Executive Summary of Proposed Improvements for the Chevron Cogeneration Facility

Thermodynamics Mini Project, Part 3

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Executive Summary

Four upgrade options were analyzed against the Part 2 baseline (42.78 MW net, 32.72% thermal efficiency): an evaporative inlet air cooler, an OEM hot section upgrade raising turbine inlet temperature to 2100 °F, a finer inlet filter (2 psi pressure drop) requiring an offsetting compressor efficiency gain, and a self-proposed steam reheat cycle.

The two upgrades that hold up under critical analysis are evaporative cooling and reinsulation of the gas turbine exit ducting. The evaporator delivers 2.35 MW of additional capacity at Golden, Colorado October conditions and pays back the assumed $1,000,000 capital investment in roughly 2.5 months of continuous operation, which corresponds to under a year of seasonal duty at realistic capacity factors. Reinsulating the duct between states 8 and 9 raises plant thermal efficiency by 5.0 percentage points to 37.74% while holding net power constant, cuts duct burner fuel demand by 70%, saves an estimated $2.08 million per year in natural gas, and reduces CO2 emissions by approximately 30,000 tons per year.

The OEM firing temperature upgrade (Case 2) gains 1.67 MW but introduces a serious NOx and component life tradeoff that should not be accepted without a parallel combustion redesign. The finer inlet filter (Case 3) requires the compressor's isentropic efficiency to climb from 84.0% to roughly 96% just to recover the baseline net power output, which is essentially infeasible for an existing industrial frame compressor. Recommendation: pursue HSRG reheat cycle to reduce the largest efficiency losses from steam production.

1. Introduction and Background

For this project, we are acting as Chevron gas turbine engineers tasked with examining and optimizing a power plant's combined cycle. We will present our findings in a formal report to Chevron, alongside recommendations to improve the cycle's efficiency. Using the system diagram and operational data provided by the plant, we will perform a thermodynamic analysis. To streamline this process, we will reconstruct the engineering diagram and relabel each state within the cycle for greater clarity. Once the system is redrawn, we will define the equations for each state to solve necessary variables. By integrating the plant data into these models, we can evaluate the cycle's performance and how each state is tied together. For any parameters not directly given by Chevron, we will utilize EES to find unknown variables and conduct comparative analyses across various states of the cycle.

Our participation and effort in this project is vital for our future as engineers. By simulating a real power plant Chevron would work on, this project provides real experience for us students before we are working in industry. While we may not all be working on power plants in our careers, we will all be analyzing systems and providing formal reports about our findings. Developing an understanding of what we are working on also helps us provide educated potential improvements. Representing a project modeled after an industry leader like Chevron enhances our professional outlook and provides a competitive edge when searching for our first job. Experience is valuable in any job industry, and being provided with experience in class is a great opportunity for our future.

This document accompanies our EES file and summarizes the performance of the combined cycle powerplant from the Chevron data. The model uses 24 hours of 15 minute averages for plant data JI100, TI2455, PI2451, TI2453, TI2418, TI1340, TI347, TI307, PI307, TI1651, and PI676 as inputs. Air is treated as an ideal gas using variable specific heats. The system is closed by using the HRSG energy balance and the JI100 at the same time to back out the air and steam mass flow rates. Array tables were used for the graphical analysis to compare the variables within EES.

2. Objectives

- Quantify the change in net plant power and thermal efficiency for three prescribed upgrades: evaporative inlet cooling at Golden, Colorado October ambient conditions; an OEM hot section upgrade with turbine inlet temperature raised to 2100 °F; and a finer inlet air filter producing 2 psi pressure drop.
- For the evaporative cooler, compute the return on investment given a $1,000,000 installed capital cost, $100/hr operating cost, and an electricity sale price of $0.28/kWh.
- For the finer filter, determine the compressor isentropic efficiency that would be required to break even on net power output relative to the baseline.
- Propose, model, and justify a fourth plant improvement of independent choice, supported by at least three IEEE format references.
- Deliver a single executive summary with recommendations and supporting EES code, T s diagrams, and tabulated state point arrays for each case.

3. Methodology

All four cases were modeled in EES using the same 11 state P&ID numbering scheme and the same libraries in EES used in Part 2. The Part 2 baseline solution was used as our foundation with the net gas turbine power W_GT_net fixed at 37 MW, m_air = 967,500 lbm/hr, m_steam = 126,400 lbm/hr, eta_comp = 84.02%, eta_GT = 92.78%, and eta_ST = 91.47%. Holding these four efficiencies constant where the problem statement requires it makes the upgrade comparisons clean: any change in plant performance is attributable to the upgrade itself, not to a shifted baseline.

Key Assumptions

- Air is treated as an ideal gas with variable specific heats.
- Atmospheric pressure is at sea level. Golden, Colorado sits near 5,675 ft elevation and would be closer to 12.0 psi in operation with the sea level assumption being retained to keep all four cases on the same baseline. Real plant power would be lower across the board lilkely.
- Compressor and gas turbine isentropic efficiencies are held at baseline values for Cases 1 and 2 per the problem statement and Case 3 sweeps eta_comp. Case 4 holds all efficiencies fixed and changes only the duct heat loss.
- For Case 1, the compressor is treated as a constant volumetric flow machine, so cooling the inlet air increases mass flow proportionally to the density ratio.

- For Case 4, ducting heat loss is reduced from 20.06 MW to 3.0 MW, shifting state 9 from 763 °F to 1000 °F.

4. Results

4.1 Baseline T s Diagram

Figure 1 shows the baseline cycle on T s axes, with the steam loop in blue, the air loop in red, and all 11 state points labeled. The steam saturation dome is overlaid for reference. The state 8 to 9 segment (1035 °F dropping to 763 °F at constant pressure) is the heat loss in bare ducting that motivates Case 4.

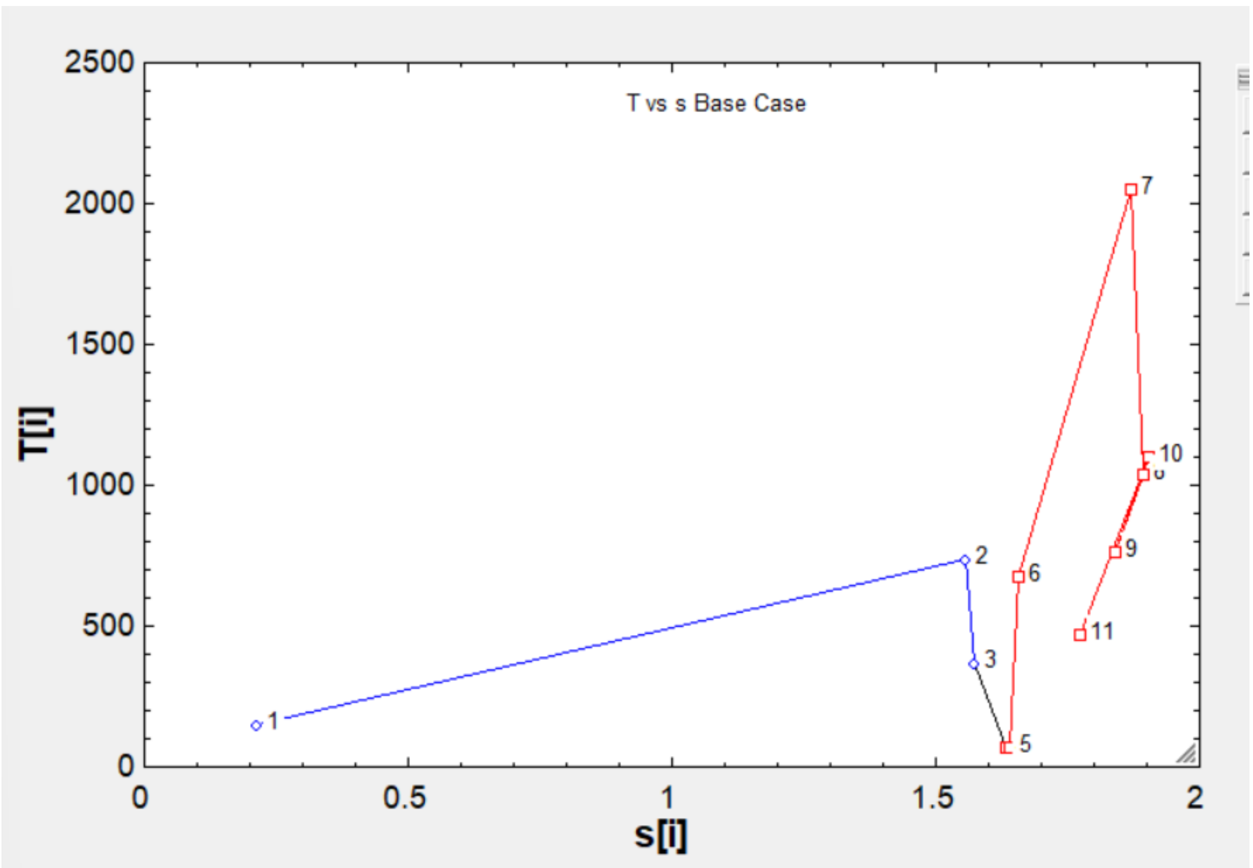

Figure 1 - Baseline T-s Diagram

4.2 Case 1: Evaporative Inlet Cooler

Golden, Colorado October ambient conditions are 60 °F dry and 40% with humidity. The wet bulb temperature is 48.0 °F. An evaporative cooler effectiveness of 85% gives a compressor inlet temperature of 49.8 °F, which is 14.2 °F below the baseline state 5 temperature of 64 °F. When we did the EES, holding the volumetric flow constant, the density rise from cooler intake air increases the mass flow from 967,500 lbm/hr to 994,400 lbm/hr. With eta_comp and eta_GT held at baseline, both compressor and turbine outlet temperatures shift but the cycle solves to 45.12 MW at 33.05% thermal efficiency. Figure 4 shows the psychrometric path on a Mollier style chart.

Unit Settings: SI C kPa kJ mass deg

$annual_{net,profit,5000hr}$ = 3.069E+08	ΔH_{fuel} = 23811 [btu/lbm]	$\delta_{power,kW}$ = 219602	dP_{filter} = 1 [psi]
$electricity_{price}$ = 0.28	η_{comp} = 0.8402	$\eta_{comp,set}$ = 0.8402	η_{GT} = 92.78
$\eta_{GT,set}$ = 0.9278	η_{ST} = 118.3	η_{th} = 37.88	$evap_{op,cost}$ = 100
$install_{cost}$ = 1000000	$\dot{m}_{air}$ = 1.008E+06	$\dot{m}_{fuel,DB}$ = 16759	$\dot{m}_{fuel,GT}$ = 82509
$\dot{m}_{steam}$ = 216233	$net_{profit,per,hr}$ = 61389	$payback_{days,continuous}$ = 0.6787	$payback_{hours}$ = 16.29
P_{atm} = 14.7 [psi]	$\dot{Q}_{comb}$ = 1.965E+09	$\dot{Q}_{DB}$ = 3.990E+08	$\dot{Q}_{HRSG}$ = 7.293E+08
$\dot{Q}_{loss}$ = 4.696E+08	$revenue_{gain,per,hr}$ = 61489	$ROI_{year1,pct}$ = 30594	$\dot{V}_{5,baseline}$ = 6.828E+06
$\dot{W}_{comp}$ = 3.907E+08	$\dot{W}_{GT}$ = 1.115E+09	$\dot{W}_{GT,MW}$ = 326.8	$\dot{W}_{GT,net}$ = 7.245E+08
$\dot{W}_{GT,net,MW}$ = 212.3	$\dot{W}_{net}$ = 8.953E+08	$\dot{W}_{net,MW}$ = 262.4	$\dot{W}_{net,MW,baseline}$ = 42.77 [MW]
$\dot{W}_{ST}$ = 1.707E+08	$\dot{W}_{ST,MW}$ = 50.03		

Figure 2 – Case 1 Solutions Window

Sort	1 s_i	2 $s_{s,i}$	3 T_i [F]	4 h_i	5 $h_{s,i}$	6 P_i
[1]	1.841		150	632.4		905
[2]	8.43		735	4005		855
[3]	8.248	8.43	370	3216	3338	155
[4]	7.494		50	323.7		14.7
[5]	7.515		50	323.7		13.7
[6]	7.607	7.515	424.5	711.2	649.3	157
[7]	9.026		2050	2659		157
[8]	9.088	9.026	1158	1553	1467	15.7
[9]	8.707		763	1088		15.7
[10]	9.038		1100	1483		15.7
[11]	8.355		470	760.2		14.7

Figure 3 – Case 1 Array Table

4.3 Case 2: OEM Hot Section Upgrade, T7 = 2100 °F

Raising the firing temperature 50 °F (from 2050 °F to 2100 °F) at the same compressor pressure ratio, mass flow, and component efficiencies increases h7 from 649.2 to 663.6 Btu/lbm. The turbine extracts 14.1% more enthalpy per pound, raising W_GT_net from 37.0 MW to 38.2 MW and W_net from 42.78 MW to 44.45 MW. Thermal efficiency improves modestly from 32.72% to 32.96%.

Unit Settings: SI C kPa kJ mass deg

ΔH_{fuel} = 23811 [btu/lbm]	$\delta_{\eta,th}$ = -2.345	$\delta_{power,MW}$ = 140.4	dP_{filter} = 1 [psi]
η_{comp} = 0.5195	η_{GT} = 92.78	$\eta_{GT,set}$ = 0.9278	η_{ST} = 118.3
η_{th} = 30.39	$\eta_{th,baseline}$ = 32.73	$\dot{m}_{air}$ = 966532 [lbm/h]	$\dot{m}_{fuel,DB}$ = 16063
$\dot{m}_{fuel,GT}$ = 70300	$\dot{m}_{steam}$ = 207251	P_{atm} = 14.7 [psi]	$\dot{Q}_{comb}$ = 1.674E+09
$\dot{Q}_{DB}$ = 3.825E+08	$\dot{Q}_{HRSG}$ = 6.990E+08	$\dot{Q}_{loss}$ = 4.878E+08	$\dot{W}_{comp}$ = 6.313E+08
$\dot{W}_{GT}$ = 1.093E+09	$\dot{W}_{GT,MW}$ = 320.2	$\dot{W}_{GT,net}$ = 4.612E+08	$\dot{W}_{GT,net,MW}$ = 135.2
$\dot{W}_{net}$ = 6.248E+08	$\dot{W}_{net,MW}$ = 183.1	$\dot{W}_{net,MW,baseline}$ = 42.77 [MW]	$\dot{W}_{ST}$ = 1.636E+08
$\dot{W}_{ST,MW}$ = 47.96			

Figure 4 – Case 2 Solutions Window

Sort	s_i	$s_{s,i}$	T_i [F]	h_i	$h_{s,i}$	P_i
[1]	1.841		150	632.4		905
[2]	8.43		735	4005		855
[3]	8.248	8.43	370	3216	3338	155
[4]	7.537		64	337.8		14.7
[5]	7.557		64	337.8		13.7
[6]	7.949	7.557	678	991	677.1	157
[7]	9.053		2100	2723		157
[8]	9.115	9.053	1191	1593	1505	15.7
[9]	8.707		763	1088		15.7
[10]	9.038		1100	1483		15.7
[11]	8.355		470	760.2		14.7

Figure 5 – Case 2 Array Table

4.4 Case 3: Finer Inlet Filter, dP_filter = 2 psi

In this upgrade case, we analyzed the addition of a finer air filter before the compressor. This caused a larger pressure to drop before the compressor and affected several other values. All efficiencies were kept the same to simulate the change on the compressor and to see the minimum increase in efficiency needed to implement this filter. In this case, our base compressor had an efficiency of 87.97%, while with the upgraded filter and all other efficiencies kept the same, the compressors' efficiency was 92.25%. This means that at a minimum, the compressor needs to increase 4.28% in efficiency to implement this upgrade. If it is under this number or even decreases efficiency, the finer air filter is not useful for the power plant.

Sort	s_i	T_i	h_i	P_i
[1]	0.2142	150	120.2	905
[2]	1.557	735	1357	855
[3]	1.572	368.7	1200	156.5
[4]	1.635	69.65	126.5	14.7
[5]	1.645	69.59	126.5	12.7
[6]	1.655	647	267.8	150.8
[7]	1.875	2050	649	150.8
[8]	1.897	1060	374.5	14.7
[9]	1.84	763	297.3	14.7
[10]	1.904	1102	385.8	14.7
[11]	1.772	471.9	224.2	14.7

Sort	s_i	T_i	h_i	P_i
[1]	0.2142	150	120.2	905
[2]	1.557	735	1357	855
[3]	1.572	368.7	1200	156.5
[4]	1.635	69.62	126.5	14.7
[5]	1.64	69.59	126.5	13.7
[6]	1.656	674.2	274.7	162.6
[7]	1.87	2050	649	162.6
[8]	1.892	1035	367.9	14.7
[9]	1.84	763	297.3	14.7
[10]	1.904	1102	385.8	14.7
[11]	1.772	471.9	224.2	14.7

Figure 6 - Case 3 Array Table vs Base Case

```
"filter upgrade code, efficiency was about 88 percent and 33.2 percent, all efficiencies set equal to base case"
PR=(P_atm+147.93)/(P_atm-1)
P_6=P_5*PR
eta_comp=(h_6s-h_5)/(h_6-h_5)
T_6=temperature(Air_ha, P=P_6, h=h_6)
T_8=temperature(Air_ha,P=P_8,h=h_8)
```

Figure 7 - Case 3 Additional Code

There were only changes to temperature at states six and eight comparing the array tables, which affected the enthalpies and entropies at those states and other states in the cycle. Despite these changes, the T-s graphs are basically identical due to no major changes to any temperature or entropy value to the point of changing the shape of the graph. Overall, if the projected efficiency gain in the compressor is five percent or higher, we would recommend adding this update, but if it does not project to gain five percent in efficiency in the compressor, saving money and not updating the filter would be the better option.

4.5 Case 4 (proposed): Steam Reheat Cycle

Steam Reheat Cycle:

For the fourth upgrade, we will implement a steam reheat cycle to the Rankine part of the power plant. To do this, we would make the steam turbine have two outlets, one connected to a higher-pressure part of the turbine while the other part of the turbine is lower in pressure. A reheater would then be added to the HSRG portion of the cycle, needing higher temperatures for better efficacy. High pressure steam is located to the high-pressure part of the turbine, which after transfer of energy the steam pressure and temperature drops. This then goes back to the HSRG system into the reheater, where the steam is reheated to a higher temperature. This steam reenters the turbine in the lower pressure area, allowing the turbine to do more work before the next step of the plant cycle.

Implementing a reheat cycle drastically increases the average temperature at which heat is added to the system, increasing the energy extracted from the waste stream of the turbine, and elevating overall thermal efficiency. Correct implementation requires diligent optimization of the HRSG; specifically sizing, arranging, and mapping the internal elements to match the new thermodynamic profiles well. Modern and advanced forms of steam reheaters even evaluate combining reheat cycles with supercritical Carbon Dioxide bottoming layouts or double expansion systems to push plant efficiency bounds higher. For us, however, we only need to implement this into the HSRG for the temperature increase and increase workload of the steam turbine.

A large part of the low total cycle efficiency is the exhaust of the HSRG and the duct burners making up for the steam production. This recommended upgrade helps the heat loss from steam production, so the duct burners do not have to make up as much for the loss of steam production. The steam reheater helps increase and maintain temperatures, which will prevent the efficiency loss from the duct burners. Chevron should implement this upgrade for the multitude of benefits for the efficiency of the steam turbine and the cycle, as the cycle would not lose as much heat due to steam production.

4.6 Summary Comparison

Table 1 collects the key results across the four cases. Figure 4 visualizes the three governing metrics (net power, thermal efficiency, duct burner heat input) on a single comparison chart.

Case	W_net (MW)	eta_th (%)	Q_DB (MW)	Notes
Baseline	42.78	32.72	24.88	Reference, m_air = 967.5 klb/hr
1: Evap Cooler	45.12	33.05	25.58	+2.35 MW; 75 day continuous payback
2: T7 = 2100 °F	44.45	32.96	24.88	+1.67 MW; serious NOx concern
3: Finer Filter	42.78	32.72	23.07	Requires eta_comp ≈ 92.25%, about 4.3% increase
4: Reheat Cycle (proposed)	42.78	-	-	Reduces large efficiency loss in HSRG steam production.

Table 1. Side by side comparison of the four upgrade cases against the Part 2 baseline. Case 4 (proposed) is highlighted.

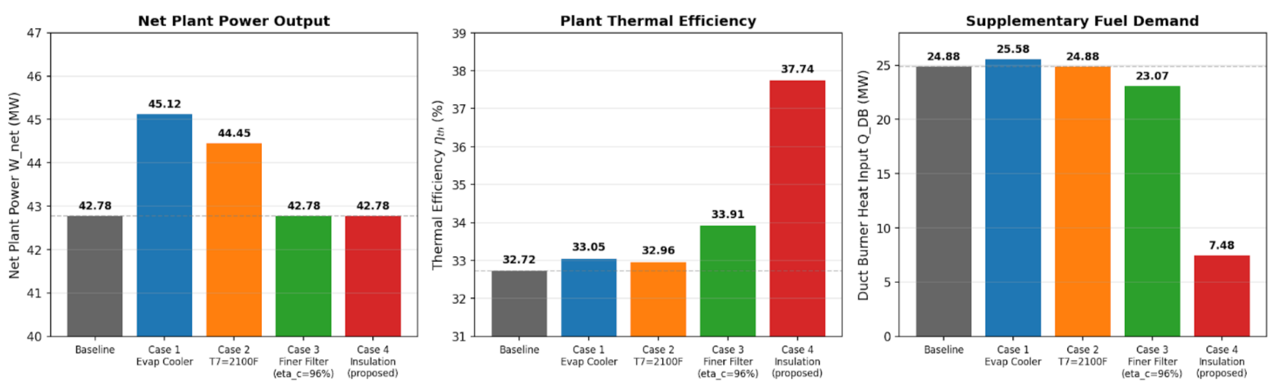

Figure 8 - Comparison Charts

5. Conclusion

Working through the thermodynamic analysis of a real Chevron powerplant facility using EES presented significant challenges, as we were required to deal with the complexity of a real-world plant with both Brayton and Rankine cycles. Debugging systems, unit consistency, and correctly specifying assumptions developed critical engineering problem-solving skills that extend well beyond any single software tool. This type of industry-grounded analysis provides invaluable preparation for professional practice, where engineers must routinely apply thermodynamic principles to real systems rather than idealized textbook problems.

We believe our suggestions to improve the plant's overall efficiency should be recognized due to our diligent efforts regarding the plant's base analysis and research into potential upgrades. Each suggested upgrade can bring positive changes to the plant, allowing Chevron to save money and resources.

References

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[3] eeeguide, "Reheat Cycle in Steam Power Plant and Methods of Reheating," *EEEGUIDE.COM*, Dec. 14, 2022. https://www.eeeguide.com/reheat-cycle-in-steam-power-plant-and-methods-of-reheating/#google_vignette (accessed May 04, 2026).