

# **ME 466 Project 2**

by

**Tolga Çatak – 2018405144**

**Ümit Henkoğlu – 2018405099**

**Muhammed Emin Yaşar – 2019405006**



**Department of Mechanical Engineering**

**Bogazici University**

**Istanbul, Turkey**

**January 2022**

**Course Instructor:** Dr. Nazlı Dönmezer

**Course Assistants:** Eren Bekiloğlu, Ertuğrul Altun

# 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 and 1 intercooler. The goal of the project is to observe and interpret changes in the effects of liquid water injection mass flow rate at the intercooler and the fuel mass flow rate at the combustion chamber to the specific power output, energy and exergy efficiencies.

The project pays attention to the air's humidity. Additionally, the combustion is represented as a reaction using methane as the fuel, and the intercooling is modeled as liquid water injection at ambient temperature. As a result, as the working fluid flows through the cycle, its composition changes. The present project focuses on reacting species and the thermodynamics of psychrometry in addition to Project 1 in this respect. To evaluate the energetic and exergetic performance of a certain Brayton cycle with intercooling, we construct a thermodynamic model in this project.

The inputs of the base case calculations to reach these data have been given to us. The ambient temperature is 300 K, the ambient pressure is 100 kPa and the ambient relative humidity is 60%. The pressure of the fluid increases 4 times in each compressor stage and the isentropic efficiencies of compressors are equal to 0.8. The temperature of the fluid which comes to intercooler is 280 K and the relative humidity of the fluid which leaves the intercooler is 90%. The temperature of the fluid leaving the combustion chamber is 1000 K and it comes out with 5% less pressure than the inlet pressure. The temperature of the fuel goes to the combustion chamber is equal to ambient temperature. In the turbine, isentropic efficiency is 90% and the output power is 175 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.

- a) Each component of the cycle is analyzed as a control volume at steady state.
- b) The working fluid is an ideal gas mixture. Assume the composition of the mixture at the first compressor inlet as a mixture of  $N_2$ ,  $O_2$  and  $H_2O$  (no  $CO_2$  and Ar, atmospheric air includes these gaseous as shown in Table 12.1) and  $y_{N_2}/y_{O_2} = 3.76$ . Therefore, dry air properties cannot be used directly throughout the engine model especially for the psychometric analysis; instead,  $c_{p,j}$  is used according to Table A.21.
- c) Each compressor stage is adiabatic.
- d) In the intercooler, liquid water is injected at  $P_2$  and all injected water evaporates.
- e) Intercooler is modeled as adiabatic
- f) Neglect effects of potential and kinetic energies.
- g) Pure methane ( $CH_4$ ) is injected to the combustion chamber at  $P_4$ , and the reaction is complete.
- h) Combustion occurs with 100% theoretical air.
- i) Heat transfer from the combustion chamber occurs at  $T_{max}$ .
- j) Turbine stage is adiabatic.
- k) No dissociation or accumulation occurs in the combustion chamber.

## 2. Methodology and Analysis

In the calculations, the sign convention is needed. Thus,  $\dot{Q}_{in}$  and  $\dot{W}_{out}$  were considered positive,  $\dot{Q}_{out}$  and  $\dot{W}_{in}$  were considered negative.

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

$$\bar{h}(T) = \bar{h}_f^0 + \int_{T_{ref}}^T \bar{c}_p(T) dT \quad (1)$$

Where  $T_{ref} = 298 K$

$$\bar{s}(T, P) = \int_{T_{ref}}^T \frac{\bar{c}_p(T)}{T} dT - \bar{R} \ln \frac{y_i P}{P_{ref}} + \bar{s}^0(T_{ref}, P_{ref}) \quad (2)$$

Where  $T_{ref} = 298 K$  and  $P_{ref} = 1 \text{ atm}$ . For  $\bar{s}^0(T_{ref}, P_{ref})$  values, Table A.25 is used.

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

$$\bar{c}_p = \bar{R}(\alpha + \beta T + \gamma T^2 + \delta T^2 + \epsilon T^3) \quad (3)$$

Since the temperature is known in state 1,  $\bar{c}_p$  values can be calculated for each temperature value using equation 3. Thus, using equations 1 and 2, molar enthalpy and molar 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 molar entropy value of state 2s is assumed to be equal to the molar 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 molar enthalpy value of state 2s was found with the help of the determined temperature and equation 1. Then, the molar 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 and 5.

$$\eta_c = \frac{h_{2s} - h_1}{h_2 - h_1} \quad (4)$$

$$\eta_t = \frac{h_5 - h_6}{h_5 - h_{6s}} \quad (5)$$

After the molar 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 molar entropy value of state 2 was calculated using the obtained temperature and equation 2.

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

For molar flow rate, we substituted turbine output power and enthalpy values of state 6 and state 5 in the equation 6 below. Since this value will be the same as our molar flow rate in state 5, we have found state 5 as well.

$$\bar{h}_5 - \bar{h}_6 = \frac{\dot{W}_t}{\dot{n}} \quad (6)$$

Then, we found the molar flow rate in combustor fuel inlet using the mole fraction of carbon dioxide. We found the molar flow rate of the combustor air inlet from the molar flow rate difference in the Combustor Fuel Inlet and the turbine inlet. The product of the mole fraction caused by the water added in the intercooler and the molar flow rate in the high-pressure compressor inlet gives the molar flow rate in the intercooler water inlet. We can find the molar flow rate in the intercooler air inlet by taking the difference of the molar flow rate of the water entering the intercooler from the molar flow rate at the intercooler outlet. Molar flow rate of the low-pressure compressor inlet is same as our molar flow rate in intercooler air inlet.

Equation 7 which can be seen below is used to calculate each state's relative humidity value. The pressures are obtained at mixture temperature and pressure.

$$\phi = \frac{p_v}{p_g} \quad (7)$$

Equation 8 was used to reach the saturation water vapor pressure value.

$$P_{\text{sat}}(T) = \frac{\exp(34.494 - \frac{4924.99}{T+237.1})}{(T+105)^{1.57}} \quad (8)$$

Where  $P_{\text{sat}}$  is in Pascals and  $T$  is in Degree Celsius.

The relative humidity value of each state can be determined using equation 6 and 7.

Equation 9 is used to determine the molar specific exergy. Equation 9 can be seen below. In the equation 9,  $T_0$  is equal to 300 K and  $P_0$  is equal to 100 kPa, which are the ambient values. The standard molar chemical exergy values were obtained from table A-26.

$$\bar{e}_f = \bar{h} - \bar{h}_0 - T_0(\bar{s} - \bar{s}_0) + \bar{e}^{\text{ch}} \quad (9)$$

For process table,  $\dot{Q}$ ,  $\dot{W}$ ,  $\dot{\sigma}_{\text{gen}}$ ,  $\Delta\dot{E}_f$ ,  $\dot{E}_D$ ,  $\epsilon$  are needed for every process. Since only process 1 to 2, process 3 to 4, and process 5 to 6 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 molar enthalpy differences of the fluid leaving the device and the fluid entering times the molar rate. For process 5 to 6,  $\dot{W}$  is equal to the molar enthalpy differences of the fluid entering the device and the fluid leaving times the molar rate. Since the cycle is operating at steady state, molar flow rate is the same for each stage.

For process 1 to 2, process 3 to 4 and process 5 to 6, the heat transfer rate is equal to zero because they are work devices. In the light of all this information, the heat transfer rate was calculated by using the molar enthalpy differences of the incoming and outgoing fluid times the molar flow rate.

Equation 10 is used to calculate the entropy generation rate. Using equation 6, the entropy generation rate of all process can be calculated.

$$\dot{\sigma}_{\text{gen}} = \dot{m}_e s_e - \dot{m}_i s_i - \frac{\dot{Q}}{T} \quad (10)$$

For process all processes, the flow exergy rate was calculated by using the molar exergy differences of the incoming and outgoing fluid times the molar flow rate thanks to equation 8.

To calculate  $\dot{E}_d$  for every state, equation 11 was used. Equation 10 can be seen below.

$$\frac{dE}{dt} = \left(1 - \frac{T_0}{T}\right) \dot{Q} - \dot{W}_{\text{cv}} + \dot{m}_i e_{\text{fi}} - \dot{m}_e e_{\text{fe}} - \dot{E}_d \quad (11)$$

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 table 1 and table 2 below.

STATE	DESCRIPTION	T (K)	P (kPa)	$\dot{n}$ (kmol/s)	$\phi$ (%)	$\bar{h}$ (kJ/kmol)	$\bar{s}$ (kJ/kmol·K)	$\bar{e}_f$ (kJ/kmol)
1	Low Pressure Compressor Inlet	300,00	100,0	9,850	60,0000	-5073,1	199,57	0,0000
2	Intercooler Air Inlet	479,97	400,0	9,850	0,4807	219,8	201,90	4612,4000
2L	Intercooler Water Inlet	280,00	400,0	0,784	0,0000	-287190,0	53,84	4410,2000
3	High Pressure Compressor Inlet	350,30	400,0	10,634	90,0000	-21412,0	193,66	3570,4000
4	Combustor Air Inlet	557,46	1600,0	10,634	2,3442	-15255,0	196,14	9079,3000
4F	Combustor Fuel Inlet	300,00	1600,0	1,011	0,0000	-74779,0	163,46	838570,0000
5	Turbine Inlet	1000,00	1520,0	11,645	0,2631	-73440,0	217,30	18429,0000
6	Turbine Outlet	577,25	100,0	11,645	0,3074	-88468,0	220,83	2603,6000

Table 1. State Table for Base Case

PROCESS	DESCRIPTION	$\dot{Q}$ (kW)	$\dot{W}$ (kW)	$\dot{\sigma}$ (kW/K)	$\Delta\bar{E}_f$ (kW)	$\dot{E}_D$ (kW)	$\epsilon$ (-)
1→2	Low P Compressor	0,0	-52135,0	23,039	45432,0	6702,8	0,8000
2+2L→3	Intercooler	0,0	0,0	28,450	-10923,0	10923,0	-
3→4	High P Compressor Inlet	0,0	-65473,0	26,336	58582,0	6891,1	0,8947
4+4F→5	Combustor	-617390,0	0,0	896,870	-729690,0	773790,0	0,2273
5→6	Turbine	0,0	175000,0	41,065	-184290,0	9287,7	0,9496
Cycle	Entire Cycle	-617390,0	57392,0	1015,800	-820890,0	807600,0	0,0677

Table 2. Process Table for Base Case

## 2.2 Parametric Analysis 1

In the first parametric analysis, intercooler outlet relative humidity ratio is changed while all other inputs are constant. This value changes from 0.05 to 1 with 0.05 increments. Cycle 2<sup>nd</sup> law thermal efficiency and back-work ratio (bwr) versus water mass flow rate are plotted in figure 1.

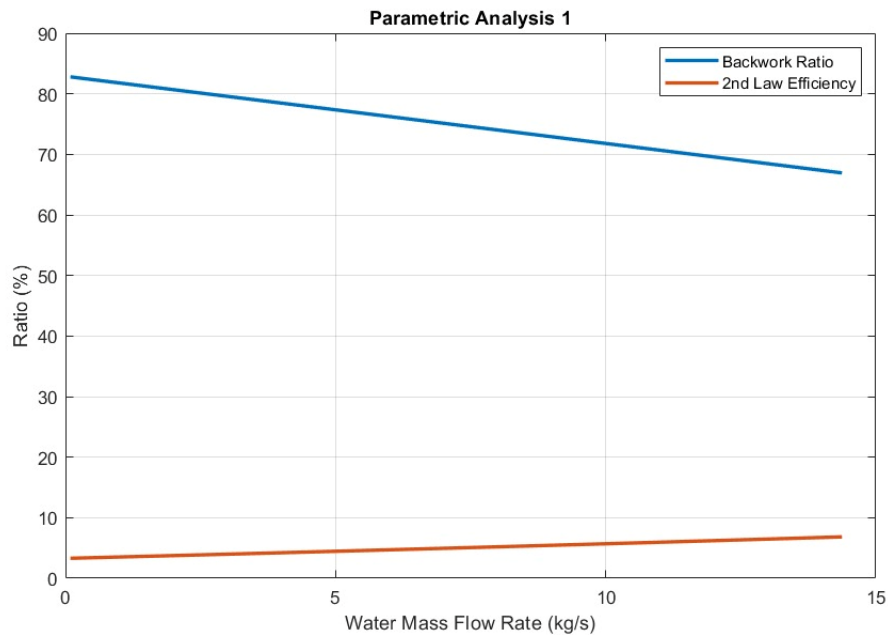


Figure 1. 2nd Law Cycle Efficiency and Back-work Ratio vs. Water Mass Flow Rate

## 2.3 Parametric Analysis 2

In the second parametric analysis, maximum cycle temperature is changed while all other inputs are constant. This value changes from 700 K to 1650 K with increments of 50 K. The reason for choosing this lower and upper bounds is explained in the discussion section. 2<sup>nd</sup> law cycle thermal efficiency and back-work ratio (bwr) versus fuel mass flow rate are plotted in figure 2.

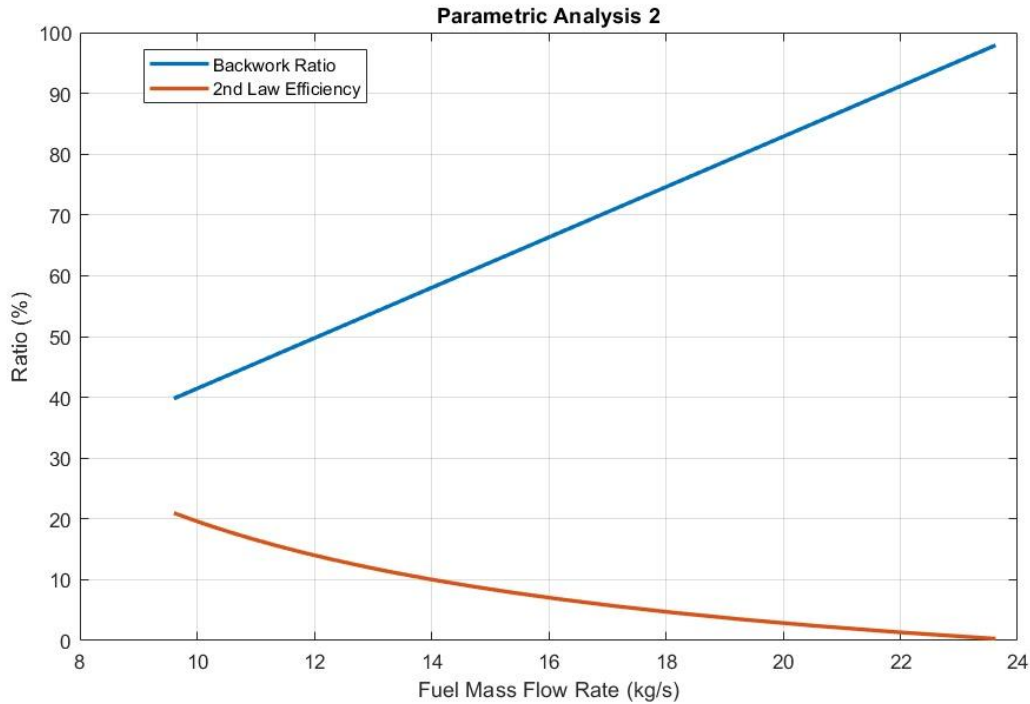


Figure 2. 2nd Law Cycle Efficiency and Back-work Ratio vs. Fuel Mass Flow Rate

## 3. Discussion

### 3.1 Explain how you set the boundaries of relative humidity and maximum cycle temperature for Parametric Analyses. What parameters did you check?

The maximum and minimum values that the relative humidity can take are 100% and 0%, respectively. In figure 1, according to these two limits, the back-work ratio was obtained in the range where the relative humidity is 0.05 and 1. When figure 1 is examined, it is seen that when the relative humidity increases, the water flow rate increases while the back-work ratio decreases. Back-work ratio is calculated by the ratio of compressor work to turbine work. In this case, the decrease in the back-work ratio increases the efficiency of the system because compressor demands less work from the work produced in the turbine.

The reason why the increase in the relative humidity increases the back-work ratio is the decrease in the temperature of the fluid. In order for the relative humidity to increase, the amount of water used in the intercooler must increase. In other words, the mass flow rate entering the intercooler should increase. If the relative humidity of the fluid coming out of the intercooler is higher than in other cases, the air is cooled more in the intercooler than in other cases. Since it will be easier for the compressor to compress the fluid whose energy decreases as it cools, the work consumed in the compressor will decrease. Therefore, the back-work ratio will increase and the efficiency of the system will increase.

In the figure 2, the back work ratio is plotted according to the 700 K and 1650 K limit values. The far left of the graph shows the maximum temperature of 1650 K, and the rightmost shows the minimum temperature of 700 K. According to the graph, as the temperature increases, the back work ratio decreases, the fuel mass flow rate decreases, and the efficiency increases. The work taken from the turbine is constant. Therefore, the hotter the fluid entering the turbine, the less mass flow rate of that fluid.

If we examine the graph, a mass flow rate below 10 kg/s is required for 1650 K. In this case, because the compressor will compress less mixture, the work it will demand will decrease. Thus, the back work ratio will decrease, and productivity will increase. For 700 K, a mass flow rate above 23 kg/s is required. In this case, the work demanded by the compressor will increase as it will compress more mixture. Thus, the back work ratio will increase, and productivity will decrease.

If the temperature of the fluid coming out of the combustion is too low, the back-work ratio will approach 100%. In this case, it will be impossible to obtain work from the system because the compressor takes the work from the turbine. In another case, the temperature of the fluid leaving the combustion could theoretically be equal to the maximum adiabatic flame temperature. According to our calculations, adiabatic flame temperature is equal to 2638 K. However, this temperature is not taken as a limit in the model because it is very difficult to reach it practically. The reason for this is mechanical strength limitations such as creep of turbine blades at these temperatures and not having enough strength.

### **3.2 Explain how you calculated the exergetic efficiencies of the compressors, the combustor, the turbine and the cycle?**

The irreversibility that generates entropy causes energy systems to lose some of their thermodynamic efficiency in the process. Exergetic models, as opposed to energy models based on the first law of thermodynamics, provide a more logical framework for the analysis and optimization of energy systems. Additionally, exergetic efficiency that is also known as second law efficiency in the literature is a more logical foundation for any assessment and optimization of energy systems than thermal efficiency when it comes to the defining of an energy system's efficiency. Exergy efficiency should be employed for thermal optimization of energy systems and irreversibilities (exergy destruction) within the energy system must be minimized.

In the exergetic efficiency of compressor calculations, equation 11 was used. The new form of equation 11 with assuming steady state, and single flow can be seen in equation 12 below.

$$\dot{W}_{cv} = \dot{m}(e_{fi} - e_{fe}) - \dot{E}_d \quad (12)$$

In the compressor, exergetic efficiency can be described as want overs pay. Thus, it can be seen in equation 13 below.

$$\epsilon_{comp} = \frac{\dot{n}(\bar{e}_{fe} - \bar{e}_{fi})}{-\dot{W}_{cv}} \quad (13)$$

In the exergetic efficiency of turbine calculations, equation 11 was used. The new form of equation 10 with assuming steady state, and single flow can be seen in equation 14 below.

$$\dot{W}_{cv} + \dot{E}_d = \dot{m}(e_{fi} - e_{fe}) \quad (14)$$

In the turbine, exergetic efficiency can be described as want overs pay. Thus, it can be seen in equation 15 below.

$$\epsilon_{turb} = \frac{\dot{W}_{cv}}{\dot{n}(\bar{e}_{fi} - \bar{e}_{fe})} \quad (15)$$

In the combustion, exergetic efficiency can be calculated with using equation 16.

$$\epsilon_{comb} = \frac{\dot{n}\bar{e}_{fe}}{\dot{n}\bar{e}_{fi}} \quad (16)$$

Considering the whole cycle, it can be seen that when the cycle is fueled, it is taken back as turbine work. Thus, the exergetic efficiency of cycle can be defined as turbine work rate over fuel exergy which can be seen in equation 17.

$$\epsilon_{cycle} = \frac{\dot{W}}{\dot{n}\bar{e}_{fuel}} \quad (17)$$

### **3.3 If you were to calculate the thermal efficiency of the system, how would you define the thermal efficiency?**

In Brayton Cycle, thermal efficiency is defined as net work divided by heat added to the system. However, there is no heat added to the system in this cycle. Although no direct heat enters the system, the heat created by the reaction can be considered. For this reason, the net work rate divided by the difference in the enthalpy values of the exiting products and the entering mixture, and the fuel can be calculated as the thermal efficiency in the system. The equation for thermal efficiency of this system can be seen in equation 18.

$$\epsilon_{cycle} = \frac{\dot{W}}{\dot{n}\bar{h}_{products} - \dot{n}\bar{h}_{mixture} - \dot{n}\bar{h}_{fuel}} \quad (18)$$



## References

- [1] H. Sayyadi, Modelling, Assessment, and Optimization of Energy Systems, Iran: Academic Press is an imprint of Elsevier, 2021.
- [2] S. B. B. Moran, Principles of Engineering Thermodynamics, John Wiley & Sons Singapore Pte. Ltd., 2012.
- [3] Y. Demirel, Nonequilibrium Thermodynamics, Lincoln: Elsevier, 2014.
- [4] C. D. H. M. Abdallah Haouam, «Thermal performance of a gas turbine based on an exergy analysis,» %1 içinde *E3S Web of Conferences* , 2019.
- [5] «Exergy Analysis of the Organic Rankine Cycle Based on the Pinch Point Temperature Difference,» *Journal of Thermal Engineering*, pp. 157-165, April 2019.