Thermodynamic Analysis of a Regenerative Reheat Rankine Cycle with a Secondary Loop

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ABSTRACT

The Rankine Cycle is fundamental to the operation of utility-scale power plants. While the simplest configuration of the cycle employs only four devices (pump, turbine, boiler, condenser), such a cycle leaves additional efficiency on the table. Greater efficiency can be achieved by adding reheat processes, feedwater heaters, and an Organic Rankine Cycle with a lower boiling point fluid. These additions increase the cost and complexity of the cycle, but also increase the efficiency. At the utility-scale, the latter is vastly more important. To analyze and optimize a system with a closed feedwater heater (C.F.H.), open feedwater heater (O.F.H.), reheat, and Organic Rankine Cycle (ORC), a script is presented based upon thermodynamic laws, project constraints, and guiding assumptions. This script is run using the Engineering Equation Solver (EES) and a number of input parameters, and outputs a measure of overall thermal efficiency. Using the model as a guide, an optimal set of state variables is reached for maximum thermal efficiency.

Nomenclature

P Pressure of fluid [kPa].
T Temperature of fluid [C].
x Quality of fluid [-]. When superheated or subcooled, the quality is defined as 100 [-] or -100 [-], respectively.
s Real specific entropy of fluid [kJ/kg-K].
s_{ideal} Ideal specific entropy of fluid [kJ/kg-K].
h Real specific enthalpy of fluid [kJ/kg].
h_{ideal} Ideal specific enthalpy of fluid [kJ/kg].
m_{main} Mass flow of main line in steam system [kg/s].
m_{hex} Mass flow of refrigerant loop [kg/s].
y Mass flow fraction released to the CFH [-].
z Mass flow fraction released to the OFH [-].
η_{th} Overall thermal efficiency [-].
η_{pump} Isentropic efficiency of pumps [-].
$\eta_{turbine}$ Isentropic efficiency of turbines [-].
Q Thermal power transferred through a device [kW].
W Work power transferred through a device [kW].
HEX Heat Exchanger.

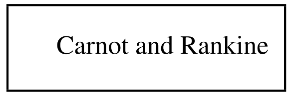

Fig. 1. Carnot and Rankine Cycles

C.F.H. Closed Feedwater Heater.
O.F.H. Open Feedwater Heater.
ORC Organic Rankine Cycle.

1 Introduction

Modern society owes itself to the study of thermodynamics, and specifically the Rankine Cycle. Standard of living depends on products made at high-scale and low-cost, and this type of production relies on cheap and accessible energy. The easiest way that humanity has found to produce energy in bulk is by using thermodynamic cycles. As such, it seems no coincidence that most innovations in thermodynamics occurred right before and during the Second Industrial Revolution [1]. The quantities of energy necessary to move humanity forward depended on an understanding of thermodynamics. While other forms of utility-scale power generation have evolved today (Photovoltaics, hydro, wind, etc.), the vast majority of power generation is still done using thermal power plants and thermodynamic cycles [1].

In order to generate work in a thermal power plant, thermodynamic cycles utilize a temperature differential created by a fuel (coal, gas, or radioactive material in a reactor). To maximize the work per heat generated (thermal efficiency), careful consideration must be taken in the selection of the cycle. The Carnot Theorem provides guidance by predicting an absolute maximum of thermal efficiency [1].

The Rankine Cycle, named after Scottish engineer William Rankine, uses the phase change of a working fluid to create isobaric processes during heating and cooling [1]. That, combined with roughly isentropic processes at the pump and turbine, makes the T-S diagram of the Rankine Cycle closely mimic that of the Carnot Cycle, resulting in decent thermal efficiency. While this is the case, there are ways to increase the efficiency further. The analysis presented uses three methods: A secondary loop, feedwater heaters, and a reheat process. Using all of these additional methods at once vastly increases the complexity of the cycle, requiring higher levels of analysis in order to maximize thermal efficiency.

2 Methodology

To perform an efficiency optimization of state variables and mass flow rates, the Engineering Equation Solver (EES) serves as a useful tool. EES provides simple lookups on state information, and can solve a system easily given enough relations are supplied. Additionally, EES allows for modulating inputs for optimization using parameter tables.

2.1 Constraints

As per the project description, the following constraints must be followed:

1. The secondary loop will use R-134a
2. Maximum turbine inlet P = 25 [MPa] and T = 600 [C]
3. Minimum turbine quality of 90%
4. Maximum reheat temperature of T = 600 [C]
5. Pumps cannot pump any vapor
6. Ambient temperature of T = 25 [C]
7. Minimum temperature differential for condenser, heat exchanger, and closed feedwater heater of 15 [C]
8. Isentropic efficiencies of pumps and turbines as 60% and 85%, respectively

Additionally, the first and second laws of thermodynamics should not be violated by our analysis. The only processes that decrease entropy must be ones that remove heat from the working fluid, and the sum of heats and works in and out should be equal to 0. These two laws provide useful assertions for the analysis.

2.2 Thermodynamic Assumptions

To simplify the analysis, the following assumptions will be made:

1. Gravitational potential and kinetic energies are small relative to the internal energies of the working fluids, and can be ignored.

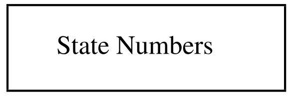

Fig. 2. State Numbers

2. All plumbing between devices contains fluid of constant state (i.e. no friction from fluid flow that would increase entropy).
3. Condensers, boilers, heat exchangers and feedwater heaters are isobaric devices.
4. The steam trap is an isenthalpic device.
5. The closed feedwater heater, boiler, and heat exchanger lose no heat to the environment.

2.3 Calculating Heats and Works

The power moving into or out of the working fluid can be measured as a difference in specific enthalpies multiplied by the mass flow rate of the fluid and the fraction of the total mass flow (dictated by y, z, and their complements). This is reasonable given the assumptions made in the analysis. Whether the power constitutes work or heat depends on the device being analyzed.

In the EES script, power direction is handled such that all values of Q and W come out as positive, given basic assumptions about which way energy should be flowing. That way, a negative sign in any work or heat quantity raises alarm. The proper signage is handled when calculating net work and thermal efficiency. Examples are shown below:

$$W_{turbine,LP2} = m_{main} * (1 - y - z) * (h[12] - h[13]) \tag{1}$$

$$W_{pump,HP} = m_{main} * (h[6] - h[5]) \tag{2}$$

$$Q_{boiler,1} = m_{main} * (h[8] - h[7]) \tag{3}$$

$$Q_{out} = m_{hex} * (h[17] - h[14]) \tag{4}$$

$$Q_{hex} = m_{hex} * (h[16] - h[15]) \tag{5}$$

The heat exchanger heat will be equal for each side, and the closed feedwater heater operates as a heat exchanger. The reheat portion of the boiler is defined as $Q_{boiler,2}$.

2.4 Isobaric Devices

For a device that is assumed isobaric, the states across the device are assigned equal pressure. An example is shown below:

$$P[14] = P[17] \tag{6}$$

For the open feedwater heater, there are four states with equal pressure.

2.5 Isenthalpic Trap

The steam trap is isenthalpic. The relation is intuitive:

$$h[3] = h[4] \tag{7}$$

2.6 Open Feedwater Heater

In addition to operating at a constant pressure, the O.F.H. will experience no change in enthalpy. As such, we can relate the enthalpies entering and exiting the O.F.H. using mass flow fractions and specific enthalpies:

$$(z * h[12]) + ((1 - y - z) * h[2]) + (y * h[4]) = h[5] \tag{8}$$

2.7 Inefficiencies of Pumps and Turbines

Using two state variables at the inlet and pressure at the outlet, it is possible to use isentropic efficiency to find outlet state information. To do this, specific entropy is determined at the input. Then, ideal enthalpy at the outlet is determined assuming ideal entropy. Finally, the real enthalpy at the outlet is determined using the isentropic efficiency. This final relation varies between pumps and turbines:

$$\eta_{pump} = \frac{h[5] - h_{ideal}[6]}{h[5] - h[6]} \tag{9}$$

$$\eta_{turbine} = \frac{h[8] - h[9]}{h[8] - h_{ideal}[9]} \tag{10}$$

2.8 Known-Goods for Optimization

With fundamental relations established, additional parameters can be set based on thermodynamic intuition:

1. Heat flow across a temperature differential produces entropy that scales with the size of the differential. As such, an optimal system will use the minimum allowable temperature differences for HEX/C.F.H./condenser operation. The outlet temperatures are set with a 15 [C] difference (the condenser is set to 15 [C] above ambient).
2. It is wasteful to cool working fluid beyond the saturation line. As such, the qualities for all three pump inlets are set to 0 [-].
3. Using the maximum temperature differential for the entire system is optimal (raises Carnot efficiency). As such, the first turbine inlet state is set to maximum allowable temperature and pressure (superheat).
4. The maximum reheat temperature is optimal for the main loop, as it drops the heat removal (hex) pressure.
5. In order to maximize the heat addition pre-boiler, the y-fraction should dump as much heat through the C.F.H. as possible. The y-fraction quality after the C.F.H. is set to 0 [-].
6. For the ORC, it is ideal to not superheat the refrigerant vapor. The pre-turbine refrigerant quality is set to 1 [-].
7. Mass flow rate of the main loop will simply scale heats and works evenly, resulting in the same overall efficiency. For simplicity, the mass flow for the water is set to 1 [kg/s].

2.9 Optimization of Remaining Parameters

From this point, there remain 4 input parameters to determine a solution (in addition to many thermodynamic state variable function calls).

In the final analysis, the following variables are chosen for optimization:

1. Pressure at the C.F.H. (P[9])
2. Pressure at the reheat section of the boiler (P[10])
3. Pressure of the O.F.H. (P[12])
4. Pressure at HEX (water-side) (P[13])

The first two parameters are optimized in tandem while the latter two are set to 200 and 100 [kPa], respectively. Once an ideal set of the first two parameters is reached, the O.F.H. pressure and then the HEX pressure are optimized.

Table of States

Fig. 3. Table of States

Final TS diagram

Fig. 4. Table of States

3 Results and Discussion

4 Conclusions

Following the methodology prescribed leads to a thermal efficiency of $\eta_{th} = 0.423[-]$, a C.F.H. flow fraction of $y = 0.082[-]$, and a O.F.H. flow fraction of $z = 0.078[-]$. Under these specifications, the ORC runs with 14.8 times the mass-flow of the main cycle. The following four input pressures are:

1. Pressure at the C.F.H: 1500 [kPa]
2. Pressure at the reheat: 1500 [kPa]
3. Pressure of the O.F.H: 340 [kPa]
4. Pressure at HEX (water-side): 35 [kPa]

Based on the constraints, assumptions, and known-goods established in the model, efficiency is maximized at about a zero pressure-drop between the C.F.H. and the reheat. Therefore, an optimized Rankine cycle of the assigned structure requires no second high pressure turbine. Removing the device would likely decrease the implementation/maintenance costs and remove a point of failure from the cycle. Moving forward, further consideration should likely be taken into maximum allowable mass-flow rates, in order to determine whether the ORC flow speed predicted my the model is truly reasonable. Additionally, a cost analysis could be performed to weigh the cost of additional devices against the gains of additional efficiency.

Acknowledgements

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References

[1] *Rankine Cycle - Steam Turbine Cycle. Nuclear Power.* Nuclear Power for Everybody.