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# Efficiency Assessment of a Combined Heat and Power Plant Using Exergy Analysis

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**Abstract**

This study conducted an exergy analysis of a cogeneration power plant utilizing gas turbines, air compressors, combustion chambers, heat recovery steam generators, heat exchangers, and pumps. The study performed an extensive exergy analysis of the system, focusing on each component's process and calculating its base efficiency while tabulating the corresponding exergy degradation. Relevant equations for mass, energy, and exergy were identified to determine optimal control volume conditions for an optimal system and boundary conditions that would enhance the design and reduce exergy destruction. The research project developed revisions and modifications necessary to the base system, utilizing available parameters and boundary conditions, to enable a second law analysis, improve the overall efficiency, and reduce irreversibility and the loss of exergy. The proposed modifications included the remodelling of the cogeneration plant by applying additional processes to utilize the excessive waste heat in the plant. The study further optimized the plant's efficiency by modifying individual system elements that yielded minimal exergy destruction to the overall design. The proposed modifications explored the best-case alteration on optimizing overall plant efficiency with minimum irreversibility compared to the initial analysis done. The technical contributions of this research project are the revisions and modifications that enabled a second law analysis and improved the overall efficiency of the cogeneration power plant.

**Keywords:** exergy destruction, cogeneration power plant, plant efficiency, second law analysis, optimization

List of Abbreviations and Symbols

| | |
|---|---|
| $\dot{m}_x$ | *Mass flow rate at x state, kg/s* |
| $E_{CV}$ | *Energy of the control volume, kJ* |
| $\dot{Q}_{CV}$ | *Heat transfer rate from/in the control volume, kW* |
| $V_i$ | *Velocity at inlet, m/s* |
| $V_e$ | *Velocity at exit, m/s* |
| $z_i$ | *Elevation of inlet from the reference line, m* |
| $z_e$ | *Elevation of exit from the reference line, m* |
| $X_x$ | *Exergy at the x state, kJ* |
| $\dot{S}_{gen}$ | *Rate of entropy generation, kW/K* |
| $r_P$ | *Pressure ratio* |
| $P_x$ | *Pressure at x state, kPa* |
| $T_x$ | *Temperature at x state, K* |
| $\eta_c$ | *Isentropic efficiency of compressor* |
| $\dot{W}_C$ | *Compressor power input, kW* |
| $N_{product}$ | *Mole fraction of products, kmol* |

| | |
|---|---|
| $\overline{h}_f^0$ | *Enthalpy of formation at reference state, kJ/kmol* |
| $\overline{h}$ | *Sensible enthalpy at the specified state, kJ/kmol* |
| $\overline{h}^0$ | *sensible enthalpy at the standard reference state of 25 $^0$C and 1 atm, kJ/kmol* |
| $N_{reactant}$ | *Mole fraction of reactants, kmol* |
| $\dot{W}_T$ | *Turbine power output, kW* |
| $\eta_T$ | *Isentropic efficiency of turbine* |
| $h_x$ | *Specific enthalpy at x state, kJ/kg* |
| $\dot{X}_{D,C}$ | *Exergy destruction at compressor, kW* |
| $\psi_x$ | *Specific flow exergy at x state, kJ/kg* |
| $s_x^0$ | *Specific entropy at x state (absolute zero as the reference temperature), kJ/kg-k* |
| $s_x$ | *Specific entropy at x state, kJ/kg-k* |
| $R$ | *Gas constant, kJ/kg-K* |
| $\eta_{II,C}$ | *Second law efficiency of compressor* |
| $\dot{X}_{D,T}$ | *Exergy destruction at turbine, kW* |
| $\eta_{II,T}$ | *Second law efficiency of turbine* |
| $\dot{X}_{D,HW}$ | *Exergy destruction at hot water HE, kW* |
| $\eta_{II,HW}$ | *Second law efficiency of hot water HE* |
| $\dot{X}_{D,SG}$ | *Exergy destruction at HRSG, kW* |
| $\eta_{II,SG}$ | *Second law efficiency of HRSG* |
| $T_0$ | *Dead state temperature, K* |
| $P_0$ | *Dead state pressure, kPa* |

**Executive Summary**

This research work presents a comprehensive exergy analysis of a cogeneration power plant, utilizing Aspen Hysys and manual calculations using MS Excel for the design and optimization, with newly developed process flow systems with critical revisions and modifications. The system comprises gas turbines, air compressors, combustion chambers, heat recovery steam generators, heat exchangers, and pumps. We focused on each component's process and calculated its base efficiency while tabulating the corresponding exergy degradation, as exergy is a more comprehensive measure of a system's thermodynamic performance than energy. Our proposed modifications aimed to reduce exergy destruction, which is a measure of the irreversibility in a system and technically, we also utilized APEN Hysys, a widely-used simulation software for process design and optimization, to model the Brayton cycles of the cogeneration power plant components. We performed advanced exergy analysis, such as calculating component-level and system-level exergy efficiencies, irreversibilities, and losses. The technical contributions of this research work are the revisions and modifications that enable a second law analysis and improve the overall efficiency of the cogeneration power plant. We improved the plant's efficiency by modifying individual system elements that yielded minimal exergy destruction to the overall design. Our results show that the viable modifications made, lead to a significant increase in the cogeneration power plant's efficiency. Summarily, this research work presents a technically sound and rigorous exergy analysis of a cogeneration power plant, utilizing Aspen Hysys for the re-design of the process. The project's outcomes provided valuable insights into the importance of exergy analysis and optimization in cogeneration power plants.

## 1. Introduction

### *1.1 Background of Study*

According to research studies, a cogeneration plant can be defined as a plant where both electricity and heat energy are utilized simultaneously (Ozkan, et al, 2012). The study also opined that cogeneration system reduces the financial requirements of energy in industries, and technically, the research work by (Ozkan, et al, 2012) stated that the most viable way to design a plant is the establishment of a cogeneration system in a way that accurately meets all the heat energy requirements which is basically more electricity power being generated more than required by the industry. Furthermore, (Ozkan, et al, 2012) emphasized the numerous benefits of utilizing cogeneration systems, and one spectacular advantage is the fact that there is no loss of transfer and by regaining the heat energy, the cost of energy is kept at a minimum because of the usage of electricity where it is actually generated. Although several benefits abide in cogeneration systems they are also not exempt from process constraints, which can be resolved through the application of thermodynamic analysis, specifically the second law analysis (Ozkan et al., 2012).

Based on the afore mentioned as regarding solving the constraints in cogeneration plant through the utilization of a thermodynamic approach, Huang, et al. carried out an exergy analysis on a cogeneration system with a steam-injected gas turbine. The research focused on determining the exergy loss and where the highest exergy loss occurred in the chamber. The approach involved taking the compressor pressure ratio, ratio of the vapour injected, temperature of the vapour, and amount of the feed water as parameters, and calculating the heat–power ratio (Huang, Hung et al., 2000). In line with this, the technical gaps and limitations outlined by (Huang, et al., 2000) were satisfactorily captured and resolved by (Bandayapadhyay, et al. 2001) where he determined the optimal design and operational requirements of a cogeneration plant by considering the heat conveyance, flow directions, and laws on heat transfer, which increased the productivity of the plant system.

As regarding exergy economic analysis, (Huang, et al.2000) further performed an exergy economic analysis of a 1000-kW gas turbine cogeneration facility. This was carried out by calculating the exergy costs for a unit product. Notably, the variables taken into consideration for the optimization in the research by (Bandayapadhyay, et al. 2001) includes the thermodynamic parameters to be determined for designing the Heat Recovery Steam Generator (HRSG). As part of quality research progression in exergy analysis, (Silveria & Tuna, 2003) carried out a research analysis that focused on the second law of thermodynamics where they tried to reduce the exergy production costs.

The developed model was initially applied to a simple rankine cycle and then to a cogeneration system with regenerator gas turbine (Silveira et al., 2003). A similar exergy analysis approach was also carried out by (Temir & Bilge, 2004), and in the research, energy balance equations were applied to each component through the second law of thermodynamics and exergy loss was thereafter calculated. In this approach by (Temir et. al., 2004), the components of the system studied included the evaporative cooler, compressor, combustion chamber, gas turbine, heat exchanger and steam boiler. Specifically, the research results in this exergy analysis showed that when the percentage of exergy destroyed by component is taken into consideration, 39.30% of loss of exergy was observed in the heat exchanger, 37.75% in the combustion chamber, 16.52% in the steam boiler, 4.64% in the gas turbine and 1.80% in the compressor, but specific research gaps were also observed in this research work (Temir et. al., 2004).

Additionally, in field applications, exergy analysis of a cogeneration plant occurs in several industries, and therefore there are several designs taken into consideration to achieve an optimal process flow and exergy analysis. In view of this, (Kamate & Gangavati, 2009) designed a steam turbine cogeneration plant where the cogeneration plant generates the required steam for the process heating, which serves the purpose of process heating because it is a heat-matched plant, and the surplus is then saved. It is critical to note that the research opined that power generation is a by-product, and based on this, the process needs saturated steam at 2.5 bar and $120^0$C, with the exhaust steam drawn at $10^o$C. This peculiar cogeneration design is developed in such a way that the surplus steam that is left over after meeting the minimum process steam demand of the plant is passed through the condenser to produce potential surplus power.

Furthermore, the research carried out by (Kamate et. al., 2009) specifically showed that a thermodynamically more accurate evaluation of a cogeneration plant can be evaluated based on exergetic efficiency. However, it is seen from the results that, there is substantial improvement in both energy and exergy efficiency of the plant carried out by (Kamate et al., 2009) and this is with increase in steam inlet pressure and temperature in both the systems. The highest energy and exergetic efficiency are 0.93 and 0.344%, respectively, at 110 bar and $545^o$C steam inlet conditions. Therefore, it is seen from the results that the improvements in performance values of plant at steam inlet conditions above 61 bar and $475^0$C are marginal in the cogeneration configuration chosen in the research, and this leaves a room for further research and analysis for a more optimal improvement.

In continuation, (Wang, Dai et al., 2009) stated that cogeneration power plant can recover the waste heats to generate electrical energy with no additional fuel consumption and thus reduce the high cost of electrical energy and $CO_2$ emissions. However, he clearly indicated that exergy analysis usually aims to determine the maximum performance of the system and also identify the equipment in which exergy loss occurs, with indications of the prospects of thermodynamic enhancement of the process under discussion, of which in this case, it is a cogeneration system. The output results from the simulation carried out by Wang, et al. vividly showed that the exergy analysis was performed to evaluate the exergy losses in the cogeneration system and it was observed that 57.9% of the total input exergy is lost: 28.1% is due to the irreversibilities in the components 3.7% to the environment in the boiler exhaust, and 26.1% in the additional boiler exhaust (Wang et al., 2009).

Although, the result and analysis proved that the biggest exergy loss due to the irreversibilities occurs in the turbine expansion process, and the condensation process causes the next largest exergy loss, it was specified that the study only conducted parametric optimization only from the viewpoint of thermodynamics, and did not consider required exergy analysis and cost under the condition of the optimum performance for cogeneration system (Wang et al., 2009). Additionally, in the research work carried out by (Ghosh, Chatterjee Paul et al., 2014), the research focus was on exergy analysis of a “conceptualized combined cogeneration plant” which employs pressurized oxygen blown coal gasifier and high-temperature, high-pressure solid oxide fuel cell (SOFC) in the topping cycle and a bottoming steam cogeneration cycle. The models of the individual components were integrated appropriately to develop the model of the combined cogeneration plant and in the research work, and the simulated performance of the plant and its components were studied by varying the “selected design” and “operating parameters” (Ghosh et al., 2014). It is remarkable to note that in the research analysis by (Ghosh et al., 2014), the exergy loss due to exhaust gas to the atmosphere through stack was quite small for the type of cogeneration plant designed and this is because most of the waste heat of exhaust gas were been utilized for either steam power or useful heating. (Geurturk & Ozop, 2014) also carried out an exergy analysis of a cogeneration plant in their study, but results indicated that the average exergy efficiencies of the boilers was considered to be 43.58%. but when the exergy efficiency of the circulation fluidized bed boilers was compared with similar boilers, it can be said that exergy efficiency of the system is low, which gives room for further simulation research (Geurturk et al., 2014).

Furthermore, in the research carried out by (Yoru, Karakoc et al.,2010), it was observed that cogeneration systems, which include gas turbines, spray dryers and exchangers, can be easily analyzed using the hourly periodic data of the system and as expected, exergy analysis is highly affected from the environment temperature (Yoru et al., 2010). Hence, based on the extensive and elaborate research analysis carried out, the system selected for advancement and re-modification in this project is the cogeneration power plant, as described by (Ozkan, et al., 2012) which employs gas turbines, air compressors, combustion chambers, heat recovery steam generators (HRSG), heat exchangers, and pumps (Ozkan et al., 2012). The research article describes its comprehensive cogeneration system, which is illustrated in Fig. 1 and Fig. 2. In this type of cogeneration system, a dual gas turbine plant primarily produces electricity while the exhaust gas was utilized for water heating and steam generation for a beer plant. In the research article, exergy analysis was conducted using second law methods and the relevant results obtained showed the total exergy degradation in (kW) and this enabled the identification of which design component has the most exergy loss within the system. However, the analysis only partially assessed the plant’s exergy degradation thus leaving a research gap of significantly improving the irreversibility found within the cogeneration plant.

## 2. Objective of Study

The examination and exergy analysis of a cogeneration plant falls under the research purview of this research work. Additionally, the investigation of the exergy destruction of a cogeneration plant is the main objective of this research and to achieve this objective of analyzing the exergy of a cogeneration power plant, all the individual equipment in the process design are being taken into consideration for analysis. Furthermore, this research project will utilize the “ASPEN” simulation software and “Python for process design and the calculation of the exergy analysis. The approach involves the design, simulation, and analysis of the exergy behaviour in a re-designed cogeneration plant by optimizing process operating conditions in the modified design. Since exergy is defined as maximum amount of work which can be produced by a system when it comes to equilibrium with a reference environment, this project would focus on thermodynamic analysis through the second law analysis of the cogeneration system.

Furthermore, this research work also aims to determine the maximum performance of the system and identify the equipment in which exergy loss occurs, with an indication of the possibilities of thermodynamic improvement of the system under consideration because exergy analysis of a complex system can be performed by analyzing each component of the system separately.

The main contributions anticipated of this research design project can be summarized as follows:

1) To perform an extensive exergy analysis on the selected cogeneration plant. Remodel the gas turbine plant and cogeneration facility while ensuring mass, energy, and exhaust temperature requirements are met. Compare the tabulated exergy loss, or degradation, on each component and identify the main cause.

2) Identify the relevant mass, energy, entropy, and exergy equations and values to determine the system's optimal condition while maintaining the objective of the facility. Distinguish the boundary conditions that would enhance the design and reduce exergy destruction.

3) Develop revisions and modifications necessary to the base system using the available parameters and boundary conditions, while maintaining the primary objective of the facility and improving both second law efficiency by 5%, and exergy reduction by 5%.

## 3. System Description

The system selected for this project is a cogeneration plant based on the article (Ozkan et al., 2012) which employs a dual gas turbine that utilizes its waste heat to produce hot water and steam for a milk, seed, and beer manufacturing plant. The twin gas turbines generate electricity at a combined capacity of 10,000kW and operates as per the Brayton cycle, which employs an air compressor that compresses air at a rate of 60,000m³/hr and includes a combustion chamber that utilizes methane gas ($CH_4$) as fuel with an Air/Fuel Ratio of 39. Although the article (Ozkan et al., 2012) incorporates evaporative coolers to condition the air entering the compressors, we did not include it as there was no information of how it had affected or contributed to the overall cycle.

A combined exhaust line is found in the outlets of the turbines which separates into two processes before releasing it into the atmosphere. The first process consumed 17.44kg/s of exhaust gas into a heat exchanger and heated 123.34kg/s of water from 419K to 430K at a pressure of 1.1MPa. The second process consumed 21.88kg/s of exhaust gas into the heat recovery steam generator (HRSG) and converted 2.36kg/s of water to steam at 478K and 1.6MPa. Lastly. Ultimately, the plant configuration is shown in the Fig. 1 which slightly deviated from the base design of the article as illustrated in Fig. 11 of the appendix.

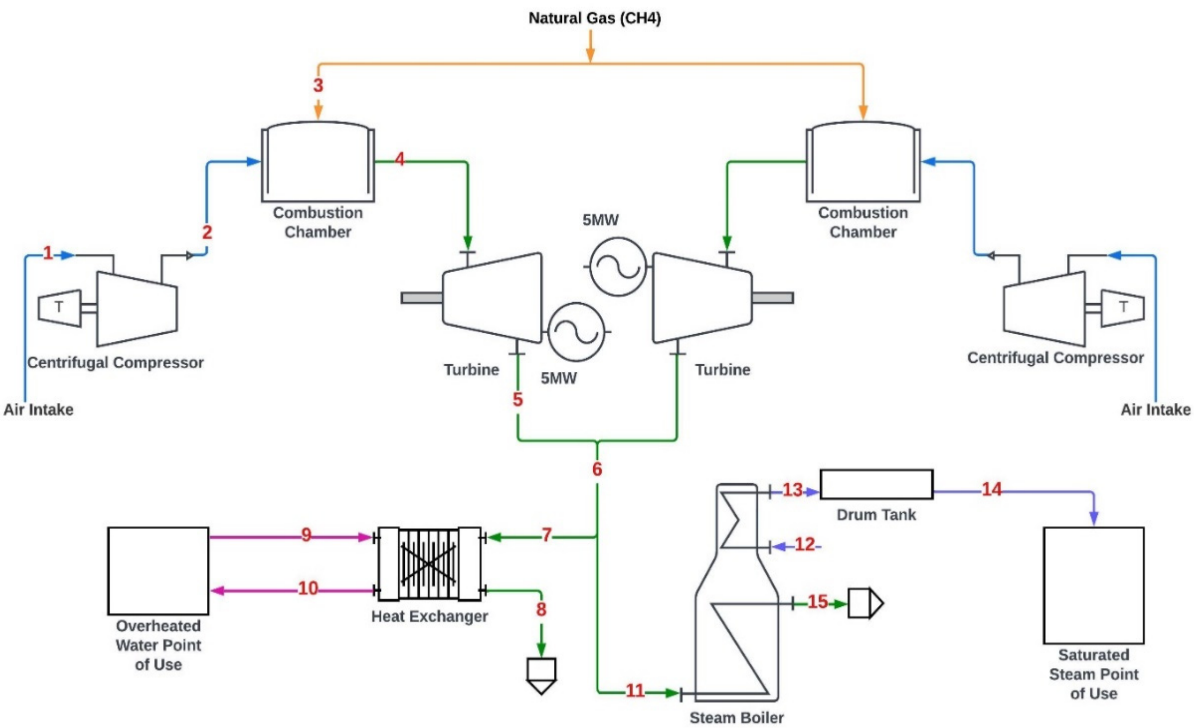


Figure 1. Configuration of the cogeneration plant

## 4. Method Statement

### *4.1 Problem Description*

The article (Ozkan et al., 2012) was missing majority of the parameters relating to the Brayton cycle which proved to be a challenge during the assessment of the amount of exergy degradation and while also proving the second law efficiencies provided. This problem enabled us to reconstruct and remodel the plant to calculate the following data while maintaining the essential requirement and output of the model plant:

- Find the temperature, pressure, enthalpy, and entropy on each point of the Brayton cycle.
- Find the new exhaust gas temperatures at point 8 and 15.
- Find exergy increase at the combustion chamber, while tabulating the useful work, heat transferred, exergy destroyed, and exergy lost at each component.
- Calculate the overall second law efficiency and corresponding thermal efficiencies.

In solving the problems described, we would ensure that the following requirements and outputs are met:

- Mass flow of air entering each compressor is maintained at 60,000m$^3$/s.
- 17.44kg/s for exhaust supplying heat to the heat exchanger, while producing 123.34kg/s of hot water at a temperature of 430K and 1.1MPa.
- 21.88kg/s for exhaust supplying heat to the HRSG, while producing 2.36kg/s of steam and 1.6MPa.
- The turbines' output work should expect 5,000 kW each while providing an input work of 1,860 kW to each compressor.

*4.2 Geometric Properties and Boundary Conditions*

The boundaries are set within each component: the compressors, combustion chambers, gas turbines, heat exchangers, and HRSG, where each is said to have a continuous flow (Ozkan, Kiziler et. al., 2010). Combustion chambers, compressors, turbines, steam generators, pipes, and other components are insulated against heat loss. Air, Natural Gas used ($CH_4$), and exhaust are assumed as ideal gases. The dead state is $T_0$ = 303K, and $P_0$ = 0.1 MPa (Ozkan et al., 2010).

*4.3 Description of the Equations (Mass, Energy, Entropy)*

The general mass balance relation for a control volume can be expressed as.

$$\frac{dm_{CV}}{dt} = \sum_{inlets} \dot{m}_i - \sum_{exits} \dot{m}_e \quad \left[\frac{kg}{s}\right] \tag{1}$$

The general energy balance relation for a control volume can be expressed as.

$$\frac{dE_{CV}}{dt} = \dot{Q}_{CV} - \dot{W}_{CV} + \sum_{inlets} \dot{m}_i \left(h_i + \frac{V_i^2}{2} + gz_i\right) - \sum_{exits} \dot{m}_e \left(h_e + \frac{V_e^2}{2} + gz_e\right) \quad [kW] \tag{2}$$

The general entropy balance relation for a control volume can be expressed as.

$$\frac{dS_{CV}}{dt} = \sum \frac{\dot{Q}_k}{T_k} + \sum_{inlets} \dot{m}_i s_i - \sum_{exits} \dot{m}_e s_e + \dot{S}_{gen} \quad \left[\frac{kW}{k}\right] \tag{3}$$

The general exergy balance relation for a control volume can be expressed as.

$$\frac{dX_{CV}}{dt} = \sum \left(1 - \frac{T_0}{T_k}\right) \dot{Q}_k - \left(\dot{W} - P_0 \frac{dV_{CV}}{dt}\right) + \sum_{inlets} \dot{m}_i \psi_i - \sum_{exits} \dot{m}_e \psi_e - T_0 \dot{S}_{gen} \quad [kW] \tag{4}$$

The process will be considered steady-flow for all devices;

$$\frac{dm_{CV}}{dt} = \frac{dE_{CV}}{dt} = \frac{dS_{CV}}{dt} = \frac{dX_{CV}}{dt} = \frac{dV_{CV}}{dt} = 0 \tag{5}$$

The change of kinetic and potential energy will be considered negligible for all devices,

$$V_i = V_e \quad \& \quad z_i = z_e \tag{6}$$

*4.4 Method of Analysis*

Based on the equations provided, each equipment is considered as an individual system with boundary conditions as stated below.

1) Compressors as the system. This will be a control volume since mass crosses the system boundary during the process. One inlet, and one exit is considered in equ. 1. In addition, the compressor assumes adequate insulation; thus, no heat crosses the control volume, so $\dot{Q}_{CV} = \dot{Q}_k = 0$ will be assumed for equ. 2 and 4.

2) Combustion chambers as the system. This will be a control volume since mass crosses the system boundary during the process. Two inlets and one exit will be considered for equ. 1.The combustion chamber will not be associated with any work, so $\dot{W}_{CV} = \dot{W} = 0$ will be assumed for equ. 2 and 4 $CH_4$ is considered the fuel for the

combustion chamber and is subjected to the chemical exergy. According to article (Ozkan et. al., 2010)., the chemical exergy can be calculated as follows.

$$\sum N_{product}(\overline{h}_f^0 + \overline{h} - \overline{h}^0)_{product} = \sum N_{reactant}(\overline{h}_f^0 + \overline{h} - \overline{h}^0)_{reactant}) \tag{7}$$

3) Turbines as the system. This will be a control volume since mass crosses the system boundary during the process. One inlet and one exit will be considered for equ. 1. No heat crosses the control volume, thus, $\dot{Q}_{CV} = \dot{Q}_k = 0$ is considered for equ. 2 and 4.

4) HRSG for steam generation and entire heat exchanger as the system for hot water generation. These will be control-volumes since mass cross the system boundary during the process. For boiler/ heat exchanger two inlets and two exits will be considered for equ. 1. In addition, both boiler and heat exchanger are well insulated hence heat cannot cross the control volume and work is not associated with these devices. Thus, $\dot{W}_{CV} = \dot{W} = 0$ & $\dot{Q}_{CV} = \dot{Q}_k = 0$ for equ. 2 and 4.

*4.5 Process Analysis*

The following flow chart Fig. 2 shows the general process of how the system was approached and evaluated to obtain the parameters necessary to conduct an exergy analysis. It starts out by recreating the cogeneration plant as it was perceived, as well as re-applying the inputs based on the model's (Ozkan et. al., 2012). requirement. The team then used a combination of software modelling and manual calculation to achieve the desired exergy tabulation and second law efficiencies from each component, as well as the overall plant. Software was utilized to simulate the plant's Brayton cycle, while the cogeneration facility was manually calculated via excel software. Results were compared with a predetermined target of 5% improvement, and remodeled if such enhancement is not achieved.

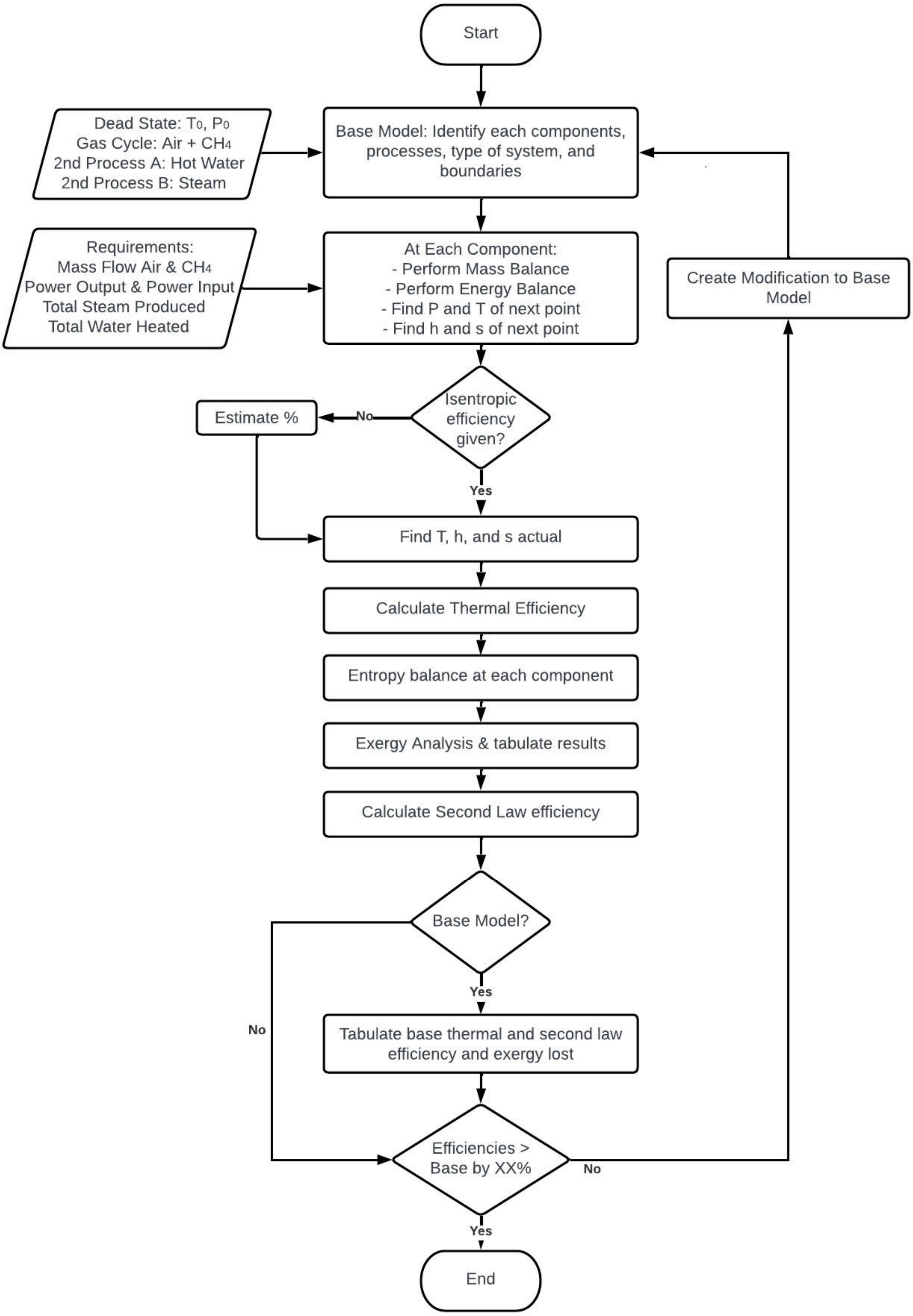


Figure 2. General process flow chart for system and the exergy analysis

## 5. Results and Discussion

### *5.1 Mass Flow Rate Analysis*

Mass flow rate on each component was prioritized to guarantee that the demanded output, such as air compressed, flue gas combusted, steam generated, and the rate of heated water was attained. Also, the information was available, or easily obtained through simple mass balance equation equ.8. Fig 3 therefore illustrates the complete mass balance of the system.

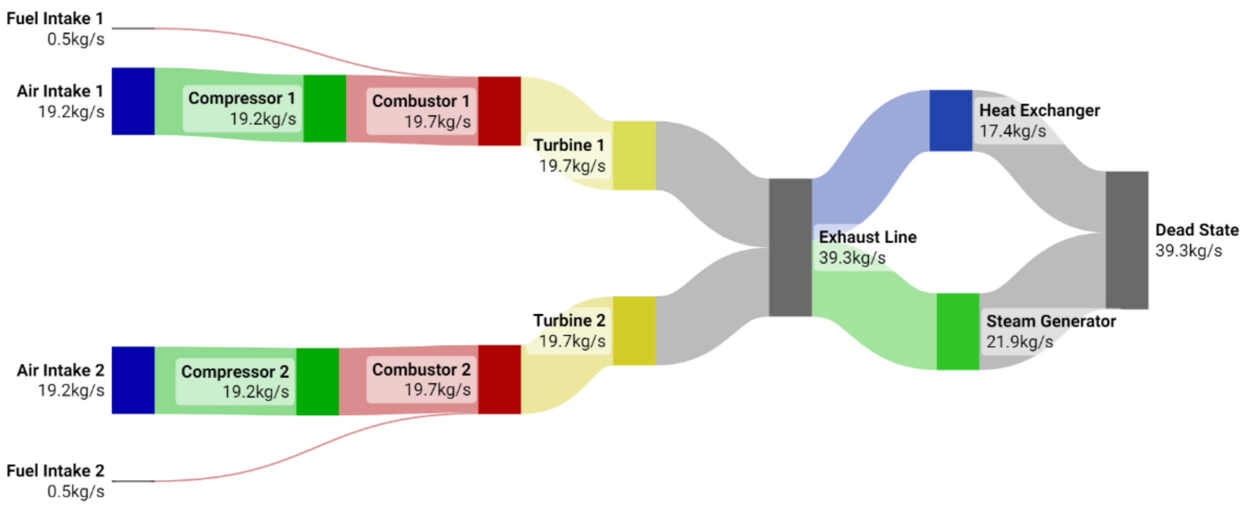


Figure 3. Mass flow rate breakdown for each equipment

### *5.2 Work Balancing*

Similar to mass balancing, the cogeneration plant's base model stated that its dual turbine is obliged to produce a total power output of 6,280kW to energize the whole manufacturing plant, including auxiliary equipment. Therefore, each of Turbine can assume work done at 3,140kW each. However, it should be noted that the Turbines' capacity is at 5,000kW each, and since the literature did not mention the work input into the air compressors, it is obvious that the surplus work was consumed by compressors as per equ. 9 for the compressor work and shown in fig. 4.

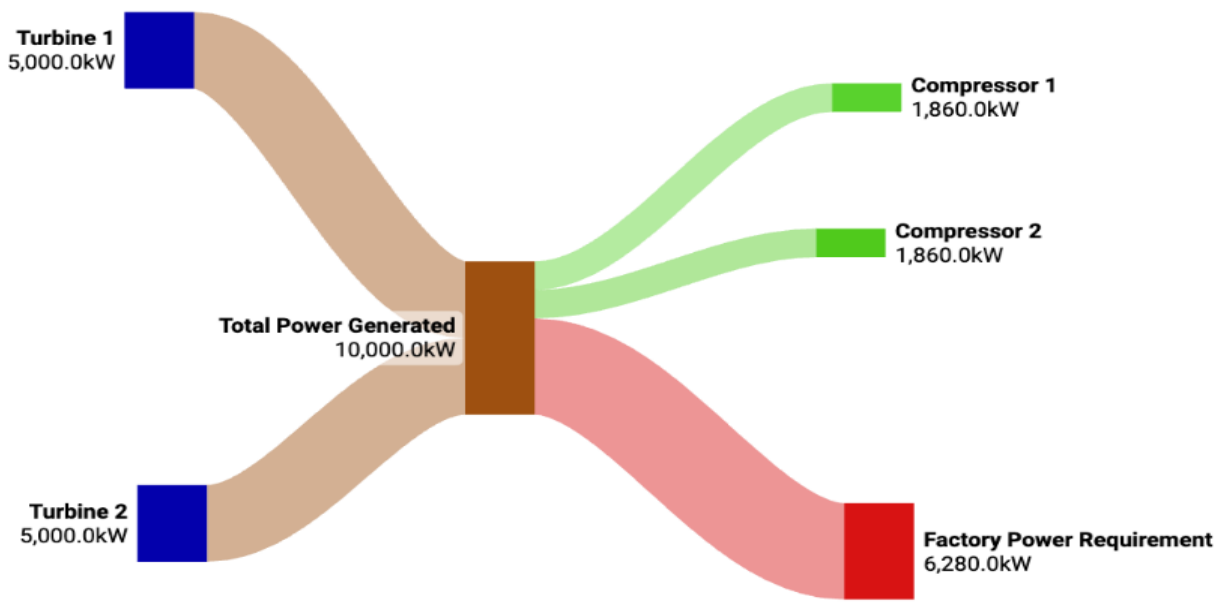


Figure 4. Power input and output for the system

*5.3 Brayton Cycle*

ASPEN HYSYSV11 software was used to simulate the dual gas turbine plant as shown in below fig 5. Since relevant information was either available or calculated, the Brayton cycle was simply remodeled.

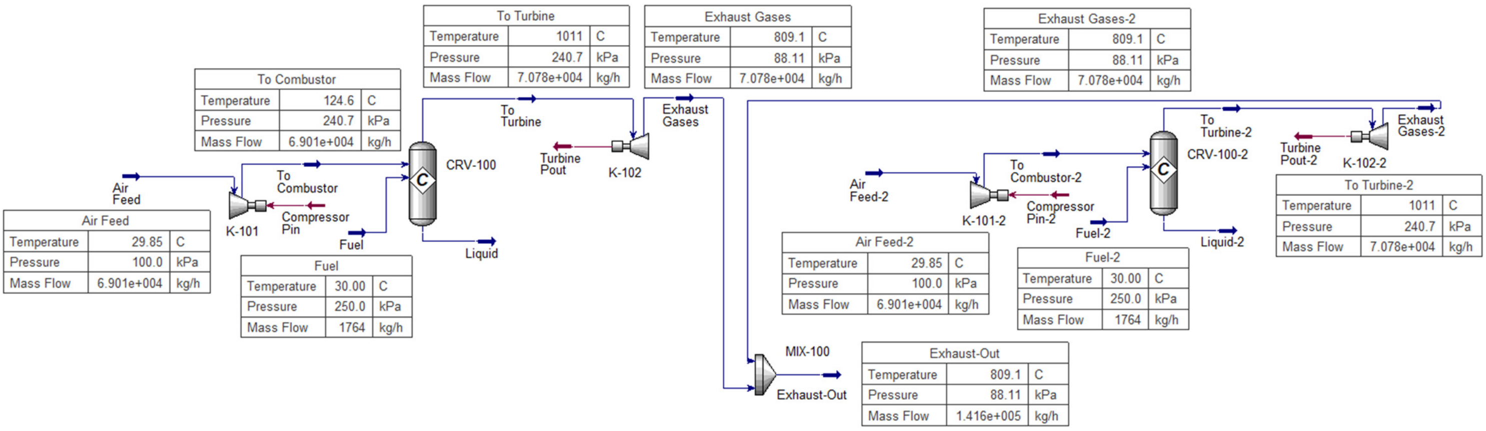


Figure 5. Dual gas turbine plant designed - Brayton cycle

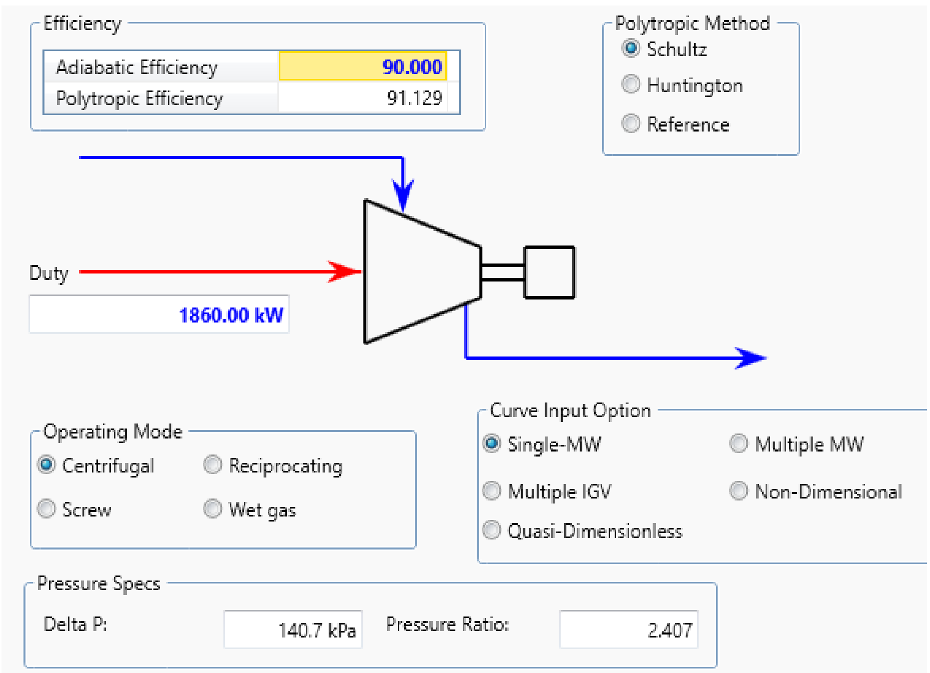


Figure 6. Simulation input and output data for the compressor

It starts by inserting data such as mass flow rate of air, intake air temperature and pressure, and work input by the compressor at point 1, then run to simulate. The program will provide the temperature and pressure of the next stage, which is required to obtain the enthalpy and entropy of point via the steam table A-17 “Ideal -gas properties of air”. Apart from those parameters, the program also provided calculated efficiencies and pressure ratios from the compressor. The next component in the simulation tool is modelling the combustion chamber. The relevant data input would be adding the fuel’s mass flow rate, Air/Fuel Ratio of 39 [$^{A}/_{F}\, ratio = kg\ \ air/kg\ \ fuel$], and provide an assumption for combustion efficiency. For our model, we assumed 85% combustion efficiency, without producing any CO on the combustion product. However, as discussed in geometric properties section, resulting flue / exhaust gases will be treated as ideal gas air. Thus, point 4 enthalpy and entropy were also obtained from steam table A-17 (Cengel, Boles et al., 2011). All resulting data was input to the gas turbine as per fig. 7, with the addition of total turbine work of 5,000kW. Upon running the program, we were able to achieve the exhaust gas temperature and calculate isentropic efficiency at point 5. Similarly, steam table A-17in (Cengel et al., 2011) was used to obtain the enthalpy and entropy at point 5. Lastly, the dual exhausts lines combined their mass flow rates, therefore point 6, 7, and 11 in fig. 1 have same enthalpy and entropy values.

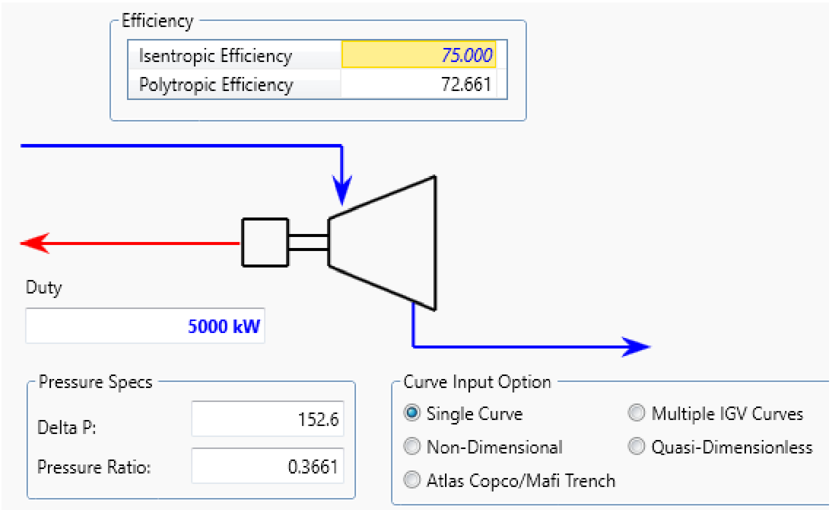


Figure 7. Gas turbine input and output parameters

*5.4 Cogeneration Facility (Heat Exchanger Section)*

Based on table 1, in the heat exchanger section, the resulting mass balances can be calculated as per equ.8, while the energy balance at the Hot Water HE can be calculated as per equ. 12. This was necessary to calculate the enthalpy at point 8 as given in fig. 1 and then using the value to extract the temperature, and entropy from same point using steam table A-17 in (Cengel et al., 2011).

*5.5 Cogeneration Facility (Steam Generator Section)*

Based on the steam generation section in table 1.0, the resulting mass balances can be calculated as per equ.8, while the energy balance at the Steam Generator HE can be calculated as per equ. 12. This was necessary to calculate the enthalpy at point 15 and use the value extract the temperature, and entropy same point using the steam using steam table A-17 in (Cengel et al., 2011). Note that enthalpy and entropy at point 12 in fig. 1 used saturated liquid value at pressure 1.6MPa, while point 13 was slightly superheated at 205°C.

## 6. Summary of Results

Table 1. Showing the value at each point, for each parameter, of the plant

| | Var. | Units | Gas Turbine | | | | | | | HE | | | | HRSG | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | 0 | 1 | 2 | 3 | 4 | 5 | 6 | 7 | 8 | 9 | 10 | 11 | 12 | 13 | 14 | 15 |
| Base Model | m | kg/s | - | 19.17 | 19.17 | 0.49 | 19.66 | 19.66 | 39.32 | 17.44 | 17.44 | 123.34 | 123.34 | 21.88 | 2.36 | 2.36 | - | 21.88 |
| | T | K | 303.00 | 303.00 | 397.60 | 303.00 | 1284.00 | 1082.10 | 1082.10 | 1082.10 | 784.63 | 419.00 | 430.00 | 1082.10 | 474.26 | 478.00 | - | 897.81 |
| | P | kPa | 100.00 | 100.00 | 240.70 | 250.00 | 240.70 | 88.11 | 88.11 | 88.11 | 88.11 | 1100.00 | 1100.00 | 88.11 | 1600.00 | 1600.00 | - | 88.11 |
| | h | kJ/kg | 303.21 | 303.21 | 398.56 | - | 1376.99 | 1140.32 | 1140.32 | 1140.32 | 805.10 | 615.37 | 662.77 | 1140.32 | 857.99 | 2802.79 | - | 930.49 |
| | s0 | kJ/kg-K | 1.71200 | 1.71200 | 1.98579 | - | 3.25877 | 3.05820 | 3.05820 | 3.05820 | 2.69655 | - | - | 3.05820 | - | - | - | 2.84582 |
| | s | kJ/kg-K | - | - | - | - | - | - | - | - | - | 1.80022 | 1.91188 | - | 2.34234 | 6.44054 | - | - |

*6.1 Exergy Analysis*

Substituting the values from table 1.0 into table 4.0 (Exergy Balance Equations), we can obtain the following:

- Exergy destruction of the Compressor from equ.14.
- Exergy increase of the combustion chamber from equ. 15.
- Exergy destruction of the Turbine from equ. 16.
- Exergy recovered and exergy destruction of the Hot Water HE from equ. 17.
- Exergy recovered and exergy destruction of the Steam Generator HE from equ. 18.

Upon tabulation of the results from the said equations, exergy summary is illustrated as per fig. 8 below.

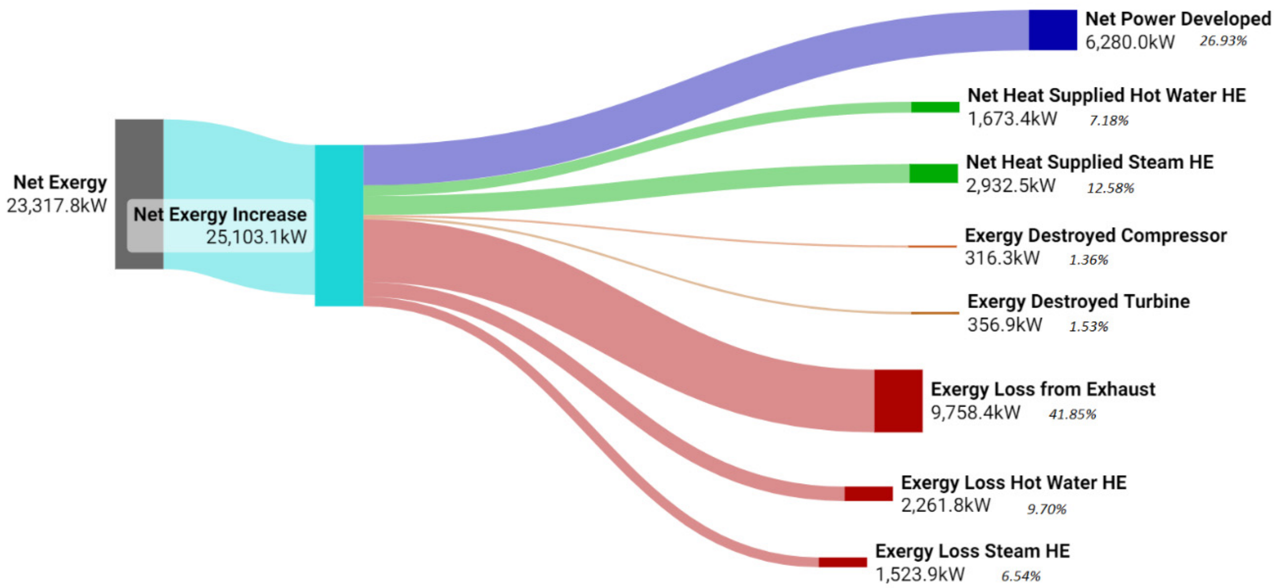


Figure 8. Net exergy increase and exergy summary of the various equipment

It should be noted that the largest exergy loss is obtained from the exhaust as the gas turbine is able to exhaust a higher temperature of waste heat at 1,082K, than the observed value by the article which was about 764.44K. There was also a significant amount exergy loss from the heat exchanger and steam generators as similar to the article's (Ozkan et al., 2012) report. Furthermore, exergy destruction coming from the turbines and compressors are the lowest due to the nature of the component.

**Error Reporting:** There was about 7.6% excess in the exergy tabulation which was due to the calculation of net exergy increase from the combustion chamber since the manual equation considered only the enthalpies and entropies of points 4 and 2, while the temperatures were based from the simulation data. It was checked that the exergies do balance if Qin does not consider point 3 which mass of fuel is injected. For this reason, we chose to represent our calculation with 25,103kw of exergy gain.

*6.2 Second Law and Thermal Efficiencies*

The second law efficiency for the plant can be defined as following equ.20.

$$\eta_{II} = (Exergy\ \ Supplied - Exergy\ \ Destroyed)/Exergy\ \ Supplied \qquad [20]$$

The overall thermal efficiency is calculated as per equ. 21 given below;

$$\eta_{Thermal} = \dot{W}_{net} \big/ \dot{Q}_{in} \qquad [21]$$

$\dot{W}_{net}$ is the sum of useful work from the entire process calculated at 16,716kW as per equ. 22

$$\dot{W}_{net} = \dot{W}_{Turbine} - \dot{W}_{Compressor} + \dot{Q}_{HE\ \ water} + \dot{Q}_{HE\ \ Steam} \qquad [22]$$

$\dot{Q}_{in}$ is the total amount of heat energy added to the system calculated at 41,650kW as per equ. 23.

$$\dot{Q}_{in} = \dot{m}_{fuel}\ \ x\ \ LHV_{CH4}\ \ x\ \ \eta_{Combustion} \qquad [23]$$

Therefore, the thermal efficiency is computed as 40.1%. The second law efficiency is calculated equ. 20, the second law efficiency is calculated as 39.0%.

*6.3 Second Law Efficiency for Components*

The corresponding components assessed their second law efficiencies $\eta_{II}$ by using equ. 14 for compressor, equ. 16 for turbine, equ. 17 for hot water heat exchanger, and equ. 18 for steam generator. After calculating the values, the subsequent fig. 10 was generated to compare the results from table 2 below.

Table 2. Components and the various second law efficiencies

| S/N | Components | Second Law Efficiency |
|---|---|---|
| **1.** | Compressor | 91% |
| **2.** | Turbine | 97% |
| **3.** | Hot Water Heat Exchanger | 43% |
| **4.** | Steam generator | 52% |
| **5.** | Overall Cogeneration Plant | 53 |

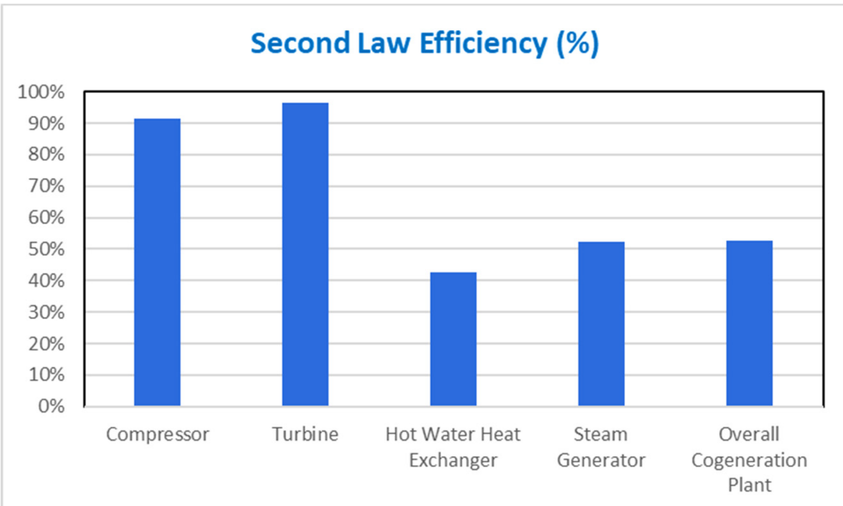


Figure 9. Plots of the components and the second law efficiencies

*6.4 Modification of the Co-generation System*

Since the exhaust gas is essential to process the secondary requirement of operating the cogeneration facility, the final iteration of the modification was focused on fully utilizing the waste heat instead of increasing the Brayton cycle efficiency. The following fig.10 was conceptualized where point 6 maintained its function by combining the exhaust gases while entering into an HRSG component. It transfers the optimum amount of waste heat and leaves at point 16. Water enters at point 17 with a rate of 4.88kg/s and transforms into steam at point 18. It then splits into two lines, where the first enters a steam turbine and produces 1,000kW of work and exhausts for steam use similar to the base model requirement for the beer plant at 2.36kg/s. The second process utilizes steam into a similar heat exchanger from the base model where hot water is produced. The steam, at point 21, then exits as compressed liquid and into a flash chamber where it acts as a pre-heat to replenished water (losses due to steam use) and then pumped back into the HRSG. The following table 3 shows the parameters of the modified cogeneration facility at each point and component.

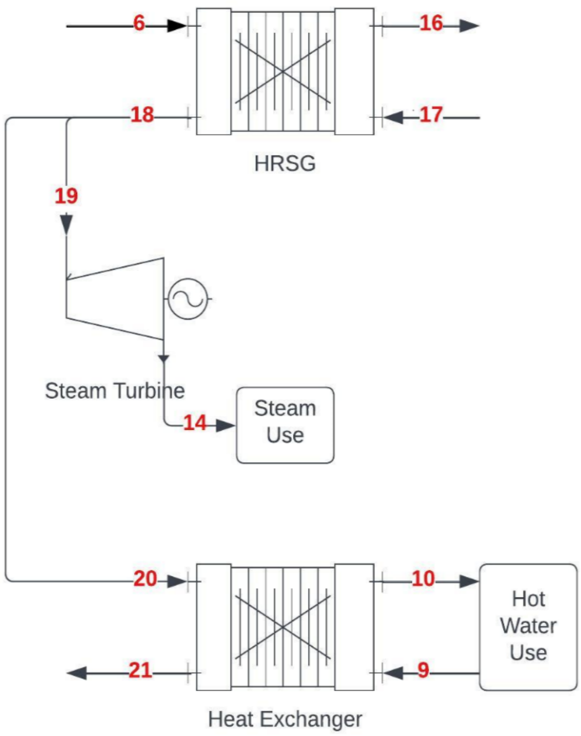


Figure 10. Modified cogeneration facility

It found that the net work and useful exergy supplied was about 10,718kW, while the total exergy lost and destroyed are 9,171kW and 3,939.7kW respectively. Using equ. 20, second law efficiency was calculated as 43.8%, and using equ. 21 calculated the thermal efficiency to be 42.2%.

Table 3. Modified cogeneration facility parameters

| | Var. | Units | Air | | | Water/Steam | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | 0 | 6 | 16 | 17 | 18 | 19 | 14 | 9 | 10 | 20 | 21 |
| Modifications | m | kg/s | - | 39.32 | 39.32 | 4.88 | 4.88 | 2.36 | 2.36 | 123.34 | 123.34 | 2.52 | 2.52 |
| | T | K | 303.00 | 1082.10 | 827.83 | 485.40 | 663.00 | 663.00 | 478.00 | 419.00 | 430.00 | 663.00 | |
| | P | kPa | 100.00 | 88.11 | 88.11 | 2000.00 | 2000.00 | 2000.00 | 1600.00 | 1100.00 | 1100.00 | 2000.00 | 2000.00 |
| | h | kJ/kg | 303.21 | 1140.32 | 852.63 | 908.47 | 3226.52 | 3226.52 | 2802.79 | 615.37 | 662.77 | 3226.52 | 906.55 |

*6.3 Conclusion*

The cogeneration power plant was remodelled to ensure the desired output for heat and power demands are met, as well as set the existing boundaries provided as per article (Ozkan et al., 2012). Based on the results, it was evident that the highest source of the exergy lost was coming from the exhaust gas since the facility did not fully utilize the remainder of the waste heat. The heat exchanger losses to heat the hot water was second highest exergy loss at about 9.7%, which was better since it was half of what the article (Ozkan et al., 2012) described. Steam generator exergy losses third highest exergy loss but was better by 25% than what the article had calculated, while the rest of the components had more or less, the expected losses based on their processes.

As for the modelled system, it was found that we could maintain the boundaries set. However, it was observed that there seems to be an imbalance between the amount of air flow and the capacities of the isentropic machines, since using both requirements yield a low pressure ratio of 2.54. As per article (DiPippo, 2008), there is a correlation between higher pressure ratio and increased overall efficiency. Based on quick analysis, reducing the mass flow of air by half, yields double the base pressure ratio with enough energy to enable the cogeneration facility. Note that most efficient gas turbines minimum pressure ratio starts at 8 to 10, therefore further modelling can improve the base configuration.

To increase second law efficiency and reduce exergy destruction while preserving the base components, adding further processes was found to be the best method. Upon multiple iterations, we established that revising the cogeneration facility to fig. 1 was the highest increase in both categories. It improved second law efficiency by 4%, increased thermal efficiency by 2%, and reduced exergy loss by 8%. A couple of iteration such as SOFC was not considered since the exhaust gas' energy and quality was essential for the secondary plant.

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Appendix

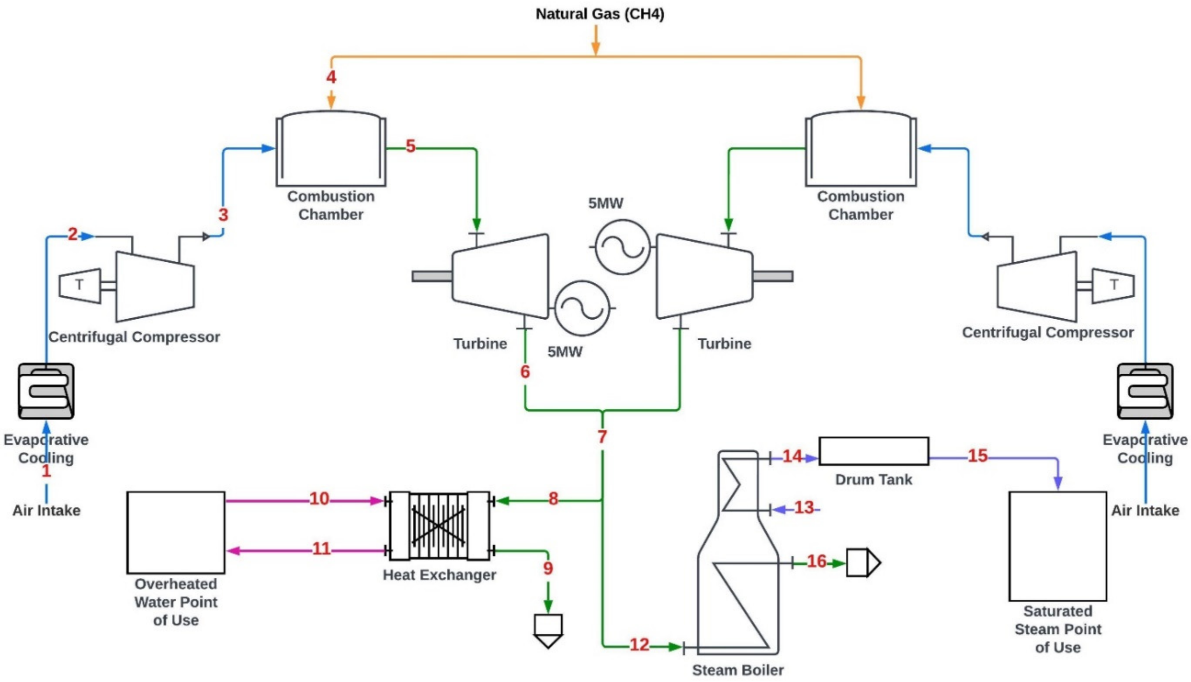


Figure 11. Base model design of the cogeneration plant

Table 4. Mass balance, energy balance and exergy balance equations

| | Mass Balance | Remarks |
|---|---|---|
| | $\dot{m}_1 = \dot{m}_2$<br>$\dot{m}_4 = \dot{m}_2 + \dot{m}_3$<br>$\dot{m}_4 = \dot{m}_5$<br>$\dot{m}_6 = 2 * \dot{m}_5$<br>$\dot{m}_6 = \dot{m}_7 + \dot{m}_{11}$<br>$\dot{m}_7 = \dot{m}_8$<br>$\dot{m}_9 = \dot{m}_{10}$<br>$\dot{m}_{11} = \dot{m}_{15}$<br>$\dot{m}_{12} = \dot{m}_{13}$ | Equ.[8] |
| **Equipment Name** | **Energy Balance** | **Remarks** |
| Compressor | $P_2 = r_P * P_1$<br>$\frac{T_{2s}}{T_1} = \left(\frac{P_2}{P_1}\right)^{\frac{k-1}{k}}$<br>$\eta_c = \frac{T_{2s} - T_1}{T_2 - T_1}$<br>$\dot{W}_C = \dot{m}_1 c_p (T_2 - T_1)$ | Equ.[9] |
| Combustion Chamber | Stoichiometric Equation<br>$CH_4 + 2(O_2 + 3.76N_2) = CO_2 + 2H_2O + 7.52N_2$ | Equ. [10] |

| | | |
|---|---|---|
| | $\sum N_{product}(\overline{h}_f^0 + \overline{h} - \overline{h}^0)_{product} = \sum N_{reactant}(\overline{h}_f^0 + \overline{h} - \overline{h}^0)_{reactant})$ | |
| Turbine | $\frac{T_{5s}}{T_4} = \left(\frac{P_5}{P_4}\right)^{\frac{k-1}{k}}$ <br> $\eta_T = \frac{T_4 - T_5}{T_4 - T_{5s}}$ <br> $\dot{W}_T = \dot{m}_4 c_p (T_4 - T_5)$ | Equ. [11] |
| Hot Water HE | $\dot{m}_7 h_7 + \dot{m}_9 h_9 = \dot{m}_8 h_8 + \dot{m}_{10} h_{10}$ | Equ. [12] |
| HRSG | $\dot{m}_{11} h_{11} + \dot{m}_{12} h_{12} = \dot{m}_{15} h_{15} + \dot{m}_{13} h_{13}$ | Equ. [13] |
| **Equipment Name** | **Exergy Balance** | |
| Compressor | $\dot{X}_{D,C} = \dot{W}_C + \dot{m}_1(\psi_1 - \psi_2)$ <br> $\dot{X}_{D,C} = \dot{W}_{C,1} + \dot{m}_1\left[(h_1 - h_2) + T_0\left(s_2^0 - s_1^0 - R\ln\left(\frac{P_2}{P_1}\right)\right)\right]$ <br> $\eta_{II,C} = \frac{\dot{m}_1(\psi_2 - \psi_1)}{\dot{W}_C}$ | Equ. [14] |
| Combustion Chamber | Stoichiometric Equation <br> $CH_4 + 2(O_2 + 3.76N_2) = CO_2 + 2H_2O + 7.52N_2$ <br> $\dot{X}_{D,CC} = T_0(\sum N_{product}\bar{s}_{product} - \sum N_{reactant}\bar{s}_{reactant})$ <br> $\bar{s}_i(T, P_i) = \bar{s}_i^0(T, P_0) - R_u \ln\left\lvert\frac{y_i P_m}{P_0}\right\rvert$ | Equ. [15] |
| Turbine | $\dot{X}_{D,T} = \dot{m}_4(\psi_4 - \psi_5) - \dot{W}_T$ <br> $\dot{X}_{D,T} = \dot{m}_4\left[(h_4 - h_5) + T_0\left(s_5^0 - s_4^0 - R\ln\left(\frac{P_5}{P_4}\right)\right)\right] - \dot{W}_T$ <br> $\eta_{II,T} = \frac{\dot{W}_T}{\dot{m}_4(\psi_4 - \psi_5)}$ | Equ.[16] |
| Hot Water HE | $\dot{X}_{D,HW} = \dot{m}_7(\psi_7 - \psi_8) + \dot{m}_9(\psi_9 - \psi_{10})$ <br> $\dot{X}_{D,HW} = \dot{m}_7[(h_7 - h_8) + T_0(s_8^0 - s_7^0)] + \dot{m}_9[(h_9 - h_{10}) + T_0(s_{10} - s_9)]$ <br> $\eta_{II,HW} = \frac{\dot{m}_9(\psi_{10} - \psi_9)}{\dot{m}_7(\psi_7 - \psi_8)}$ <br> $\psi_7 = (h_7 - h_0) - T_0(s_7 - s_0)$ <br> $\psi_8 = (h_8 - h_0) - T_0(s_8 - s_0)$ <br> $\psi_9 = (h_9 - h_0) - T_0(s_9 - s_0)$ <br> $\psi_{10} = (h_{10} - h_0) - T_0(s_{10} - s_0)$ | Equ. [17] |

| | | |
|---|---|---|
| HRSG | $\dot{X}_{D,SG} = \dot{m}_{11}(\psi_{11} - \psi_{15}) + \dot{m}_{12}(\psi_{12} - \psi_{13})$<br>$\dot{X}_{D,SG} = \dot{m}_{11}[(h_{11} - h_{15}) + T_0(s_{15}^0 - s_{11}^0)] + \dot{m}_{12}[(h_{12} - h_{13}) + T_0(s_{13} - s_{12})]$<br>$\eta_{II,SG} = \frac{\dot{m}_{12}(\psi_{13} - \psi_{12})}{\dot{m}_{11}(\psi_{11} - \psi_{15})}$<br>$\psi_{11} = (h_{11} - h_0) - T_0(s_{11} - s_0)$<br>$\psi_{12} = (h_{12} - h_0) - T_0(s_{12} - s_0)$<br>$\psi_{13} = (h_{13} - h_0) - T_0(s_{13} - s_0)$<br>$\psi_{15} = (h_{15} - h_0) - T_0(s_{15} - s_0)$ | Equ. [18] |

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**Authors contributions**

Wilson Ekpotu contributed to investigation, methodology, conceptualization, and writing – original draft. Abir Hossain Mridul performed investigation, conceptualization, writing – original draft, Paul Mikii Abuel contributed to writing – review & editing, simulation analysis, and review. Additionally, Paul Mikii participated in investigation and visualization and Abir Hossain Mridul contributed to research analysis, review, and project administration. All authors have thoroughly read and given their approval for the final manuscript to be published.

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**Data sharing statement**

No additional data are available.