MECH-3020 Design Project

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Section 1: Problem Statement

The James H Miller Electric Generator Plant is the largest power plant in terms of Megawatt capacity in Alabama. The plant produces 700 MW for each of its 4 units and burns over 30,000 tons of coal per day. In this project, the challenge is to determine how modifications affect the cycle efficiency, turbine work, and overall feasibility of the system. By analyzing how pressure selections and component configurations change the thermodynamic states of the cycle, this project aims to identify a more efficient and practical design for power production and reduction of needed coal.

Section 2: Base Plant

The base plant analyzed in this project is a simplified Rankine cycle with one stage of reheat based on the given operating conditions for the reference plant. This configuration serves as the baseline for all later comparisons, allowing the effects of design changes such as varying reheat pressure and adding feedwater heaters to be evaluated against a consistent reference case. The base plant cycle layout is shown in Figure 1, the operating conditions are summarized in Table 1, the corresponding temperature-entropy diagram is shown in Figure 2.

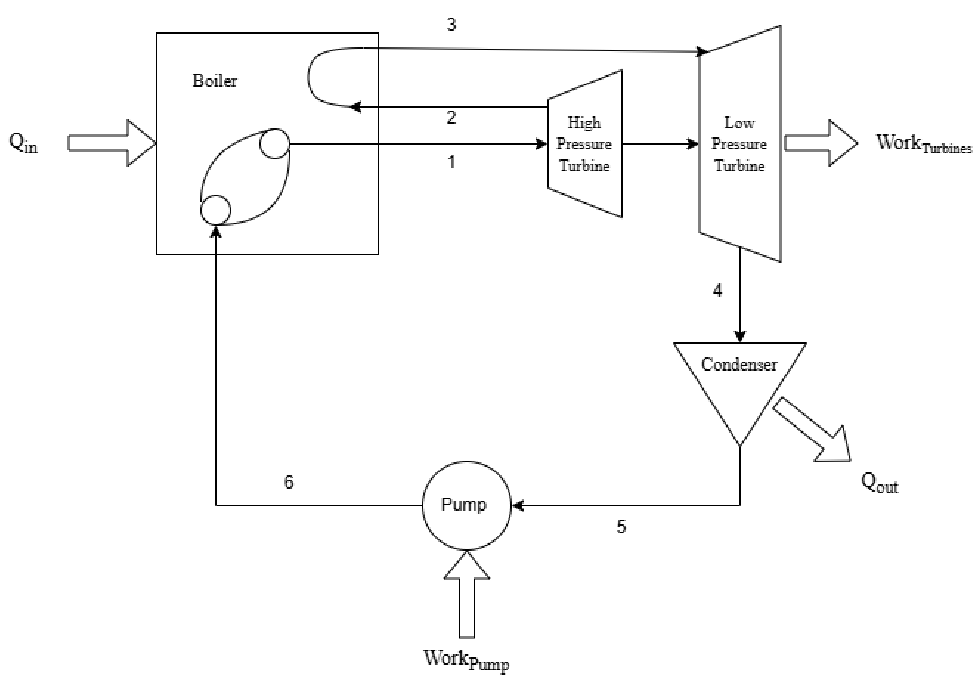

Figure 1: Base Plant Cycle Diagram

Table 1: Base Plant Operating Parameters

	State	Pressure (psi)	Temperature(F)	Enthalpy ($\frac{BTU}{lb}$)
1	SHV	2400	1000	1460.58
2	SHV	800	695	1335.75
3	SHV	800	1000	1511.9
4	SM	14.7	212	1140.738
5	SL	14.7	212	180.15
6	CL	2400	244	189.375

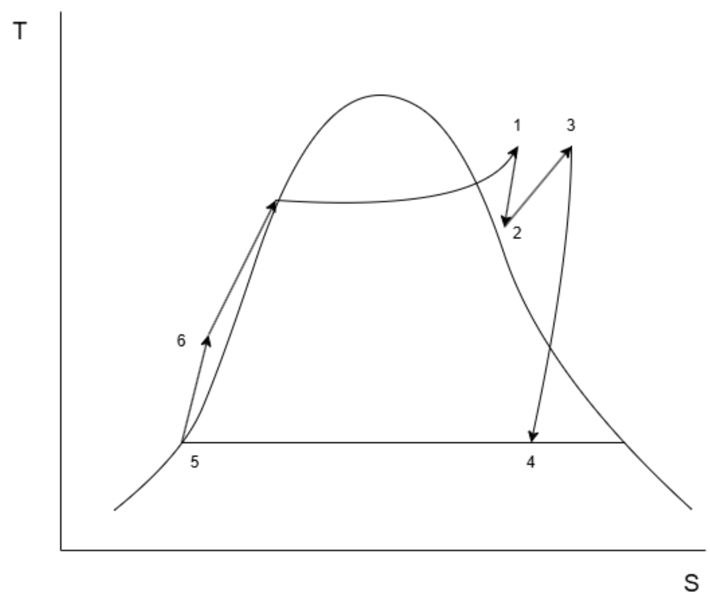

Figure 2: Temperature-Entropy Table of Base Cycle

The base plant was then assessed for critical performance metrics. The first metric recorded was the efficiency using the following equation:

$$\eta_{th} = \frac{W_{net}}{Q_{in}}$$

Or in terms of the enthalpies, h, of states shown in figure 1:

$$\eta_{th} = \frac{(h_1 - h_2) + (h_3 - h_4) - (h_6 - h_5)}{(h_1 - h_6) + (h_3 - h_2)}$$

The mass flow rate was calculated using the following:

$$\dot{m} = \frac{\dot{W}_{net}}{w_{net}}$$

The heat into and rejected by the system were found by performing energy balances of the boiler and condenser respectively.

Boiler Energy balance:

$$Q_{in} = \dot{m}(q_{primary} + q_{reheat}) = \dot{m}((h_1 - h_6) + (h_3 - h_2))$$

Condenser Energy balance:

$$Q_{out} = \dot{m}(q_{out}) = \dot{m}(h_4 - h_5)$$

All the critical performance metrics for the base plant were recorded in table 2 below.

Table 2: Base Plant Performance Metrics

Efficiency, η (%)	Mass Flowrate, $\dot{m}$ $(\frac{lbm}{hr})$	Heat Rejected, $\dot{Q}_{out}(\frac{BTU}{hr})$	Heat in, $\dot{Q}_{in}$ $(\frac{BTU}{hr})$
33.632	4908049.802	4714612643.658	7103691141.951

Section 3: Design Parameters

The design space considered in this project was restricted to modifications that could improve the thermal performance of the cycle without requiring a complete redesign of the existing plant. The allowable changes included:

- Varying the reheat pressure.
- Adding one or combinations of feedwater heaters.
- Evaluating combinations of these modifications.

These options were selected because they have a direct influence on the thermodynamic behavior of the Rankine cycle while remaining representative of practical plant-level improvements.

To ensure that all proposed designs remained realistic and consistent with the limitations of the base plant, several constraints were imposed on the analysis.

- The total mass flow rate of any modified cycle must remain within 15% of the base-cycle mass flow rate due to the limitations of the existing piping infrastructure.
- All original plant operating parameters remain fixed, with the exception of the reheat pressure, which was treated as a design variable in this study.
- The total heat rejected to the river must not exceed the heat rejected by the base plant, since any increase in waste heat discharge would violate the operational and environmental limits of the system.

Collectively, these design parameters and constraints established the acceptable range of cycle modifications evaluated in this project.

Section 4: Design One

Based on the reheat-cycle guidance presented by Habib[1], the optimum reheat pressure for a single-reheat Rankine cycle is approximately one-fourth of the maximum boiler pressure. Using this same guideline for the present study, a reheat pressure of 600 psi was selected because the boiler pressure of the base plant is 2400 psi, and 600 psi corresponds to 25% of the boiler pressure. This provided a literature-supported basis for evaluating whether lowering the reheat pressure from the base case could improve cycle performance.

The base cycle was analyzed with a modified reheat pressure of 600 psi in order to evaluate the effect of reduced reheat pressure on cycle performance. All other plant operating conditions were kept the same as those of the original base plant so that the influence of reheat pressure could be isolated directly. Since the cycle configuration itself did not change, Figure 1 and Figure 2 remain the same as those presented for the base plant. The updated operating conditions for this modified case are summarized in Table 3, while the corresponding performance results, including the critical cycle metrics, are presented in Table 4.

Table 3: Cycle Parameters with 600 psi Reheat Pressure

	State	Pressure (psi)	Temperature (°F)	Enthalpy ($\frac{BTU}{lb}$)
1	SHV	2400	1000	1460.58
2	SHV	600	695	1307.525
3	SHV	600	1000	1517.8
4	SM	14.7	212	1162.323
5	SL	14.7	212	180.15
6	CL	2400	244	189.375

Shown in table 3, enthalpy values for states 1, 5, and 6 remain the same as they are not affected by the change in reheat pressure. State 2 enthalpy value decreases by 28.225 btu/lb, state 3 enthalpy increases by 5.9 btu/lb, and state 4 enthalpy increases by 21.585 btu/lb. The changes in enthalpy values result in the following performance metrics for the new cycle with 600 psi reheat section:

Table 4: Performance Metrics for Reheat Pressure at 600 psi

Efficiency, η (%)	Mass Flowrate, $\dot{m}$ ($\frac{lbm}{hr}$)	Heat Rejected, $\dot{Q}_{out}$ ($\frac{BTU}{hr}$)	Heat in, $\dot{Q}_{in}$ ($\frac{BTU}{hr}$)
33.703	4784786.638	4699488493.064	7088566991.358

Shown in table 4, the design goal of increasing the efficiency of the cycle was accomplished. The efficiency increased by .213% and as a result reducing the amount of coal needed. All design constraints were satisfied as well. The mass flowrate only decreased by 2.511% which meets the max percent difference criteria of 15%. Finally, the heat rejected by the condenser was decreased by .321%.

While the efficiency of the improved Rankine Cycle was only increased by a small amount, the cost of this modified system would be significantly lower than incorporating new mechanical equipment or a complete redesign.

Section 5: Design Two

A second modified-cycle concept was evaluated by incorporating an open feedwater heater operating at 200 psi while maintaining the reheat pressure at 800 psi. This configuration was selected as an initial regenerative design intended to improve cycle efficiency by preheating the feedwater before it entered the boiler, while avoiding extraction at excessively high pressure. A pressure of 200 psi was chosen as a reasonable starting point because, although increasing feedwater-heater pressure can strengthen regenerative heating, extracting steam at progressively higher pressures eventually reduces the amount of steam available for further turbine expansion and can negatively affect net cycle performance. Prior studies on regenerative Rankine cycles show that feedwater heater and extraction pressures are important design variables that must be selected or optimized based on the balance between improved feedwater heating and the loss of turbine work [2], which supports using 200 psi as an initial pressure for evaluation in the present study.

Figure 3 shows the modified Rankine Cycle with the addition of the open feed water heater at 200 psi. Table 5 has the respective values for each stage of the cycle and Figure 4 plots the states on a temperature-entropy diagram.

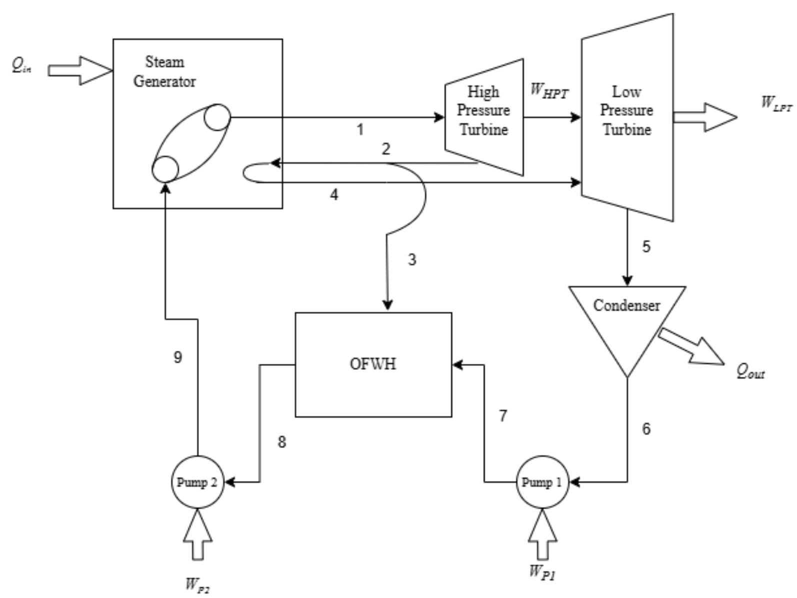

Figure 3: Modified Cycle with Open Feedwater Heater at 200 psi

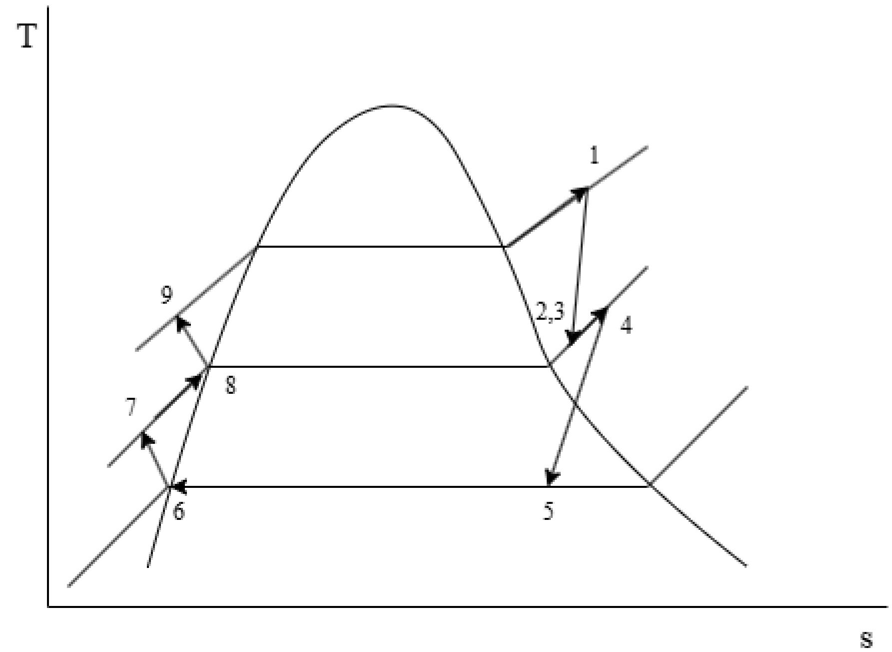

Figure 4: Temperature-Entropy Table of a Regenerative Rankine Cycle with Reheat

Table 5: Modified Cycle Parameters for Open Feedwater Heater at 200 psi

	State	Pressure (psi)	Temperature (°F)	Enthalpy ($\frac{BTU}{lbm}$)
1	SHV	2400	1000	1460.58
2	SHV	800	695	1335.75
3	SHV	200	695	1335.75
4	SHV	800	1000	1511.9
5	SM	14.7	212	1140.749
6	SL	14.7	212	180.15
7	CL	200	381.86	181.05
8	SL	200	522.03	355.6
9	CL	2400	662.146	365.1

The critical performance metrics for this modified cycle with an open feedwater heater were recorded in table 6 below. The efficiency of this cycle differs from the previous cycles. The efficiency of this cycle is found with equations:

$$\eta_{th} = \frac{(h_1 - h_2) + (h_4 - h_5) - (h_7 - h_6) - (h_9 - h_8)}{(h_1 - h_9) + (h_4 - h_2)}$$

The heat into and rejected by the system also differ from the base system equations. For the open feedwater cycle, the energy balance equations of the boiler and condenser equate to:

Boiler Energy balance:

$$Q_{in} = \dot{m}(q_{primary} + q_{reheat}) = \dot{m}((h_4 - h_2) + (h_1 - h_9))$$

Condenser Energy balance:

$$Q_{out} = \dot{m}(q_{out}) = \dot{m}(h_5 - h_6)$$

All the critical performance metrics for the base plant were recorded in table 6 below.

Table 6:Performance Metrics for Modified Cycle with Open Feedwater Heater at 200 psi

Efficiency, η (%)	Mass Flowrate, $\dot{m}$ $(\frac{lbm}{hr})$	Heat Rejected, $\dot{Q}_{out}$ $(\frac{BTU}{hr})$	Heat in, $\dot{Q}_{in}$ $(\frac{BTU}{hr})$
34.52	5557698.767	4530913590	6919992090

When adding an open feedwater heater, the efficiency of the plant cycle increased by 2.65%. The mass flowrate increased significantly; however, it stays within the constraint of 15% change at 13.24%. The heat rejected into the river decreased by 3.9%.

The performance metrics for this modified system show great potential, however, the cost effects will be taken into consideration for the final decision shown in the next section.

Section 6: Cost

The ratio for tons of coal per BTU of the system was found by finding the ratio from the base plant using:

$$\frac{\frac{tons\ of\ coal}{day}}{\dot{Q}_{in}}$$

Using the equation above, the ratio for tons of coal per btu was found to be:

$$4.39912144 \times 10^{-8}\ \frac{tons\ of\ coal}{BTU}$$

Using this ratio, the amount of coal used per day for each design modification is listed in table 7.

Table 7: Coal Usage Comparison

	Base Plant	Lower Reheat Pressure	Open Feedwater Heater
Tons of coal per day	7500	7484.032	7306.053
Percent Difference (%)	0	.21%	2.59%

According to the U.S Energy Information Administration [3], the average cost of coal delivered to U.S power plants was approximately $53 per ton, varying heavily by region and coal type. Using that as an average, the daily cost of fuel can be determined by using:

$$Daily\ Fuel\ Cost = \left(\frac{tons}{day}\right) \times \left(\frac{\$}{ton}\right)$$

Table 8: Daily Cost of Coal

	Base Plant	Lower Reheat Pressure	Open Feedwater Heater
Daily Fuel Cost	$397,500	$396,653.70	$387,220.89

Assuming the base plant boiler can operate over a range of pressure settings, the cost of reducing the reheat pressure can be considered negligible aside from a small labor expense associated with adjustment and evaluation. In contrast, implementing an open feedwater heater requires additional capital investment, including the cost of the feedwater heater itself, an added pump to raise the feedwater back to boiler pressure after extraction, and the associated installation labor. Labor cost estimates were based on data from the U.S. Bureau of Labor Statistics [4], while equipment cost estimates for the feedwater heater and pump were obtained from industry and vendor information sources [5][6].

Table 9: Cost of Equipment and Labor Comparison

	Lower Reheat Pressure	Open Feedwater Heater
Equipment	0	$200,000
Labor	$300	$25,000
Total	$300	$225,000

Section 7: Environmental Effects

In addition to economic considerations, the environmental impact of the power plant must be evaluated. As a coal-burning Rankine cycle, the base plant produces significant greenhouse gas emissions, primarily in the form of carbon dioxide. The carbon footprint of the plant is directly tied to the amount of coal consumed. Since the base plant generates 700 MW of power, it results in substantial daily coal usage and high emissions. These emissions can contribute to environmental concerns such as climate change and air quality degradation, making them important to consider when designing modifications to reduce coal consumption and improve overall efficiency.

On average, burning one ton of coal produces approximately 2.07 tons of carbon dioxide. The amount of carbon dioxide produced can be determined from the calculated coal consumption.

$$Emmisions\ of\ CO_2 = Coal\ Burned * 2.07$$

Table 10: Emmisions Comparison

Design Cycle	Emission Estimate
Base	18750
Low Pressure Base	18710.08
OFWH	18265.13

Section 8 : Decision Matrix

To compare the proposed plant configurations in a structured and objective manner, a decision matrix was developed using three key evaluation criteria: cost, environmental impact, and thermal efficiency. Each plant design was assessed based on its expected implementation expense, its effect on fuel consumption and emissions, and its ability to convert heat input into useful power output. By considering these factors together, the decision matrix provides a balanced method for identifying which modification offers the most practical overall improvement rather than focusing on performance in only one area. The scale follows the scale criteria of 1 = worst, 5= best.

Table 11: Decision Matrix

	weight	Base Plant	Lower Reheat Pressure	Open Feedwater Heater
Cost	40%	5	4	1
Efficiency	40%	1	2	5
Environmental Effects	20%	1	3	4
Total		2.6	3	3.2

Section 9: Conclusion

Based on the results of the decision matrix in table 11, the open feedwater heater configuration was selected as the preferred design despite its higher initial cost. While this option scored lower in the cost category because of the added expense of the feedwater heater, pump, and installation labor, it performed more favorably in both efficiency and environmental impact. The increase in thermal efficiency reduced the amount of coal required to produce the same power output, which in turn lowered fuel consumption and the associated emissions. Because the decision matrix considered all three criteria together, the feedwater heater option provided the strongest overall balance of performance and long-term benefit.

Although reducing the reheat pressure offered a lower-cost modification, the efficiency improvement from that change alone was relatively small compared to the gains achieved with the open feedwater heater. As a result, the fuel savings associated with the reheat-pressure change were not as significant. The feedwater heater option, on the other hand, requires a larger upfront investment but produces greater annual savings through reduced coal use. This means that, over time, the additional capital cost can be recovered, and the return on investment becomes more favorable than the lower-cost alternative.

For this reason, the open feedwater heater was determined to be the more practical improvement for the plant when evaluated from both a thermodynamic and economic perspective. Even though the initial implementation cost is higher, stronger efficiency gains and greater reduction in fuel usage make it a more advantageous long-term solution. The decision matrix supports this conclusion by showing that the higher upfront cost is outweighed by improved overall plant performance and the potential for a greater financial return over the operating life of the system.

References

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