PERFORMANCE ANALYSIS OF A COMBINED CYCLE GAS TURBINE UNDER VARYING OPERATING CONDITIONS

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***Abstract* :**

The combined cycle gas turbine integrates the Brayton cycle as topping cycle and the steam turbine Rankine cycle as bottoming cycle in order to achieve higher thermal efficiency and proper utilization of energy by minimizing the energy loss to a minimum. In this work, the effect of various operating parameters such as maximum temperature and pressure of Rankine cycle, turbine inlet temperature and pressure ratio of Brayton cycle on the net output work and thermal efficiency of the combine cycle are investigated. The outcome of this work can be utilized in order to facilitate the design of a combined cycle with higher efficiency and output work. A MATLAB simulation has been carried out to study the effects and influences of the above mentioned parameters on the efficiency and work output.

Key-words :

Brayton Cycle; Rankine Cycle; Combined Cycle Gas Turbine; Efficiency; Pressure ratio.

1. INTRODUCTION:

The combined cycle gas turbine (CCGT) technology has attracted much attention by the researchers in the last few decades by utilizing the Brayton cycle gas turbine and Rankine cycle steam turbine with air and water as working fluids to achieve efficient, reliable, and economic power generation. Currently overall thermal efficiencies up to 60% are confirmed by the foremost manufacturers from the sector as state of the art and special modifications have been proposed to improve the overall thermal efficiencies more than 60%. Higher efficiency level in CCGT up to 65% can be achieved by incorporating enhancements in the gas turbine technology such as higher compression ratio, turbine inlet temperature and ambient condition. The improvement in HRSG design, the use of advanced thermodynamic plant configurations such as two shaft gas turbine, combining intercooler cycle and regenerative cycle may be adopted to improve the thermal efficiency.

The most commonly used means in industrial practice for generation of mechanical power is the utilization of gas and steam turbines. Different means have been employed by a lot of researchers to get better thermal efficiency of the turbines. Combining two or more thermodynamic cycles result in improved overall efficiency, reducing fuel costs. In stationary power plants, a widely used combination is a gas turbine operating by the Brayton cycle whose hot exhaust powers a steam power plant operating by the Rankine cycle. This is called a Combined Cycle Gas Turbine (CCGT) plant, and can achieve a thermal efficiency of around 60%, in contrast to a single cycle steam power plant which is limited to efficiencies of around 35-42%. When combined, the gas turbine (GT) Brayton cycle and the steam turbine (ST) power plant Rankine cycle complement each other to form an efficient CCGT. The Brayton cycle has a high source temperature and rejects heat at a temperature that is conveniently used as the energy source for the Rankine cycle plant [1].

The gas turbine is a quite complex equipment and its performance and reliability are governed by many a number of parameters. The major variables that affect the gas turbines include type of application, plant location and site configuration, plant size and efficiency, type of fuel, enclosures and plant operation mode like base or peaking [2]. The performance of the gas turbine relies mostly on the efficiency achieved at the compressor of the turbine. The performance and reliabilty improvement of the gas turbine is dependent on the maximum temperature tolerance of the first stage blades and is also reliant on inter stage cooling at the compression stage [3].

Many researchers have focussed on improvement in the modeling of CCGT power plant system utilizing the Brayton Cycle gas turbine and Rankine Cycle steam turbine with air and water as working fluids. Many of these methods such as use of air cooler [4], regenerative steam injection [5], effusive blade cooling techniques [6], use of desiccant-based evaporative cooling [7] or absorption chillers [8] are commonplace. Kaushika et al. [9] studied the optimum performance of a CCGT by modeling and simulation. The best combination of process parameters of steam leaving the steam generator was determined at part load operation for optimum performance of the CCGT. Khaliq and Kaushik [10] developed a simulator of the combined-cycle co-generation power plant by a mathematical model for power plant modeling. The simulator was consisting of two parts out of which the first one was a simulation of fluid flow in the power plant and the second part was a simulation of the control system of the plant.

The thermal impact of operating conditions on the performance of a combined cycle gas turbine was investigated by Ibrahim and Rahman [11]. They have studied effects of varying the operating conditions such as ambient temperature, compression ratio, turbine inlet temperature, isentropic compressor and turbine efficiencies, and mass flow rate of steam on the overall efficiency and total output power of the CCGT. Rai et. al. [12] presented a model of gas turbine for optimizing the power of CCGT by varying different operating parameters. According to their studies the efficiency of a gas turbine which ranges from 28% to 33% can be raised to about 60% by recovering some of the low grade thermal energy from the exhaust gas for steam turbine process.

This work presents a matlab simulation based study of variation of the work output and thermal efficiency on operating parameters such as maximum temperature, maximum pressure, pressure ratio and turbine inlet temperature. The CCGT consists of a topping Brayton cycle and the bottoming cycle is a steam tubine Rankine cycle.

2. THERMODYNAMIC MODELLING OF THE CCGT CYCLE

A combined cycle gas turbine power plants having Bray-ton cycle based topping cycle and Rankine cycle based bot-toming cycle has been considered for the present study and analysis of variation in work out and thermal efficiency for different operating conditions. Figure 1 [13] shows the schematic of a conventional CCGT with topping Brayton cycle and bottoming Rankine cycle. Generally the Gas turbine power plants consist of four major components namely the compressor, combustion chamber, turbine and generator. According to the basic principle of the CCGT, the air is compressed by the air compressor and transferred to the combustion chamber where it combines with liquid or gaseous fuel for producing high-temperature flue gas through the process of combustion. Hot gases leaving the combustion chamber expands in the turbine thereby producing output work and finally discharges to the atmosphere. The waste exhaust gas temperature from gas turbine decreases as it flows into the heat recovery steam generator (HRSG), which consists of superheater, evaporator and economizer. Then the HRSG supplies a steam for the steam turbine in producing electricity. The temperature-entropy plot of a commonly used CCGT cycle is presented in Figure 2 [13]. The systematic modeling of a CCGT is presented in following steps.

The concept of compressor polytropic efficiency can be developed considering the definition of the isentropic or overall compressor efficiency with reference to the pressure ratio.

The ideal and actual processes on the temperature-entropy diagram for an actual and ideal Brayton cycle are shown in Figure 3 [2]. Let the compressor and turbine efficiency is η_c and η_T respectively.

2.1 Air Compressor Model:

The compressor efficiency can be determined taking into account the deviation of the actual cycle from the ideal cycle. The compression ratio across the compressor is given as

$$r_p = \frac{P_2}{P_1} \tag{1}$$

where r_p is the compression ratio, P_1 is the initial pressure before compression and P_2 is the pressure after compression.

The isentropic efficiency of compressor (η_c) can be given as

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

where T_1 is the compressor inlet temperature, T_2 is the temperature at the end of isentropic compression and $T_{2'}$ is the temperature at the end of actual compression.

The same principle can be applied to the gas turbine expansion process to determine the turbine efficiency.

$$\eta_T = \frac{T_3 - T_{4'}}{T_3 - T_4} \tag{3}$$

where,

η_T = Isentropic efficiency of turbine
T_3 = Turbine inlet temperature
T_4 = Turbine exit temperature in isentropic expansion process.
$T_{4'}$ = Turbine exit temperature during actual expansion process.

The actual temperature at the outlet of compressor can be calculated taking into account the compressor efficiency (η_c) and the specific heat ratio for air (γ_a).

$$T_{2'} = T_1\left(1+\frac{r_p^{\frac{\gamma_a-1}{\gamma_a}}-1}{\eta_c}\right) \tag{4}$$

These relationships can be further expressed as [1]

$$R_{pa} = \frac{r_p^{\frac{\gamma_a-1}{\gamma_a}}-1}{\eta_c} \tag{5}$$

$$R_{pg} = 1-\frac{1}{r_p^{\frac{\gamma_g-1}{\gamma_g}}} \tag{6}$$

The compressor work neglecting the blade cooling (W_c) can be calculated as

$$W_c = \frac{C_{pa}\times T_1\left(r_p^{\frac{\gamma_a-1}{\gamma_a}}-1\right)}{\eta_m\times\eta_c} = \frac{C_{pa}\times T_1\times R_{pa}}{\eta_m} \tag{7}$$

where η_m is the mechanical efficiency of the compressor and C_{pa} is the specific heat of air. The specific heat of air can be determined by considering the empirical relations depending on the working range of temperature.[14]

2.2 Combustion Chamber Model:

The combustion chamber is a component of gas turbine where combustion takes place. The combustion chamber is fed with high pressure air which is being heated at constant pressure before being passed through the nozzle guide vanes to the turbine. The basic principle of operation of a combustion chamber is based on the energy balance principle. Applying the principle of energy balance [15],

$$m_a C_{pa} T_{2'} + m_f \times LHV + m_f C_{pf} T_f = \left(m_a + m_f\right) C_{pg} \times TIT \tag{8}$$

where m_f is the mass flow rate of fuel (kg/s), m_a is the mass flow rate of air(kg/s), LHV is low heating value, *TIT* is the turbine inlet temperature, C_{pf} is the specific heat of fuel, and T_f is the temperature of the fuel. The specific heat of the flue gas can be determined from the empirical relations available in literature. However in this work, the relationship developed by Naradasu et. al. is taken into consideration [16].

$$C_{pg} = 1.8083 - 2.3127\times10^{-3}T + 4.045\times10^{-6}T^2 - 1.7363\times10^{-9}T^3 \tag{9}$$

The air fuel ratio can be expressed as

$$\frac{m_f}{m_a} = \frac{C_{pg}\times TIT - C_{pa}T_1\left(1+R_{pg}\right)}{LHV + C_{pf}\times T_f - C_{pg}\times TIT} \tag{10}$$

2.3 Gas Turbine Model:

A gas turbine, also known as combustion turbine has an upstream rotating compressor coupled to a downstream turbine, and a combustion chamber in-between. Fresh atmospheric air flows through a compressor that brings it to higher pressure. Energy is then added by spraying fuel into the air and igniting it so the combustion generates a high-temperature flow. This high-temperature high-pressure gas enters a turbine, where it expands down to the exhaust pressure, producing a shaft work output in the process.

The exhaust gas temperature of the gas turbine can be expressed as

$$T_{4'} = T_{3'}\left[1-\eta_T\times\left(1-\frac{1}{r_p^{\frac{\gamma_g-1}{\gamma_g}}}\right)\right] \tag{11}$$

The total output work of the turbine (W_T) is expressed as

$$W_T = C_{pg}\times TIT\times\eta_T\times R_{pg}/\eta_m \tag{12}$$

Hence the net work output of the gas turbine is

$$W_{net} = W_T - W_C \tag{13}$$

The total heat supplied is given as

$$Q_1 = C_{pg}\left[TIT - T_1\left(1+R_{pa}\right)\right] \tag{14}$$

Hence the thermal efficiency of the gas turbine can be determined as

$$\eta_{th} = \frac{W_{net}}{Q_1} \qquad (15)\backslash$$

2.4 Steam Turbine Model

The behavior of the subsystems of a steam turbine cycle can be captured in terms of the mass and energy conservation equations, semi-empirical relations and thermodynamic state conservation. The system dynamics is represented by a number of lumped models for each subsections of turbine. There are many dynamic models for individual components, which are simple empirical relations between system variables with a limited number of parameters and can be validated for the steam turbine by using real system responses. In this work, a single pressure heat recovery steam generator (HRSG) is considered here as a common type for the combined cycle gas turbine power plant. Applying energy balance, the heat available with exhaust gases (Q_a) from the gas turbine can be given as

$$Q_a = m_g \times C_{pg} \times \left(T_{g1} - T_{g4}\right) \times h_{1f} \tag{16}$$

where T_{g4} is the exhaust temperature from HRSG and h_{1f} is the heat loss factor. Generally the heat loss factor varies within a range from 0.98 to 0.99 [17]. Similarly the energy balance can be applied to determine the temperature of the exhaust hot gases exit from the HRSG.

$$m_s\left(h_{sh} - h_{w1}\right) = m_g C_{pg}\left(T_{g1} - T_{g4}\right) \tag{17}$$

The steam obtained from HRSG expands to the condenser pressure in the steam turbine. The energy balance of steam turbine as represented in Figure 1 gives

$$W_s = m_s\left(h_3 - h_4\right) \tag{18}$$

The heat rejected in the condenser is given as

$$Q_c = m_w\left(h_4 - h_1\right) \tag{19}$$

Similarly, the pump work can be determined as

$$W_p = m_w v_{f1}\left(P_{sh} - P_c\right) \tag{20}$$

The net output work of the steam turbine plant is

$$W_{snet} = W_s - W_p \tag{21}$$

Thus the overall thermal efficiency ($\eta_{overall}$) of the combined cycle is

$$\eta_{overall} = \frac{W_{net} + W_{snet}}{Q_a} \tag{22}$$

3. RESULTS AND DISCUSSION

The effect of various operating conditions such as maximum temperature, turbine inlet temperature, pressure ratio and maximum pressure on the output work and thermal efficiency of the cycle have been considered in this work. The MATLAB script is utilized to investigate the required objectives.

Figure 3 represents the variation of output work with maximum pressure for different cases of maximum steam turbine cycle temperature. In this case the pressure ratio remains fixed at 7 while the turbine inlet temperature is fixed at 1500K. Three different conditions of steam turbine cycle maximum temperatures are considered at 600, 750 and 900K. As observed from the simulation results, it is found that the net work output of the cycle increases with increase in steam turbine cycle temperature. For the above mentioned pressure ratio and gas turbine inlet temperature, the output work is around 975kJ/kg for steam turbine temperature of 900K, the same falls to 945kJ/kg at steam turbine temperature of 600K while the maximum pressure is 25MPA. However it can be observed that for all ranges of temperature, output work increases with maximum pressure of the cycle. Figure 5 represents the variation of thermal efficiency with maximum pressure under same operating conditions. It can be observed that for a maximum pressure of 25Mpa, the thermal efficiency corresponding to maximum steam turbine cycle temperature of 900K reaches approximately 67.5% while for 600K it attains around 65.3%. Similarly at a lower magnitude of maximum pressure of 5Mpa, the same parameters are 65.4% and 63% respectively. Thus it can be observed that the variation of thermal efficiency is not much significant with maximum steam cycle temperature for maximum pressure of the cycle.

Figure 6 represents the behaviour of work output with pressure ratio for different values of gas turbine inlet temperature while the maximum pressure of the cycle and the maximum temperature of steam turbine cycle are fixed. As observed from the Figure, for a particular value of gas turbine inlet temperature, the outwork remains almost constant with pressure ratio. In this work the pressure ratio is considered within a range from 4 to 10. However the output work indicates a significant increase in magnitude for a pressure ratio when the turbine inlet temperature increases. For a pressure ratio of 4, the output work increases from 385 to 980kJ/kg when turbine inlet temperature increases from 1100K to 1900K. For turbine inlet temperature of 1500K, the same attains a value of 68kJ/kg. This indicates that turbine inlet temperature has a significant effect on the output work of the CCGT cycle. As compared to output work, the thermal efficiency follows almost a similar behaviour with turbine inlet temperature and pressure ratio as shown in Figure 7. However unlike output work, the pressure ratio has a significant effect on thermal efficiency unlike that of output work. For turbine inlet temperature of 1100K, the efficiency increases from 53.6% to 59.5% over a pressure ratio of 4 to 10. Simultaneously, the efficiency varies from 59.5% to 66% when the turbine inlet temperature increases from 1100K to 1900K. Thus by increasing the pressure ratio and turbine inlet temperature, the overall thermal efficiency can be increased significantly. However other operating parameters have to be taken into consideration regarding increase in pressure ratio and turbine inlet temperature.

Figure 8 represents the variation of output work with maximum steam cycle temperature for different maximum cycle pressure while turbine inlet temperature and pressure ratio are kept constant. It can be observed from the Figure that there is not much variation in output work for maximum cycle pressure for a particular value of maximum steam cycle temperature. There is approximately a difference of 3 to 4 kJ/kg in output work over a range of maximum pressure from 3 to 27 MPa. But the output work increases from 940 to 982 kJ/kg when the maximum temperature varies from 550 to 950K for maximum cycle pressure of 3Mpa. Similar behaviour of thermal efficiency can be observed for maximum temperature corresponding to maximum cycle pressure as shown in Figure 9. The thermal efficiency varies with a margin of approximately 1.5% for maximum pressure variation from 3 to 27 Mpa. But for maximum pressure of 3 Mpa, the efficiency increases by 3 to 4% for the considered range of temperature.

Figure 10 represents the effect of turbine inlet temperature on output wotk for three different values of compression ratio for constant values of maximum cycle pressure and maximum steam cycle temperature. It can be observed that the variation in output work for compression ratio is not much significant. There is approximately a difference of 20 to 25kJ of output work is observed for difference in pressure ratio from 4 to 10. For pressure ratio of 4, the output increases from 310 to 920 kJ/kg when the turbine inlet temperature increases from 1100K to 1900K. Similar trend is observed for other values of pressure ratios. Thus the turbine inlet temperature significantly affects the output work. Figure 11 represents the variation of thermal efficiency with turbine inlet temperature for constant values of maximum cycle pressure and maximum steam cycle temperature. The efficiency has been considered for pressure ratio values 4, 7 and 10. Unlike output work, distinctive effect of pressure ratio on efficiency is observed. For turbine inlet temperature of 1100K, the efficiency increases from 59.5% 64.6% when pressure ratio is varied from 4 to 10. Similarly for same range of pressure ratio, efficiency increases from 65.8 to 69.8% when turbine inlet temperature is 1900K. In a similar manner, significant effect of turbine inlet temperature on efficiency is observed for a particular pressure ratio. For pressure ratio 4, the efficiency increases from 59.5% to 65.8% over a range of turbine inlet temperature from 1100K to 1900K. Almost a similar trend is obtained for pressure ratio 7 and 10. However for pressure ratio 10, an abrupt trend of efficiency is observed within turbine inlet temperature from 1600K to 1800K. This may be due to some computational error. Besides it can be clearly observed that the overall thermal efficiency of a CCGT cycle can be improved significantly by increasing the turbine inlet temperature keeping in view the mechanical constraints.

5. CONCLUSION:

A model of CCGT was developed and variation of output work and thermal efficiency by varying various operating parameters were studied. The simulated modeling results show that the maximum cycle pressure, maximum temperature, pressure ratio and turbine inlet temperature have significant effects on the performance of the CCGT.

For constant pressure ratio and turbine inlet temperature, outwork and thermal efficiency are strongly influenced by maximum cycle pressure and maximum temperature of steam turbine cycle.

For constant maximum cycle pressure and maximum temperature of steam turbine, the output work is not much affected by pressure ratio, but it is strongly affected by turbine inlet temperature. However under similar operating conditions, efficiency is significantly affected by both pressure ratio and turbine inlet temperature.

For constant turbine inlet temperature and pressure ratio, both output work and efficiency increases with maximum cycle pressure and maximum temperature of steam turbine cycle. However a significant increase occurs in output work with maximum steam turbine cycle temperature.

For maximum cycle pressure and maximum temperature of steam turbine temperature being maintained constant, both the output work and thermal efficiency are not much affected by pressure ratio, but significantly affected by turbine inlet temperature.

Hence keeping in view the mechanical constraints the above mentioned parameters can be monitored in order to achieve higher thermal efficiency and output work of a combine cycle gas turbine.

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Figures:

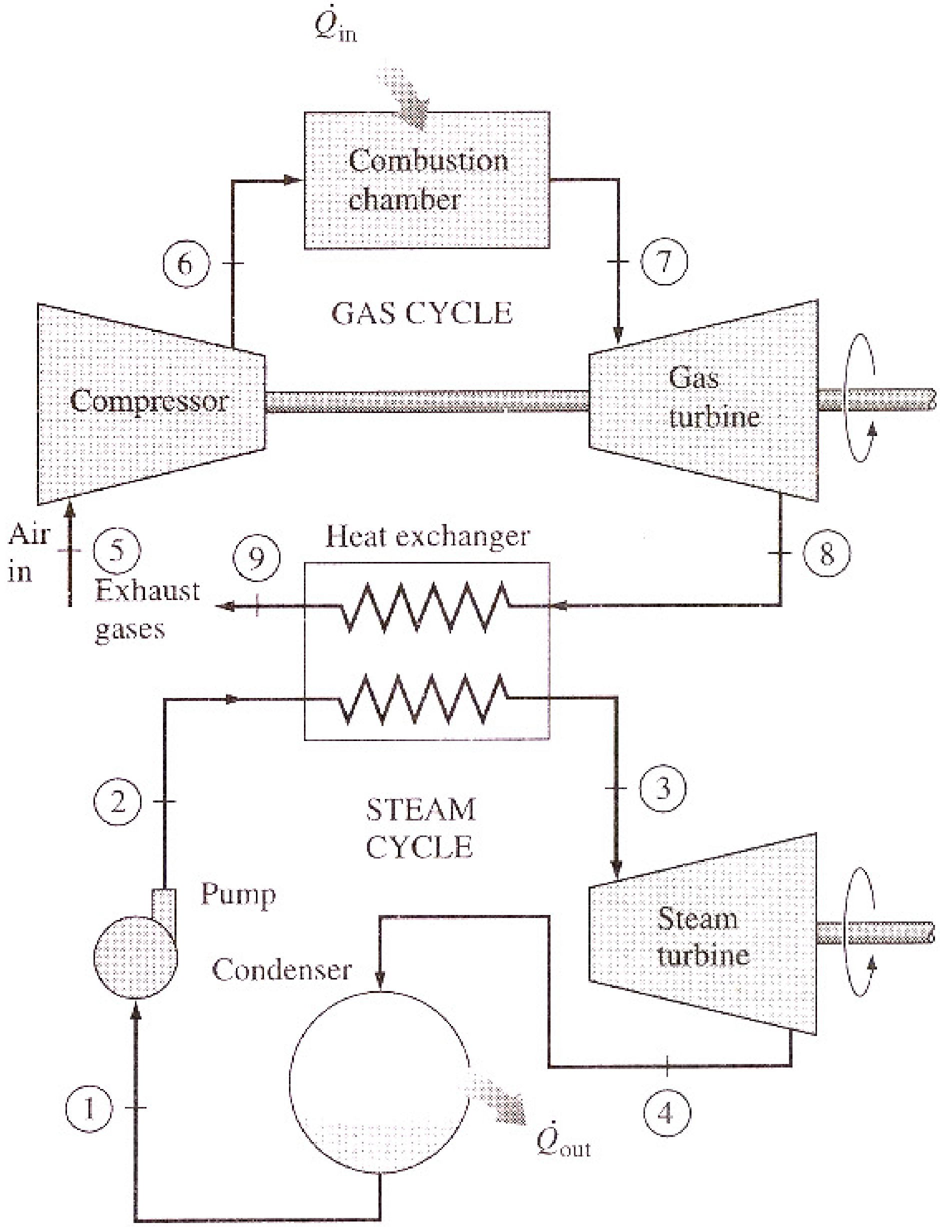

Figure 1: schematic diagram of a conventional CCGT cycle

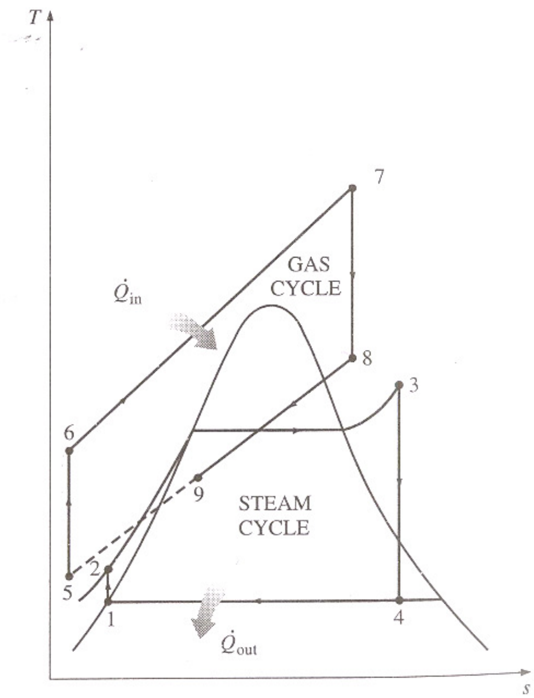

Figure 2 : Temperature-entropy plot of a CCGT cycle

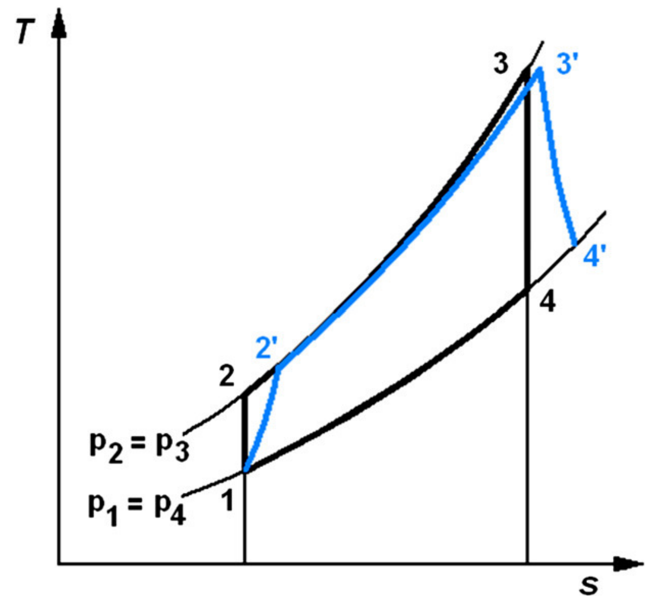

Figure 3 : Temperature-entropy diagram for an actual and ideal Brayton cycle

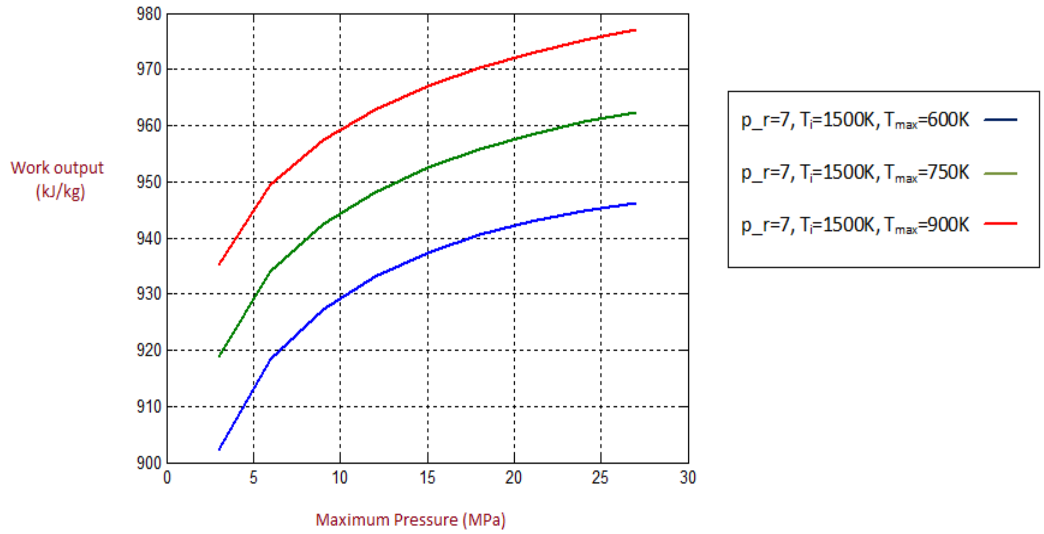

Figure 4 : Variation of Work Output with Maximum pressure

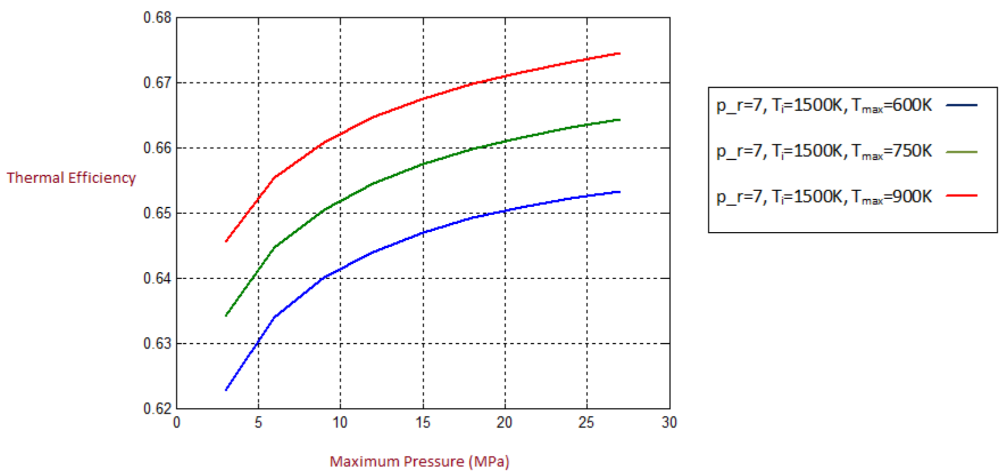

Figure 5 : Variation of Thermal Efficiency with Maximum pressure

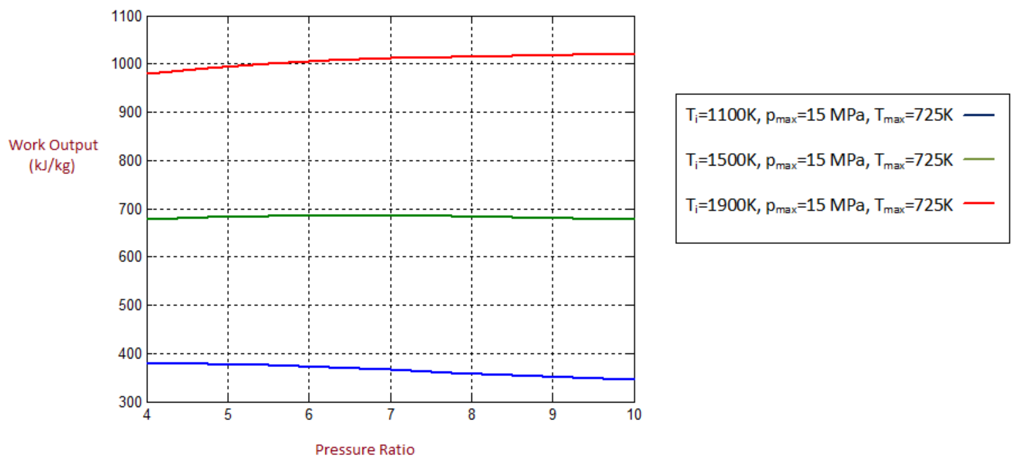

Figure 6 : Variation of Work Output with pressure ratio

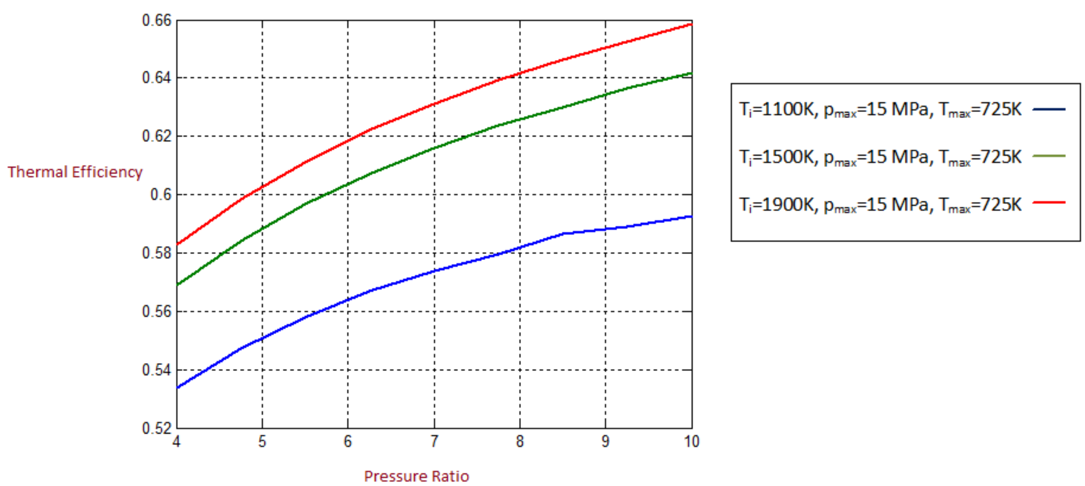

Figure 7 : Variation of Thermal Efficiency with pressure ratio

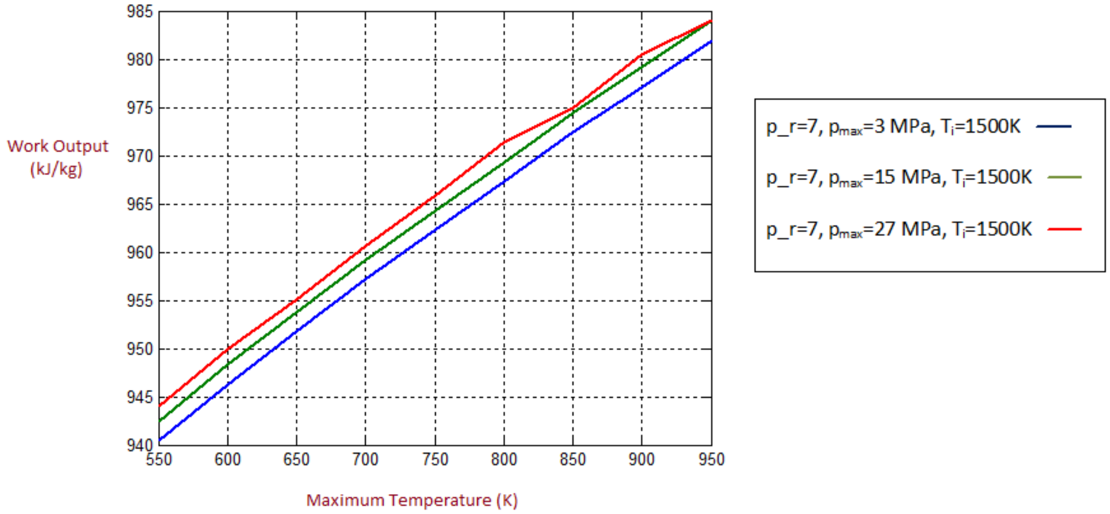

Figure 8 : Variation of Work Output with Maximum temperature

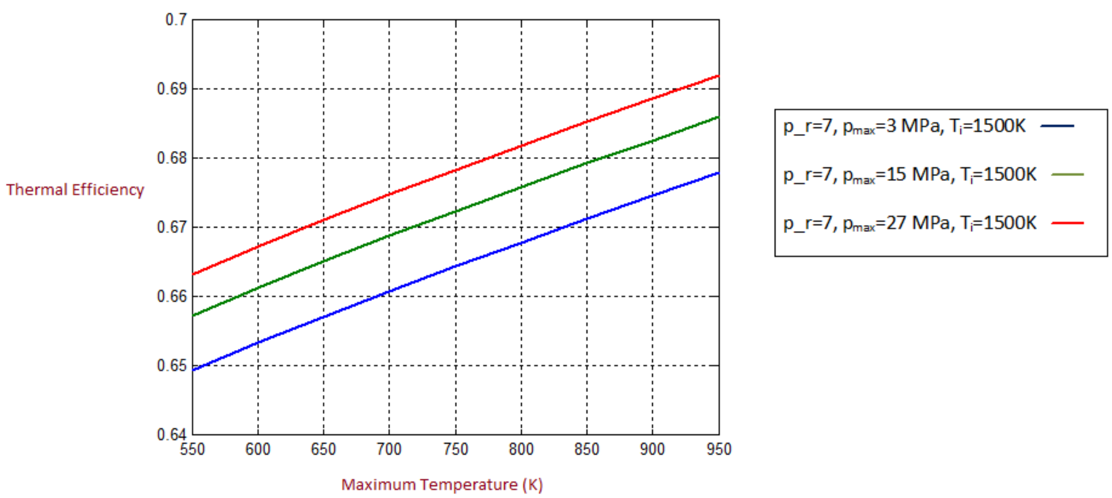

Figure 9 : Variation of Thermal Efficiency with Maximum temperature

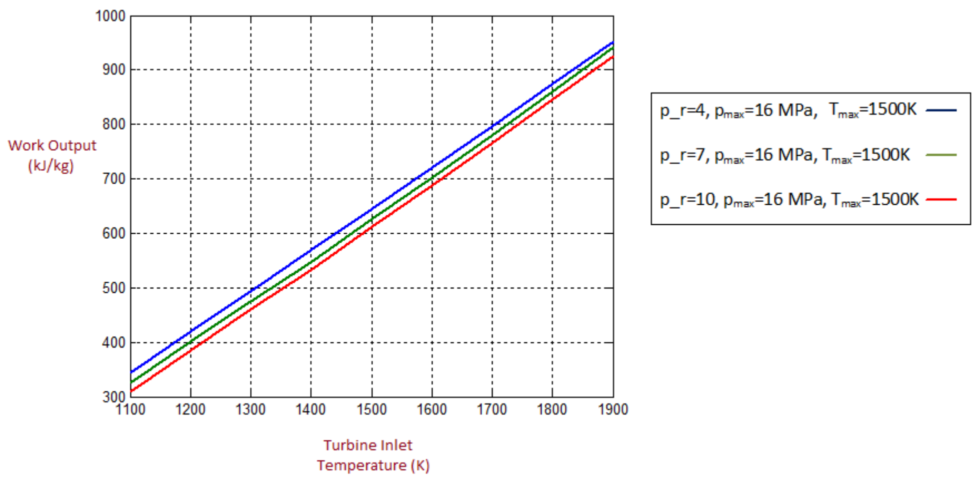

Figure 10 : Variation of Work Output with Turbine inlet temperature

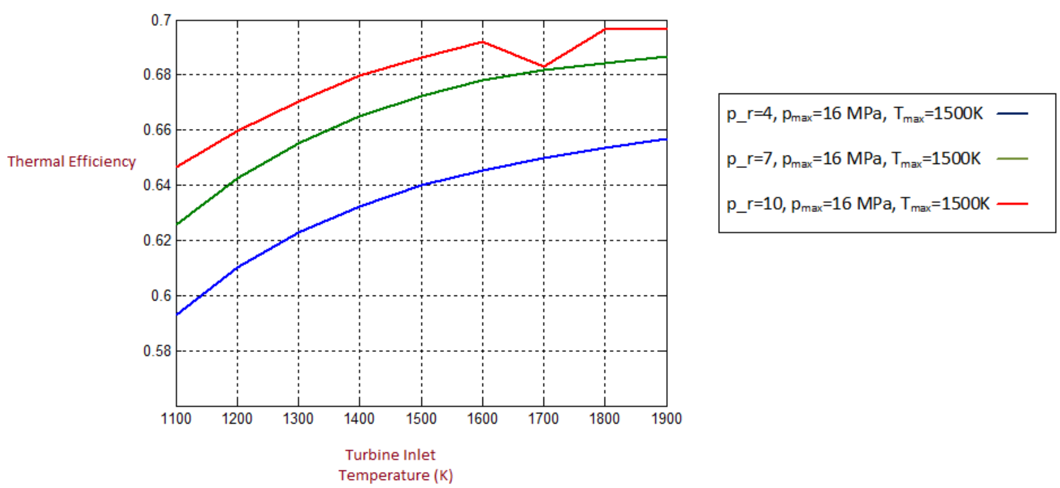

Figure 11 : Variation of Thermal Efficiency with turbine inlet temperature