Analysis of Rankine Heat Engine Cycle and Optimization of Cycle Efficiency Between Maximum and Minimum Working Pressures

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Abstract— The analysis of Rankine Heat Engine Cycle illuminates the crucial factors affecting the performance. By systematically varying of boiler and condenser pressures, we can identify optimal pressure ranges that significantly enhance the cycle's efficiency. The valuable findings hold immense practical value in power plant design and operation, where maximizing efficiency translates directly to minimizing both operational costs and environmental impact. Continued advancements in the Rankine cycle, or exploration of alternative power generation cycles, are essential to achieve the highest possible efficiency and ensure long-term sustainability. Analysis and plotting of Rankine Cycle using MATLAB eases the understanding while producing the expected accurate results.

Keywords— Rankine Heat Cycle, MATLAB, Optimization, Maximization of Efficiency, Power Generation Cycles

I. OBJECTIVES OF ANALYSIS

This OEL aims to analyse and optimize the performance of a Rankine cycle heat engine for maximum efficiency within given operating constraints. The main targets of the analysis ase:

- Analyze the cycle for various combinations of boiler and condenser pressures.
- Maintain constant degrees of superheating and subcooling across all simulations.
- Develop characteristic curves for the following parameters:
- Identify the operating conditions (boiler and condenser pressures) that lead to maximum cycle efficiency within the given constraints.
- Utilization of MATLAB interface for analysis of the cycle for different pressure combinations and generate T-s diagrams for visualization.

II. INTRODUCTION TO RANKINE CYCLE

The Rankine cycle is an idealized thermodynamic cycle which effectively describes the process by which the common use heat engines, predominantly steam turbines or reciprocating steam engines, allow mechanical work to be extracted from a fluid as it moves between a heat source and heat sink. The Rankine cycle is named after William John Macquorn Rankine, a Scottish polymath professor at Glasgow University.

Deliberating much into the Rankine cycle, it closely describes the process by which steam engines are usually found in thermal power generation plants, which utilize the thermal energy of a fuel or other heat source to generate electricity. Generally, the common use heat sources include combustion of fossil fuels such as coal, natural gas, and oil, use of mined resources for nuclear fission, renewable fuels like biomass and ethanol, and energy capture of natural sources such as concentrated solar power and geothermal energy. Heat sinks, which serve to transfer the excessive heat generated to colder surfaces predominantly comprise of ambient air above or around a facility and bodies of water such as rivers, ponds, and oceans.

Generally, the operation of Rankine engines is within a closed loop for which the working fluid (fuel) is reutilized for numerous repetitions. The water vapor with condensed droplets often seen swirling from power stations is created by the cooling systems (not directly from the closed-loop Rankine power cycle). This is generally referred to as exhaust heat Q_{out} flowing out of the lower side of the cycle. Generally, the purpose of cooling towers is to act as a gigantic heat exchanger, which play into action by the absorption of latent heat of vaporization and consequently expelling the vapours into the environment.

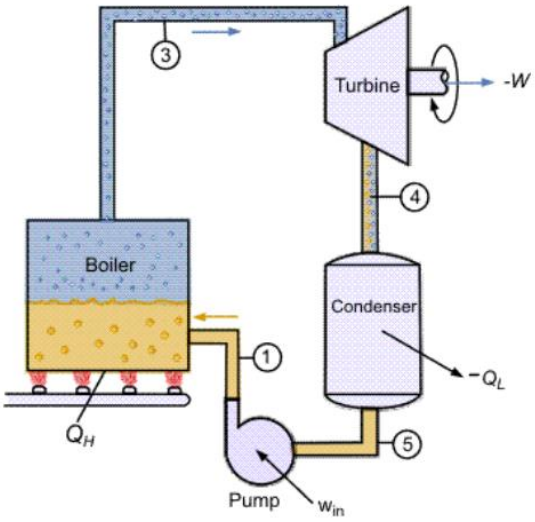


Figure 1 General Schematic of Rankine Cycle

III. PROCESSES IN RANKINE CYCLE

A Rankine Cycle operates in four different processes, which are as follows: -

- Isentropic Compression Process (Curve 1-2)
- Isothermal Heat Addition Process (Curve 2-3)
- Isentropic Expansion Process (Curve 3-4)
- Isothermal Heat Rejection Process (Curve 4-1)

A. Isentropic Compression Process (Curve 1-2):

In this process, the pump begins the cycle by pumping the working fluid from region of lower to higher pressure. As the fluid is a liquid at this stage, the pump requires little input energy. Compression of the fluid is performed by the pump.

B. Isothermal Heat Addition Process (Curve 2-3):

In this process, high-pressure liquid enters the boiler where it undergoes heating under constant pressure condition by an external heat source to transit towards dry saturated vapour (superheated vapours). Regarding the required input energy, it can successfully be evaluated by referring to EES/MATLAB, graphs, or charts.

C. Isentropic Expansion Process (Curve 3-4):

In the isentropic expansion process, the dry saturated vapour (superheated vapours) undergo expansion when passed through a turbine, consequently generating power. In this stage, a significant decrease in the temperature and pressure of the vapour is observed and some condensation may occur. The output in this process can be easily calculated using the chart or tabular data.

D. Isothermal Heat Rejection Process (Curve 4-1):

As the name itself suggests, this stage involves colling of steam into vapours. The wet vapour enters a condenser, where it is condensed at a constant pressure to become a saturated liquid.

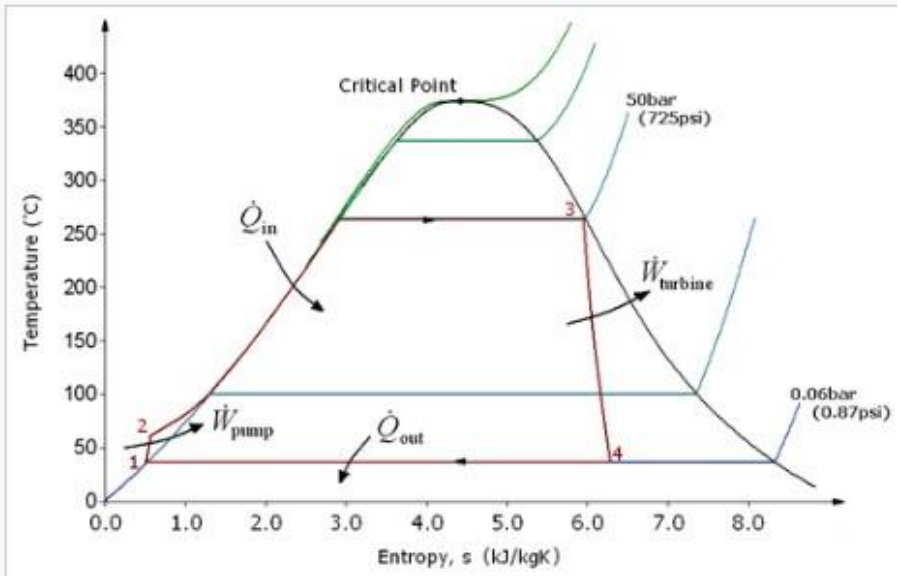


Figure 2 T–s diagram of a typical Rankine cycle operating between pressures of 0.06 bar and 50 bar.

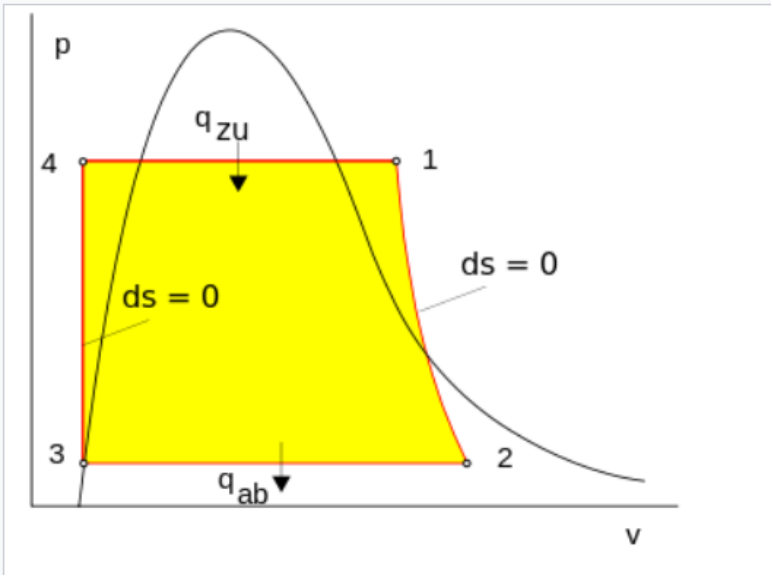


Figure 3 Pressure-Volume Curve of Rankine Cycle

It must be well pertained that the actual vapor power cycle differs from the ideal Rankine cycle owing to the presence of irreversibility factors, dominantly consequent of fluid friction and heat loss to the surroundings. Due to the dominance of fluid friction, pressure drops in the boiler, the condenser, and the piping between the components are observed and, consequently, the steam leaves the boiler at a lower pressure; heat loss reduces the net output work, thus heat addition to the steam in the boiler is required to maintain the same level of net work output.

IV. VARIABLES AND EQUATIONS INVOLVED IN RANKINE CYCLE

Table 1 Variables and Equations Involved in Rankine Cycle

Variables	Explanation
$\dot{Q}$	Heat flow rate (energy per unit time)
$\dot{m}$	Mass Flow Rate
$\dot{W}$	Mechanical power consumed by or provided to the system (energy per unit time)
η_{therm}	Thermodynamic efficiency of the process (net power output per heat input, dimensionless)
$\eta_{\mathrm{pump}}, \eta_{\mathrm{turb}}$	Isentropic efficiency of the compression (feed pump) and expansion (turbine) processes, dimensionless
h_1, h_2, h_3, h_4	The "specific enthalpies" at indicated points on the T–s diagram
h_{4s}	The final "specific enthalpy" of the fluid if the turbine were isentropic
p_1, p_2	The pressures before and after the compression process

$$\frac{\dot{Q}_{\mathrm{in}}}{\dot{m}} = h_3 - h_2,$$

Equation 1 Heat Flow Rate (Into the System)

$$\frac{\dot{Q}_{\mathrm{out}}}{\dot{m}} = h_4 - h_1,$$

Equation 2 Heat Flow Rate (Outside System)

$$\frac{\dot{W}_{\mathrm{pump}}}{\dot{m}} = h_2 - h_1,$$

Equation 3 Work done by the Pump

$$\frac{\dot{W}_{\mathrm{turbine}}}{\dot{m}} = h_3 - h_4.$$

Equation 4 Work done by Turbine

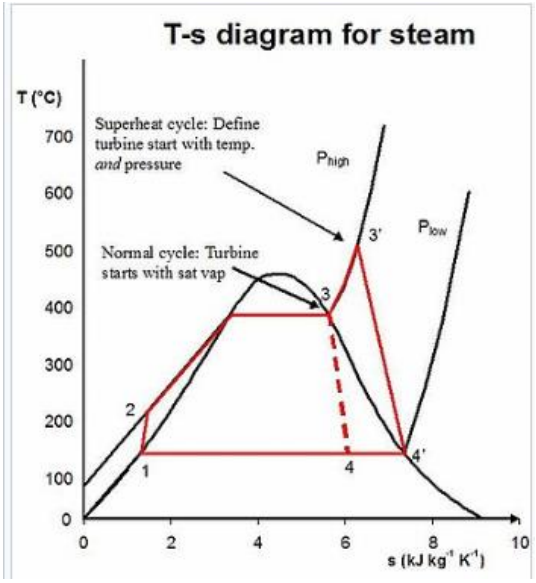
$$\eta_{\mathrm{therm}} = \frac{\dot{W}_{\mathrm{turb}} - \dot{W}_{\mathrm{pump}}}{\dot{Q}_{\mathrm{in}}} \approx \frac{\dot{W}_{\mathrm{turb}}}{\dot{Q}_{\mathrm{in}}}$$

Equation 5 Thermal Efficiency Formula for Rankine Cycle

V. REAL RANKINE CYCLE (NON-IDEAL RANKINE CYCLE)

Referring to the real power-plant cycle, the compression action and the expansion action performed by the pump and turbine respectively are not isentropic in nature. Put differently, these processes are non-reversible in nature, and entropy is increased significantly during the two processes. This somewhat increases the power required by the pump and decreases the power generated by the turbine.

In particular, the efficiency of the steam turbine shall significantly decrease owing to water-droplet formation and minute extent of cavitation. Upon the condensation of water, the droplets hit the turbine blades at relatively high speed, causing pitting and erosion, gradually decreasing the life of turbine blades and efficiency of the turbine. Superheating the steam an optimum solution to the problem. Referring to the cycle diagram, state 3 is at a border of the two-phase region of steam and water, so after expansion the steam will be very wet. By superheating, state 3 will move to the right (and up) in the diagram and hence produce a drier steam after expansion.



Equation 6 Non-Ideal Rankine Cycle (With Superheating Action)

VI. RANKINE CYCLE WITH REGENERATION

After the working fluid emerges from the condenser (most likely subcooled liquid) it is heated by steam which is brought from the hot portion of the cycle by tap system. Referring to the diagram attached below, the fluid at 2 is mixed with the fluid at 4 (isobaric condition dominates as the pressures are constant) to end up with the saturated liquid at 7. This is called "direct-contact heating". The Regenerative Rankine cycle (with minor variants) is commonly used in real power stations. Another variation sends bleed steam from between turbine stages to feedwater heaters to preheat the water on its way from the condenser to the boiler. These heaters do not mix the input steam and condensate, function as an ordinary tubular heat exchanger, and are named "closed feedwater heaters". The heat cycle input temperature is increased significantly by the regeneration process by elimination of the heat addition from the boiler/fuel source at the relatively low feedwater temperatures that would exist without regenerative feedwater heating. This improves the efficiency of the cycle, as more of the heat flow into the cycle occurs at higher temperature.

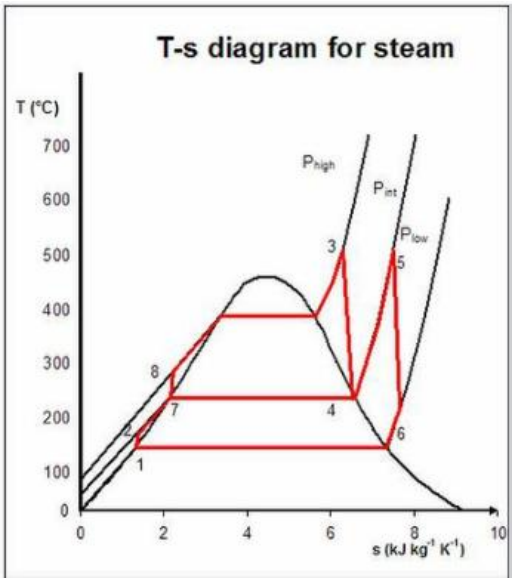


Figure 4 Regenerative Rankine Cycle

VII. INEFFICIENT BEHAVIOUR OF RANKINE CYCLE AND POSSIBLE SOLUTIONS

The efficiency of the Rankine cycle is greatly restricted by the relatively high value of heat of vaporization of different fluids. There must exist an adequate cycle for reutilization of fuel continuously. Hence, water is the

best working fluid for the cycle due to extent of availability. Upon the condensation of water in condenser, water vapours carry away the excessive generated heat which may be observed escaping through the cooling towers. This waste heat is necessary in any thermodynamic cycle. Due to this condensation step, the pressure at the turbine outlet is lowered. This means the pump requires less work to compress the water resulting in higher overall efficiencies.

Aforementioned are some reasons why the Rankine Cycle has relatively less efficiency:

- The combustion temperature for gases is usually around 1000°C and 2000°C, which is considerably higher than the highest vapor temperatures. The transfer of heat across a large temperature difference increases the entropy hence more disorder within the system.
- Considering the feasible temperatures, which are relatively easy to be achieved, the combustion factor is ruled out due to technicalities.

Due to the finite value of heat transfer through the condenser system, the condensation shall take place at a temperature higher than the temperature of the cooling medium. Again, heat transfer occurs across a temperature difference, causing the generation of entropy. The deposition of dirt in condensers during operation with cooling water reduces the efficiency.

Following are some methodologies which maybe adopted in order to improve the overall efficiency of the system:

A. Increasing the Pressure Difference:

Increase in the pressure difference helps to improve the efficiency of the system. The area under the curve of P-V plot shall significantly increase, leading to the rise in the efficiency of the system.

B. Superheating and Reheating:

The irreversibility of any process is reduced if it is performed as close as possible to the temperatures of the high temperature and low temperature reservoirs. This is achieved by operating the condenser at sub atmospheric pressure. The temperature in the boiler is limited by the saturation pressure. Further increase in temperature is possible by superheating the saturated vapor. This has the additional advantage that the vapor quality after the turbine is increased and, therefore the erosion of the turbine blades is reduced. It is quite common to reheat the vapor after expansion in the high-pressure turbine and expand the reheated vapor in a second, low pressure turbine.

C. Feedwater Reheating:

The cold liquid leaving the feed pump is mixed with the saturated liquid in the boiler and/or re-heated to the boiling temperature. The resulting irreversibility reduces the efficiency of the boiler. According to the Carnot process, the highest efficiency is reached if heat transfer occurs isothermally. To preheat the feed liquid to its saturation temperature, bleed vapor from various positions of the turbine is passed through external heat exchangers (regenerators), as shown in Figure 5. Ideally, the temperature of the bleed steam should be as close as possible to the temperature of the feed liquid.

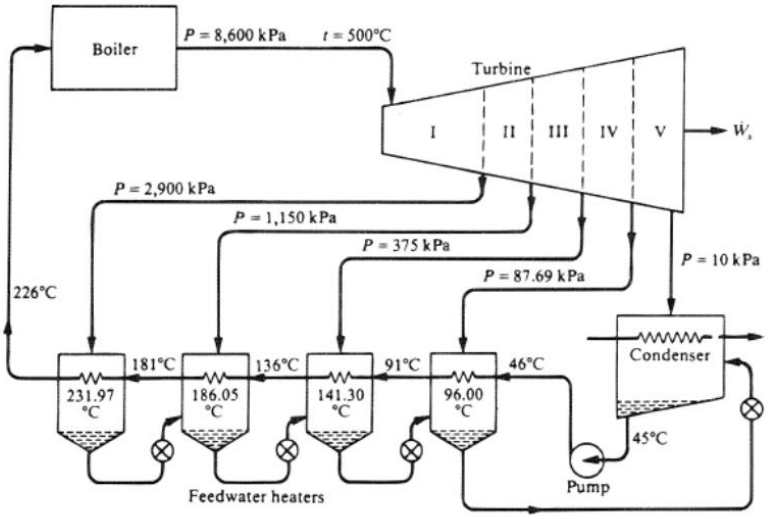


Figure 5 Regenerative feed liquid heating.

VIII. CONSTRAINTS IN WORK DONE BY VARYING BOILER/CONDENSER PRESSURES

There are certain reasons why increasing the boiler and/or condenser pressures render no useful services in increasing the magnitude of work done. Certain reasons shall be analysed as follows in appropriate manner:

A. *Material limitations:*
Extremely high-pressure systems require very strong and expensive components to handle the stress.

B. *Pump work:*
Increasing boiler pressure also necessitates a larger pressure increase in the pump (1-2). This consumes more energy, reducing overall efficiency.

C. *Moisture content in turbine:*
High-pressure expansion (3-4) can lead to excessive moisture in the low-pressure turbine stages. This moisture can damage turbine blades due to erosion.

D. *Condenser limitations:*
Extremely low condenser pressure is desirable, but it's limited by the ambient temperature and the cost of removing non-condensable gases.

Thermodynamic constraints also play a significant role in the regard. Some points are asserted as follows:

E. *Carnot Limit*:
Even with ideal components, there's a theoretical limit to efficiency (Carnot efficiency) defined by the temperature difference between the boiler and condenser. Increasing pressure alone doesn't necessarily improve this temperature difference.

F. *Isentropic Efficiency:*
Real pumps and turbines aren't perfectly isentropic (processes 1-2 and 3-4). Increasing pressure can exacerbate these inefficiencies, reducing actual work output.

Mechanical aspects also come into play in decreasing the yield of work done.

G. *Lubrication challenges:*
High-pressure environments can make it difficult to properly lubricate moving parts in pumps and turbines, leading to increased wear and tear.

H. *Maintenance costs:*
High-pressure systems require more frequent and specialized maintenance, raising operational costs.

I. *Environmental impact:*
Higher pressure systems can potentially lead to larger leaks, which can be environmentally hazardous if the working fluid is toxic or flammable.

IX. OPTIMIZATION TECHNIQUES

Bringing into aspects for improving the yield work, certain optimization techniques may be brought into consideration. Some of them shall be discussed as follows:

- *Superheating steam (state 3):*
Heating steam beyond its saturation temperature (point 3) reduces moisture content at the turbine exit, preventing blade erosion.
- *Reheat cycle:*
Adding heat midway through the expansion (3-4) improves efficiency by utilizing more thermal energy.
- *Cooling tower technology:*
Efficient cooling towers help achieve lower condenser pressure, raising the theoretical efficiency limit.
- *Open Feedwater Heater:*
By extraction of some steam (state x) between the turbine stages (3-4) and utilizing it to preheat the feedwater (state 1) entering the boiler (point x to 1) certainly can help in reducing the amount of heat required in the boiler, improving efficiency.

X. CODE INITIALIZATION

The following MATLAB code simulates the Rankine cycle and computes its performance characteristics. This is a breakdown of the initializations:

A. Adding of Path:

```
addpath('C:\Users\SURFACE BOOK\Downloads\XSteam_Matlab_v2.6\')
```

This line includes the path to the XSteam library in MATLAB's search path. This library includes functions for determining the thermodynamic parameters of water, which are required for Rankine cycle computations.

B. User Inputs:

C.

```
cond_pressure = "Enter condenser pressure (bar): ";
cond_pressure = input(cond_pressure);
boiler_pressure = "Enter boiler pressure (bar): ";
boiler_pressure = input(boiler_pressure);
```

These lines ask the user to enter the condenser pressure and boiler pressure for the Rankine cycle simulation. The input function takes the prompt string and stores the user's input in the corresponding variables (cond_pressure and boiler_pressure).

D. Constants:

```
max_temp = 300;  % Maximum vapor temperature in °C
mass_flow = 0.975;  % Mass flow rate in kg/s
```

Here, two constants are defined:

- max_temp: This sets the maximum temperature of the steam in the cycle (300°C for this case).
- mass_flow: This defines the mass flow rate of the working fluid (which is water) in the cycle (being 0.975 kg/s).

E. State Point Calculation:

The code then computes the characteristics of water at various state stages in the Rankine cycle using the XSteam library functions. It is used to calculate the points on the Rankine cycle.

The code makes use of XSteam functions for each state (1 to 4), which represents a distinct point in the cycle. These functions take pressure or other factors as input and provide results like as temperature, specific enthalpy, volume, and entropy. State 2a serves as an intermediary step for theoretical calculations but is not an actual state in the cycle. The setup code will start by calculating the parameters of saturated liquid water at condenser pressure (State 1). It will then proceed to an ideal expansion scenario (State 2a), by calculating the theoretical temperature and enthalpy if the liquid from state 1 expands to the boiler pressure with not heat loss nor gain. With that, it then estimates the parameters of the boiler outflow (State 2b), assuming saturated liquid water at boiler pressure. State 3 focuses on the boiler input, assuming that the temperature is equal to the boiler's saturation temperature. Finally, it estimates the enthalpy of superheated vapor (State 3b) using a predefined maximum temperature and boiler pressure. At the turbine outlet (state 4), The high-pressure, high-temperature steam from the boiler expands and cools down. This expansion doesn't happen perfectly, so the exiting fluid becomes a mix of liquid water and vapor, rather than just dry steam. The code calculates the quality of this vapor (dryness fraction) and its total heat content (enthalpy) to understand how much work the turbine was able to extract from the steam, and with that the calculation of graph points on the Rankine cycle is initialized.

F. Calculations:

The code now calculates key parameters that tell us how well the cycle is functioning as a thermodynamic cycle. First, it will determine the heat input (energy added in the boiler) and heat rejection (energy lost in the condenser). The code will then calculate how much the turbine does the work as the working fluid expands, and also the network output, which considers both turbine work and any pumping work required. Efficiency is an important parameter of how well the thermodynamic cycle works. *The* code will now calculate thermal efficiency, which indicates how much of the heat input is converted into usable work output. Finally, it will analyse the quality of the steam that exits the turbine, displaying the vapour concentration of the mixture. Higher quality (more vapour) results in more efficient work extraction.

G. Graphing and Plotting:

The code section of graph points is very critical for producing smooth lines in T-s and P-h diagrams. By setting `num_points` to 1000, the code will guarantee that these graphs have a high degree of information. Specific entropy and enthalpy values for water are calculated

iteratively across a wide temperature and pressure range. This repeated algorithm fills up arrays with exact property data, establishing the working for accurate and aesthetically pleasing graphs

Moving on to the Rankine cycle points, the code organises each stage's temperature, entropy, pressure, and enthalpy information. It transforms temps to Kelvin for uniformity and prepares the data for visualising the Rankine cycle on diagrams. This rigorous organisation and preparation of data makes it easy to comprehend how the system operates and functions.

H. Output:

Finally, the code prints the user-entered condenser and boiler pressures, the maximum vapor temperature, and the calculated performance parameters of the Rankine cycle (heat in/out, turbine/net work, boiler work ratio, quality, and efficiency).

XI. MATHEMATICAL CODE

```
>> addpath('C:\Users\SURFACE BOOK\Downloads\XSteam_Matlab_v2.6\')
cond_pressure = "Enter condenser pressure (bar): ";
cond_pressure = input(cond_pressure);
boiler_pressure = "Enter boiler pressure (bar): ";
boiler_pressure = input(boiler_pressure);
% Constants
max_temp = 300;
mass_flow  =  0.975;
% Calculations of Points on the Rankine
Cycle % AT State 1
t1  =  XSteam('Tsat_p',  cond_pressure);
hl = XSteam('hL_p', cond_pressure);
vl = XSteam('vL_p', cond_pressure);
sl = XSteam('sL_p',
cond_pressure); % AT State 2a
t2a = XSteam('T_ps', boiler_pressure, sl);
hg = XSteam('h_pT', boiler_pressure, t2a);
sg = sl;
% AT State 2b
t2b = XSteam('Tsat_p', boiler_pressure);
sb = XSteam('sL_p', boiler_pressure);
% At State
3a  t3a  =  t2b;
sv = XSteam('sV_p',
boiler_pressure); % At State 3b
t3b = max_temp;
hg_superheated = XSteam('h_pT', boiler_pressure, t3b);
sg_superheated   =   XSteam('s_pT',   boiler_pressure,
t3b); % AT State 4
t4 = t1;
s4 = sg_superheated;
x4 = XSteam('x_ps', cond_pressure, s4);
h4 = XSteam('h_ps', cond_pressure,
s4); % Calculations
q_in = mass_flow * (hg_superheated - hg);
q_out = mass_flow * (h4 - hl);
w_turbine = mass_flow * (hg_superheated - h4);
w_net = mass_flow * (q_in - q_out);
bwr = w_pump / w_turbine; quality_4 =
100 * x4;
efficiency = 100 * (1 - (q_out /
q_in)); % Graph
% Graph Points
num_points = 1000;
temps = zeros(1, num_points); sv_vals
= zeros(1, num_points); sl_vals =
zeros(1, num_points); pressures =
zeros(1, num_points); %zeros is for
matrix filled with zeros
```

```
hv_vals = zeros(1, num_points);
hl_vals = zeros(1, num_points);
for i = 1:num_points
% T-s
temp = 0 + (i - 1) * (600 - 0) / (num_points - 1);  % 0 to 600°C
sv_val = XSteam('sV_T', temp);
sl_val = XSteam('sL_T', temp);
temps(i) = temp + 273;
sv_vals(i) = sv_val;
sl_vals(i) = sl_val;
% P-h
pressure = 0 + (i - 1) * (10 - 0) / (num_points - 1);
hv_val = XSteam('hV_p', pressure); hl_val =
XSteam('hL_p', pressure);
pressures(i) = pressure;
hv_vals(i) = hv_val;
hl_vals(i) = hl_val;
end
% Rankine Cycle Graph Points
cycle_temps = [t1 + 273, t2a + 273, t2b + 273, t3a + 273, t3b + 273, t4 +
273, t1 + 273];
cycle_entropy = [sl, sg, sb, sv, sg_superheated, s4, sl];
cycle_pressures = [cond_pressure, boiler_pressure, boiler_pressure,
cond_pressure, cond_pressure];
cycle_enthalpy = [hl, hg, hg_superheated, h4, hl];
% Plotting the T-s and P-h
Graphs figure;
subplot(1, 2, 1); %subplot is to create and manage plots in the grid of
subplot
hold on;
plot(sv_vals, temps, 'cyan', 'LineWidth', 1);
plot(sl_vals, temps, 'cyan', 'LineWidth', 1);
plot(cycle_entropy, cycle_temps, 'g', 'LineWidth', 2);
ylabel('Temperature (K)');
xlabel('Entropy (kJ/kg*K)');
title('Temperature-Entropy Diagram');
legend('Vapor', 'Liquid', 'Rankine Cycle');
grid on;
subplot(1, 2, 2);
hold on;
plot(hv_vals, pressures, 'cyan', 'LineWidth', 1);
plot(hl_vals, pressures, 'cyan', 'LineWidth', 1);
plot(cycle_enthalpy, cycle_pressures, 'g', 'LineWidth', 2);
ylabel('Pressure (bar)');
xlabel('Enthalpy (kJ/kg)');
title('Pressure-Enthalpy Diagram');
legend('Vapor', 'Liquid', 'Rankine Cycle');
grid on;
% Output
fprintf('\nCondenser Pressure: %.2f bar\n', cond_pressure);
fprintf('Boiler Pressure: %.2f bar\n', boiler_pressure);
fprintf('Maximum Vapor Temperature: %.2f K\n\n', max_temp + 273);
fprintf('Parameter | Value (W)\n'); fprintf('------------------
---------\n'); fprintf('Heat In | %.2f\n', q_in); fprintf('Heat
Out | %.2f\n', q_out); fprintf('Turbine Work | %.2f\n',
w_turbine); fprintf('Net Work | %.2f\n', w_net); fprintf('Boiler
Work Ratio | %.2f\n', bwr); fprintf('Quality | %.2f%%\n',
quality_4); fprintf('Efficiency | %.2f%%\n', efficiency);
```

XII. TABULAR DATA

In lieu of parameters set forth for analysis, about 100 observations were taken into account, half for which the boiler pressure was assumed to be constant and for the other half the condenser pressure was kept constant.

Table 2 Observation Number 01 for Constant Condenser Pressure

BP	CP	Wout	Back Ratio	Efficiency
4.5	2.75	118.73	-0.03	0.0483
4.675	2.75	127.31	-0.03	0.051
4.85	2.75	135.5	-0.03	0.051
5.025	2.75	139.49	-0.03	0.0583
5.2	2.75	150.81	-0.03	0.0614
5.375	2.75	157.90	-0.03	0.0643
5.55	2.75	164.86	-0.03	0.0671
5.725	2.75	171.47	-0.03	0.0698
5.9	2.75	177.83	-0.03	0.0724
6.075	2.75	183.95	-0.03	0.0749
6.25	2.75	189.84	-0.03	0.074

Table 3 Observation Number 02 for Constant Condenser Pressure

BP	CP	Wout	Back Ratio	Efficiency
4.50	2.75	118.31	-0.03	0.0481
4.67	2.75	126.90	-0.03	0.0516
4.85	2.75	135.10	-0.03	0.0550
5.03	2.75	142.92	-0.03	0.0582
5.20	2.75	150.41	-0.03	0.0612
5.38	2.75	157.59	-0.03	0.0642
5.55	2.75	164.47	-0.03	0.0670
5.72	2.75	171.08	-0.03	0.0697
5.90	2.75	177.44	-0.03	0.0723
6.08	2.75	183.56	-0.03	0.0748
6.25	2.75	189.35x	-0.03	0.0772

Table 4 Observation Number 03 for Constant Condenser Pressure

BP	CP	Wout	Back Ratio	Efficiency
4.50	2.76	117.90	-0.03	0.0480
4.67	2.76	126.49	-0.03	0.0515
4.85	2.76	134.69	-0.03	0.0548
5.03	2.76	142.52	-0.03	0.058
5.20	2.76	150.01	-0.03	0.611
5.38	2.76	157.19	-0.03	0.064
5.55	2.76	164.08	-0.03	0.068
5.72	2.76	170.69	-0.03	0.0695
5.90	2.76	177.05	-0.03	0.0721
6.08	2.76	183.18	-0.03	0.0746
6.25	2.76	189.08	-0.03	0.0771

Table 5 Observation Number 04 for Constant Condenser Pressure

BP	CP	Wout	Back Ratio	Efficiency
4.50	2.77	117.49	-0.03	0.0478
4.67	2.77	126.09	-0.03	0.0513
4.85	2.77	130.70	-0.03	0.0547
5.03	2.77	142.12	-0.03	0.0579
5.20	2.77	149.61	-0.03	0.0609
5.38	2.77	156.80	-0.03	0.0639
5.55	2.77	163.69	-0.03	0.0667
5.72	2.77	170.30	-0.03	0.0694
5.90	2.77	176.67	-0.03	0.0720
6.08	2.77	182.80	-0.03	0.0745
6.25	2.77	188.70	-0.03	0.0769

Table 6 Observation Number 05 for Constant Condenser Pressure

BP	CP	Wout	Back Ratio	Efficiency
4.50	2.78	116.25	-0.03	0.0473
4.67	2.78	124.86	-0.03	0.0508
4.85	2.78	134.69	-0.03	0.0548
5.03	2.78	142.52	-0.03	0.058
5.20	2.78	150.01	-0.03	0.611
5.38	2.78	157.19	-0.03	0.064
5.55	2.78	164.08	-0.03	0.068
5.72	2.78	170.69	-0.03	0.0695
5.90	2.78	177.05	-0.03	0.0721
6.08	2.78	183.18	-0.03	0.0746
6.25	2.78	189.08	-0.03	0.0771

Table 7 Observation Number 06 for Constant Condenser Pressure

BP	CP	Wout	Back Ratio	Efficiency
4.50	2.79	115.84	-0.03	0.0472
4.67	2.79	124.46	-0.03	0.0507
4.85	2.79	132.67	-0.03	0.0540
5.03	2.79	140.52	-0.03	0.0572
5.20	2.79	148.03	-0.03	0.0603
5.38	2.79	155.22	-0.03	0.0632
5.55	2.79	162.13	-0.03	0.0661
5.72	2.79	168.76	-0.03	0.0688
5.90	2.79	175.13	-0.03	0.0714
6.08	2.79	181.27	-0.03	0.0739
6.25	2.79	187.19	-0.03	0.0763

Table 8 Observation Number 07 for Constant Condenser Pressure

BP	CP	Wout	Back Ratio	Efficiency
4.50	2.79	115.43	-0.03	0.0470
4.67	2.79	124.05	-0.03	0.0505
4.85	2.79	132.27	-0.03	0.0539
5.03	2.79	140.12	-0.03	0.0571
5.20	2.79	147.63	-0.03	0.0601
5.38	2.79	154.83	-0.03	0.0631
5.55	2.79	161.74	-0.03	0.0659
5.72	2.79	168.37	-0.03	0.0686
5.90	2.79	174.75	-0.03	0.0712
6.08	2.79	180.89	-0.03	0.0738
6.25	2.79	186.81	-0.03	0.0762

Table 9 Observation Number 08 for Constant Condenser Pressure

BP	CP	Wout	Back Ratio	Efficiency
4.50	2.79	115.03	-0.03	0.0468
4.67	2.79	123.65	-0.03	0.0504
4.85	2.79	131.87	-0.03	0.0537
5.03	2.79	139.72	-0.03	0.0569
5.20	2.79	147.24	-0.03	0.0600
5.38	2.79	154.44	-0.03	0.0629
5.55	2.79	161.35	-0.03	0.0658
5.72	2.79	167.98	-0.03	0.0685
5.90	2.79	174.37	-0.03	0.0711
6.08	2.79	180.51	-0.03	0.0736
6.25	2.79	186.43	-0.03	0.0760

Table 10 Observation Number 09 for Constant Condenser Pressure

BP	CP	Wout	Back Ratio	Efficiency
4.50	2.80	114.62	-0.03	0.0467
4.67	2.80	123.24	-0.03	0.0502
4.85	2.80	131.47	-0.03	0.0536
5.03	2.80	139.33	-0.03	0.0568
5.20	2.80	146.85	-0.03	0.0598
5.38	2.80	154.05	-0.03	0.0628
5.55	2.80	160.96	-0.03	0.0656
5.72	2.80	167.60	-0.03	0.0683
5.90	2.80	173.98	-0.03	0.0709
6.08	2.80	180.13	-0.03	0.0735
6.25	2.80	186.06	-0.03	0.0759

Table 11 Observation Number 10 for Constant Condenser Pressure

BP	CP	Wout	Back Ratio	Efficiency
4.5	2.75	118.73	-0.03	0.0483
4.5	2.755	118.31	-0.03	0.0481
4.5	2.76	117.90	-0.03	0.048
4.5	2.765	117.49	-0.03	0.0478
4.5	2.77	117.08	-0.03	0.0476
4.5	2.775	116.66	-0.03	0.0475
4.5	2.78	116.25	-0.03	0.0473
4.5	2.785	115.84	-0.03	0.0472
4.5	2.79	115.84	-0.03	0.0472
4.5	2.795	115.03	-0.03	0.0469

It is pertinent to assert that the values of condenser pressure (CP) and boiler pressure (BP) are taken into account in bars, the Output Work is taken into account as joules (J).

XIII. GRAPHING

In order to obtain relevant graphs for the above generated data, predominantly for the data with maximum efficiency, following code lines were brought into consideration:

```
clc; clear; close all;
figure(1);
grid on; hold on;
CycleEfficiency = [0.074 0.072 0.071 0.0769 0.0768 0.0766 0.0763 0.0762
0.0760 0.0759 0.0469];
CP = [2.75 2.75 2.76 2.77 2.77 2.77 2.78 2.79 2.79 2.80 2.795];
```

```
BP = [6.25 6.25 6.25 6.25 6.25 6.25 6.25 6.25 6.25 6.25 4.5];
length Wout
length CP
length BP
plot(CycleEfficiency,CP,'k-')
plot(CycleEfficiency,BP,'b-')
title('CycleEfficiency vs CP(black) and BP(Blue)')
xlabel('CE')
ylabel('CP/BP')
```

XIV. GRAPHING DATA

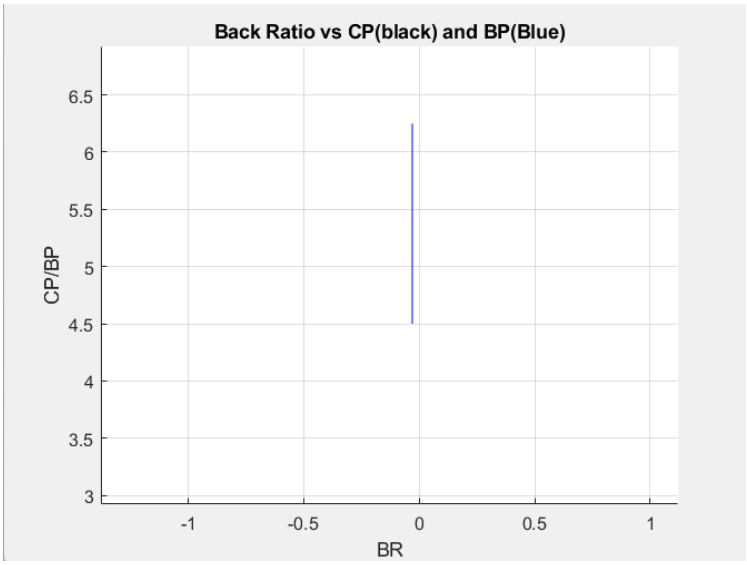


Figure 6 Back Ratio vs. Boiler Pressure plot

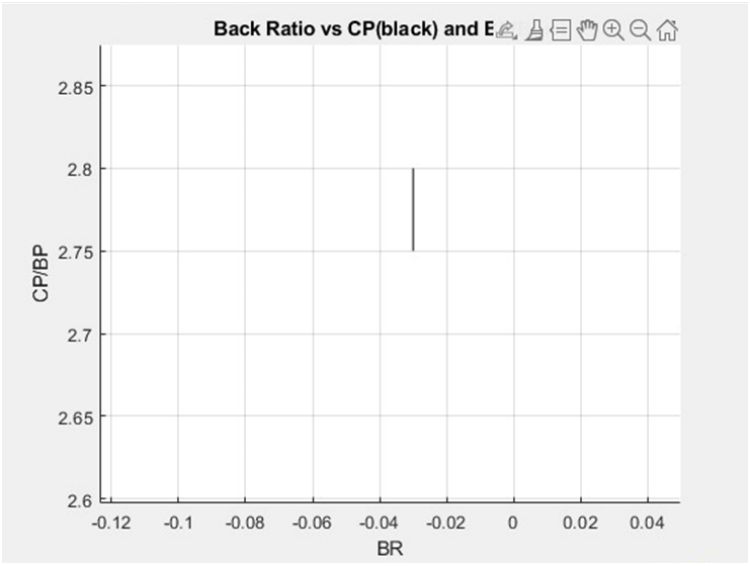


Figure 7 Back Ratio vs. Condenser Pressure plot

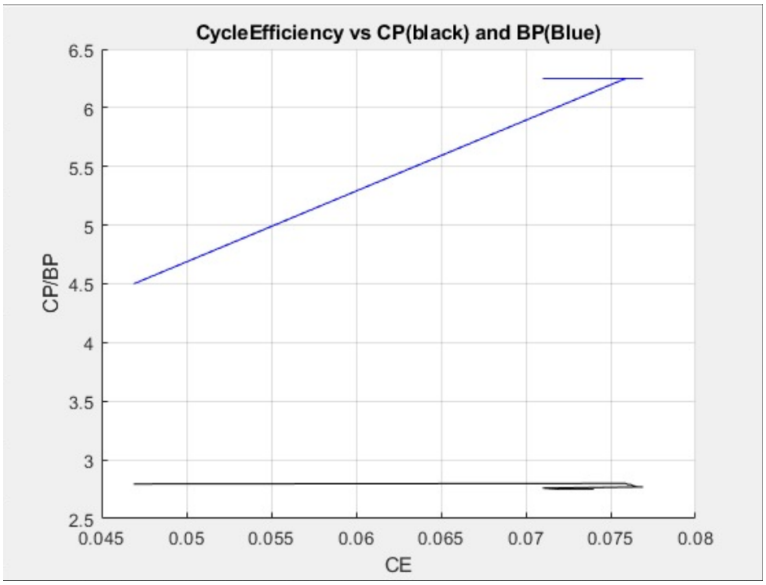


Figure 7 Cycle Efficiency vs. Condenser Pressure and Boiler Pressure plot

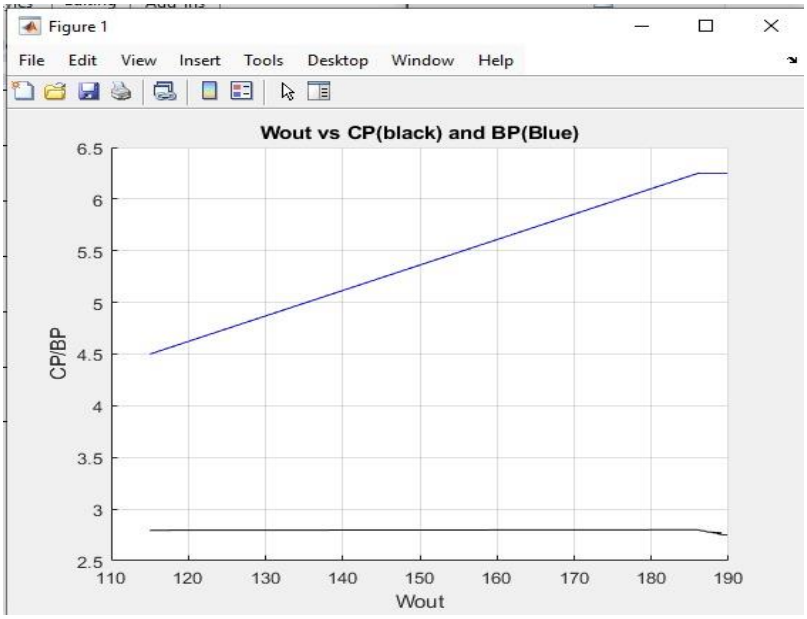


Figure 8 Work Output vs. Condenser Pressure and Boiler Pressure plot

Analysing the Graphs, a total of 4 graphs as shown above were devised. The code used involved encompasses functions such as setting graph parameters, use of arrays to plot the points, griding, and distinction of colour and nature of lines used for each curve. The values were taken from the highest value (basically the last iteration) from each iteration table. Furthermore, analysing each graph all curves were strictly linear with minor discrepancy at the ends due to a sudden change in the otherwise quantized change of values. Values of BP were throughout the same.

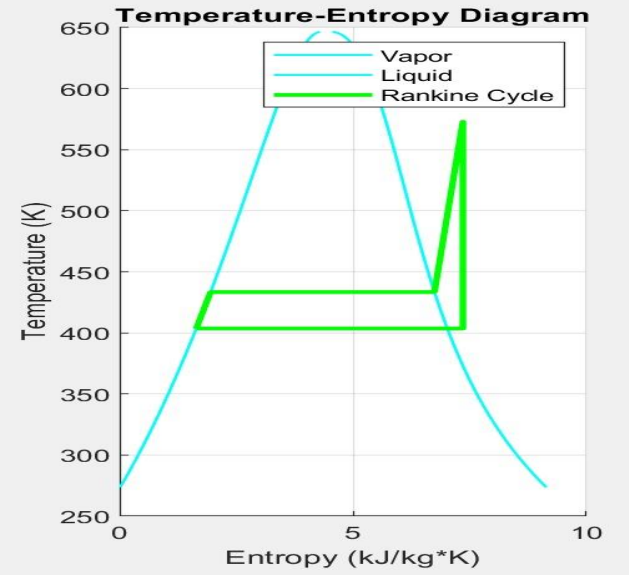


Figure 9 Temperature-Entropy Diagram

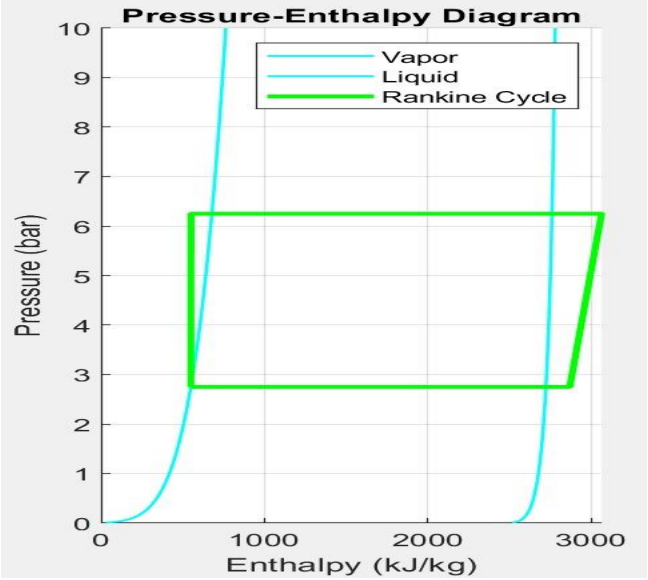


Figure 7 Pressure-Enthalpy Diagram

XV. RESULTS AND DISCUSSION

The variation in boiler and condenser pressures gave insightful interpretation regarding the performance of Rankine heat engine cycle. The effect on some key parameters i.e back ratio, efficiency and work output were studied by changing the pressure combinations. Characteristic curves depicting the relationship between work output and pump/boiler inlet pressures provided valuable insights into the cycle's behaviour. It was observed that variations in pressure exerted notable influences on the work output of the turbine and pump, with optimal pressure ranges yielding higher efficiencies. Moreover, the back ratio exhibited distinct trends, indicating the relationship between condenser and boiler pressures in dictating cycle performance. Moreover, the study emphasized the significance of boiler and condenser pressures in influencing cycle efficiency. By identifying optimal pressure ranges, the cycle's performance can be maximized, leading to enhanced energy conversion, and reduced operational costs.
costs.

XVI. CONCLUSIONS

In conclusion, the analysis and optimization of the Rankine Heat Engine Cycle highlighted the importance of pressure parameters in determining cycle efficiency. Through exploration and visualization of characteristic curves, valuable insights were gained into the thermodynamic behavior of the cycle. The findings of this study hold significant practical implications for power plant design and operation, where maximizing efficiency is

important. Moving forward, continued advancements in cycle optimization are essential to ensure long-term sustainability and environmental stewardship in power generation processes.

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