E}_{x\,in}$ and $\dot{E}_{x\,out}$ are the inlet and outlet exergy flow rate, respectively (kJ/s), and $\dot{E}_{dest}$ is the exergy destruction (kJ/s).

On the compressor (Eq. 12):

$$\dot{E}_{x1} + \dot{W}_c = \dot{E}_{x2} + \dot{E}_{dest} \quad (12)$$

where $\dot{E}_{x1}$ and $\dot{E}_{x2}$ are the exergy flow rate at states 1 and 2, respectively (kJ/kg), $\dot{W}_c$ is the exergy transfer by compressor work (kJ/kg), and $\dot{E}_{dest}$ is the exergy destruction (kJ/s).

In the combustion chamber (Eq. 13):

$$\dot{E}_{x2} + \dot{E}_{x5} = \dot{E}_{x3} + \dot{E}_{dest} \quad (13)$$

where $\dot{E}_{x2}$, $\dot{E}_{x3}$, $\dot{E}_{x5}$ are the exergy flow rate at states 2, 3, 5, respectively (kJ/kg) and $\dot{E}_{dest}$ is the exergy destruction (kJ/s).

On the turbine (Eq. 14):

$$\dot{E}_{x3} + \dot{E}_{x4} = \dot{W}_{GT} + \dot{E}_{dest} \quad (14)$$

where $\dot{E}_{x3}$ and $\dot{E}_{x4}$ are the exergy flow rate at states 3 and 4, respectively (kJ/kg), $\dot{W}_{GT}$ is the exergy transfer by gas turbine work (kJ/kg), and $\dot{E}_{dest}$ is the exergy destruction (kJ/s).

These equations contain detailed information about the components of a gas turbine power plant, specifically the air compressor, combustion chamber, and turbine. The exergy destruction of each component can be estimated by using the equation for that component.

Analysing each component in terms of energy and exergy helps identify the components with the lowest and highest efficiency. Eq. 15-Eq. 17 are used to determine the exergy efficiency of each component.

On Compressor:

$$\eta_{x,C} = 1 - \left(\frac{\dot{E}_{dest}}{\dot{W}_C + \dot{E}_{x1}}\right) \quad (15)$$

where $\eta_{x,C}$ is the exergy efficiency of the compressor, $\dot{E}_{dest}$ is the exergy destruction (kJ/s), $\dot{W}_c$ is the exergy transfer by compressor work (kJ/kg), and $\dot{E}_{x1}$ is the exergy flow rate at state 1 (kJ/kg).

In the combustion chamber:

$$\eta_{x,CC} = 1 - \left(\frac{\dot{E}_{dest}}{\dot{E}_{x2} + \dot{E}_{x5} + \dot{E}_{x3}}\right) \quad (16)$$

where $\eta_{x,CC}$ is the exergy efficiency of the combustion chamber, $\dot{E}_{dest}$ is the exergy destruction (kJ/s), and $\dot{E}_{x2}$, $\dot{E}_{x3}$, $\dot{E}_{x5}$ are the exergy flow rate at states 2, 3, and 5, respectively (kJ/kg).

On the turbine:

$$\eta_{x,GT} = 1 - \left(\frac{\dot{E}_{dest}}{\dot{W}_{GT} + \dot{E}_{x3}}\right) \quad (17)$$

where $\eta_{x,GT}$ is exergy efficiency of the gas turbine, $\dot{E}_{dest}$ is the exergy destruction (kJ/s), $\dot{W}_{GT}$ is the exergy transfer by gas turbine work (kJ/kg), and $\dot{E}_{x3}$ is the exergy flow rate at state 3 (kJ/kg).

The overall energy and exergy efficiencies of a gas turbine power plant can be determined using Eq. 18-Eq. 19.

$$\eta_{x,Overall} = \frac{\dot{W}_{net,GT}}{\dot{E}_{x,f}} \quad (18)$$

where $\eta_{x,Overall}$ is the overall exergy efficiency, $\dot{W}_{net,GT}$ is the exergy transfer by gas turbine work (kJ/kg), and $\dot{E}_{x,f}$ is the fuel exergy flowrate (kJ/kg).

$$\eta_{Overall} = \frac{\dot{W}_{net,GT}}{\dot{m}_{Fuel}LHV} \quad (19)$$

where $\eta_{Overall}$ is the overall energy efficiency, $\dot{W}_{net,GT}$ is the gas turbine work (kJ/s), $\dot{m}_{Fuel}$ is the fuel mass flow rate (kg/kg), LHV is the low heating value (kJ/kg).

In this study, thermodynamic modelling relied on the first and second laws of thermodynamics. Precise calculations were crucial in conducting exergy analysis to pinpoint the primary source of irreversibility within the system. The ambient temperature also played a role in the dissipation of exergy and the overall efficiency of the system.

3 Results and Discussion

3.1 Effect of Cycle Load on Heat Rate Gas Turbine

Fig. 4 illustrates the relationship between cycle load and gas turbine heat rate. Variations in the load cycle have a noteworthy impact on the heat rate, which is a crucial measure of a gas turbine's efficiency. Lower heat rates result from gas turbines running closer to peak efficiency during high load periods. Conversely, the gas turbine's efficiency diminishes during low-load or frequent load changes due to higher heat rates resulting from transient operations. Empirical data from existing gas turbine power plants indicates a positive correlation between heat rate degradation and load variability. Specifically, power plants experiencing frequent and sudden load fluctuations exhibit a substantial increase in heat rate, attributed to the additional fuel required to maintain the turbine's pressure and temperature.

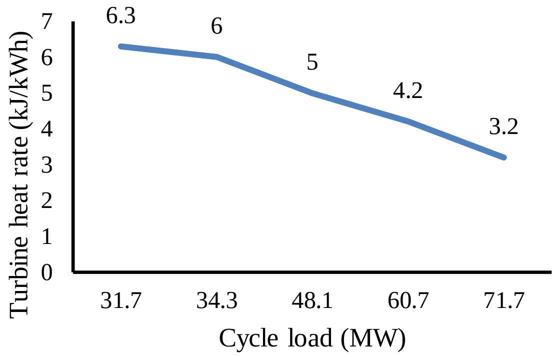

Fig. 4. Effect of cycle load on gas turbine heat rate.

As can be observed from Fig. 4, the gas turbine heat rate results indicate that there is a consistent decrease in heat rate as the load increases. This tendency has also been discussed by Liu et al., [18]. The change in cycle load from 31.7 kW to 34.3 kW has shown a gentle slope with 4.7 % heat rate drop. A steep decline occurs when there is a change in the cycle load from 34.3 MW to 71.7 MW with a decrease of 53.3 %. A lower heat rate value signifies a higher thermal efficiency of the gas turbine, whereas a higher heat rate value indicates a decrease in the gas turbine's thermal efficiency. Therefore, a lower heat rate means less energy is required to produce 1 kWh.

3.2 Effect of Cycle Load on Compressor and Gas Turbine Work

The load cycle determines the operating conditions and work requirements for both the turbine and compressor in a gas turbine engine. Understanding this relationship is essential for optimising performance, controlling fuel consumption, and ensuring reliable operation under varying load conditions. In Fig. 5, the relationship between cycle loads and the actual work of the turbine and compressor is depicted. The figure shows that the increase in cycle load directly correlates with the increase in turbine and compressor work. At the lowest cycle load, it was observed that the actual work of the turbine is 47.2% higher than that of the compressor. The work difference between the turbine and compressor grows as the cycle load increases. The result reveals that the work of the turbine is 55.8% higher when the cycle load reaches 71.7 MW. Significant amount of work produced by the turbine is used to drive the compressor. The relationship between compressor work and turbine work is known as the back work ratio and typically falls around 0.5 or between 0.40 and 0.6 for gas turbine engines.

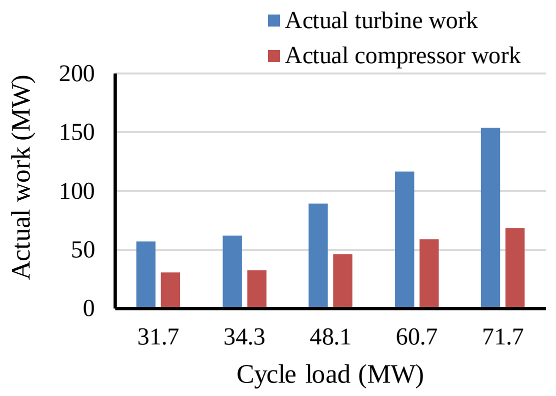

Fig. 5. Effect of cycle load on actual work of turbine and compressor.

3.3 Comparison of Thermal and Exergy Efficiency

The overall performance and sustainability of GTPP systems are significantly affected by the relationship between the load cycle and the thermal and exergy efficiencies. Thermal efficiency is typically low at the start of the operational phase because of incomplete combustion and suboptimal operating temperatures [19]. In addition, significant exergy destruction resulting from rapid changes in temperature and pressure compromises exergy efficiency. As the engine reaches a stable operating state, thermal efficiency improves by operating at the intended temperature and pressure ratios, promoting more complete combustion and effective energy utilization. This phase also maximises exergy efficiency by maintaining a stable environment with minimal irreversibilities. Transient load changes can cause variations in both thermal and exergy efficiencies. Rapid load increases may temporarily reduce thermal efficiency as it takes time to achieve optimal combustion conditions, and increased entropy generation may affect exergy efficiency. Conversely, as the load gradually decreases during shutdown, thermal efficiency decreases as the system moves away from ideal conditions, and exergy efficiency also decreases due to increased exergy destruction. Understanding these correlations is vital for devising strategies to enhance the efficiency and sustainability of gas turbine engines across different load cycles.

In Fig. 6, it is evident that the thermal and exergy efficiency of the Gas Turbine Power Plant (GTPP) rises with the increase in cycle load. The overall thermal efficiency of the turbine falls within the 40-44% range, as depicted in Fig. 6. This falls within the normal range in the context of Brayton's basic design, where the heat loss retains some usable energy.

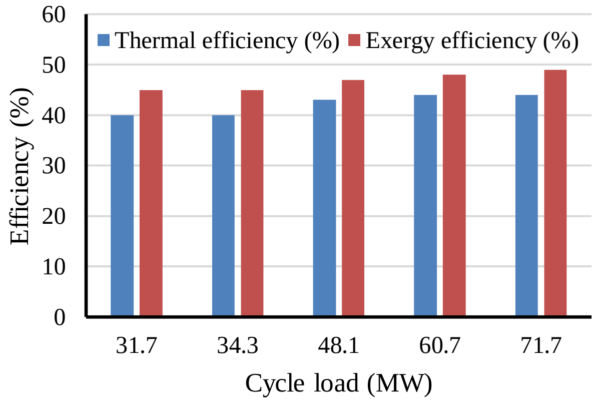

Fig. 6. Thermal and exergy efficiency of the gas turbine.

Thus, it becomes essential to conduct exergy analysis to ascertain the potential efficiency of the gas turbine generating cycle, by understanding the exergy rate, exergy destruction, and exergy efficiency for each component of the gas turbine. Upon reviewing the thermal and exergy efficiency, it becomes apparent that there is potential to increase the GTPP's thermal efficiency by 5%. This can be achieved by maximising the heat generated in the combustion chamber, perhaps through redesigning the power plant cycle in the compressor, combustion chamber, and gas turbine to achieve better results.

4 Conclusion

In conclusion, the relationship between cycle load and gas turbine heat rate was crucial for determining efficiency. Lower heat rates indicated higher efficiency, while higher heat rates suggested a decrease in efficiency. Operating conditions and work requirements for the turbine and compressor were influenced by the load cycle, which impacts overall performance. Understanding these connections was essential for optimising performance, controlling fuel consumption, and ensuring reliable operation under different load conditions. By analysing thermal and exergy efficiencies, improvements could be made to enhance the sustainability and efficiency of gas turbine engines throughout various load cycles. It is important to consider these factors when developing strategies to increase efficiency and sustainability.

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