ME466 Project I

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1. INTRODUCTION

In this project, a thermodynamic model of a Brayton cycle which can be seen in figure 1 has been examined. As seen in Figure 1, the Brayton cycle consists of 2 compressors, 1 turbine, 1 combustion chamber, 1 regenerator and 1 intercooler. The goal of the project is to observe and interpret changes in engine inlet volumetric flowrates, specific power outputs, back-work ratios and cycle thermal efficiencies under different ambient temperatures and pressure ratios.

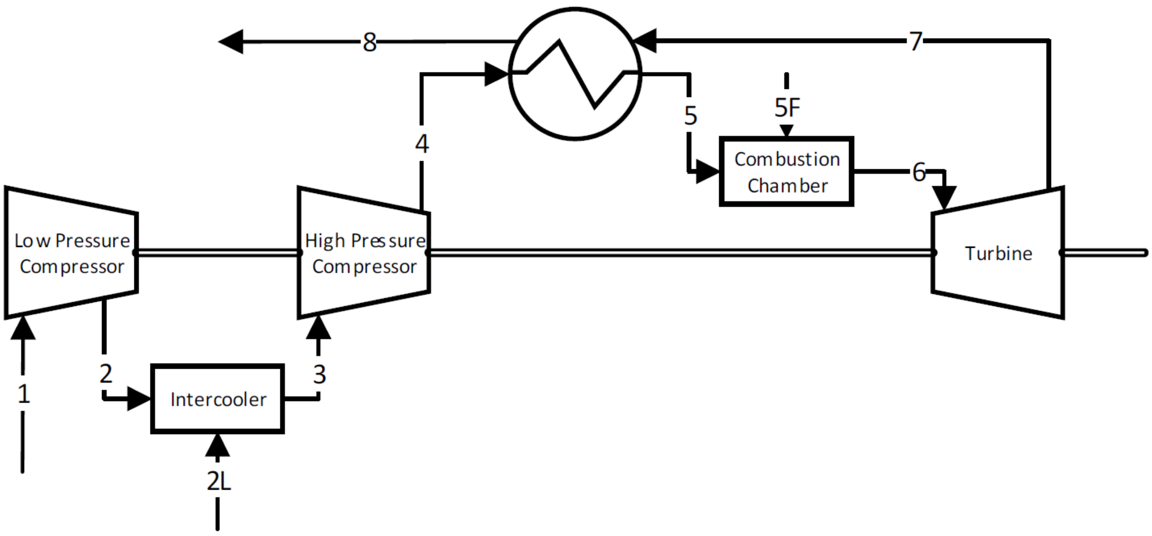

Figure 1. Schematic of the Brayton cycle with regeneration and intercooling modeled

The inputs of the calculations to reach these data have been given to us. The ambient temperature is 300 K, and the ambient pressure is 100 kPa. In each compressor stage, the pressure of the fluid increases 4 times in each compressor stage. The isentropic efficiencies of compressors are equal to 0.88. In the regenerator, the effectiveness is equal to 0.88. The temperature of the fluid leaving the combustion chamber is 1000 K and it comes out with 5% less pressure than the inlet pressure. In the turbine, 5% of the power produced is lost as heat loss, and the turbine's isentropic efficiency is 90%. Finally, the net output power produced in this cycle is equivalent to 110 MW.

In order to fill in the information in the state table and process table, some assumptions must be made. The assumptions made can be seen below.

- Each component of the cycle is analyzed as a control volume at steady state.
- The working fluid is air, and it can be assumed as an ideal gas.
- Each compressor stage is adiabatic.
- Intercooler exit temperature is the same as ambient temperature.
- Intercooling is modelled as heat loss to the surroundings which occurs at the intercooler exit temperature.
- Neglect the pressure drop in the regenerator.
- Regenerator stage is adiabatic.

- Neglect effects of potential and kinetic energies
- Combustion is modelled as heat transfer to combustion chamber which occurs at 1000 Kelvin.
- Heat loss from the turbine occurs at the turbine exit temperature.
- The heat transfer occurs at $(T_5 + T_8)/2$ between cold and hot sides through the regenerator.
- The theoretical heat exchanger from state 8 to state 1 is modelled as heat loss to the surroundings which occurs at T_1.

2. METHODOLOGY AND ANALYSIS

The following equations 1 and 2 are used to calculate enthalpy and entropy for different states.

$$h(T) = \int_{T_{ref}}^{T} c_p(T)dT + h(T_{ref}) \tag{1}$$

where h = 0 at T_{ref} = 0 Kelvin.

$$s(T,P) = \int_{T_{ref}}^{T} \frac{c_p(T)}{T} dT - R \ln \frac{P}{P_{ref}} + s(T_{ref}, P_{ref}) \tag{2}$$

where $s(T_{ref}, P_{ref})$ = 1.70203 kJ/kg.K at T_{ref} = 300 Kelvin and P_{ref} = 1 atm.

To obtain enthalpy and entropy values from equation 1 and 2, we need c_p values for every temperature within $T_{ref}$to T. It can be seen in the equation 3, which is used to obtain c_p values at every temperature.

$$\frac{\overline{c_p}}{\bar{R}} = \alpha + \beta T + \gamma T^2 + \delta T^2 + \varepsilon T^3 \tag{3}$$

Since the temperature is known in state 1, c_p values can be calculated for each temperature value using equation 3. Thus, using equations 1 and 2, enthalpy and entropy values can be reached. However, the temperature is unknown in state 2. Therefore, the values of state 2s must be calculated first. First, the entropy value of state 2s is assumed to be equal to the entropy value of state 1. Then, in equation 2, this value is written on the left side of the equation and the inverse of the integral is taken. Thus, the temperature of state 2s was determined. The enthalpy value of state 2s was found with the help of the determined temperature and equation 1. Then, the enthalpy value of state 2 was found by substituting all the obtained data in the isentropic efficiency formula. Isentropic efficiency formula for compressor 1 and turbine can be seen in equation 4.

$$\eta_c = \frac{h_{2s} - h_1}{h_2 - h_1} \tag{4}$$

$$\eta_t = \frac{h_6 - h_7}{h_6 - h_{7s}} \tag{5}$$

After the enthalpy value of State 2 is found, it is written to the left of equation 1 and the inverse of the integral is taken. Thus, the temperature of state 2 is obtained. The entropy value of state 2 was calculated using the obtained temperature and equation 2.

The same process was applied for state 3, 6 with known temperature and state 4, 5, 7, 8 with unknown temperature, and finally, temperature, enthalpy and entropy values were found for all states.

For process table, $\dot{Q}$, $\dot{W}$, $\dot{\sigma}_{gen}$ and η_{cycle} are needed for every process. Since only process 1 to 2, process 3 to 4, and process 6 to 7 have work devices, $\dot{W}$ is zero in other processes. For process 1 to 2, process 3 to 4, $\dot{W}$ is equal to the enthalpy differences of the fluid leaving the device and the fluid entering times the mass flow rate. For process 6 to 7, $\dot{W}$ is equal to the enthalpy differences of the fluid entering the device and the fluid leaving times the mass flow rate. Since the cycle is operating at steady state, mass flow rate is the same for each stage.

For process 1 to 2 and process 3 to 5, the heat transfer rate is equal to zero because they are work devices. However, in process 6 to 7 the heat transfer rate is not equal to zero despite the fact that the turbine is a work device. The reason of that there is a heat loss in the turbine. In the light of all this information, the heat transfer rate was calculated by using the enthalpy differences of the incoming and outgoing fluid times the mass flow rate .

Equation 6 is used to calculate the entropy generation rate. The inlet and exit mass flow rate are the same.

$$\dot{\sigma}_{gen} = \dot{m}_e s_e - \dot{m}_i s_i - \frac{\dot{Q}}{T} \qquad (6)$$

Using equation 6, the entropy generation rate of all process can be calculated.

Mass flow rate is calculated by dividing the given cycle work to specific cycle work. Finally, the volumetric flow rate of each stage is found from the ideal gas equation since the gas is assumed to be ideal.

After all these formulas were implemented in MATLAB and the required values were calculated, the following tables and graphics were obtained.

2.1 Base Case

In the base case, the state and process tables can be seen in tables 1 and 2.

STATE	DESCRIPTION	T (K)	P (kPa)	ṁ (kg/s)	V̇ (m^3/s)	h (kJ/kg)	s (kJ/kg.K)	k (-)
1	Low Pressure Compressor Inlet	300.00	100.00	1016.10	874.79	303.21	1.7058	1.4002
2	Intercooler Inlet	480.98	400.00	1016.10	350.63	487.98	1.7850	1.3885
3	High Pressure Compressor Inlet	300.00	400.00	1016.10	218.70	303.21	1.3080	1.4002
4	Regeneration Inlet	480.98	1600.00	1016.10	87.66	487.98	1.3879	1.3885
5	Combustor Inlet	531.33	1600.00	1016.10	96.83	539.86	1.5197	1.3835
6	Turbine Inlet	1000.00	1520.00	1016.10	191.84	1048.60	2.1902	1.3364
7	Turbine Outlet	538.15	100.00	1016.10	1,569.20	546.93	2.2994	1.3828
8	Regeneration Outlet	487.87	100.00	1016.10	1,422.60	495.06	2.2019	1.3879

Table 1. State Table for Base Case

PROCESS	DESCRIPTION	Q (kW)	W (kW)	σ̇ (kW/K)	η (-)
1→2	Low P Compressor	0.00	-187740.00	80.42	-
2→3	Intercooler	-187740.00	0.00	141.14	-
3→4	High P Compressor Inlet	0.00	-187740.00	81.20	-
4→5	Regeneration (cold)	52710.00	0.00	30.55	-
5→6	Combustor	516940.00	0.00	164.25	-
6→7	Turbine	-24274.00	485480.00	156.06	-
7→8	Regeneration (hot)	-52710.00	0.00	4.42	-
8→1	Theoretical HX	-194930.00	0.00	145.67	-
Cycle	Entire Cycle	110000.00	110000.00	803.72	0.2128

Table 2. Process Table for Base Case

2.2 Parametric Analysis 1

In the first parametric analysis, ambient temperature is changed while all other inputs are constant. The temperature changes from 255 K to 315 K with 10 K increments.

Cycle thermal efficiency, back-work ratio (bwr), specific power output and engine inlet volumetric flow rate (State 1) versus ambient temperature with and without the regenerator are plotted in figures 2, 3, 4 and 5.

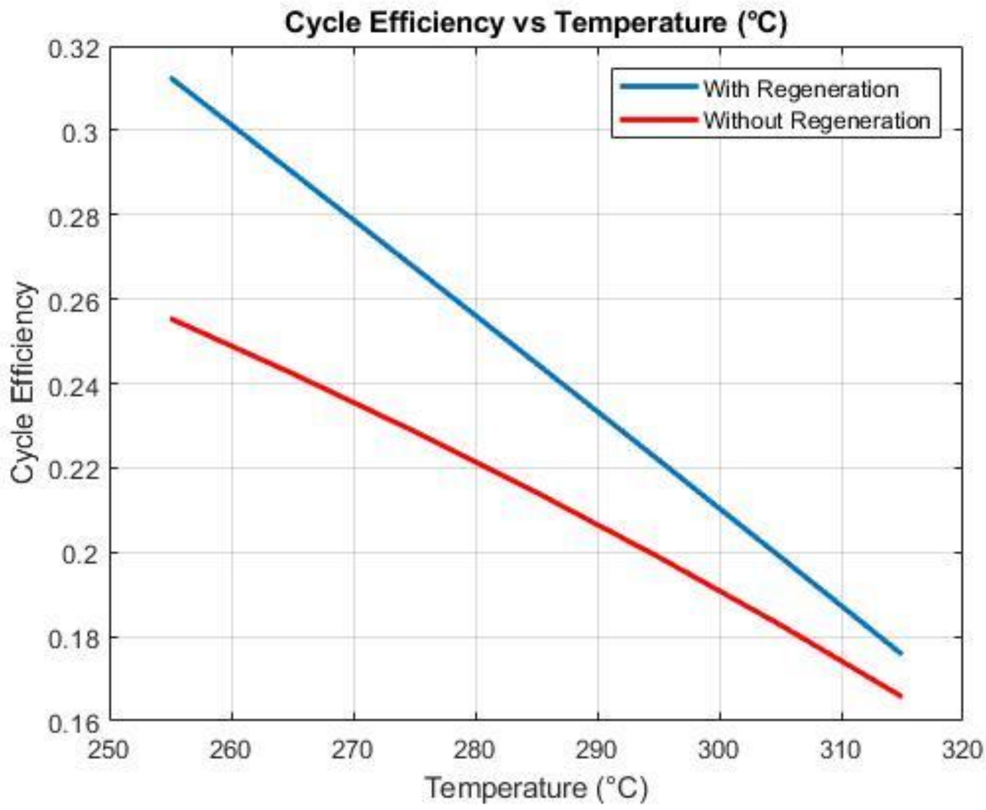

Figure 2. Cycle Efficiency vs. Temperature Plot

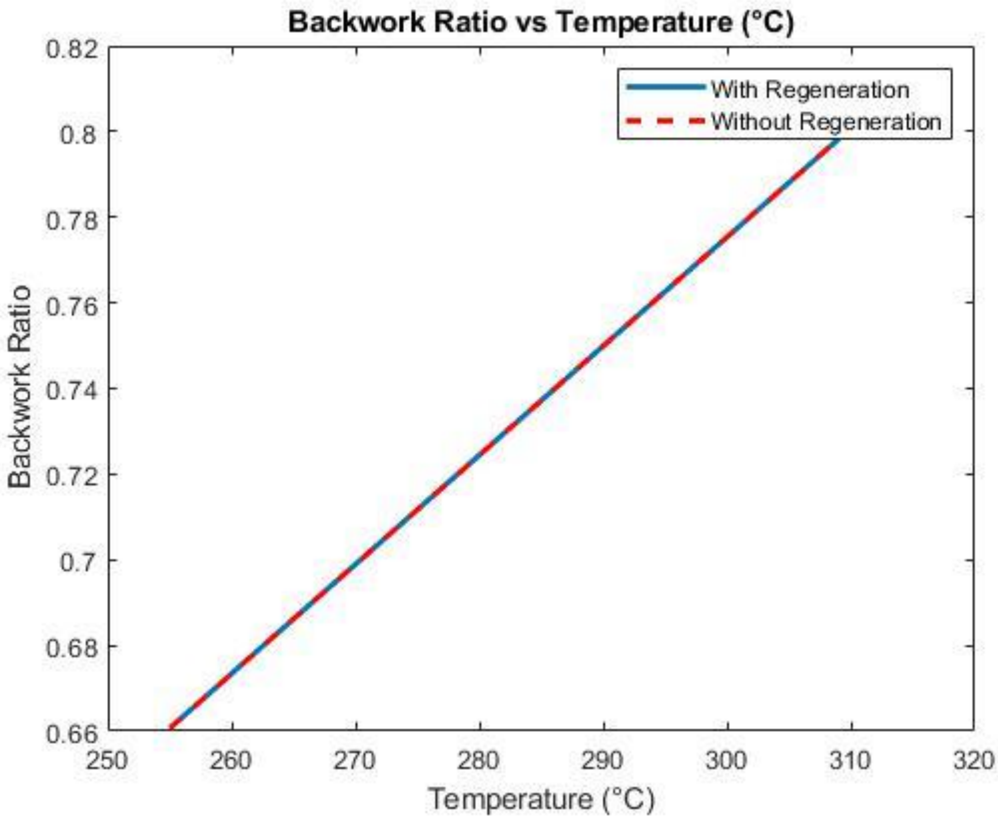

Figure 3. Backwork Ratio vs. Temperature Plot

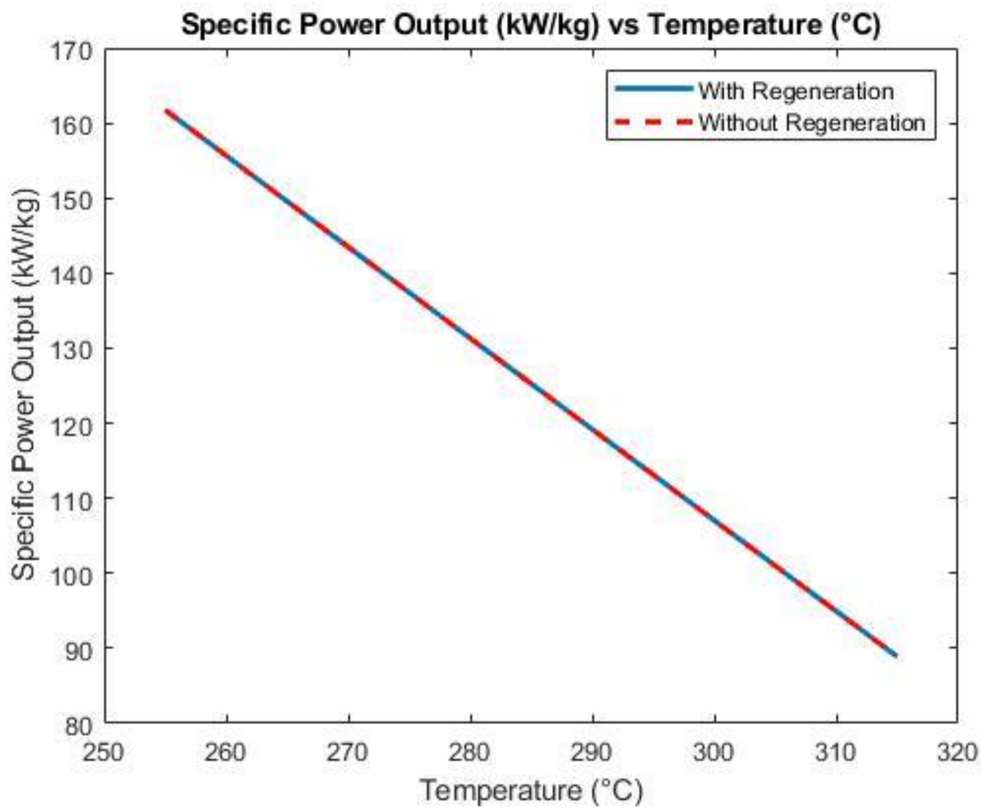

Figure 4. Specific Power Output vs. Temperature Plot

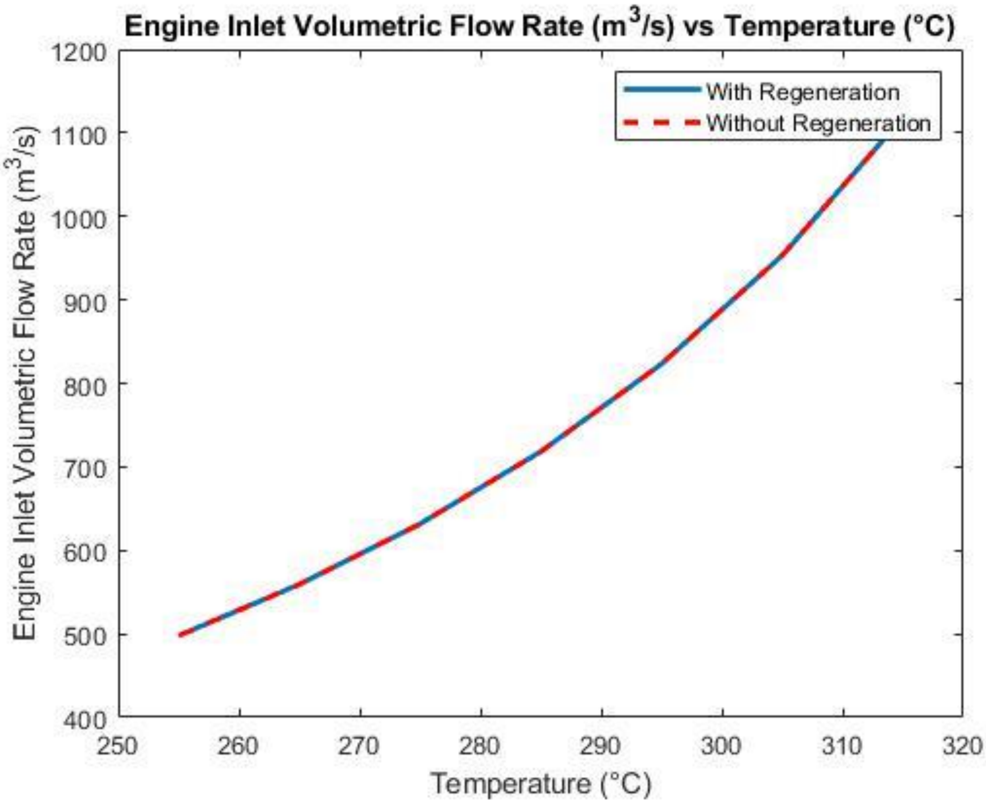

Figure 5. Engine Inlet Volumetric Flow Rate vs. Temperature Plot

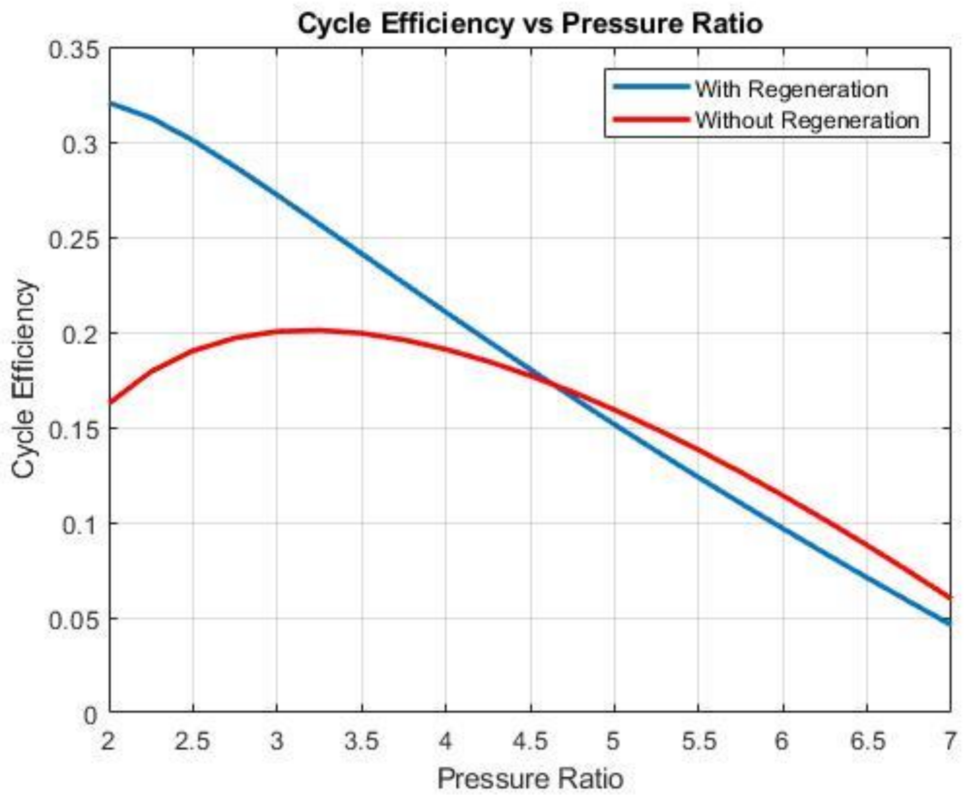

Figure 6. Cycle Efficiency vs. Pressure Ratio Plot

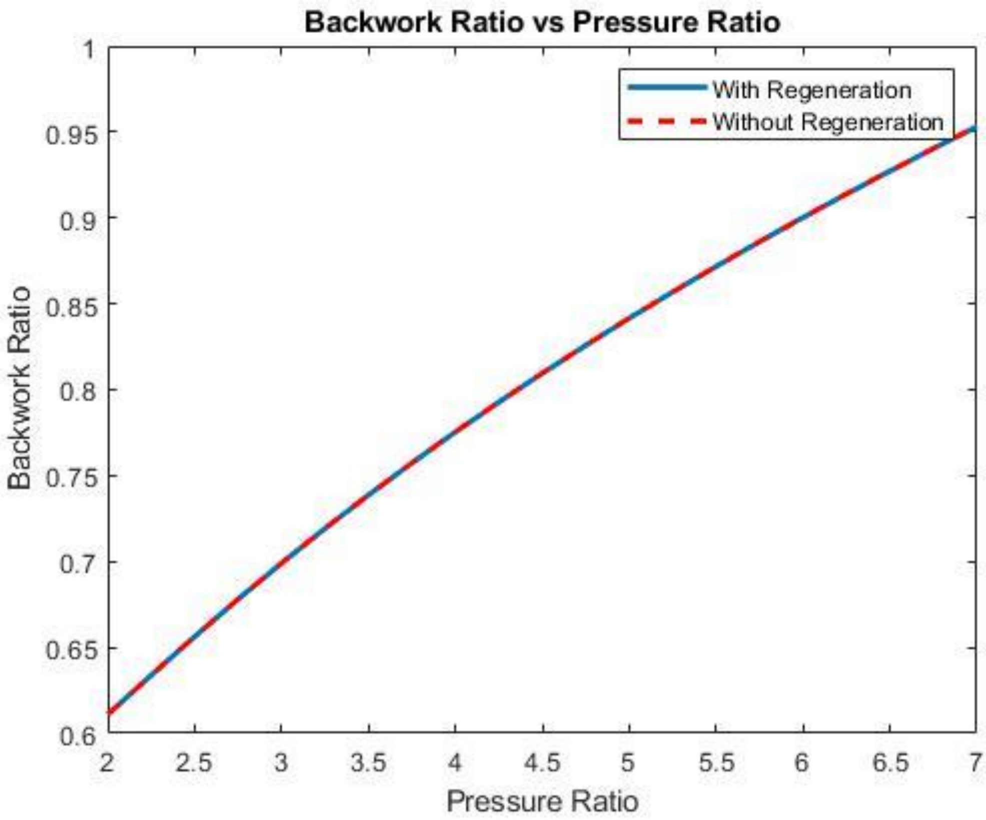

Figure 7. Backwork Ratio vs. Pressure Ratio Plot

2.3 Parametric Analysis 1

In the second parametric analysis, pressure ratio is changed while all other inputs are constant. The temperature pressure ratio changes from 2.0 to 7.0 with increments of 0.25. The reason choosing this lower and upper bounds is explained in the discussion section.

Cycle thermal efficiency, back-work ratio (bwr), specific power output and engine inlet volumetric flow rate (State 1) versus pressure ratio with and without the regenerator are plotted in figures 6, 7, 8 and 9.

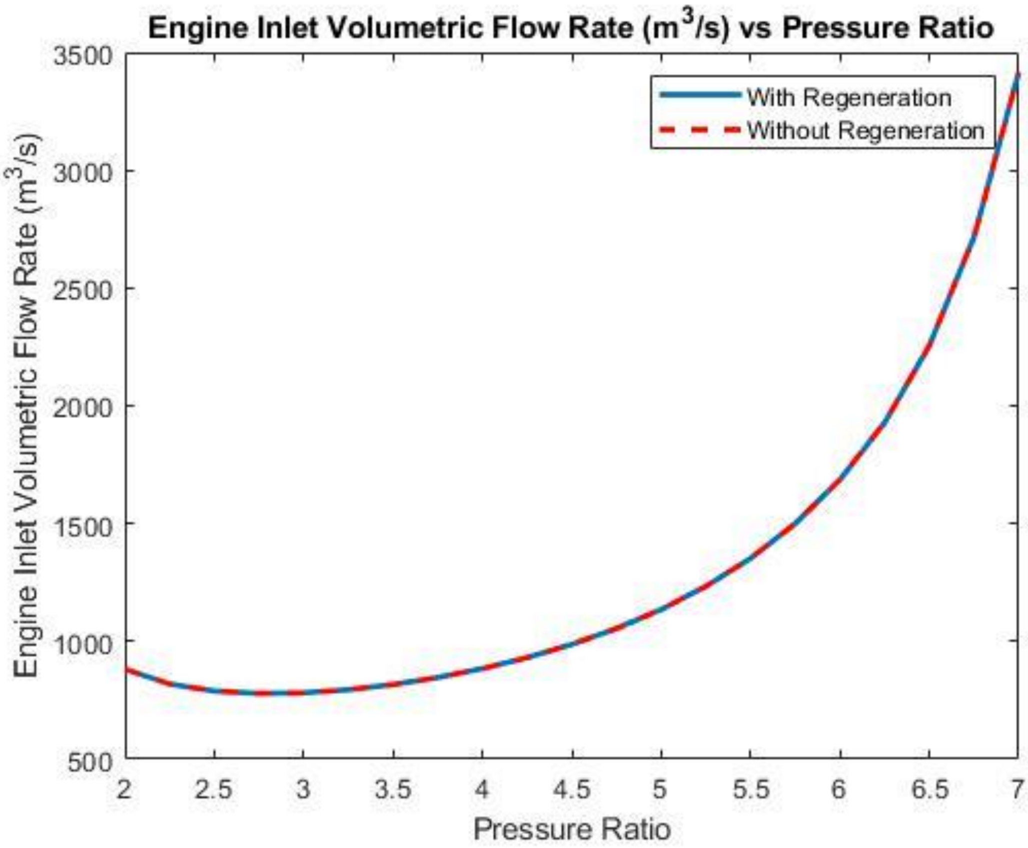

Figure 8. Engine Inlet Volumetric Flow Rate vs. Pressure Ratio Plot

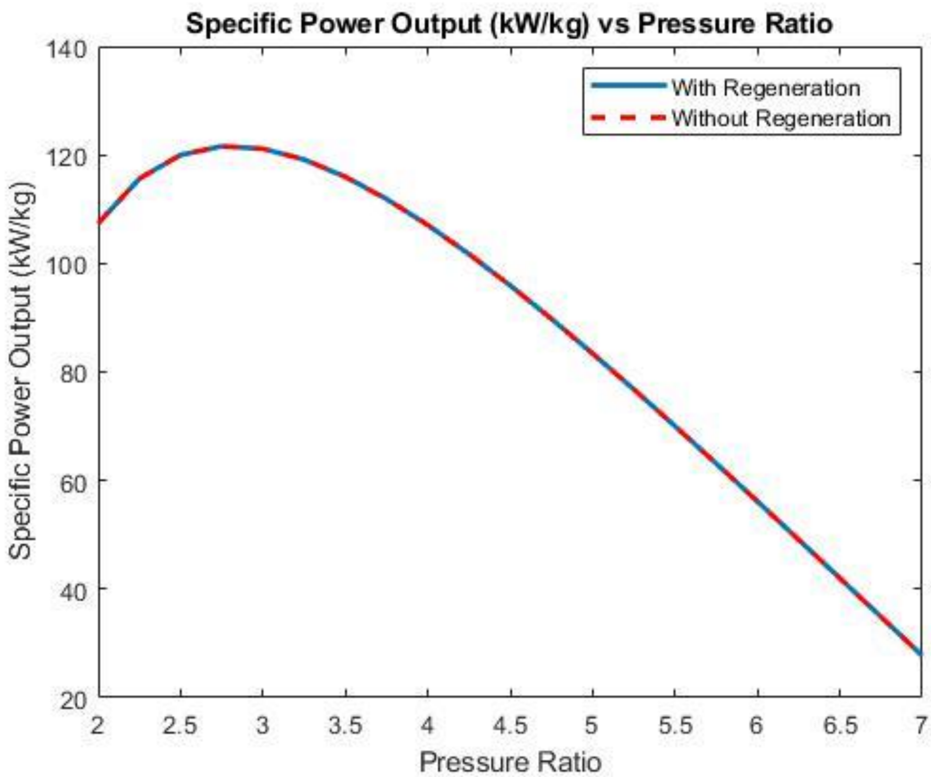

Figure 9. Specific Power Output vs. Pressure Ratio Plot

3. DISCUSSION

3.1 What is typical backwork ratio (bwr) of gas turbines? Discuss, bwr values that you obtained according to the typical range.

The energy required to run the compressor is supplied from the turbine itself. The ratio of this energy used in the compressor to the energy produced in the turbine gives the back work ratio. Since the energy required for compression is higher in gas turbines than in vapor systems, back work ratio is higher in gas turbines. This is because compressing fluids are easier than compressing vapor. Thus, more work is required. Typical back work ratio for gas power cycles is between 20% and 80%. Our back work ratio is 76,38% for the base case. For the parametric analysis, the back work ratio is increasing with the increasing ambient temperature since it is harder to compress fluid with higher temperature and compressors require more work. This can be seen from figure 3. Additionally, the back work ratio is increasing with the increasing pressure ratio as well since it requires more work to compress to higher pressure values (with higher ratios). This can be seen from figure 9. Our values agree with the back work ratio values of the typical cycles of this type.

3.2 What is the suitable range of pressure ratio of each compressor stage ($r_{p,i}$)? Explain, how the upper and lower boundaries for this range are found.

The suitable range of pressure ratio of each compressor stage ($r_{p,i}$) is between 2.0 and 7.0. Outside the limits, the results don't make sense physically. As can be seen from figure 10, when the ratio is below 2.0, cycle efficiency when there is no regeneration is larger than 1, which is physically and thermodynamically impossible. This is against the second law of thermodynamics.

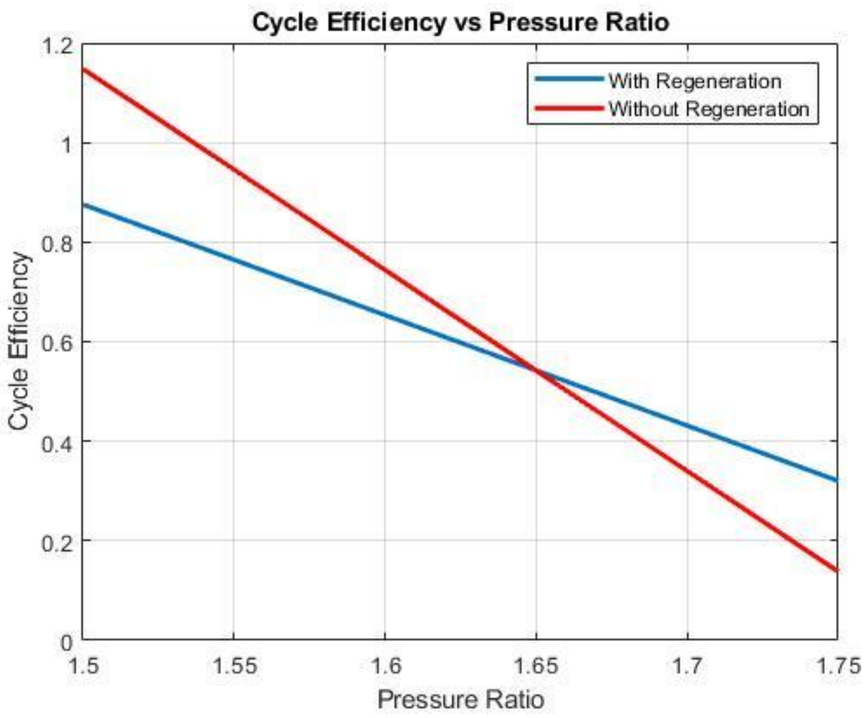

Figure 10. Cycle Efficiency vs. Pressure Ratio Plot

Additionally, when the ratio is above 7.75, the thermal efficiency is smaller than 0, the back work ratio is larger than 1, the volumetric flow rate is lower than 0, and the specific work output is also lower than 0 as seen in figures 11, 12, 13 and 14. These values also don't make sense. We found the upper range by simply changing the pressure range input and then investigated the results and eliminated the ones that didn't make sense.

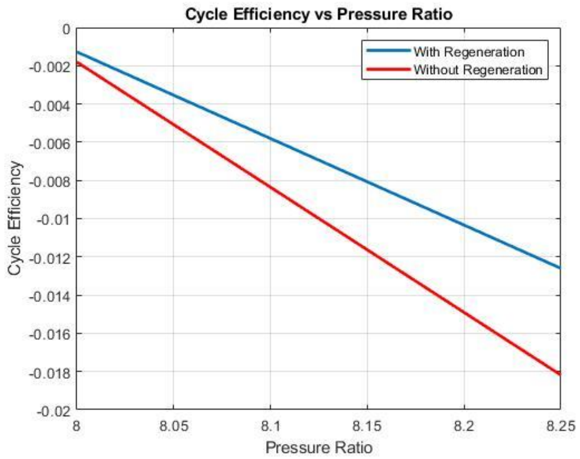

Figure 11. Cycle Efficiency vs. Pressure Ratio Plot

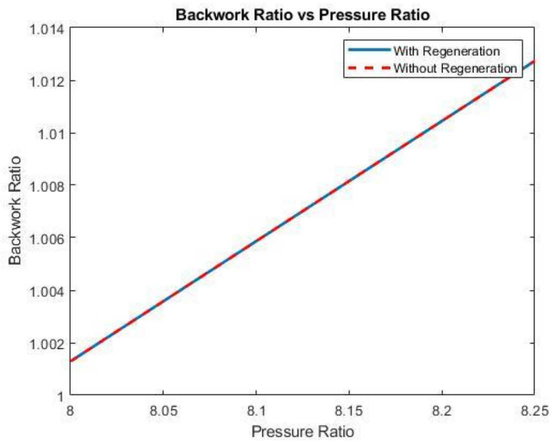

Figure 12. Back Work Ratio vs. Pressure Ratio Plot

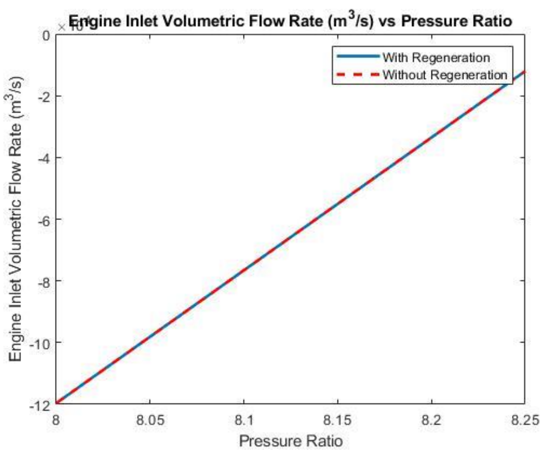

Figure 13. Engine Inlet Volumetric Flow Rate vs. Pressure Ratio Plot

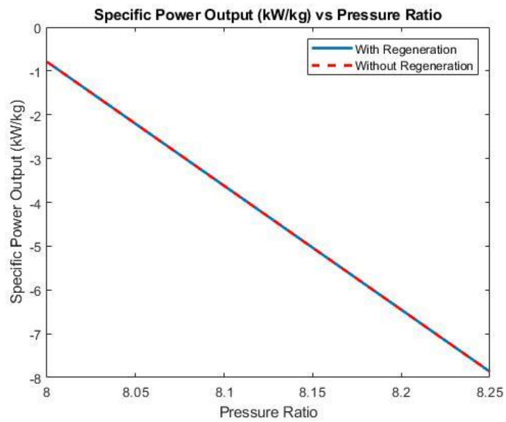

Figure 14 Specific Power Output vs. Pressure Ratio Plot

3.3 Discuss the effect of regenerator on bwr, specific power output and thermal efficiency.

A basic gas turbine's turbine exhaust temperature is typically much higher than the surrounding air's temperature. Significantly, if hot turbine exhaust gas were to be released directly into the environment, its thermodynamic utility (exergy) would be permanently lost. Utilizing a regenerator, a type of heat exchanger that allows air to be prepared before entering a combustor to reduce the amount of fuel that must be consumed in the combustor, is one approach to make use of this potential. Therefore, it is very usual for the efficiency to be excessive in the case of using the regenerator in the figure 2.

The reason why efficiency differs depending on the ambient temperature is the energy consumed in combustion and the work used in the compressor. When the ambient temperature is low, more energy must be given to the fluid in combustion to obtain the desired power output. From another point of view, when the ambient temperature is low, the energy required to compress the fluid will be less. In other words, the back work ratio will decrease because the work rate obtained from the turbine is constant. If we look at the figure 2, the effect of the temperature change of the fluid is more important in terms of compressor and can create larger changes in efficiency. Regenerator increases the efficiency of the existing system to a certain extent. It does this by using the temperature of the expelled fluid. When ambient temperature increases, the heat transfer rate will decrease as the temperature difference between the discharged fluid and the fluid coming out of the 2nd compressor will decrease. Therefore, the efficiency of the regenerator will decrease as the ambient temperature increases.

When the regenerator is not used, the value of the pressure ratio is important for the cycle efficiency. The figure 6 shows that if the pressure ratio is less than a certain value, efficiency increases with increasing pressure ratio, while if the pressure ratio exceeds a certain value, efficiency first increases and then starts to decrease with increasing pressure ratio. With further increase in pressure ratio, the increase in pressure ratio results in a decrease in efficiency. Thus, there is an optimum value for pressure ratio. When the regenerator is used, the efficiency will decrease almost linearly as the pressure ratio increases.

Specific power output is the ratio of energies produced and consumed per unit fluid by turbine and compressor. The turbine produces the same output power per unit fluid in every ambient temperature condition. However, as the ambient temperature increases, the specific power output will decrease as the compressor will consume more energy. The figure 9 shows that if the pressure ratio is less than a certain value, specific power output increases with increasing pressure ratio, while if the pressure ratio exceeds a certain value, specific power output first increases and then starts to decrease with increasing pressure ratio. With further increase in pressure ratio, the increase in pressure ratio results in a decrease in specific power output. Thus, there is an optimum value for pressure ratio.

As the pressure ratio increases, the compressors will have to compress the fluid more. Therefore, the energy consumed in compressors will increase. However, although it has little effect, increasing the pressure ratio will increase the work to be obtained in the turbine. Therefore, the back work ratio increases as the pressure ratio increases.

Since the back work ratio and specific power output values depend on the enthalpy values of the fluid entering and leaving the turbine and compressor, the regenerator does not affect these values for different ambient temperature and pressure ratios.

4. REFERENCES

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