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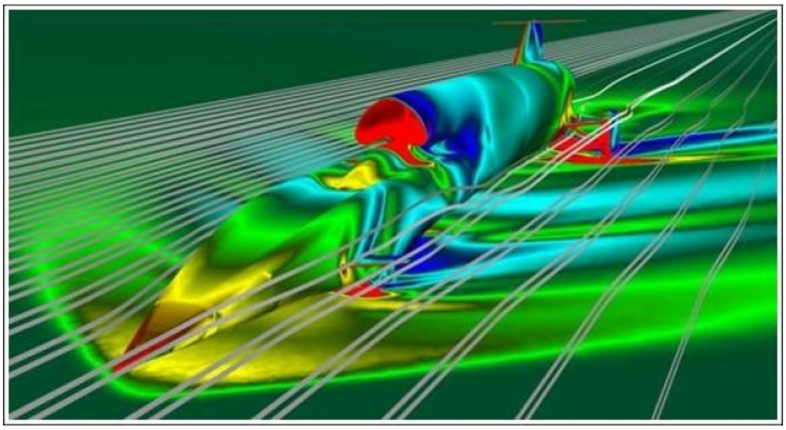
BSc Mechanical Engineering

Applied Thermodynamics



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Summary

This report is based on the findings of the following systems and other related objectives.

Compression Cycle

Critical analysis and the evaluation of internal combustion engines, by applying techniques that relate to the performance characteristics, and then after discussing enhancements of the system.

Rankine Cycle

Critical analysis of the design and application of power plants covering all aspects associated.

Refrigeration Cycle

Applying the techniques and parameters used in the design and operating principles of air refrigeration systems.

Methodology

Throughout this report, ability of the systems and comparisons will be researched via thermodynamics methodology, calculated, examined, and subjected to critically analysed conclusions.

Advice

Use Navigation Tool, see View Settings (Microsoft Word).

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Introduction

Thermodynamics

Thermodynamics derives from two Greek ideas: therme meaning hot or heat, and dynamikos meaning power or powerful. Thermodynamics is the study of heat related to matter in motion.

Thermodynamics is used in the design of machines for example engines or refrigeration, turbines and compressors, it involves many factors and environmental parameters. This report touches briefly on certain aspects of thermodynamics (Sole, 2018).

Laws

The first law, also known as Law of Conservation of Energy, states that energy cannot be created or destroyed in an isolated system (Lumen, 2018).

The second law of thermodynamics states that the entropy of any isolated system always increases (Lumen, 2018).

The third law of thermodynamics. The entropy of a perfect crystal at absolute zero is exactly equal to zero (Lumen, 2018 (2)).

Zeroth Law

Zeroth **law of thermodynamics**: If two thermodynamic systems are each in thermal equilibrium with a third, then they are in thermal equilibrium with each other (Atkins, 2007).

This report will answer the questions given as coursework using the laws of thermodynamics to investigate the criteria set by University of Derby.

The report will demonstrate an understanding of the systems, I will talk about some aspects and others will be calculated given 3300 word-limit I can only touch on subjects considering the coursework questions.

Internal Combustion Engines

Table 1 Internal Combustion Engine

Overview of Reciprocating Engines
Otto Cycle – The Ideal Cycle for SI Engines
Diesel Cycle – The Ideal Cycle for CI Engines
Criteria of Performance
Performance Characteristics
Examples

Thermal Power Plant

Table 2 Thermal Power Plant

Rankine Cycle
Rankine Cycle's Efficiencies
Specific Steam Consumption
Rankine Cycle with Superheat
The Enthalpy-Entropy (h-s) Chart

The Reheat Cycle
The Regenerative Cycle
Open and Closed, Feedwater Systems
Principles of Thermal Power Plant Design
Power Plant Efficiency and Waste

Refrigeration Cycles

Table 3 Refrigeration Cycle

Reversed Heat Engine Cycle
Coefficient of Performance (COP)
Reversed Carnot Cycle vapour- Compression Refrigeration Cycles
Use of Throttle Valve
Condition at Compressor Inlet
Undercooling of Condenser Vapour
The Pressure-Enthalpy (p-h) Diagram
The Use of the Flash Chamber
Examples
Principles of Refrigeration Design and Power
Refrigeration Cycle Efficiency

Formal Report
Part 1

Reciprocating Engine Cycle

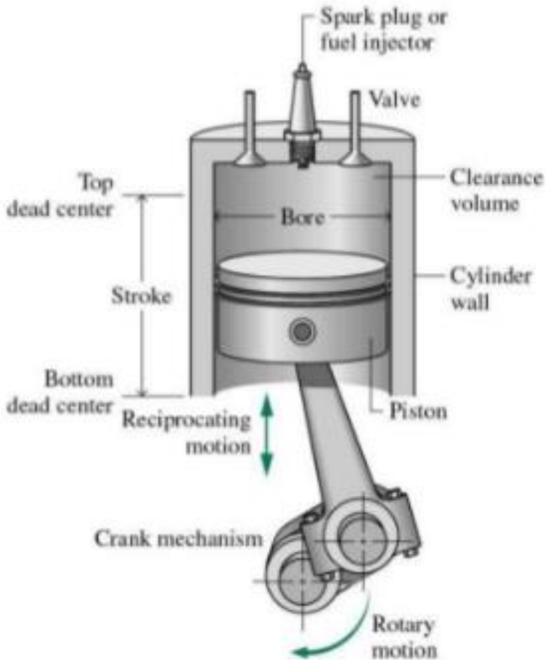

Figure 1 Reciprocating Engine Cycle (AboKhalil, 2013)

There are two types of systems to study: the Otto Cycle and the Diesel Cycle.

Diesel

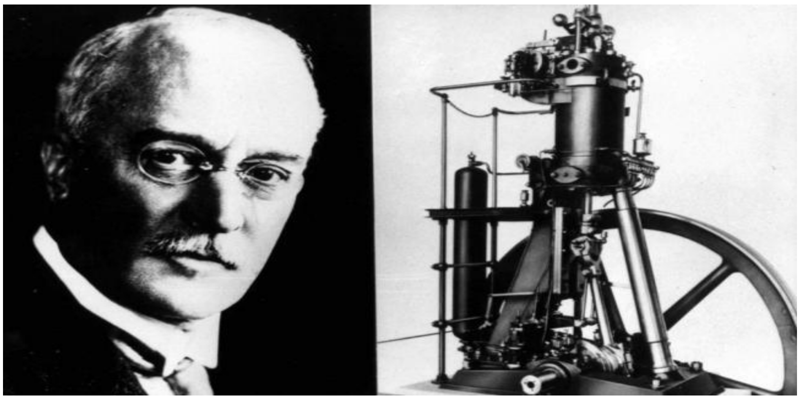

Figure 2 Rudolf Diesel (LaVache, 2016)

In the early 1890's Rudolf Diesel invented a compression ignition, internal combustion engine. Early types of diesel engines were large and operated at low speeds due to the limitations of their compressed, air-assisted, fuel injection systems (Jääskeläinen, 2013).

Otto

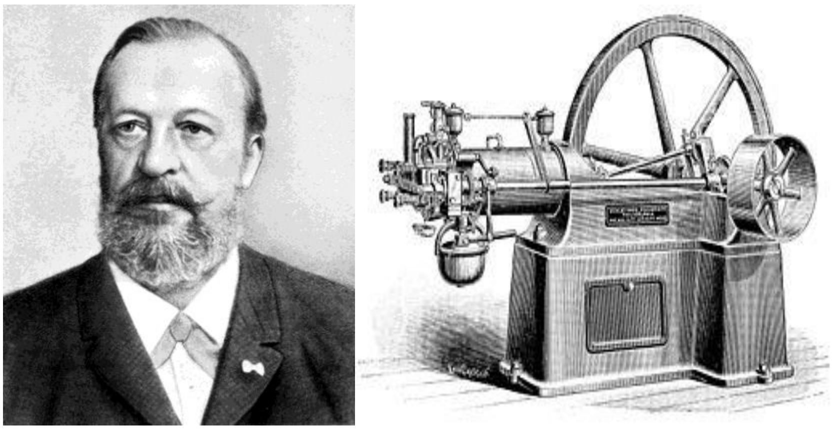

Figure 3 Nikolaus Otto (DeMotor, 2017)

Born in 1832, in Germany, Nikolaus Otto was an engineer, in 1861, developed a gasoline-petrol powered engine and later invented the internal combustion engine using the four-stroke system, an alternative to steam engines. The four-stroke cycle is referred to as the 'Otto Cycle' despite being patented by Aplhonse Beau de Rochas (Biography.com, 2014).

Difference

The difference between an Otto cycle and diesel is that in an Otto Cycle, the heat transfer is at a constant volume, whereas a Diesel cycle operates when heat transfer occurs at a constant pressure.

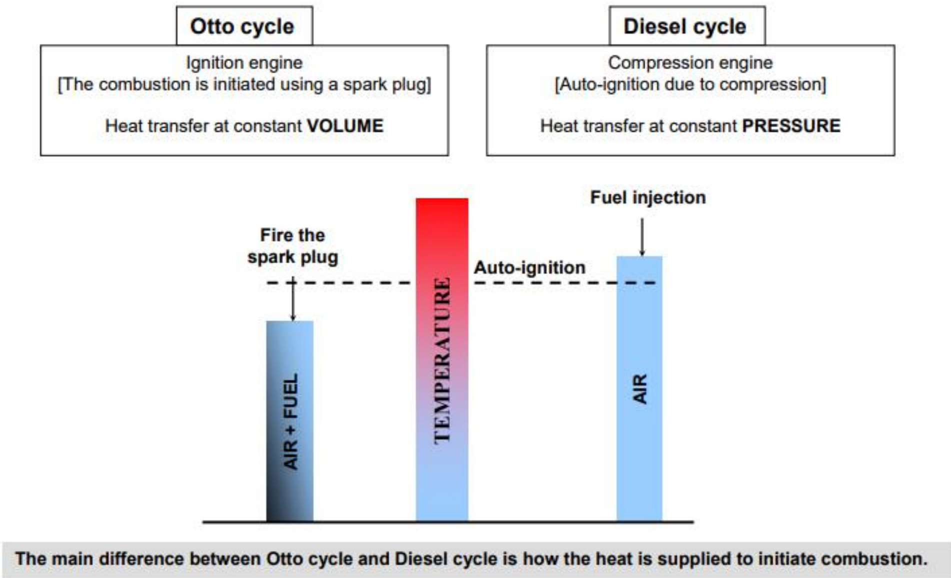

Figure 4 Diesel and Otto Differences (Kadem, 2007)

Fuel

Diesel and petrol are both Hydrocarbons made up of the same elements. The difference between them is the length of chain, with diesel having a longer chain than petrol.

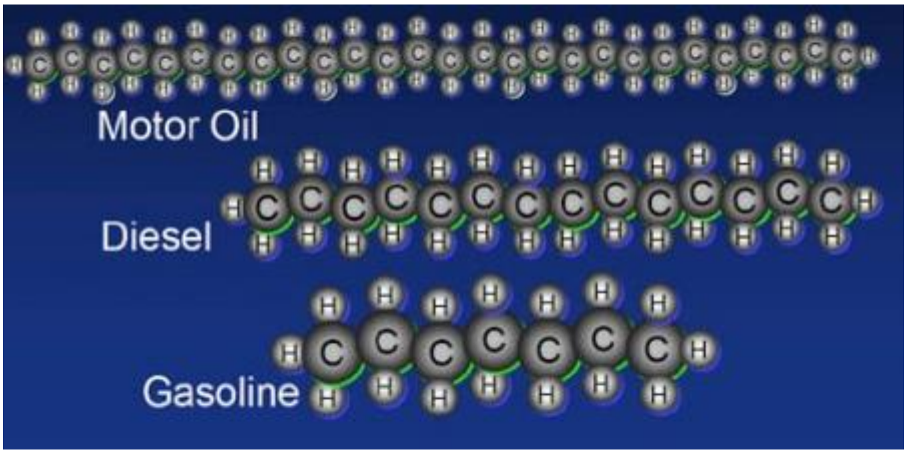

Figure 5 Molecular Structure Diesel Vs Petrol (Mehta, 2015)

In an Otto Cycle, fuel is mixed with air using a spark plug to ignite the fuel mix. The diesel cycle operates using auto ignite due to compression when fuel is injected into the air chamber. The Diesel Cycles compression ignition happens at a lower temperature than petrol and air combustion. Petrol withstands a higher temperature than diesel and needs to be ignited. Petrol vapours are higher and readier at room temperature than diesel vapours, but at lower temperatures, the diesel vapours are higher and readier. (Gorupec, 2006).

Identical systems at lower compression rates suggest that the Otto cycle is the most efficient, as parts used weigh less giving better efficiency overall. The compression area is greater in the diesel cycle and extra parts are required for fuel injection pulsing which adds weight to the engine. An Otto cycles compression ratio is between 7-12.5.

For example, using ideal engine cycle parameters, a compression ratio of 10 and a cut-off ratio (r_c) = 2, the following results were obtained.

Table 4 Results Otto Diesel Cycle

Cycle	**Efficiency**
The Otto cycle	60.2%
The Diesel cycle	53.7%

(Kadem, 2007)

The Diesel cycle has the highest compression ratio 22 and it has the highest efficiency at 64.7%. The diesel cycle has a higher fuel efficiency as it combusts all fuel through pressure, whereas the Otto cycle spark ignites but does not burn all the fuel (Kadem, 2007).

Fuel injection in a diesel engine starts when the piston approaches Top Dead Centre (TDC) and continues during the first part of the power stroke. The combustion process takes place over a longer interval than the Otto; the combustion process in the ideal Diesel cycle is approximated as a constant-pressure heat-addition process and that is the only difference between the Otto and Diesel cycles.

Ideal Diesel Cycle

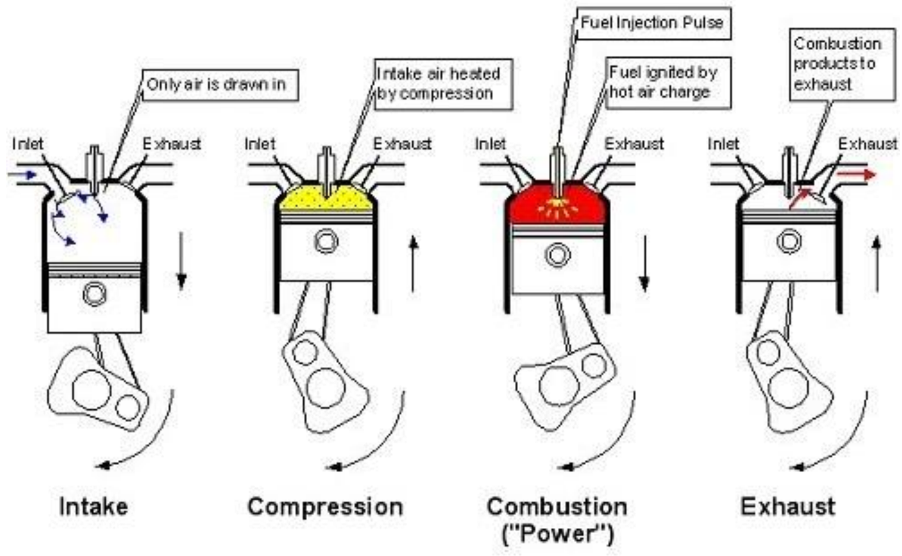

Figure 6 Diesel Compression Cycle (Petersen, 2014)

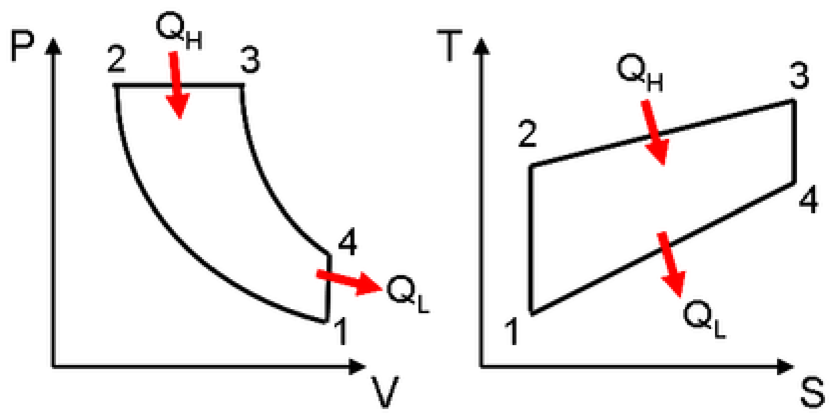

Figure 7 PV and TS Diagram (Wijaya, 2009)

Table 5 Table showing Compression Cycle Process for Diesel

1-2 Isentropic Compression
3-4 isentropic expansion
4-1 constant volume heat rejection

Ideal Diesel Calculations

$Q_{in} - W_{out} = U_3 - U_2 \rightarrow Q_{in}$

$P_2 = (V_3 - V_2) + (U_3 - U_2)$

$= h_3 - h_2 = C_p(T_3 - T_2)$

And

$Q_{out} = U_1 - U_4 \rightarrow Q_{out} = U_4 - U_1 \, C_V(T_4 - T_3)$

$Q_{in} = U_3 - U_2 = C_V(T_3 - T_2)$

Then the Thermal Efficiency under cold air becomes.

η_{th} Diesel $= \frac{W_{net}}{Q_{in}} = 1 - \frac{Q_{out}}{Q_{in}} = 1 - \frac{T_4 - T_1}{y(T_3 - T_2)} = 1 - \frac{T_1(T_4/T_1)}{yT_{2(}T_3/T_2)}$

Cut off ratio

r$_c$ = = $\frac{V_3}{V_2} = \frac{U_3}{U_2}$

Definition for process 1-2 and 3-4 thermal efficiency reduces to.

$\eta_{th\ Diesel} = 1 - \frac{1}{r^{y-1}}\left[\frac{(r_c^y - 1}{y(r_c - 1}\right]$ (Sole, 2018)

Ideal Otto Calculations

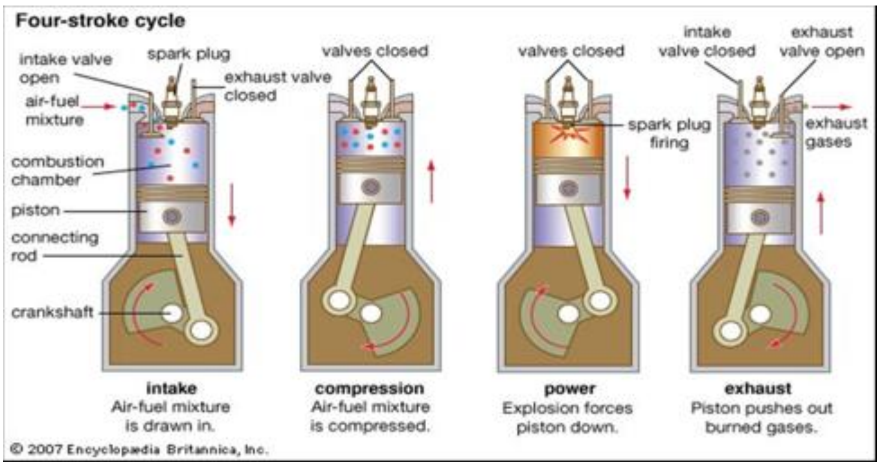

Figure 8 Otto Four Stroke picture (Sole, 2018)

PV and TS diagram

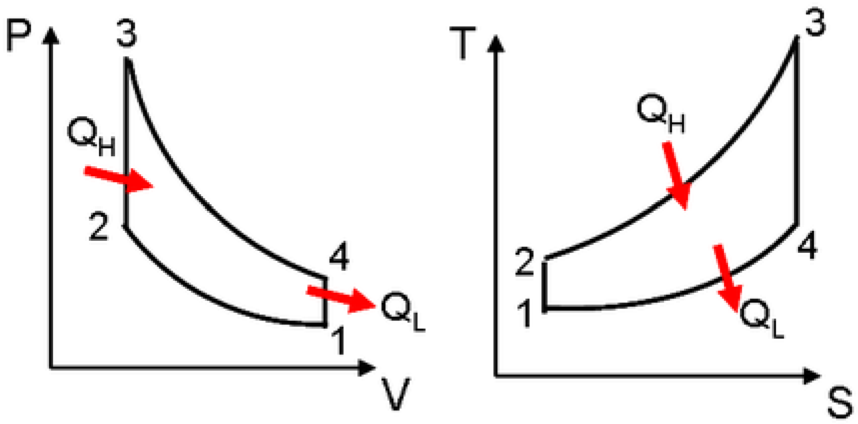

Figure 9 PV and TS diagram/ Otto (Wijaya, 2009)

Table 6 Table showing process Otto

1-2 Isentropic Compression
2-3 Constant volume heat addition
3-4 Isentropic expansion
4-1 Constant volume heat rejection

Note: Otto Cycle has two sets of isochoric and two sets of adiabatic processes.

Isochoric – A process in which the specific volume remains constant. (n = ∞) (Sole, 2018))

Adiabatic – Pressure and volume change but no heat is added or lost (Using an insulated cylinder) (n = γ)) (Sole, 2018).

Thermal Efficiency 2
(Ideal Air Standard Cycle Efficiency Otto)

$$\eta = \frac{W_{net}}{Q_{in}} \quad = 1 - \frac{Q_{out}}{Q_{in}} \quad = 1 - \frac{\dot{m}C_v(T_4 - T_1)}{\dot{m}C_v(T_3 - T_2)} \quad = 1 - \frac{(T_4 - T_1)}{(T_3 - T_2)}$$

Process (1) to (2) $\frac{T_2}{T_1} = \left(\frac{V_1}{V_2}\right)^{\gamma-1} = r_v^{\gamma-1}$

Process (3) to (4) $\frac{T_3}{T_4} = \left(\frac{V_4}{V_3}\right)^{\gamma-1} = r_v^{\gamma-1}$

Where r_v is the volume compression ratio $r_v = \frac{V_1}{V_2} = \frac{V_4}{V_3}$

It follows that $\frac{T_2}{T_1} = \frac{T_3}{T_4}$ $\quad T_3 = \frac{T_2 T_4}{T_1}$ $\quad$ and $\quad \frac{T_4}{T_1} = \frac{T_3}{T_2}$ $\quad T_4 = \frac{T_3 T_1}{T_2}$

And that $\eta = 1 - \frac{T_4 - T_1}{T_3 - T_2} = 1 - \frac{\frac{T_3 T_1}{T_2} - T_1}{\frac{T_2 T_4}{T_1} - T_2} = 1 - \frac{T_1\left(\frac{T_3}{T_2} - 1\right)}{T_2\left(\frac{T_4}{T_1} - 1\right)}$

Since $\frac{T_4}{T_1} = \frac{T_3}{T_2}$ then $\left(\frac{T_4}{T_1} - 1\right) = \left(\frac{T_3}{T_2} - 1\right)$

$$\eta = 1 - \frac{T_1}{T_2} \qquad = 1 - \frac{T_4}{T_3} \qquad = 1 - \frac{1}{r_v^{\gamma - 1}}$$

$\eta = 1 - r_v^{1-\gamma}$ (Sole, 2018)

Calculation coursework

Air at 120Kpa and 22℃ is compressed reversibly and adiabatically. The air is then heated at a constant volume to 1650℃. The air expands reversibly and adiabatically back to the original volume and is cooled at constant volume back to the original temperature and pressure. The compression ratio is 7.

Table 7 Table Parameters Otto

C_v = 0.718KJ/Kg
Y = 1.35(air) y = cp/cv
Solved for a mass of 1 kg
Using absolute temperatures
Bars converted to pascal

Calculate the following

Table 8 Calculation table Otto

The Thermal Efficiency
The Heat Output per kg of Air
The Network Output per kg of Air
The maximum Cycle Pressure

Symbols, Facts and Units for all Coursework Calculations

Table 9 Symbols, facts and units

T = Temperature
r_v = (Compression Ratio)
c_r = cut off ratio
η_{th} = 1 – $1^{1\text{-}y}$
Solved for a mass of 1 kg
$\dot{m}$ = flow rate (kg/s)
M = mass

p = pressure
V = Volume
n = Index of expansion or compression
C = Constant
kJ = kilojoules
Kg = kilograms
kW = kilowatt
MW = Megawatt
MJ = Mega joule
K = Kelvin
m^3 = volume
Mpa = Mega pascal
pa = pascal
ϕ and Q = heat
Wnett = Work Done

(System 1)

Thermal Efficiency

Polytropic/Adiabatic Process (sole 2018)

$PV^n=C$

n = Y

$Y = C_P/C_V = 1.35$

$\eta = 1 - r_v^{1-y} \qquad = 1 - 7^{1-1.35} = 0.493$

0.493 x 100 = 49.3

Thermal Efficiency = 49.3%

The heat input per kg of air

$T_2 = T_1 (\frac{v_1}{v_2})^{y-1} = 295 (7^{0.35}) = 582.9K$

$T_2 = 582.9K$

$Q_{in} = mc_v (t_3 - t_2) = 1 \times 0.718(1923-582.9) = 962.2$ Kj/kg

The network output per kg of air

$W_{net} = n^{th} Q_{in} = 0.49 \times 962.2 = 471.5$ kJ/kg

The maximum Cycle pressures

$\frac{p_1 v_1}{t_1} = \frac{p_3 v_3}{t_1 v_3}$

$P_3 = \frac{p_1 v_1 t_3}{t_3}$

$P_3 = \frac{120000 x v_1 x 1923}{295} x7 = 5475661.017$

$= 5.5 Mpa$

Last, T_4 and Q_{out} to complete the chart (Sole, 2018)

$$T_4 = \frac{T_3}{r_v^{\gamma-1}}$$

$$T_4 = \frac{1923}{7^{1.35-1}}$$

$$T_4 = 973.184 \text{ K}$$

$$Q_{out} = \dot{m} C_v (T_4 - T_1)$$

$$Q_{out} = 1 \times 718 \times (973.184 - 295)$$

$$Q_{out} = 486936.93 \text{ J/kg}$$

$$Q_{out} = 486.93 \text{ kJ/kg}$$

Example Calculations for generator with power output 7.7MW (Sole, 2018))

Find mass flow rate to give power output of 7.7 MW

$$\text{Power output} = \text{Net work output} \times \text{Mass flow rate}$$

$$(7.7 \times 10^6) =$$

$$\text{Mass flow rate} = \frac{7.7 \times 10^6}{471.5}$$

$$\text{Mass flow rate} = 16.3 \text{ kg/s}$$

$$Q_{in} = 962191.8 \times 16.3 = 15.6 \text{ MJ}$$

$$Q_{out} = 486936.112 \times 16.3 = 7.93 \text{ MJ}$$

$$\eta Q_{in} = 471.5 \times 16.3 = 7.68 \text{ MJ}$$

ii) The mass flowrate of air = 2.3kg/s, Determine the actual value for

(System 2)

Heat in

$Q_{in} = \dot{m}c_v\,(t_3 - t_2) = 2.3 \times 0.718(1923\text{-}582.9) = 2213.5$ kJ/kg

Work

$W_{net} = n^{th}\,Q_{in} = 0.49 \times 2213.5 = 1084.6$ kJ/kg

The maximum Cycle Pressure retest for system 2

$$\frac{p_1v_1}{t_1} = \frac{p_3v_3}{t_1v_3}$$

$$P_3 = \frac{p_1v_1t_3}{t_3}$$

$$P_3 = \frac{120000xv_1x1923}{295xv_3}x7 = 5475661.017$$

$$= 5.5Mpa$$

iii) What Methods could be employed to improve the heat in and hence work out to improve the system.

To help solve for efficiency we adjust calculations, when an increase in flow rate from 1kg to 2.3kg occurs, it increases the amount of work done and heat required and has no effect on pressure or efficiency. If we then look at the Thermal efficiency calculation of the Otto Cycle, the density of air Y = 1.35 and the compression ratio 7 if we increase these values.

For example:

Table 10 Example values

Y = 1.40
Compression = 8.5/1
2.3 kg/s

(System 3)

Thermal Efficiency

$\eta_{th} = 1 - r_v^{1-y} \qquad = 1 - 8.5^{1-1.40} = 0.575$

=57.5% Efficient

Heat Input per kg of Air

$T_2 = T_1\,(\frac{v_1}{v_2})^{y-1} = 295\,(8.5^{0.40}) = 694.3K$

$T_2 = 694.3K$

$Q_{in} = mc_v (t_3 - t_2) = 2.3 \times 0.718(1923-694.3) = 2029$ Kj/kg

Network Output

$W_{net} = n^{th} Q_{in} = 0.57 \times 2029 = 1156.5$ kJ/kg

The Maximum Cycle Pressure

$$\frac{p_1 v_1}{t_1} = \frac{p_3 v_3}{t_1 v_3}$$

$$P_3 = \frac{p_1 v_1 t_3}{t_3}$$

$$P_3 = \frac{120000 x v_1 x 1923}{295 x v_3} x 8.5 = 6649016.949$$

$$= 6.6 Mpa$$

Results

Results show the values obtained when calculating the 3 systems.

Table 11 System Results

Otto System	Thermal Efficiency η_{th} %	Heat Input (kJ/kg)	Network Output (kJ/kg)	Cycle Pressure (Mpa)
System 1 (1kg/s)	49%	962.2	471.5	5.5
System 2 (2.3 kg/s)	49%	2213.5	1084.6	5.5
System 3 (2.3 kg/s)	57%	2029	1156.5	6.6

Conclusion

System 3, when compared to the results of System 1 and 2, has been improved. The compression ratio and density of air have been increased to 8.5 and 1.40 respectively. Calculations suggest a more efficient system with an improved Thermal Efficiency of 57%, reduced heat input per kg of air at 2029 KJ/kg and an increase Network Output at 1156.5 KJ/kg, it produces more pressure than systems 1 and 2 and therefore material selection should be critical in design as previously mentioned material adds weight to engines reducing efficiency.

Part 2

Steam Power Plants Operating Principles

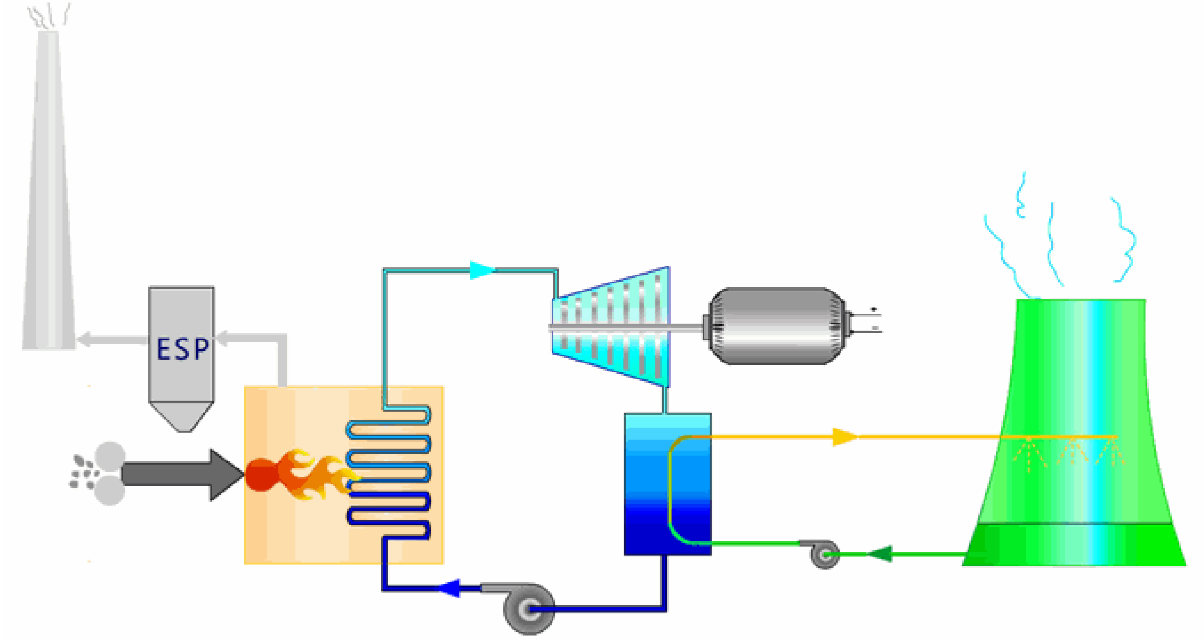

Figure 10 Steam Power Plant (Panakkal, 2016)

Power plants:

Power plants are closed systems and use the same fluid continually. In the first stage, water fills into the boiler, which covers the entire surface area of heat transfer. Inside of the boiler, water is heated by hot gases of combustion the fuel mixed with air then turns the water into vapor. Steam is then produced by the boiler. The pressure and temperature are directed to do work on the turbine to produce mechanical power in the form of rotation. Remaining steam then flows into the condenser, to be cooled with cooling water that turned to water. The Condensate water is used again as feed water for the boiler (Barange, 2017).

Superheating:

Superheating to High Temperatures (Increase TH) Superheating steam increases network output and efficiency of the cycle. And decreases moisture content of the steam at the turbine exit. The temperature to which steam can be superheated is limited by material (Bahrami 2017).

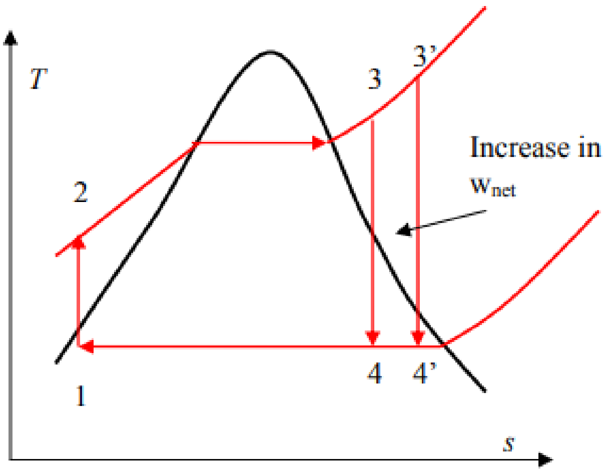

Figure 11 Superheat T-S diagram (Bahrami, 2017)

Regenerative Rankine Cycle:

Regeneration process in steam power plants is done by extracting steam from a turbine at various stages and feeding that steam into a heat exchanger where the feedwater is heated. These heat exchangers types are called feedwater heater (FWH). FWH or regenerators. also help removing the air that leaks in at the condenser de-airing the feedwater. There are two types, open and closed (Bahrami 2017).

Open (Direct-Contact) Feedwater Heaters:

An open FWH is a mixing chamber where steam extracted from turbine mixes with the feedwater exiting the pump. The mixture leaves the heater as a saturated liquid at the heater pressure (Bahrami 2017).

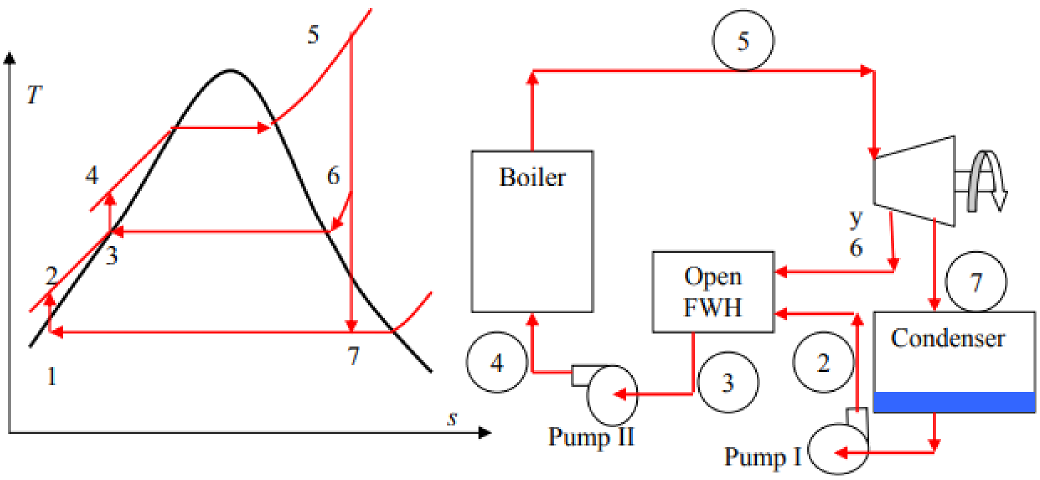

Figure 12 Open Feed and Regeneration (Bahrami, 2017)

Closed Feedwater Heaters:

Closed FWH, heat is shifted from the extracted steam to the feedwater with no mixing. This means two streams can be at different pressures. In an ideal closed FWH, feedwater is heated to the exit temperature of the extracted steam, which leaves the heater as a saturated liquid at extraction pressure (Bahrami, 2017).

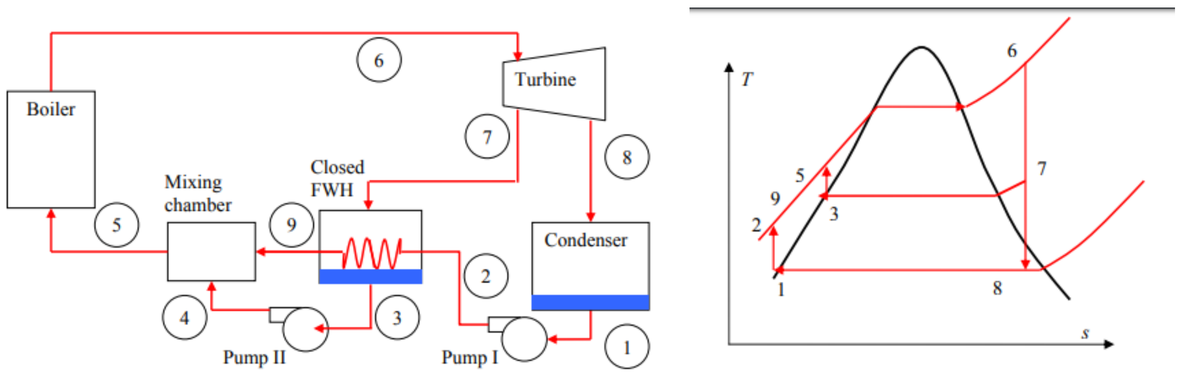

Figure 13 Closed Feed and Regeneration (Bahrami, 2017)

Rankine Cycle Calculation

i) A steam Power Plant operates to the Rankine Cycle. The following is its data.

Facts

Table 12 Rankine Facts

Flow rate 39kg/s
Boiler pressure = 60 bar (6Mpa)
Steam temperature from boiler = 350℃
Condenser pressure 0.04 bar
x = dryness factor
Using steam tables

Assuming isentropic expansion and pumping, determine the following

Table 13 Rankine Question

The power output of the turbine
The power input to the pump
The heat input to the boiler
The heat rejected in the condenser
The thermal efficiency of the cycle

Rankine Cycle

Rankine Schematic Diagram

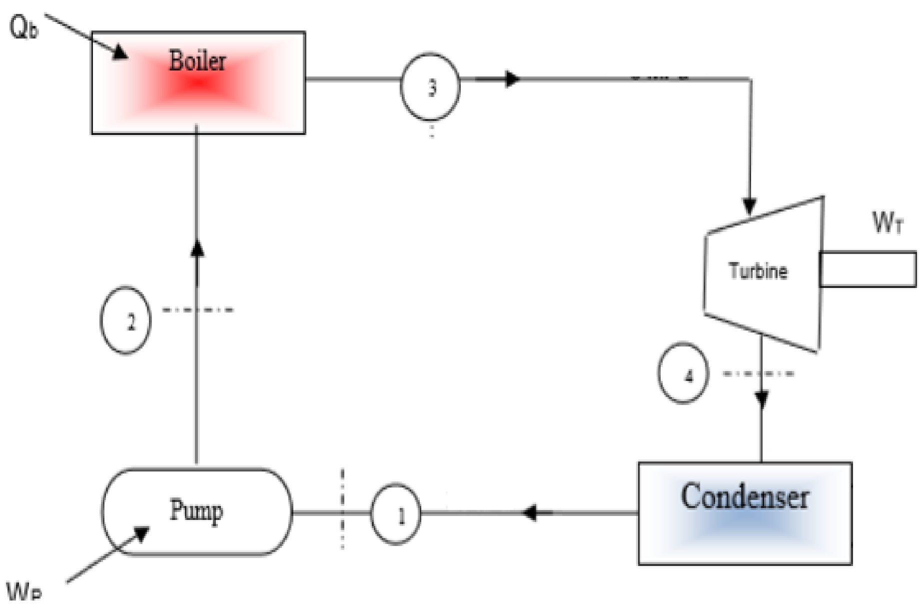

Figure 14 Rankine Schematic Diagram

Ideal T-S diagram

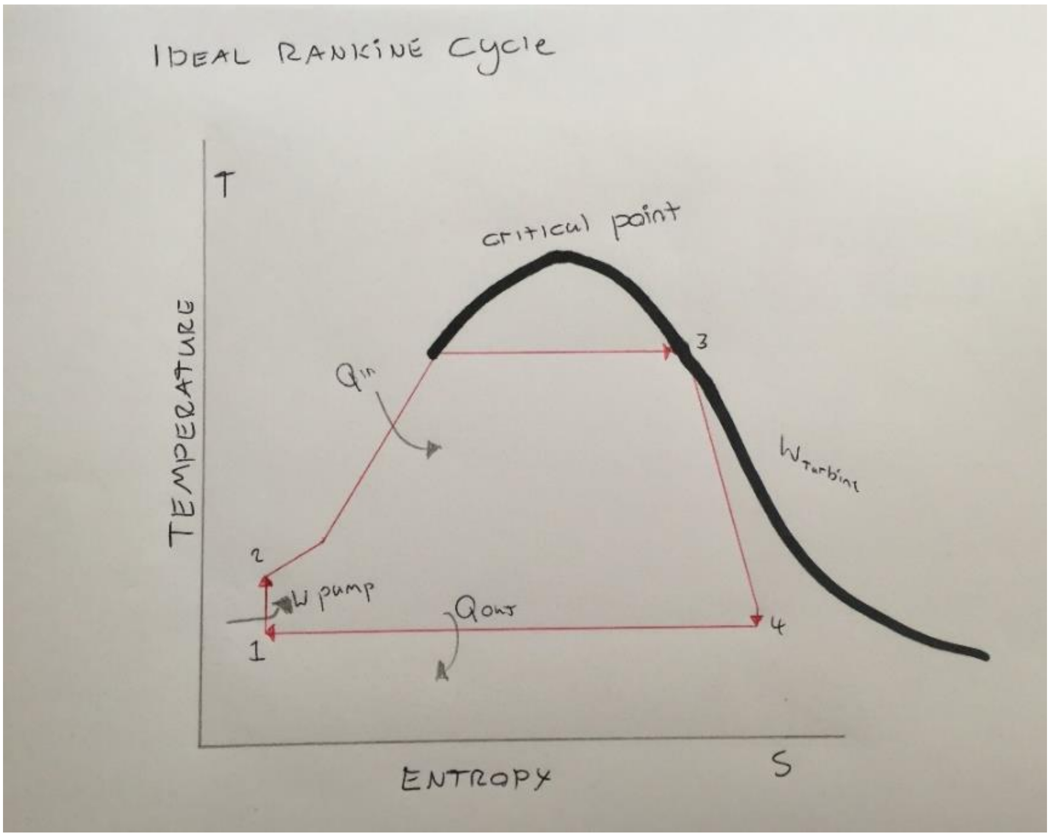

Figure 15 T-S Rankine diagram

The power output of the turbine

First, we need to obtain correct values for the system from the superheated steam table, 60 bars = 6 Mpa, at 350℃.

P=6.0 MPa (275.6°C)

Temp	volume	energy	enthalpy	entropy
°C	v(m^3/kg)	u(kJ/kg)	h(kJ/kg)	s(kJ/kg.K)
Sat.	0.0325	2589.9	2784.6	5.890
300	0.0362	2668.4	2885.5	6.070
350	0.0423	2790.4	3043.9	6.336
400	0.0474	2893.7	3178.2	6.543
450	0.0522	2989.9	3302.9	6.722
500	0.0567	3083.1	3423.1	6.883
600	0.0653	3267.2	3658.7	7.169
700	0.0736	3453.0	3894.3	7.425
800	0.0817	3643.2	4133.1	7.658
900	0.0896	3838.8	4376.6	7.875
1000	0.0976	4040.1	4625.4	8.079

Figure 16 Superheated Steam Table (Urieli, 2018)

Table 14 Steam Table results

h_g @ 350℃ Enthalpy = 3043.9 (kJ/kg) k
s_g @ 350℃ , Entropy = 6.336 (kJ/kg) k

Saturation data for the condenser at 0.04 bar = 0.004Mpa

Saturation Properties for Steam - Pressure Table (1 kPa - 1 MPa)

Pressure	Temp	volume (m^3/kg)		energy (kJ/kg)		enthalpy (kJ/kg)			entropy (kJ/kg.K)		
MPa	°C	vf	vg	uf	ug	hf	hfg	hg	sf	sfg	sg
0.001	6.97	0.00100	129.18	29.3	2384.5	29.3	2484.4	2513.7	0.1059	8.8690	8.9749
0.0012	9.65	0.00100	108.67	40.6	2388.2	40.6	2478.0	2518.6	0.1460	8.7622	8.9082
0.0014	11.97	0.00100	93.90	50.3	2391.3	50.3	2472.5	2522.8	0.1802	8.6720	8.8522
0.0016	14.01	0.00100	82.74	58.8	2394.1	58.8	2467.7	2526.5	0.2100	8.5935	8.8035
0.0018	15.84	0.00100	74.01	66.5	2396.6	66.5	2463.4	2529.9	0.2366	8.5242	8.7608
0.002	17.50	0.00100	66.99	73.4	2398.9	73.4	2459.5	2532.9	0.2606	8.4620	8.7226
0.003	24.08	0.00100	45.65	101.0	2407.9	101.0	2443.8	2544.8	0.3543	8.2221	8.5764
0.004	28.96	0.00100	34.79	121.4	2414.5	121.4	2432.3	2553.7	0.4224	8.0510	8.4734
0.006	36.16	0.00101	23.73	151.5	2424.2	151.5	2415.1	2566.6	0.5208	7.8082	8.3290
0.008	41.51	0.00101	18.10	173.8	2431.4	173.8	2402.4	2576.2	0.5925	7.6348	8.2273
0.01	45.81	0.00101	14.67	191.8	2437.2	191.8	2392.1	2583.9	0.6492	7.4996	8.1488
0.012	49.42	0.00101	12.36	206.9	2442.0	206.9	2383.4	2590.3	0.6963	7.3886	8.0849
0.014	52.55	0.00101	10.69	220.0	2446.1	220.0	2375.8	2595.8	0.7366	7.2945	8.0311
0.016	55.31	0.00102	9.431	231.6	2449.8	231.6	2369.0	2600.6	0.7720	7.2126	7.9846
0.018	57.80	0.00102	8.443	242.0	2453.0	242.0	2363.0	2605.0	0.8036	7.1401	7.9437
0.02	60.06	0.00102	7.648	251.4	2456.0	251.4	2357.5	2608.9	0.8320	7.0752	7.9072
0.03	69.10	0.00102	5.228	289.2	2467.7	289.3	2335.2	2624.5	0.9441	6.8234	7.7675
0.04	75.86	0.00103	3.993	317.6	2476.3	317.6	2318.5	2636.1	1.0261	6.6429	7.6690
0.06	85.93	0.00103	2.732	360.0	2489.0	359.9	2293.0	2652.9	1.1454	6.3857	7.5311
0.08	93.49	0.00104	2.087	391.6	2498.2	391.7	2273.5	2665.2	1.2330	6.2009	7.4339
0.1	99.61	0.00104	1.694	417.4	2505.6	417.5	2257.4	2674.9	1.3028	6.0560	7.3588
0.12	104.78	0.00105	1.428	439.2	2511.7	439.4	2243.7	2683.1	1.3609	5.9368	7.2977
0.14	109.29	0.00105	1.2366	458.3	2516.9	458.4	2231.6	2690.0	1.4110	5.8351	7.2461
0.16	113.30	0.00105	1.0914	475.2	2521.4	475.4	2220.6	2696.0	1.4551	5.7463	7.2014
0.18	116.91	0.00106	0.9775	490.5	2525.5	490.7	2210.7	2701.4	1.4945	5.6676	7.1621

Figure 17 Saturation Table (Urieli, 2018)

Table 15 Saturation Table results

s_f = 0.4224 (kJ/kg) k
s_{fg} = 8.0510 (kJ/kg) k
h_f = 121.4 (kJ/kg) k
h_{fg} = 2432 (kJ/kg) k

Calculations

All relevant data has been selected for the Rankine Cycle, calculations are as follows.

Turbine power output

$\dot{m}\,(h_3-h_4)$

h_3 = 3043.9 kJ/kg (Superheated Steam table, Enthalpy at 6Mpa, at 350℃)

Dryness factor

$x = \frac{s_g - s_f}{s_{fg}} = \frac{6.336-0.4224}{8.0510} = 0.734$

x = 0.734

Enthalpy of Turbine

$h_4 = h_f + x\, h_{fg}$

$121.4 + (0.734 \times 2432) = 1906.4$

$h_4 = 1906.4$ kJ/kg

Output of turbine

$\dot{m}\,(h_3 - h_4)$

$39(3043.9 - 1906.4) = 44362.5$ kW

$P_{output} = 44.4$ Mw

The power input to the pump

$P_{in} = \dot{m} v \Delta p$

Volume of water = $0.001 m^3/kg$

$\dot{m} v\,(p_2 - p_1) \times 10^5$

$39(0.001)\,(60 - 0.04) \times 10^5$

$P_{in} = 233.8$ kW

The heat input to the boiler

$h_2 = 121$kj/kg @ 0.04 bar saturated steam table (specific Enthalpy)

$\phi_{in} = \dot{m}\,(h_3 - h_2)$

$39(3043.9 - 121.4) = 113977.5$ kJ/kg

= 114 MJ/Mg

The heat rejected in the condenser

$h_f = h_1$ at 0.04 bar = 121.4 kJ/kg

$h_4 = 1906.4$

$\phi_{out} = \dot{m}\,(h_1 - h_4)$

$39(121.4 - 1906.4)$

$= - 69615$ kW

$= - 69.6$ MW

The Thermal Efficiency of the Cycle

$P_{output} = h_3 – h_4$

$3043.9 – 1906.4 = 1137.5$ kJ/kg

$\phi_{in} = h_3 – h_f$

$3043.9 – 121.4 = 2922.5$

$\eta_{th} = p_{out} / \phi_{in} = 1137.5/2922.5 = 0.389 X 100 = 39$

= 39 % Thermal efficient

or

$\eta_{th} = 1 – \phi_{in} / \phi_{out}$

$= 1 - 69.6/114 = 0.389$

= 0.39 % Thermal efficient

Specific Steam Consumption

$\phi in - \phi out = pnett$

$114 – 69.6 = 44.4$ MW

$SSC = p_{nett} / \dot{m}$

$45/39 = 1.1538$ MW/Kgs or MJ/kg

Rankine Cycle Operation

The pressure of the saturated liquid that is leaving the condenser is at state 1-2. It is then subjected to a raise in an adiabatic reversible process via the pump to state number 2. At this point it enters the boiler. The compressed liquid is then heated at constant pressure, until it reaches a saturated liquid state 2-3 at a constant pressure and temperature until the liquid that has vaporized becomes saturated vapor which occurs at state number 3.

Additional heat is added to superheat, the saturated vapor at a constant pressure, and its temperature then rises to state 3-4. The superheated vapor then enters a Turbine and expands in an adiabatic and reversible process towards low pressure held by the condenser, which is indicated as state 4. And at stage 4-1, the condenser then converts the vapor that leaves the turbine to liquid by extracting all the heat from it.

P-V - H-S Diagram Rankin Cycle

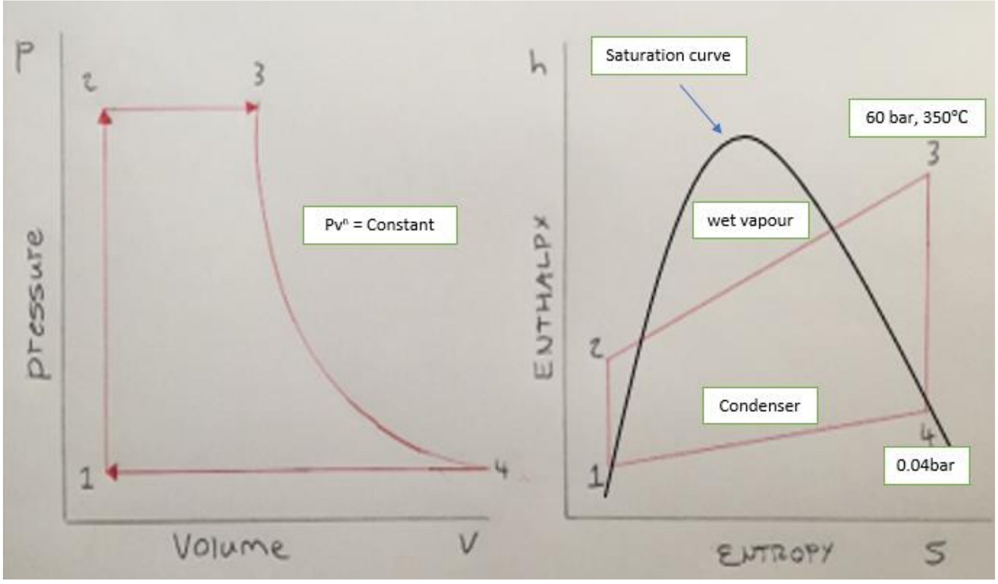

Figure 18 P-V and H-S Diagram Rankine

ii)To improve the Thermal Efficiency of the above Rankine Cycle redesign it to use reheat. Draw a temperature/Entropy diagram and show all your calculations to determine the increase in Thermal Efficiency.

Reheat Rankine Calculation

T-S Diagram Results

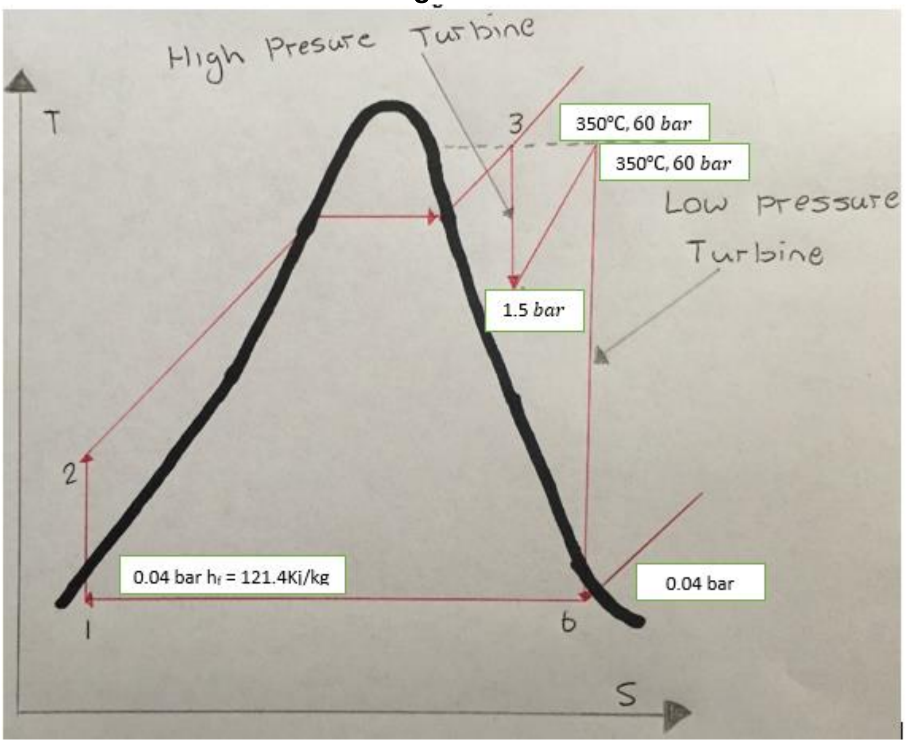

Figure 19 T-S Diagram Reheat Rankine

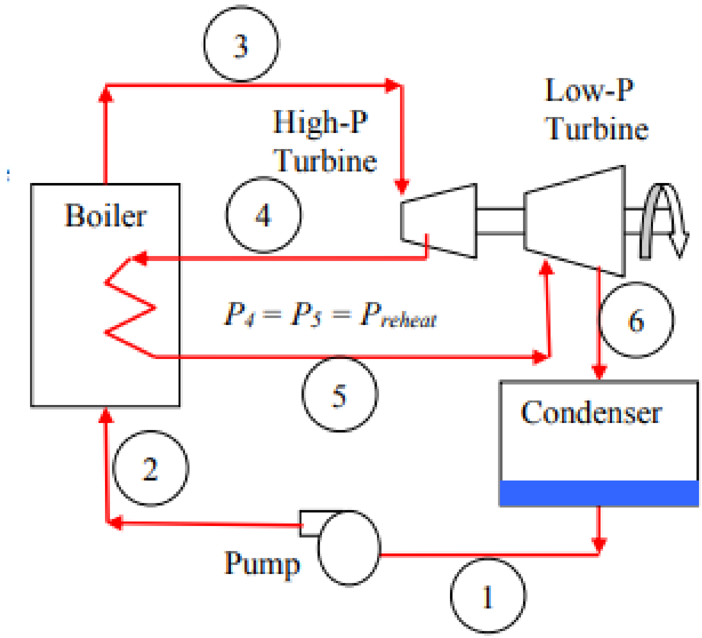

Figure 20 Rankine Reheat diagram (Bahrami, 2017)

Table 16 Rankine Facts Table

Flow rate 39kg/s
Boiler pressure = 60 bar (6Mpa)
Steam temperature from boiler = 350℃
Condenser pressure 0.04 bar
X = dryness factor
Assume a value of ¼ of the inlet pressure and temperature of the turbine = 1.5Mpa

Position 3 (60 bar, 350°C)

Superheated Steam Chart

Table 17 Rankine Superheat System results 6 bar

$h_g = 3043.9 \text{ kJ/kg}$
$s_g = 6.336 \text{ kJ/kg k}$

Position 4 (15 bar, 350°C)

The Ideal Reheat Rankine cycle without given values for the High-Pressure Turbines return reheat section should be a ¼ of the pressure intake (Turns, 2006).

60 bars Divided by 1/4 = 15 bar or 1.5Mpa.

Reheating the Rankine cycle relieves work done by the boiler, sending a small amount of superheated steam back into the boiler reducing the work done by the boiler. This improves the thermal efficiency of the cycle, it does not increase work done on the turbine, also interpolation techniques relay on obscure values between Temperatures. We have a set temperature. The best method in this case would be to average the values for pressures 1.4 and 1.6. Once values are obtained, accuracy of values will be demonstrated using an online calculator.

Saturation Properties for Steam - Pressure Table (1 MPa - 22.064 MPa)

Pressure	Temp	volume (m^3/kg)		energy (kJ/kg)		enthalpy (kJ/kg)			entropy (kJ/kg.K)		
MPa	°C	vf	vg	uf	ug	hf	hfg	hg	sf	sfg	sg
1	179.88	0.00113	0.1944	761.4	2582.7	762.5	2014.6	2777.1	2.1381	4.4469	6.5850
1.2	187.96	0.00114	0.1633	797.0	2587.8	798.3	1985.4	2783.7	2.2159	4.3058	6.5217
1.4	195.04	0.00115	0.1408	828.4	2591.8	830.0	1958.8	2788.8	2.2835	4.1840	6.4675
1.6	201.37	0.00116	0.1237	856.6	2594.8	858.5	1934.3	2792.8	2.3435	4.0764	6.4199
1.8	207.11	0.00117	0.1104	882.4	2597.2	884.5	1911.4	2795.9	2.3975	3.9800	6.3775
2	212.38	0.00118	0.0996	906.1	2599.1	908.5	1889.8	2798.3	2.4468	3.8922	6.3390
3	233.85	0.00122	0.0667	1004.7	2603.2	1008.3	1794.9	2803.2	2.6455	3.5401	6.1856
4	250.35	0.00125	0.0498	1082.5	2601.7	1087.5	1713.3	2800.8	2.7968	3.2728	6.0696
6	275.58	0.00132	0.0325	1206.0	2589.9	1213.9	1570.7	2784.6	3.0278	2.8623	5.8901
8	295.01	0.00139	0.0235	1306.2	2570.5	1317.3	1441.4	2758.7	3.2081	2.5369	5.7450
10	311.00	0.00145	0.0180	1393.5	2545.2	1408.1	1317.4	2725.5	3.3606	2.2554	5.6160
12	324.68	0.00153	0.0143	1473.1	2514.3	1491.5	1193.9	2685.4	3.4967	1.9972	5.4939
14	336.67	0.00161	0.0115	1548.4	2477.1	1571.0	1066.9	2637.9	3.6232	1.7495	5.3727
16	347.35	0.00171	0.0093	1622.3	2431.8	1649.7	931.1	2580.8	3.7457	1.5006	5.2463
18	356.99	0.00184	0.0075	1699.0	2374.8	1732.1	777.7	2509.8	3.8718	1.2343	5.1061
20	365.75	0.00204	0.0059	1786.4	2295.0	1827.2	585.1	2412.3	4.0156	0.9158	4.9314
22.064	373.95	0.00311	0.00311	2015.7	2015.7	2084.3	0	2084.3	4.4070	0	4.4070

Figure 21 Saturation Table (NIST Chemistry WebBook, 2008)

$$\mathbf{example} = \frac{\mathbf{h_{f+}\,h_f}}{\mathbf{2}} = 844\ \mathrm{kJ/kg\ k}$$

Saturated Steam Table Results

Table 18 Saturated Steam Table Pressure Results

Mpa	s_f	s_{fg}	h_f	h_{fg}
1.4	2.2835	4.1840	830	1958.8
1.6	2.3435	4.0764	858.5	1934.3

Mean Results

Table 19 Mean Average Results

$s_f = 2.3135\ \mathrm{kJ/kg\ k}$
$s_{fg} = 4.1302\ \mathrm{kJ/kg\ k}$
$h_f = 844\ \mathrm{kJ/kg\ k}$
$h_{fg} = 1946.55\ \mathrm{kJ/kg\ k}$

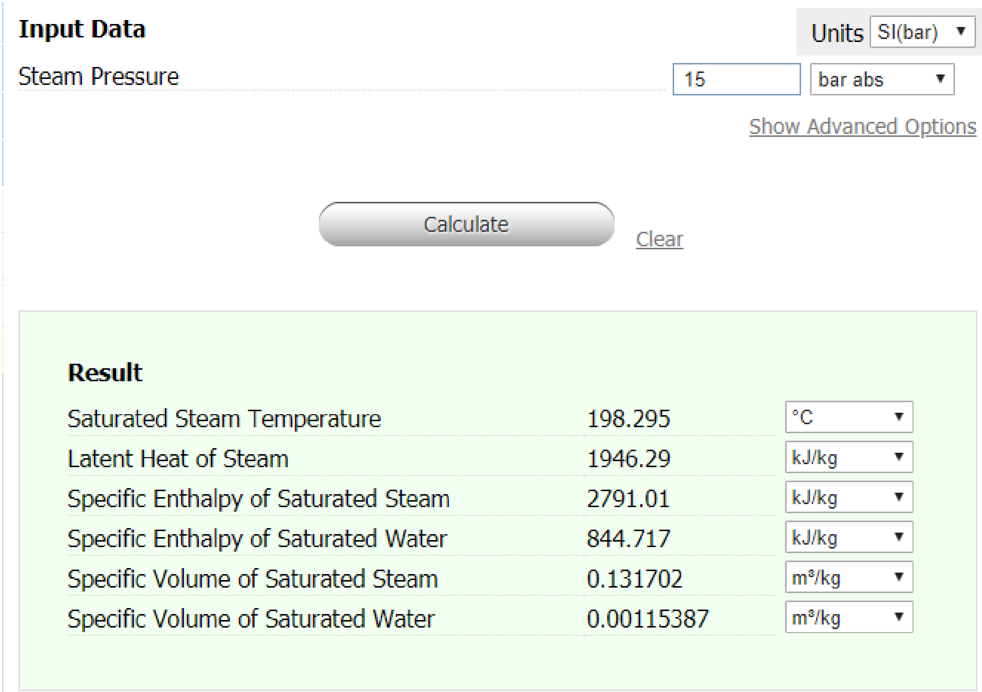

Figure 22 Online Calculator (TLV.com, 2018)

From results obtained, it shows the values have been averaged. The online calculator allows us to see pressure results at 1.5Mpa (15 bar), specific enthalpy of saturated water steam (h_f), and the latent heat of steam (h_{fg}) are the same as the values, give or take rounding and steam table types used.

$$\text{Dryness fraction } (x_4) = \frac{s_g - s_f}{s_{fg}}$$

$$\text{Dryness fraction } (x_4) = \frac{6.336 - 2.3135}{4.1302}$$

$\text{Dryness fraction } (x_4) = \ 0.97$ (good dryness fraction)

$$h_4 = h_f + x_4 h_{fg}$$

$$h_4 = 844 + (0.97 \times 1946.55)$$

$$h_4 = 2732.15 \text{ kJ/kg}$$

Position 5

The results need to be averaged from both superheated steam charts for 1.4 – 1.6 Mpa to obtain values for 1.5Mpa.

P=1.40 MPa (195.0°C)			
volume	energy	enthalpy	entropy
v(m^3/kg)	u(kJ/kg)	h(kJ/kg)	s(kJ/kg.K)
0.1408	2591.8	2788.8	6.468
0.1430	2602.7	2803.0	6.498
0.1636	2698.9	2927.9	6.749
0.1823	2785.7	3040.9	6.955
0.2003	2869.7	3150.1	7.138
0.2178	2953.1	3258.1	7.305
0.2351	3037.0	3366.1	7.459
0.2522	3121.8	3474.8	7.605
0.2860	3295.1	3695.4	7.873
0.3195	3474.4	3921.7	8.118
0.3529	3660.2	4154.3	8.346
0.3861	3852.7	4393.3	8.559
0.4193	4051.7	4638.8	8.759

Figure 23 Superheated Steam Table 1.40Mpa (NIST Chemistry WebBook, 2008 (2))

Table 20 Superheated Steam Table 1.40Mpa Values

$h_g = 3150.1$ kJ/kg
$s_g = 7.138$ kJ/kg K

P=1.60 MPa (201.4°C)				
Temp	volume	energy	enthalpy	entropy
°C	v(m^3/kg)	u(kJ/kg)	h(kJ/kg)	s(kJ/kg.K)
Sat.	0.1237	2594.8	2792.8	6.420
225	0.1329	2645.1	2857.8	6.554
250	0.1419	2692.9	2919.9	6.675
300	0.1587	2781.6	3035.4	6.886
350	0.1746	2866.6	3146.0	7.071
400	0.1901	2950.7	3254.9	7.239
450	0.2053	3035.0	3363.5	7.395
500	0.2203	3120.1	3472.6	7.541
600	0.2500	3293.9	3693.9	7.810
700	0.2794	3473.5	3920.5	8.056
800	0.3087	3659.5	4153.3	8.283
900	0.3378	3852.1	4392.6	8.497
1000	0.3669	4051.2	4638.2	8.697

Figure 24 Superheated Steam Table 1.6Mpa (NIST Chemistry WebBook, 2008 (3))

Table 21 Superheated Steam Table 1.60Mpa Values

$h_g = 3146$ kJ/kg
$s_g = 7.071$ kJ/kg K

Saturated steam table

$$\mathbf{example} = \frac{\mathbf{h_{g+}\,h_g}}{\mathbf{2}} = 3148.05 \text{ kJ/kg k}$$

Table 22 Steam Table Results

Mpa	h_g	s_g
1.4	3150.1 kJ/kg	7.138 kJ/kg K
1.6	3146 kJ/kg	7.071 kJ/kg K

Table 23 Steam Table Mean Results, 1.5Mpa

$h_g = 3148.05$ kJ/kg
$s_g = 7.1045$ kJ/kg K

Position 5

Is at 15 bars (1.5Mpa), the reheat pressure into the boiler is a ¼ of the high-pressure turbine inlet pressure 60 bar (6Mpa), when no value is given.

Position 6 (0.04 bar)

Table 24 Saturation Table Condenser Values

$s_f = 0.4224$ kJ/kg K
$s_{fg} = 8.0510$ kJ/kg K
$h_f = 121.4$ kJ/kg
$h_{fg} = 2432.3$ kJ/kg K

Saturation Properties for Steam - Pressure Table (1 kPa - 1 MPa)

Pressure	Temp	volume (m^3/kg)		energy (kJ/kg)		enthalpy (kJ/kg)			entropy (kJ/kg.K)		
MPa	°C	vf	vg	uf	ug	hf	hfg	hg	sf	sfg	sg
0.001	6.97	0.00100	129.18	29.3	2384.5	29.3	2484.4	2513.7	0.1059	8.8690	8.9749
0.0012	9.65	0.00100	108.67	40.6	2388.2	40.6	2478.0	2518.6	0.1460	8.7622	8.9082
0.0014	11.97	0.00100	93.90	50.3	2391.3	50.3	2472.5	2522.8	0.1802	8.6720	8.8522
0.0016	14.01	0.00100	82.74	58.8	2394.1	58.8	2467.7	2526.5	0.2100	8.5935	8.8035
0.0018	15.84	0.00100	74.01	66.5	2396.6	66.5	2463.4	2529.9	0.2366	8.5242	8.7608
0.002	17.50	0.00100	66.99	73.4	2398.9	73.4	2459.5	2532.9	0.2606	8.4620	8.7226
0.003	24.08	0.00100	45.65	101.0	2407.9	101.0	2443.8	2544.8	0.3543	8.2221	8.5764
0.004	28.96	0.00100	34.79	121.4	2414.5	121.4	2432.3	2553.7	0.4224	8.0510	8.4734
0.006	36.16	0.00101	23.73	151.5	2424.2	151.5	2415.1	2566.6	0.5208	7.8082	8.3290
0.008	41.51	0.00101	18.10	173.8	2431.4	173.8	2402.4	2576.2	0.5925	7.6348	8.2273
0.01	45.81	0.00101	14.67	191.8	2437.2	191.8	2392.1	2583.9	0.6492	7.4996	8.1488
0.012	49.42	0.00101	12.36	206.9	2442.0	206.9	2383.4	2590.3	0.6963	7.3886	8.0849
0.014	52.55	0.00101	10.69	220.0	2446.1	220.0	2375.8	2595.8	0.7366	7.2945	8.0311
0.016	55.31	0.00102	9.431	231.6	2449.8	231.6	2369.0	2600.6	0.7720	7.2126	7.9846
0.018	57.80	0.00102	8.443	242.0	2453.0	242.0	2363.0	2605.0	0.8036	7.1401	7.9437
0.02	60.06	0.00102	7.648	251.4	2456.0	251.4	2357.5	2608.9	0.8320	7.0752	7.9072
0.03	69.10	0.00102	5.228	289.2	2467.7	289.3	2335.2	2624.5	0.9441	6.8234	7.7675
0.04	75.86	0.00103	3.993	317.6	2476.3	317.6	2318.5	2636.1	1.0261	6.6429	7.6690
0.06	85.93	0.00103	2.732	360.0	2489.0	359.9	2293.0	2652.9	1.1454	6.3857	7.5311
0.08	93.49	0.00104	2.087	391.6	2498.2	391.7	2273.5	2665.2	1.2330	6.2009	7.4339
0.1	99.61	0.00104	1.694	417.4	2505.6	417.5	2257.4	2674.9	1.3028	6.0560	7.3588
0.12	104.78	0.00105	1.428	439.2	2511.7	439.4	2243.7	2683.1	1.3609	5.9368	7.2977
0.14	109.29	0.00105	1.2366	458.3	2516.9	458.4	2231.6	2690.0	1.4110	5.8351	7.2461
0.16	113.30	0.00105	1.0914	475.2	2521.4	475.4	2220.6	2696.0	1.4551	5.7463	7.2014
0.18	116.91	0.00106	0.9775	490.5	2525.5	490.7	2210.7	2701.4	1.4945	5.6676	7.1621

Figure 25 Saturation of Steam, Pressure Table (NIST Chemistry WebBook, 2008 (4))

$$\text{Dryness fraction } (x_6) = \frac{s_g - s_f}{s_{fg}}$$

$$\text{Dryness fraction } (x_6) = \frac{7.0145 - 0.4224}{8.0510}$$

Dryness fraction $(x_6) = 0.81$

$h_6 = h_f + x_4 h_{fg}$

$h_6 = 121.4 + (0.81 \times 2432)$

$h_6 = 2091.32 \text{ kJkg}$

Position 1 (0.1 bar)

$h_f = 121.4 \text{ kJ/kg}$

Position 2 (0.1 bar)

Specific volume of water at 0.1 bar $= 0.001010 \text{ m}^3/\text{kg}$

$W_{p\ in} = V_1(p_2 - p_1)$

$W_{p\ in} = 0.00100 \times (60 - 0.4) \times 10^5$

$W_{p\ in} = 5960 \text{ J/kg}$

$W_{p\ in} = 6 \text{ kJ/kg}$

$h_2 = h_1 + W_{in}$

$h_2 = 121.4 + 6$

$h_2 = 127.4$

Thermal efficiency $(\eta_{th}) = \frac{\phi_{out}}{\phi_{in}}$

Thermal efficiency (η_{th})

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

Thermal efficiency $(\boldsymbol{\eta_{th}})$

$$= \frac{(3043.9 - 2732.15) + (3364.8 - 2091.32) - (127.4 - 121.4)}{(3043.9 - 127.4) + (3364.8 - 2732.15)}$$

$$(\eta_{th}) = \frac{311.75 + 1273.48 - 6}{2916.5 + 632.65}$$

$$(\eta_{th}) = \frac{1591.23}{3549.15}$$

$$(\eta_{th}) = 0.44$$

$$(\eta_{th}) = 44\%$$

Conclusion:
There is a 5% increase in thermal efficiency of the reheat Rankine Cycle when compared to standard Rankine Cycle. This is because heat from the HP turbine is fed back into the boiler which reduces the work done by the boiler.

Part 3

Calculation Coursework

i) A refrigeration system uses refrigerant, the upper and lower temperature are 35℃ and -18℃ respectively, find the following:

Table 25 Questions

Refrigeration Effect
Draw and fully label the temperature/entropy diagram
Coefficient of performance (COP)
Reversed Carnot Efficiency

Facts

Table 26 Facts

x = dryness factor
Using Refrigerant 717 Table

Ammonia 717 Table

Saturation Values							50 K		100 K	
T [°C]	p_s [bar]	v_g [m³/kg]	h_f [kJ/kg]	h_g [kJ/kg]	s_f [kJ/kg K]	s_g [kJ/kg K]	h [kJ/kg]	s [kJ/kg K]	h [kJ/kg]	s [kJ/kg K]
−50	0.4089	2.625	−44.4	1373.3	−0.194	6.159	1479.8	6.592	1585.9	6.948
−45	0.5454	2.005	−22.3	1381.6	−0.096	6.057	1489.3	6.486	1596.1	6.839
−40	0.7177	1.552	0	1390.0	0	5.962	1498.6	6.387	1606.3	6.736
−35	0.9322	1.216	22.3	1397.9	0.095	5.872	1507.9	6.293	1616.3	6.639
−30	1.196	0.9633	44.7	1405.6	0.188	5.785	1517.0	6.203	1626.3	6.547
−28	1.317	0.8809	53.6	1408.5	0.224	5.751	1520.7	6.169	1630.3	6.512
−26	1.447	0.8058	62.6	1411.4	0.261	5.718	1524.3	6.135	1634.2	6.477
−24	1.588	0.7389	71.7	1414.3	0.297	5.686	1527.9	6.103	1638.2	6.444
−22	1.740	0.6783	80.8	1417.3	0.333	5.655	1531.4	6.071	1642.2	6.411
−20	1.902	0.6237	89.8	1420.0	0.368	5.623	1534.8	6.039	1646.0	6.379
−18	2.077	0.5743	98.8	1423.7	0.404	5.593	1538.2	6.008	1650.0	6.347
−16	2.265	0.5296	107.9	1425.3	0.440	5.563	1541.7	5.978	1653.8	6.316
−14	2.465	0.4890	117.0	1427.9	0.475	5.533	1545.1	5.948	1657.7	6.286
−12	2.680	0.4521	126.2	1430.5	0.510	5.504	1548.5	5.919	1661.5	6.256
−10	2.908	0.4185	135.4	1433.0	0.544	5.475	1551.7	5.891	1665.3	6.227
−8	3.153	0.3879	144.5	1435.3	0.579	5.447	1554.9	5.863	1669.0	6.199
−6	3.413	0.3599	153.6	1437.6	0.613	5.419	1558.2	5.836	1672.8	6.171
−4	3.691	0.3344	162.8	1439.9	0.647	5.392	1561.4	5.808	1676.4	6.143
−2	3.983	0.3110	172.0	1442.2	0.681	5.365	1564.6	5.782	1680.1	6.116
0	4.295	0.2895	181.2	1444.4	0.715	5.340	1567.8	5.756	1683.9	6.090
2	4.625	0.2699	190.4	1446.5	0.749	5.314	1570.9	5.731	1687.5	6.065
4	4.975	0.2517	199.7	1448.5	0.782	5.288	1574.0	5.706	1691.2	6.040
6	5.346	0.2351	209.1	1450.6	0.816	5.263	1577.0	5.682	1694.9	6.015
8	5.736	0.2198	218.5	1452.5	0.849	5.238	1580.1	5.658	1698.4	5.991
10	6.149	0.2056	227.8	1454.3	0.881	5.213	1583.1	5.634	1702.2	5.967
12	6.585	0.1926	237.2	1456.1	0.914	5.189	1586.0	5.611	1705.7	5.943
14	7.045	0.1805	246.6	1457.8	0.947	5.165	1588.9	5.588	1709.1	5.920
16	7.529	0.1693	256.0	1459.5	0.979	5.141	1591.7	5.565	1712.5	5.898
18	8.035	0.1590	265.5	1461.1	1.012	5.118	1594.4	5.543	1715.9	5.876
20	8.570	0.1494	275.1	1462.6	1.044	5.095	1597.2	5.521	1719.3	5.854
22	9.134	0.1405	284.6	1463.9	1.076	5.072	1600.0	5.499	1722.8	5.832
24	9.722	0.1322	294.1	1465.2	1.108	5.049	1602.7	5.418	1726.3	5.811
26	10.34	0.1245	303.7	1466.5	1.140	5.027	1605.3	5.458	1729.6	5.790
28	10.99	0.1173	313.4	1467.8	1.172	5.005	1608.0	5.437	1732.7	5.770
30	11.67	0.1106	323.1	1468.9	1.204	4.984	1610.5	5.417	1735.9	5.750
32	12.37	0.1044	332.8	1469.9	1.235	4.962	1613.0	5.397	1739.3	5.731
34	13.11	0.0986	342.5	1470.8	1.267	4.940	1615.4	5.378	1742.6	5.711
36	13.89	0.0931	352.3	1471.8	1.298	4.919	1617.8	5.358	1745.7	5.692
38	14.70	0.0880	362.1	1472.6	1.329	4.898	1620.1	5.340	1748.7	5.674
40	15.54		371.9	1473.3	1.360	4.877	1622.4	5.321	1751.9	5.655

Figure 26 Ammonia 717 Table Saturation Values (Rogers & Mayhew, 1995)

Note: The mean values of 34℃ and 36℃ were used to determine the values for 35℃

Position 3

35℃

Table 27 Position 3, h_f Value

$h_{f2} = 347.4\ \mathrm{kJ/kg} = h_4 = h_f$
$h_2 = h_4 = h_f = 347.4\ \mathrm{kJ/kg}$

Position 2

35℃

Table 28 Mean Values of 35-36℃

$h_{g2} = 1471.3 \text{ kJ/kg K}$
$s_{g2} = 4.9295 \text{ kJ/kg K}$

Position 1

-18℃

Table 29 Values at -18℃

$s_{f1} = 0.404 \text{ kJ/kg K}$
$s_{g1} = 5.593 \text{ kJ/kg K}$

Dryness Factor

$$\text{Dryness fraction } (x) = \frac{s_1 - s_{f1}}{s_{g1} - s_f}$$

$$(x) = \frac{4.9295 - 0.404}{5.593 - 0.404} = 0.872133359$$

$$s_1 = s_f + x\left(s_g + s_f\right)$$

$$4.9295 = 0.404 + 0.872133359\ (5.953 + 0.404)$$

$$h_1 = h_f + x(h_g - h_f)$$

$$h_1 = 347.4 + 0.872133359(1471.3 - 347.4)$$

$$h_1 = 1327.59 \text{ kJ/kg}$$

Refrigeration effect

COP = $\frac{Q_{in}}{W_{nett}}$ = $\frac{Q_{41}}{W_{12}}$

$$Q_{41} = h_1 - h_4$$

$$Q_{41} = 1327.59 - 347.4$$

Q_{41} = 980.1

Network Expended

$$W_{12} = h_2 - h_1$$

$$W_{12} = 1471.3 - 1327.59$$

$$W_{12} = 143.71$$

$$COP_R = \frac{h_1 - h_4}{h_2 - h_1}$$

$$COP_R = \frac{1327.59 - 347.4}{1471.3 - 1327.59}$$

$$COP_R = 6.82$$

Corresponding Carnot Efficiency

$$T_c = -18 + 273 = 255\ K$$

$$T_H = 35 + 273 = 308\ K$$

$$COP = \frac{T_c}{T_H - T_c}$$

$$COP = \frac{255}{308 - 355}$$

$COP_R = 4.81$ (Sole, 2018)

Conclusion

The Carnot is a less efficient Method.

Vapour - Compression Refrigeration Cycles

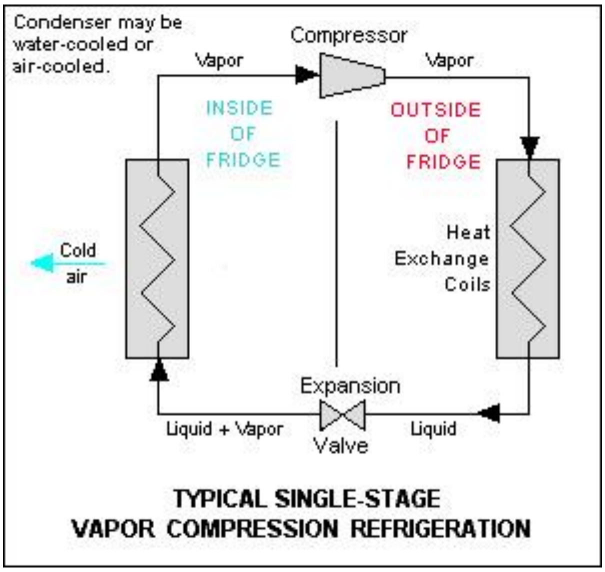

Figure 27 Vapour Compression Cycle (FridgeFilters.com, 2018)

Refrigerant has a very low boiling point between -26 °C (or -15 °F). gas is compressed by a compressor, it then heats up, wanting to boil and expand. then it is fed through the large heat-exchangers coil at the back of a refrigeration system which allows the heat to disperse evenly. Through the coils, refrigerant cools and condenses into liquid form. The cooled refrigerant flows through the expansion valve of the evaporator and evaporates into a gas. Gas then travels through coils inside the fridge around food compartments. Moving through the system it expands further capturing heat inside the fridge, which gets carried back out to the back of the fridge. Gas then re-enters the compressor, and the cycle begins again (FridgeFilters.com, 2018).

Throttle Valve

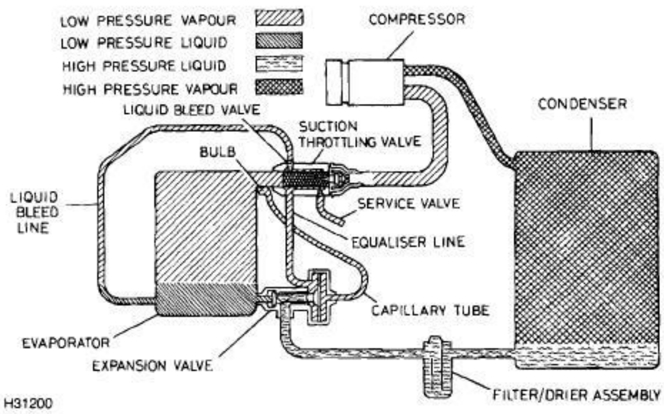

Figure 28 Throttle Valve (Auto Aircon, 2015)

A Throttling devise is associated with a refrigeration system and air conditioning system but not the evaporator compressor, or condenser (Khemani, 2009). For example, Refrigerant 717 exits a compressor at high temperature and pressure and enters the condenser. When 717 leaves the condenser, it is at medium to high pressure, it then enters the throttling valve. Within the throttle valve the pressure and the temperature of 717 is reduced suddenly. The throttle valve where the temperature of the 717 is reduced, is then able to produce the cooling effect in the evaporator of the refrigeration system or by cooling the coil of an air conditioner. The throttling valve controls the amount of the refrigerant that should enter the evaporator depending on the refrigeration load, the change in Enthalpy across a throttling valve is = 0.and value h_4 can be found from h_3 (Khemani, 2009).

Flash Chamber

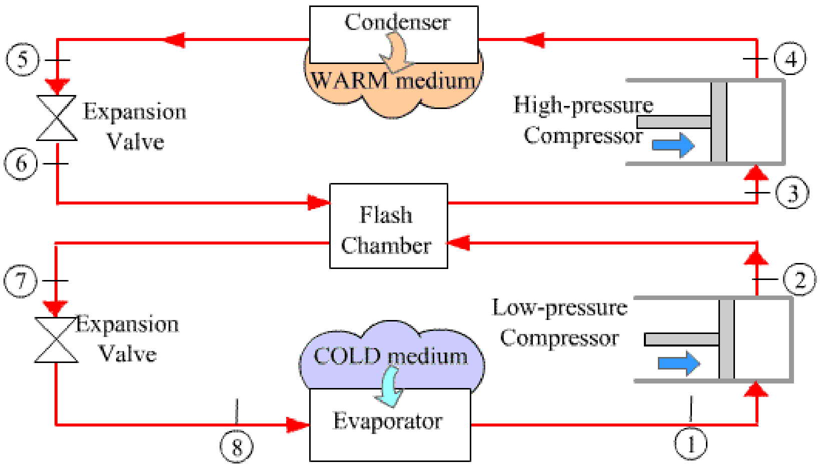

Figure 29 Flash Chamber (Bhattacharjee, 2010)

Flash chamber is used in multistage refrigeration systems and exists between expansion valve and evaporator coil. When refrigerant is passed through expansion valve in which pressure and temperature drops during which some amount of refrigerant in liquid form converts to vapour. The vapour is sent along with low temperature refrigerant into the evaporator, efficiency decreases due to reduction of contact surface between liquid refrigerant and evaporator internal surfaces reducing the overall efficiency of a system (Nanguluri, 2015).

Condition at Compressor Inlet

Temperature

Inlet Temperature produces large changes in performance. In cold weather, a centrifugal can deliver much more weight flow of air than in warm weather if the drive is sized to provide the additional power required.

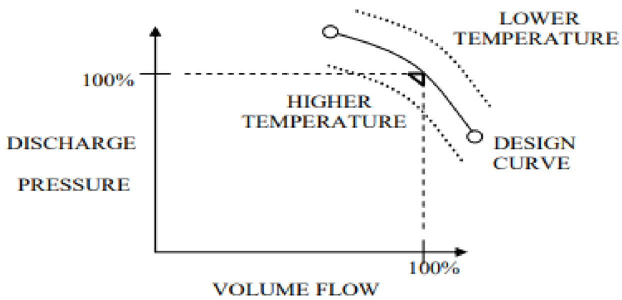

Figure 30 Density of air increases with reduction of air temperature (Stasyshan, 2018)

Lower Temperature

Table 30 Lower Tempreture

Increases the surge pressure.
Increases the maximum capacity (weight flow) at a given discharge pressure.
Increases power consumption (horsepower)

Higher Temperature

Table 31 Higher Tempreture

Decreases the surge pressure
Decreases the maximum capacity (weight flow) at a given discharge pressure
Decreases power consumption (horsepower)

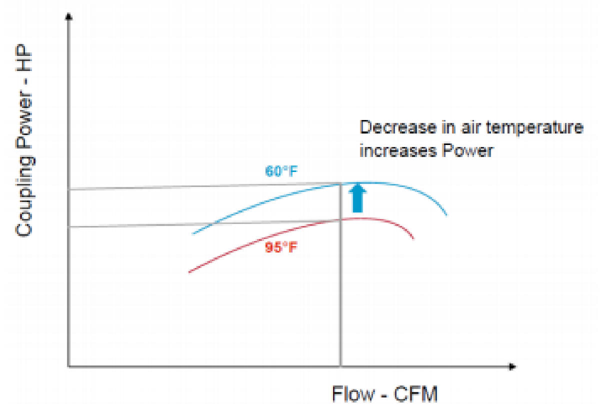

Figure 31 How inlet temperature affects power (Stasyshan, 2018)

Pressure

A decrease in inlet pressure will reduce the density of the air at the compressor intake. As with higher temperatures, it will result in lower free air delivery and power. Changes in inlet pressure can be caused by fouled inlet filters or changing barometric pressure. The same goes for the available turndown lower intake pressure it will result in smaller available turndown.

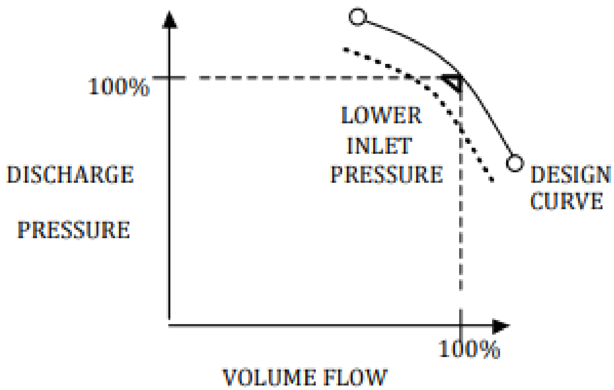

Figure 32 How inlet pressure impacts centrifugal compressor performance (Stasyshan, 2018)

Lower inlet pressure

Table 32 Lower Inlet Pressure

Decreases the discharge pressure along the entire curve
Decreases the maximum capacity (weight flow)
Decreases power consumption or horsepower (due to reduced weight flow).

(Compressed Air and Gas Institute, 2015)

Undercooling/Sub Cooling

Table 33 Under-Cooling Attributes

As a condenser undercooling increases, COP undergoes a maximum
COP maximizing undercooling is not a strong function of thermodynamic properties
Certain refrigerants benefit more than others when condensing
COP increase due to condenser undercooling is a function of thermodynamics

About

After condensation, refrigerant is cooled below the saturation temperature before entering the expansion valve. This is done to increase the COP. Generally, the refrigerant is superheated after compression and sub-cooled before throttling to increase the COP. the process of under cooling is done by circulating more quantity of cooling water through the condenser or by using water colder than main circulating water or by employing a heat exchanger (Mechanical Stuff4U, 2018).

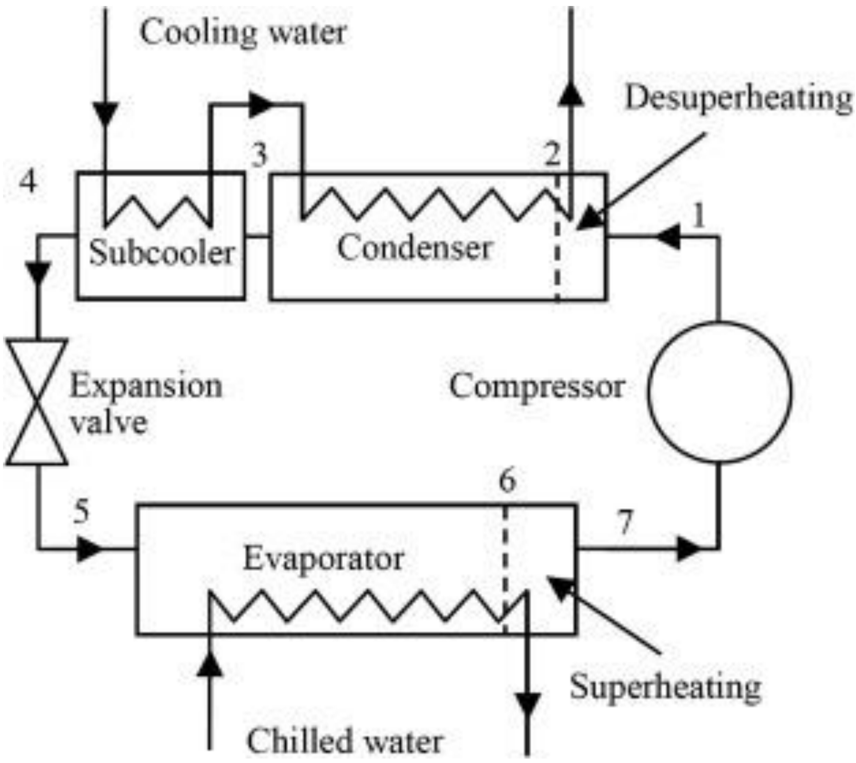

Figure 33 Under-Cooling (Min-Hsiung & Rong-Hua, 2015)

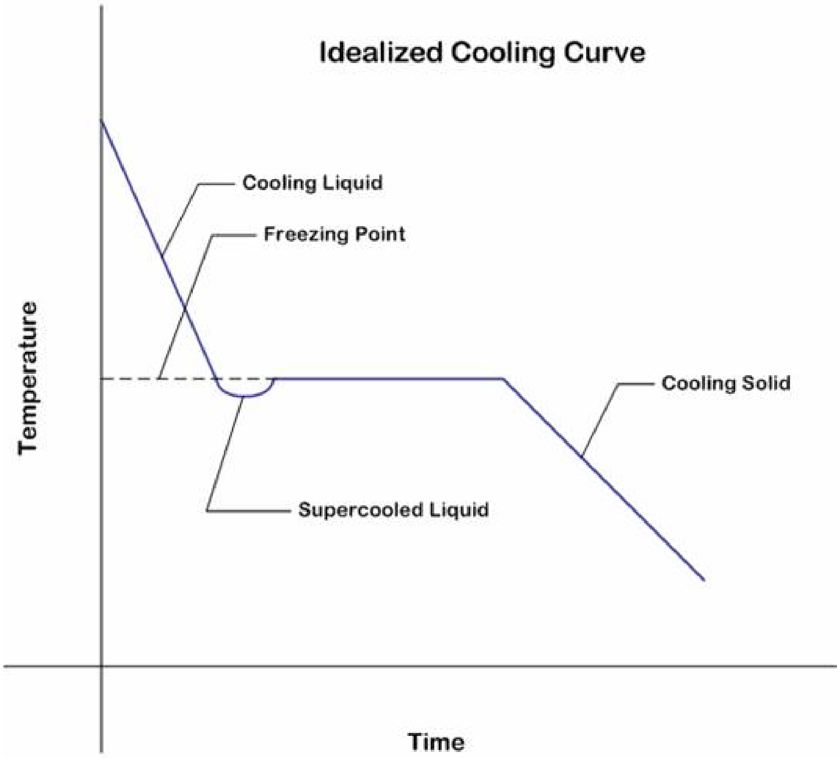

Figure 34 Idealised Cooling Curve (StackExchange, 2015)

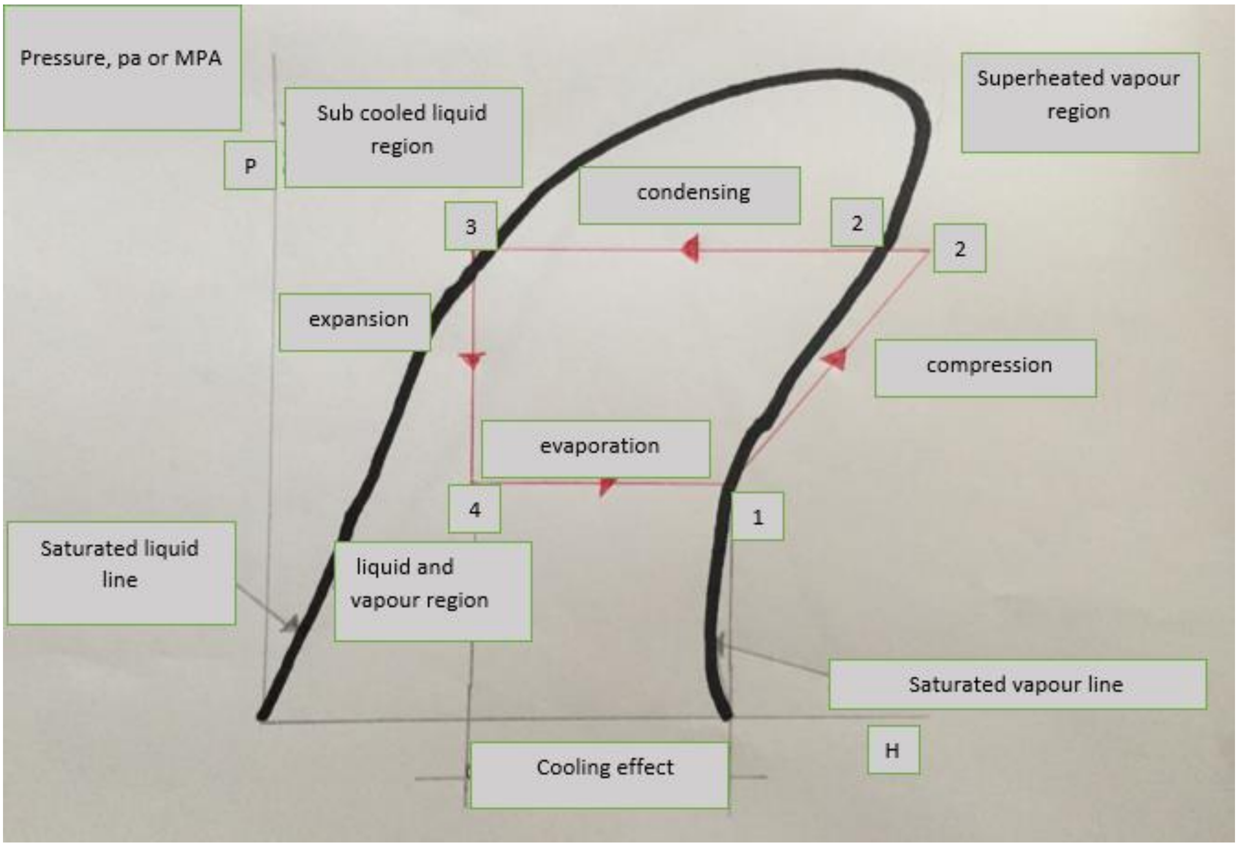

Figure 35 The Pressure-Enthalpy (p-h) Diagram

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