

Parametric study of a two-shaft gas turbine cycle model of power plant

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Power Plant

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Abstract. In this paper, the parametric study of a two shafts gas turbine cycle model of the power plant was proposed. The power output, compression work, specific fuel consumption and thermal efficiency are evaluated with respect to the cycle temperature and compression ratio for a typical set of operating conditions. Two shafts gas turbine cycle with realistic parameters is modeled. The computational model was developed utilizing the MATLAB codes. Turbine work found to be decreases as ambient temperature increases as well as the thermal efficiency decreases. It can be seen that the thermal efficiency and power output increases linearly with increases of compression ratio while decreases of ambient temperature. The power of the simulated two shafts gas turbine reach to 135MW, which is higher than the simple gas-turbine cycle (Baiji gas turbine power plant, power < 131MW). The specific fuel consumption increases with increases of ambient temperature as well as the lower turbine inlet temperature. Even though at the lower turbine inlet temperature is decrement the thermal efficiency dramatically and the power output increases linearly with increases of compression ratio and decreases the ambient temperature.

1. Introduction

In a two-shaft gas turbine having an adjustable nozzle between the compressor high-pressure turbine and the load turbine, a control system is provided in which a biasing signal is added to the fuel temperature control increasing its base reference so that on an increase in called for speed or load the gas turbine is able to dynamically respond in an improved manner [1]. The steady-state temperature limits are exceeded transiently without damage to the gas turbine until the system returns to its new steady-state condition. Also provided for the two-shaft gas turbine is acceleration limiting control whereby the overall gas turbine is allowed to accelerate at the maximum allowable value of either of the two shafts. Further provision is made for a blow off valve control whereby excess energy from a regenerator is diverted from the turbines so that the speed and load control will remain in command, thus preventing over speed conditions and/or subsequent shutdown [2]. The gas turbine has experienced phenomenal progress and growth since its first successful development in the 1930s. The early gas turbine build in the 1940s and even 1950s had simple-cycle efficiencies of about 17 percent

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because of the low compressor and turbine efficiencies and low turbine inlet temperature due to metallurgical limitations of those times [3]. Therefore, gas-turbines found only limited use despite their versatility and their ability to burn a variety of fuel. The efforts to improve the cycle efficiency concentrated in three areas [4]:

- i. Increasing the turbine inlet (or firing) temperatures
- ii. Increasing the efficiencies of turbo machinery components.
- iii. Adding modifications to the basic cycle.

Gas turbines manufactured by General Electric in the early 1990s had a pressure ratio of 13.5 and generated 135.7 MW of net power with a thermal efficiency of 33% in a simple cycle operation. A more recent gas turbine manufactured by General Electric used a turbine inlet temperature of 1425 °C and produced up to 282 MW while achieving a thermal efficiency of 39.5% in the simple cycle mode [5]. Accordingly, gas turbines are rapidly becoming the choice for current and future power generation systems because they offer efficient fuel conversion, reduced cost of electricity, low installation and maintenance cost, can be put in service with minimum of delay time, occupy less room than other plants for the same capacity, and the ability to consume wide range of hydrocarbon fuels [6-7]. Their exhaust gases are relatively less pollutant to the environment, and can be used for preheating air before entering the combustion chamber, or for district heating as in combined heat and power plants [8].

Gas turbine performance can be qualified with respect to its efficiency, power output, specific fuel consumption and work ratio. There are several parameters that affect its performance, such as the compressor pressure ratio, the combustion inlet temperature and the turbine inlet temperature (TIT) [9]. Obviously, these parameters have been actually improved by various gas turbine manufactures, as it has been mentioned above. Meanwhile, operating parameters such as ambient temperature, altitude, humidity, inlet and exhaust losses also affect the performance of gas turbine units. Basically, gas turbines draw air directly from the atmosphere. Its performance is undoubtedly changed by anything that would affect the air density or mass flow rate of the air intake into the compressor. Consequently, a parametric study on the effect of operation conditions requires managing the operation conditions of the system. Thus, the aim of the present work is to develop a strategy to determine the performance of two shaft gas turbine power plant utilizing the effect of operating conditions.

2. Thermodynamic Modelling

The basic of a two shaft gas turbine is consists of a compressor where air is compressed adiabatically and directed to combustion chamber where the fuel is burned with the air, the maximum cycle temperature occurring at state 3 as a result from combustion. The exhaust gases that product from combustion process expanded adiabatically in the high pressure turbine, part from developed work in the high pressure turbine being used to drive the compressor, the remainder being delivered to drive the gas turbine generator. The additional turbine used to drive the gas turbine generator and can call power turbine, so that the gases can expand back to the pressure at the compressor inlet. The two shaft gas turbine illustrated in figure 1, where the compressor is keyed on the high pressure turbine shaft, whereas, the low pressure turbine is keyed on a second shaft that is mechanically separated from the first and activates the gas turbine user. The first assembly is called the gas generator and the second turbine is called the power turbine.

It is assumed that the compressor efficiency (η_c) and turbine efficiency (η_t). The ideal processes and actual processes are represented in dashed line and continuous line, respectively, on the T - s diagram as shown in figure 2. These parameters in terms of temperature are defined as [10].

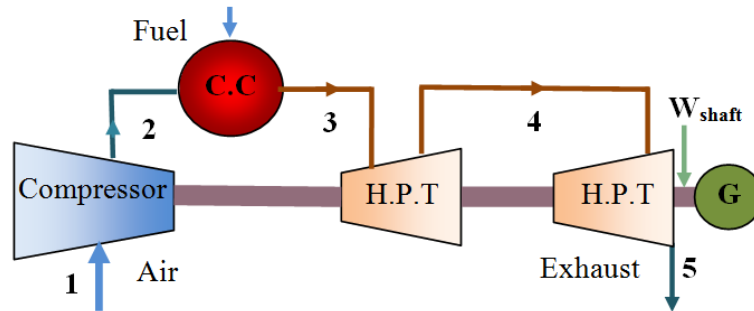


Figure 1. Schematic diagram of two shaft gas turbine

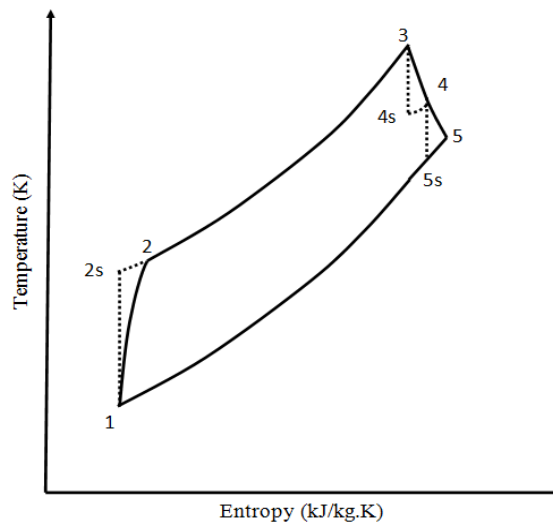


Figure 2. $T-s$ diagram of two shaft gas turbine cycle.

$$\eta_c = \frac{T_{2s} - T_1}{T_2 - T_1} \tag{1}$$

$$\eta_{HPT} = \frac{T_3 - T_4}{T_3 - T_{4s}}, \quad \eta_{LPT} = \frac{T_4 - T_5}{T_4 - T_{5s}} \tag{2}$$

$$\eta_{LPT} = \eta_{HPT} = \eta_t \tag{3}$$

where η_{HPT} and η_{LPT} represent the high and low pressure turbine efficiency respectively.

The final temperature of the compressor is calculated from Eq. (4) [11]:

$$T_2 = T_1 \left(1 + \frac{r_p^{\frac{\gamma_a - 1}{\gamma_a}} - 1}{\eta_c} \right) \tag{4}$$

Therefore it can be simplified these relations by Eq. (5):

$$Rpa = \frac{r_p^{\frac{\gamma_a-1}{\gamma_a}} - 1}{\eta_c}, \quad Rpg = \left(1 - \frac{1}{(r_p)^{\frac{\gamma_g-1}{\gamma_g}}} \right) \quad (5)$$

where $\gamma_a = 1.4$ and $\gamma_g = 1.33$.

The work of the compressor (W_c) when blade cooling is not taken into account can be calculated as:

$$W_c = \frac{C_{pa} \times T_1 \left(r_p^{\frac{\gamma_a-1}{\gamma_a}} - 1 \right)}{\eta_m \times \eta_c} = \frac{C_{pa} \times T_1 \times Rpa}{\eta_m} \quad (6)$$

where C_{pa} is the specific heat of air which can be fitted by Eq. (7) for the range of $200K < T < 800K$ and η_m is the mechanical efficiency of the compressor and turbine [11]:

$$C_{pa} = 1.0189 \times 10^3 - 0.13784 T_a + 1.9843 \times 10^{-4} T_a^2 + 4.2399 \times 10^{-7} T_a^3 - 3.7632 \times 10^{-10} T_a^4 \quad (7)$$

where $T_a = \frac{T_2 - T_1}{2}$ in Kelvin.

The specific heat of flue gas (C_{pg}) is given by Naradasu et al. [11]:

$$C_{pg} = 1.8083 - 2.3127 \times 10^{-3} T + 4.045 \times 10^{-6} T^2 - 1.7363 \times 10^{-9} T^3 \quad (8)$$

From the energy balance in the combustion chamber is expressed as:

$$\dot{m}_a C_{pa} T_2 + \dot{m}_f \times LHV + \dot{m}_f C_{pf} T_f = (\dot{m}_a + \dot{m}_f) C_{pg} \times TIT \quad (9)$$

where, \dot{m}_f is fuel mass flow rate (kg/s), \dot{m}_a is air mass flow rate (kg/s), LHV is low heating value, $T_3 = TIT$ = turbine inlet temperature, C_{pf} is specific heat of fuel and T_f is temperature of fuel. After manipulating Eq. (8), the fuel air ratio (f) is expressed as Eq. (10)

$$f = \frac{\dot{m}_f}{\dot{m}_a} = \frac{C_{pg} \times TIT - C_{pa} T_2}{LHV - C_{pg} \times TIT} \quad (10)$$

The exhaust gases temperature from high gas turbine is given by Eq. (11) [12].

$$T_4 = TIT \left(1 - \eta_t \times \left(1 - \frac{1}{r_p^{\frac{\gamma_g-1}{\gamma_g}}} \right) \right) = TIT (1 - \eta_t \times Rpg) \quad (11)$$

The exhaust gases temperature from low gas turbine is given by Eq. (12).

$$T_5 = T_4 \left[1 - \eta_t \left(1 - \frac{T_{5s}}{T_4} \right) \right] \quad (12)$$

where T_{5s} = adiabatic temperature at exit low pressure turbine, is then defined as Eq. (13).

$$T_{5s} = \frac{\left[T_3 - C_{pa} T_1 \left(\frac{r_p^{\gamma_a}}{\eta_c} \right) / C_{pg} \eta_m \right] \times \left[TIT / \left\{ TIT - C_{pa} T_1 \left(\frac{r_p^{\gamma_a}}{\eta_c} \right) / C_{pg} \eta_m \eta_t \right\} \right]}{r_p^{\frac{\gamma_a-1}{\gamma_a}}} \quad (13)$$

The net work is expressed as in (14) [13]:

$$W_{net} = C_{pg} (T_4 - T_5) \quad (14)$$

In the combustion chamber, the heat supplied by the fuel is equal to the heat absorbed by air, Hence,

$$Q_{add} = C_{pg} (TIT - T_4) \quad (15)$$

The power output is,

$$\text{Power} = \dot{m}_g \times W_{net} \quad (16)$$

where \dot{m}_g = exhaust gases mass flow rate.

The specific fuel consumption is expressed as (17):

$$\text{SFC} = \frac{3600}{\text{AFR} \times W_{net}} \quad (17)$$

Further the thermal efficiency of the cycle is then expressed as in (18).

$$\eta_{th} = \frac{W_{add}}{Q_{net}} \quad (18)$$

4. Results and Discussion

The cycle was modelled using the thermodynamic analysis for the simple gas turbine and two shaft gas turbine. The pressure losses are assumed negligible in this study. The effects of compression ratio across the compressor, turbine inlet temperature, ambient temperature and the air fuel ratio on the first-law efficiency and power are obtained by the energy-balance approach using MATLAB software.

4.1. Effect of Ambient Temperature

Figure 3 shows the effect of ambient temperature and air fuel ratio on thermal efficiency of two shaft gas turbine cycle. Turbine inlet temperature, compression ratio and the components efficiency are of 1425 K, 12 and 0.9 respectively. It can be seen that the thermal efficiency decreases with increases of ambient temperature while increases of air fuel ratio. The specific work of the compressors increases as the ambient temperature increases [10]. Thus the thermal efficiency for the two shaft gas turbine cycles is reduced. It can be noticed that the gain of thermal efficiency decrease of 3.6% with increases of air fuel ratio from (40-60). The density of air increases when ambient temperature decreases, which causes an increase in the air mass flow

rate. The power and heat supplied to the combustion chamber increases with the gas mass flow rate increases. However, the power output increases is more than the increases in the heat supplied in the combustion chamber. The simulated thermal efficiency of a two shaft gas turbine is higher than the thermal efficiency of simple gas turbine cycle as shown in figure 4.

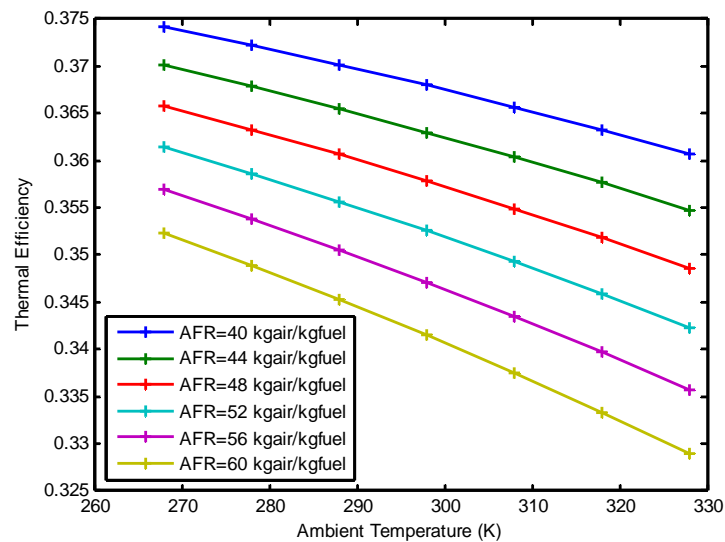


Figure 3. Effect of ambient temperature and air fuel ratio on thermal efficiency of two shaft gas turbine cycle.

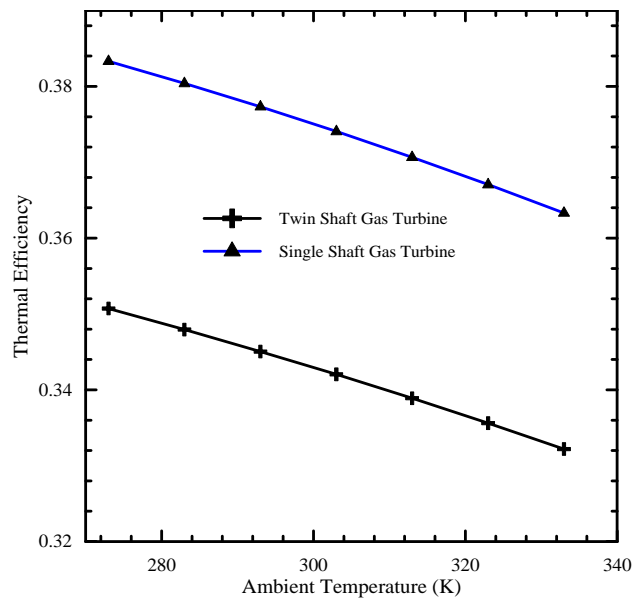


Figure 4. Effect of ambient temperature on thermal efficiency of two shaft and simple gas turbine cycle.

Figure 5 shows the variation of gas turbine power output with air fuel ratio as well as ambient temperature. It is noticed that the power output decreases linearly with increase of ambient temperature as well as increases the air fuel ratio. The simulated power output of a two shaft gas turbine is higher than the simple gas turbine cycle and practical data from Baiji gas turbine power plant as shown in figure 6. Figure 7 shows the effect of ambient temperature and air fuel ratio on specific fuel consumption. It can be seen that the specific fuel consumption increase linearly with increase of both ambient temperature and air to fuel ratio.

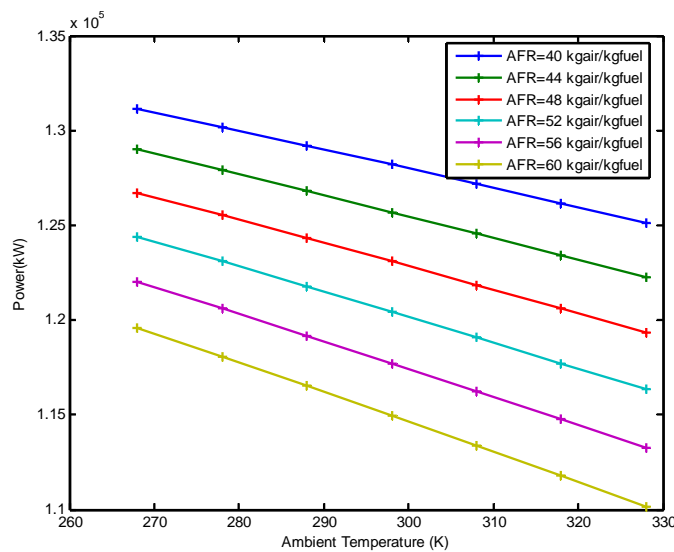


Figure 5. Effect of ambient temperature and air fuel ratio on power output of two shaft gas turbine cycle.

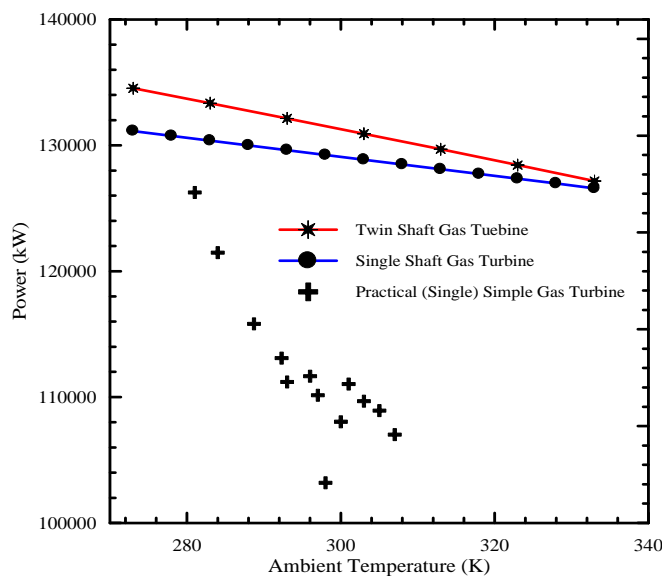


Figure 6. Effect of ambient temperature and air fuel ratio on power output of different gas turbine cycles.

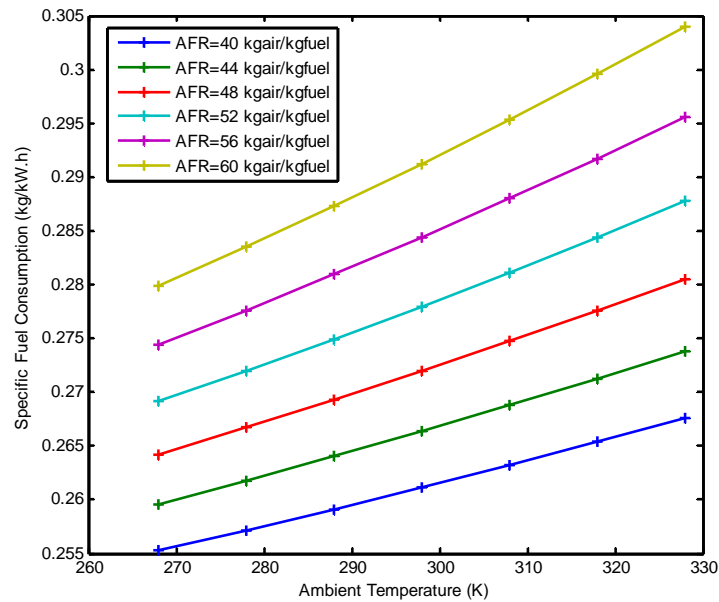


Figure 7. Effect of ambient temperature and air fuel ratio on specific fuel consumption of two shaft gas turbine cycle.

4.2. *Effect of Compression Ratio*

Figure 8 shows the variation of two shaft gas turbine cycle thermal efficiency with compression ratio and ambient temperature. The thermal efficiency increases with increases of compression ratio as well as lower ambient temperature. However, the variation of efficiency is insignificant at lower compression ratio. The thermal efficiency at high compression ratio decreases from 43.8% to 41% when the ambient temperature increases from 268 to 328K.

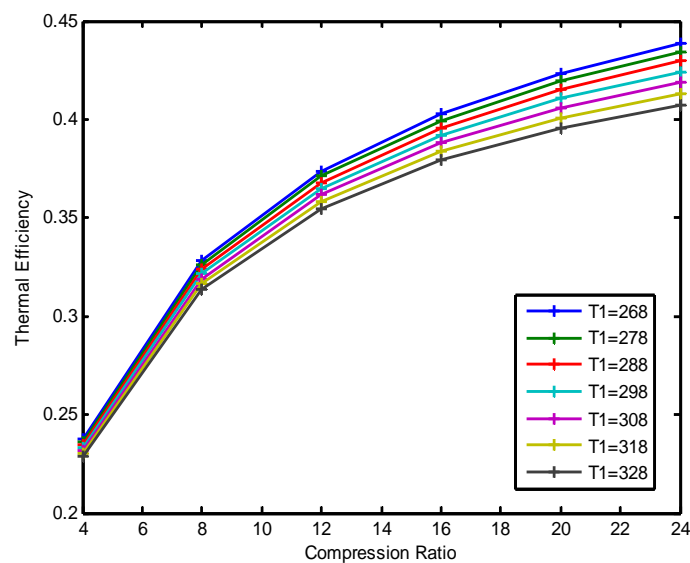


Figure 8. Effect of compression ratio and ambient temperature on thermal efficiency of two shaft gas turbine cycle.

The variation of two shaft gas turbine cycle power output with compression ratio and ambient temperature shown in figure 9. It can be seen that the power output insignificant at lower compression ratio and it significance at high compression ratio. The power output at high compression ratio decreases from 149MW to 137MW when the ambient temperature increases from 268 to 328K. Figure 10 shows effect of compression ratio and ambient temperature on specific fuel consumption of two shaft gas turbine cycle. The specific fuel consumption decreases from 0.4 to 0.22 kJ/kW.K when the compression ratio increases from 4 to 24. The effect of variation of SFC is more significance at higher ambient temperature than lower temperature.

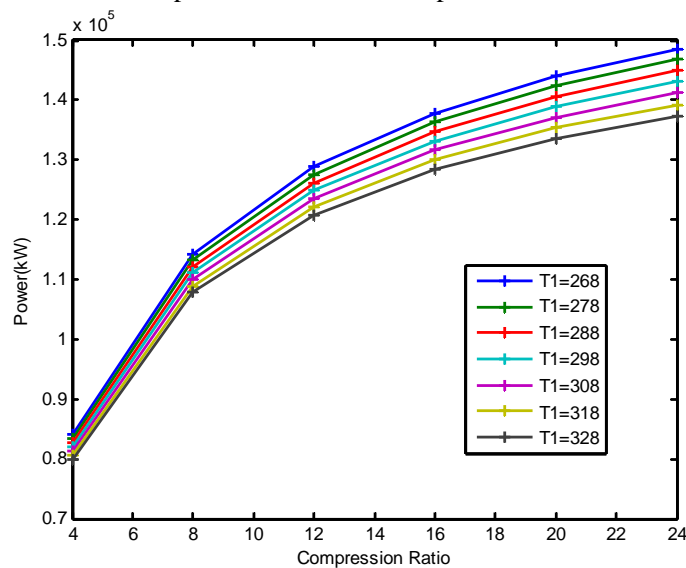


Figure 9. Effect of compression ratio and ambient temperature on power output of two shaft gas turbine cycle.

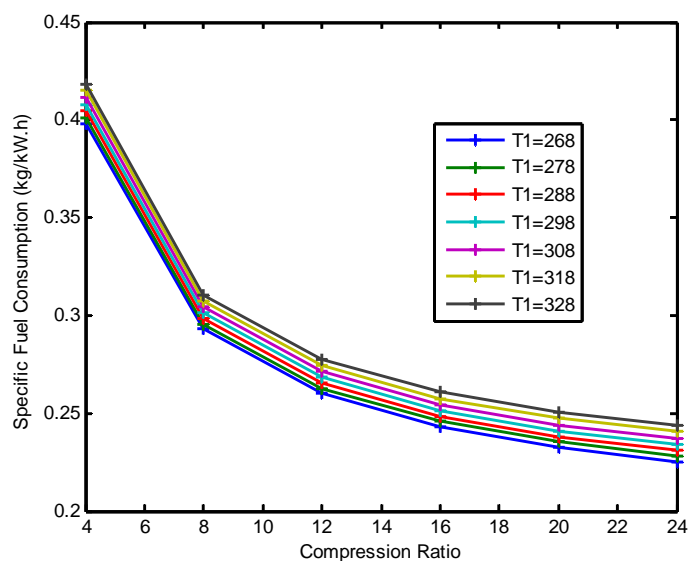


Figure 10. Effect of compression ratio and ambient temperature on specific fuel consumption of two shaft gas turbine cycle.

4.3. Effect of compressor and Turbine Isentropic Efficiency

Figure 11 shows the effect of the compressor isentropic efficiency on thermal efficiency for different ambient temperature. The thermal efficiency increases linearly with increase of the compressor isentropic efficiency. This means that the thermal losses reduced in compressor which leads to increase the power output. However, the variation of thermal efficiency is significance at lower compressor isentropic efficiency. The thermal efficiency decreases with increase of ambient temperature.

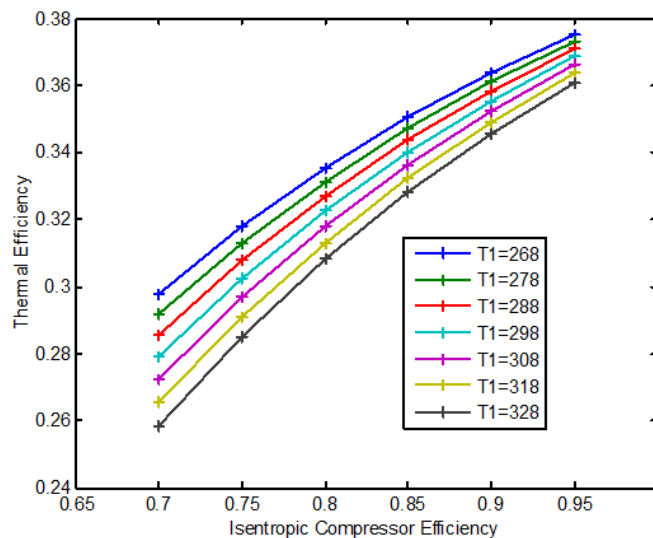


Figure 11. Effect of isentropic compressor efficiency and ambient temperature on thermal efficiency of two shaft gas turbine.

Figure 12 shows the effect of the turbine isentropic efficiency on thermal efficiency for different ambient temperature. The thermal efficiency increases linearly with increase of the turbine isentropic efficiency, however the variation thermal efficiency is more significant as lower isentropic efficiencies. The thermal efficiency decreases from 29.6% to 25.7% when the ambient temperature increases from 268K to 328K with effect of isentropic compressor efficiency, however the thermal efficiency decreases from 21.5% to 16.7% with effect of isentropic turbine efficiency for the same range of air to fuel ratio.

4.4. Effect of Turbine Inlet Temperature and exhaust temperature

Figure 13 represents the variation of exhaust temperature on thermal efficiency for several ambient temperature and air fuel ratio. It is noticed that the thermal efficiency increases with decrease of the ambient temperature and air fuel ratio while the thermal efficiency decreases with increase of the exhaust temperature. Figure 14 represents the variation of thermal efficiency against the turbine inlet temperature as well as compression ratio and air fuel ratio. The air fuel ratio varies of 40-52 (kgair/kgfuel) and compression ratio varies of 4-28. Thermal efficiency are strongly related with turbine inlet temperature, the efficiency increases when turbine inlet temperature increases and also decreases the air to fuel ratio caused increases of thermal efficiency and turbine inlet temperature.

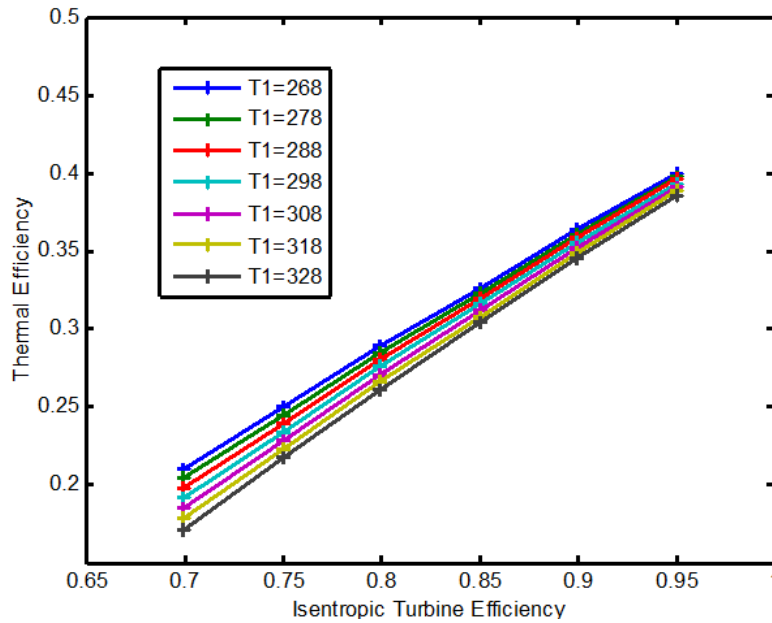


Figure 12. Effect of isentropic turbine efficiency and ambient temperature on thermal efficiency of two shaft gas turbine.

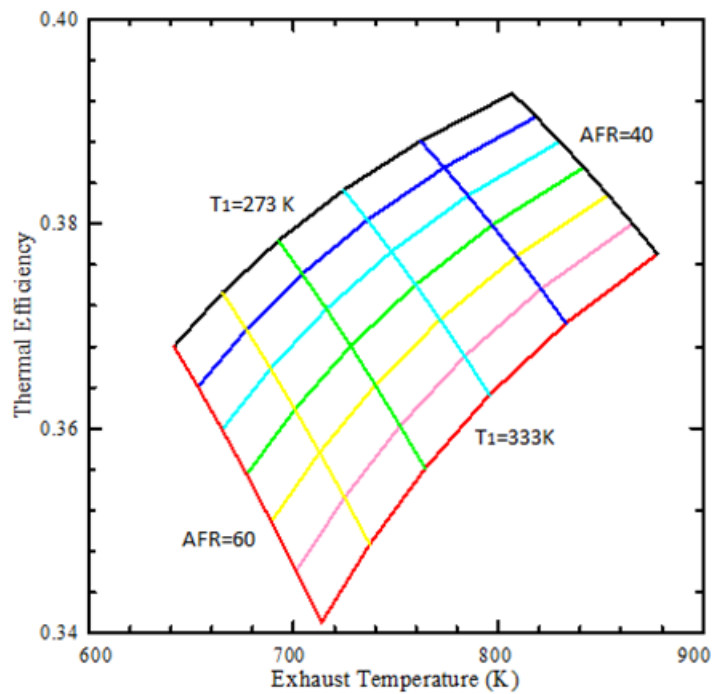


Figure 13. Effect of exhaust temperatures on thermal efficiency for ambient temperature and air to fuel ratio.

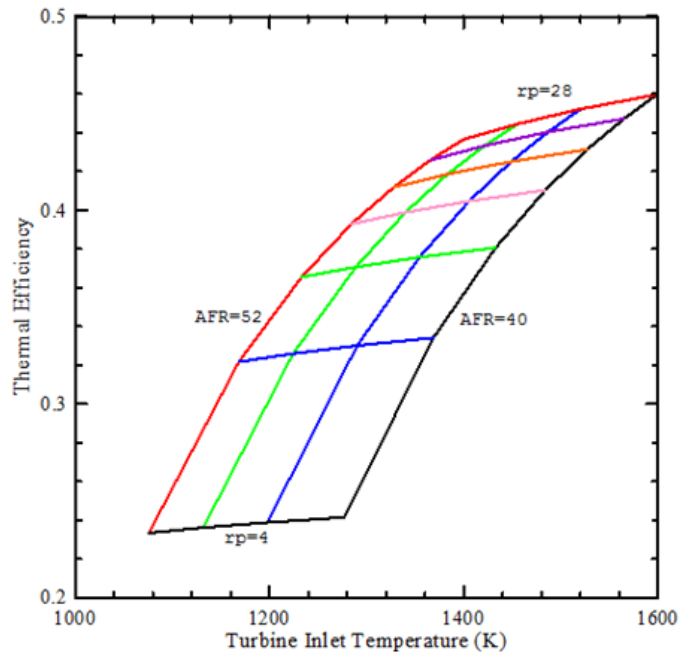


Figure 14. Variation of thermal efficiency against TIT, compression ratio and air fuel ratio.

5. Conclusions

The simulated modelling results shows that the influence of ambient temperature, compression ratio, air to fuel ratio, isentropic components efficiencies, turbine inlet temperature are significantly effect on performance of two shaft gas turbine power plant. The results were summarized as follows:

1. The compression ratios, ambient temperature, air fuel ratio as well as the isentropic efficiencies are strongly influence on the thermal efficiency of the two shaft gas turbine power plant.
2. The thermal efficiency and power output decreases linearly with increase of ambient temperature as well as the air to fuel ratio.
3. The peak efficiency, power and specific fuel consumption occur at when compression ratio increased in the gas turbine power plant.

The power of the simulated two shaft gas turbine is higher than the simple gas turbine cycle and real data from Baiji gas turbine power plant.

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