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Steam Motor Laboratory Report

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Summary

In this experiment, a TD 1050 steam motor was used to compare the ideal Rankine cycle with the real Rankine cycle. The importance of steam power in engineering was highlighted and an original solution to improve the efficiency of the Rankine cycle was proposed. The Willans line was used to approximate a mechanical loss of approximately 88 W in the steam motor. Additionally, the inlet pressure was plotted on the same graph to evaluate how efficiently the engine converts steam energy into mechanical power. Analysing the specific steam consumption in relation to engine power led to the conclusion that a steam motor's power can be increased by reducing its specific steam consumption. A Sankey diagram was created to visualise the energy flow within the system, providing a clear overview of losses throughout the cycle. The experimental procedure was clarified with differences between the setup used and a traditional TD 1050 steam motor highlighted. An explanation of the calorimeter was provided, and the isenthalpic device was used to calculate a steam dryness fraction of 0.977 at the boiler outlet. Improvements to the cycle were suggested, addressing the key losses and inefficiencies as well as experimental uncertainties. It was determined that the ideal cycle operated at an efficiency of 8 % while the real cycle achieved only 1.5 %.

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Introduction

A vapour cycle is a thermodynamic process that generates a useful net power output for mechanical applications [1]. Thermodynamic principles dominate large-scale power generation across the world and thus, the continued quest for higher thermal efficiencies has resulted in innovative modifications to the basic vapour power cycle [2]. One original vapor power scheme makes use of the natural temperature difference existing between the surface of the ocean and its deeper levels. An Ocean Thermal Energy Conversion (OTEC) plant harnesses this temperature gradient – typically around 20K – by drawing water at a depth of 300m. Under ideal conditions, typical inlet and outlet values of the OTEC plant would be 28°C and 26°C for the boiler and 7°C and 10°C for the condenser. Despite its potential, the OTEC plant suffers from very low efficiencies of approximately 3%, making their capital cost very high and their application not common [3]. Figure 1 shows the arrangement of such a plant:

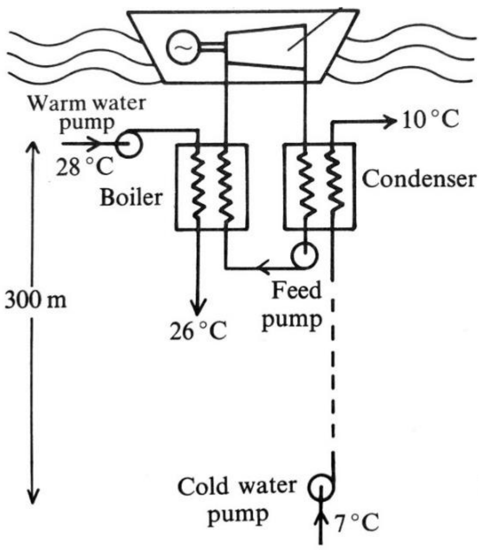

Figure 1 depicts an OTEC plant coupling thermodynamic processes with nature, reducing the heat addition to this thermodynamic process and increasing the efficiency of the cycle [4].

During this steam motor laboratory, the aim is to determine the performance of the TD1050 steam motor and compare its efficiency with the ideal Rankine cycle.

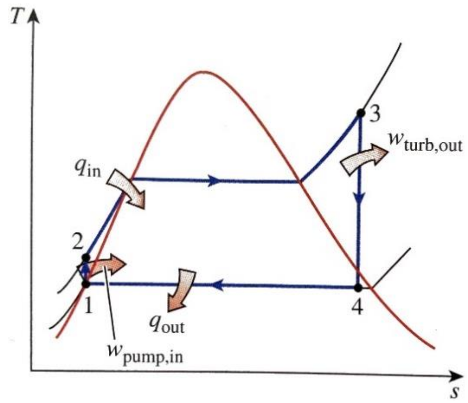

Figure 2 represents the ideal Rankine cycle with no inefficiencies or losses [5].

An important metric to determine the optimal efficiency of the Rankine cycle is the specific steam consumption.

Equation 1, the Specific Steam Consumption Equation.

$$Specific\ Steam\ Consumption \left(\frac{kg}{kWh}\right) = \frac{Steam\ Flow \left(\frac{kg}{h}\right)}{Power\ Output\ (kW)}$$

A throttling calorimeter will be used to determine the enthalpy of the steam leaving the boiler. Understanding how the calorimeter works will be another objective of this lab. Using equation 3, we can determine the dryness fraction from values calculated by the isenthalpic device.

Equation 2: Determines the dryness fraction after the boiler

$$x_{in} = \frac{h_{out} - h_{f,in}}{h_{fg,in}}$$

The Willans line will also be used to estimate the mechanical losses of the steam engine and the Specific Steam Consumption against engine power will reinforce this claim.

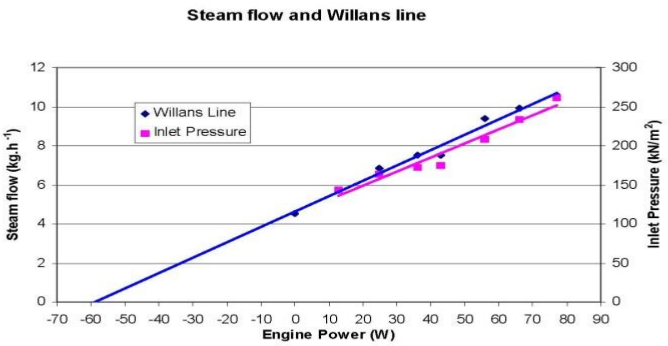

Figure 3: the Willans line as shown by the lecture slides for this lab.

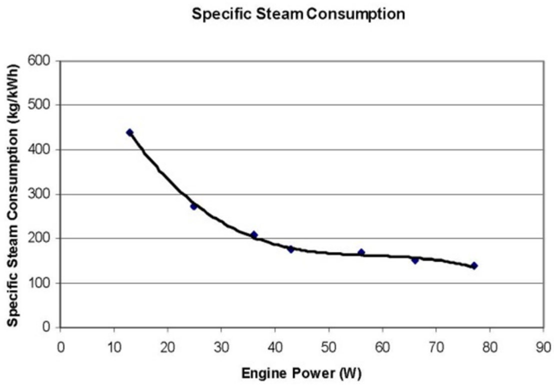

Figure 4: the Specific Steam Consumption against engine power as shown in the lecture slides for this lab.

Finally, the energy balance of the Rankine cycle will be depicted using a Sankey diagram to visualise the flow of energy throughout the process.

Equation 3, the steady flow energy equation for the steam plant may be written as:

$$\dot{W}\_eng = Q\_b - Q\_loss,b - Q\_loss,eng - Q\_loss,cond - Q\_cond + \dot{m}\ h_2 - \dot{m}\ h_1$$

Experimental Apparatus and Procedure

In this experiment the TecQuipment Ltd's TD 1050 steam motor and energy conversion unit was used. One aspect of the steam motor used during the lab was the exclusion of a feed pump. In figure 5, we can clearly see the feed pump, highlighted in red, that was not used for the experiment. Alternatively, the boiler was simply treated as the pump as well as the boiler.

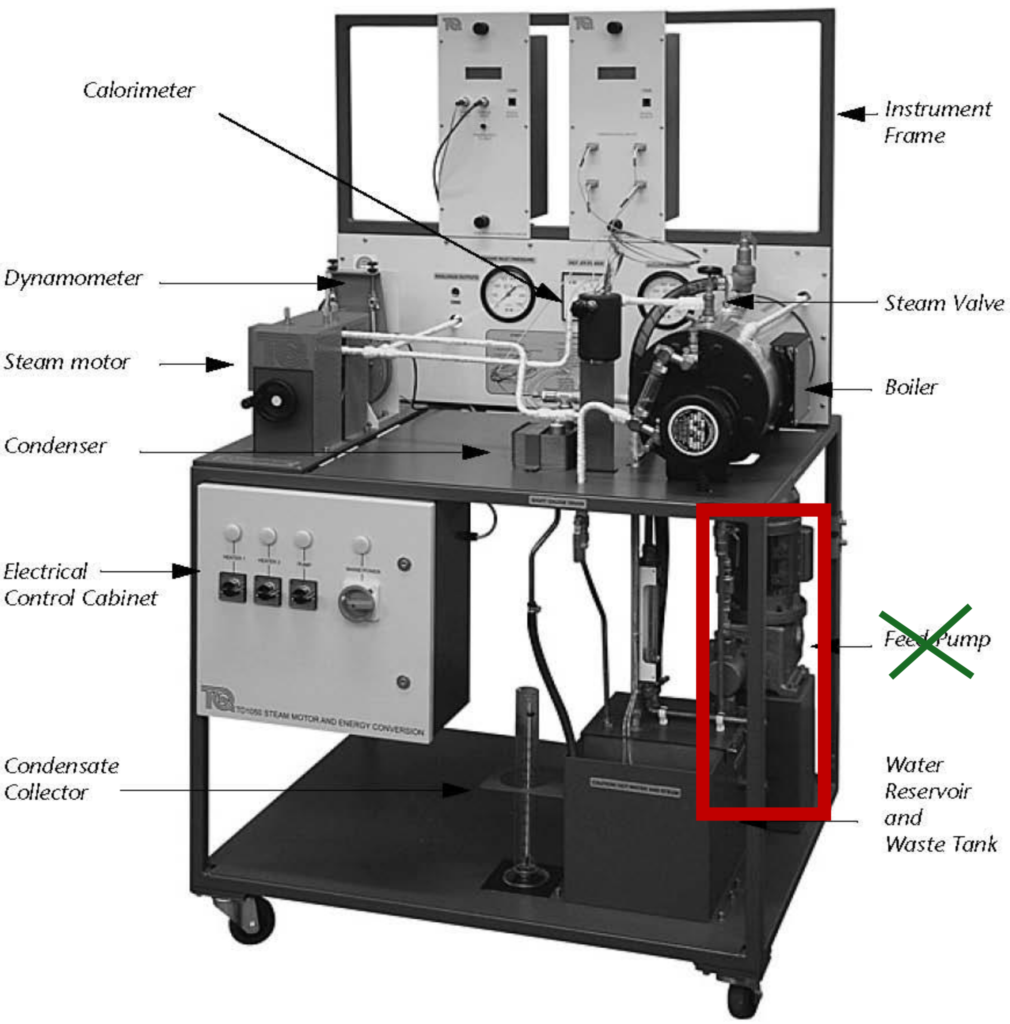

Figure 5 explicitly shows the feed pump on the traditional TD 1050 set up that was not used during the steam lab.

Additional considerations to make from this set up were that unlike the ideal Rankine cycle, this cycle operates in an open loop fashion. As we can see from Figure 2, in the ideal Rankine cycle, the working fluid flows from point 4 back to point 1. In this set up, the working fluid is added to the system at point 1, and at point 4, the working fluid is extracted into the condenser collector. For a high efficiency engine, it is essential to keep the condenser pressure as low as possible, However, where space is limited such as in our case, an open cycle is used [7]. Furthermore, the traditional Rankine cycle extracts mechanical work from the working fluid through a turbine, whereas the TD 1050 uses a steam engine instead.

The procedure of this experiment adhered to the following:

1. Switching on the heaters (isolator and individual) until a boiler pressure of 300 kPa and a motor inlet pressure of 60 kPa is reached
2. Turning the motor overhead piston, driven by an external connecting link from a vertical shaft that is driven by a bevel gear at the crankshaft, to engage the motor.
3. Using the steam valve on top of the boiler to regulate the steam flow from the boiler to the motor
4. Switching off one of the electrical boiler heaters to stabilise its pressure at 230 kPa
5. Adjusting the brake dynamometer to apply a load on the drum increasing the torque on the motor crankshaft in steps of ≈ 0.05 Nm
6. Recording heater power, boiler temperature and pressure, motor inlet pressure, motor speed, motor power, cooling water temperature, flow rate and condensate flow rate
7. Recording the condensate flow rate in a measuring cylinder over periods of 60 seconds
8. Opening the calorimeter valve and recording the boiler steam temperature and calorimeter temperature
9. Shutting down the rig

During the experiment, the following variables were controlled:

- Atmospheric pressure
- Ambient temperature
- Heater power
- Boiler pressure
- Boiler temperature
- Cooling water inlet temperature
- Cooling water flow rate
- Motor speed

The following was the independent variable:

- Torque

The following were the dependent variables:

- Condensate flow rate
- Motor inlet pressure
- Cooling water outlet temperature
- Motor power
- Dryness fraction

To determine the dryness fraction of the system, a calorimeter was used to calculate the enthalpy of the steam leaving the boiler. The calorimeter expands steam from the boiler to atmospheric pressure, allowing us to determine the enthalpy of the liquid-vapour mixture leaving the boiler that is essential to calculate the dryness fraction. The device is isenthalpic, meaning no work is done on the fluid and the heat transferred is negligible. Therefore, the enthalpy of the working fluid entering the calorimeter must be equal to the enthalpy of the fluid leaving the calorimeter. Using equation 2, we can therefore determine the dryness fraction.

Results

Elliot Routier							14/02/2025					Lab Group 10
Atmospheric Pressure: 1 bar				Cooling Water Flow Rate: 1 LPM								
Dryness fraction: 0.977				Boiler Steam Pressure: 230 kPa Boiler Steam Temperature: 136°C Calorimeter Temperature: 101.8°C								
Torque	Motor Speed	Motor Power		Heater Power	Condensate Flow		SSC	Boiler	Motor Inlet	Temperatures T1, T3, T4		
Nm	rev/min	W	kW	kW	L/min	kg/h	kg/kWh	kPa	kPa	Boiler (°C)	Cooling Water Inlet (°C)	Cooling Water Outlet (°C)
0.00	1640	0	0.000	3	0.045	2.70	0.00	230	62	136.0	17.0	29.9
0.03	1600	5	0.005	3	0.050	3.00	600.00	230	75	136.0	16.8	30.8
0.09	1620	16	0.016	3	0.054	3.24	202.50	230	82	136.0	16.8	30.8
0.15	1610	25	0.025	3	0.064	3.84	153.60	230	120	135.7	16.8	35.8
0.20	1620	34	0.034	3	0.069	4.14	121.76	230	120	136.3	16.9	34.7
0.26	1660	46	0.046	3	0.071	4.26	92.61	230	130	136.2	16.9	35.0
0.31	1685	55	0.055	3	0.077	4.62	84.00	230	140	136.2	17.0	36.6
0.36	1655	64	0.064	3	0.082	4.92	76.88	230	160	135.8	17.1	40.9
0.54	1530	88	0.088	3	0.093	5.58	63.41	230	220	135.9	17.5	46.5

Table 1 tabulates the data recorded during the experiment

The table lists the ranging the independent variable from 0.00 to 0.54 in steps of ≈ 0.05 Nm while recording the dependant variables of motor power, condensate flow, motor inlet pressure and boiler pressure. The table also presents the control variables while executing some necessary calculations such as Specific Steam Consumption, motor power conversion into kW and condensate flow into kg/h.

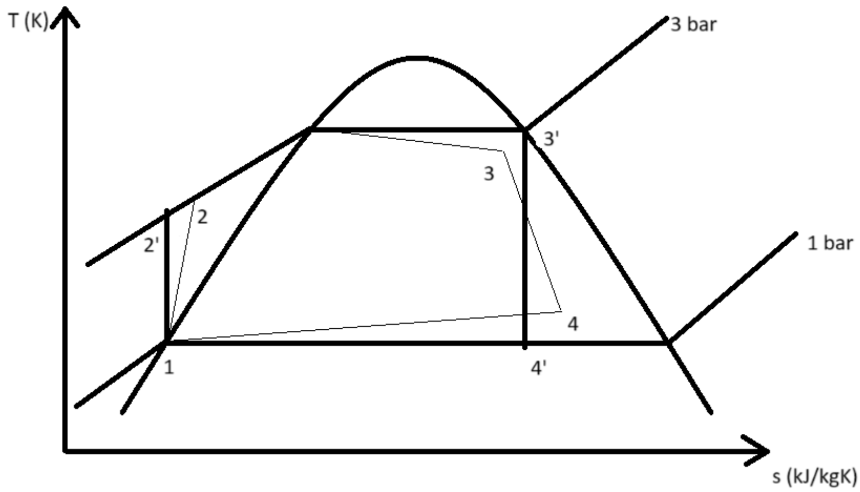

Figure 6 shows the ideal and real Rankine cycles. The real Rankine cycle is shown as the thick line, and the actual Rankine cycle is shown as the thin line.

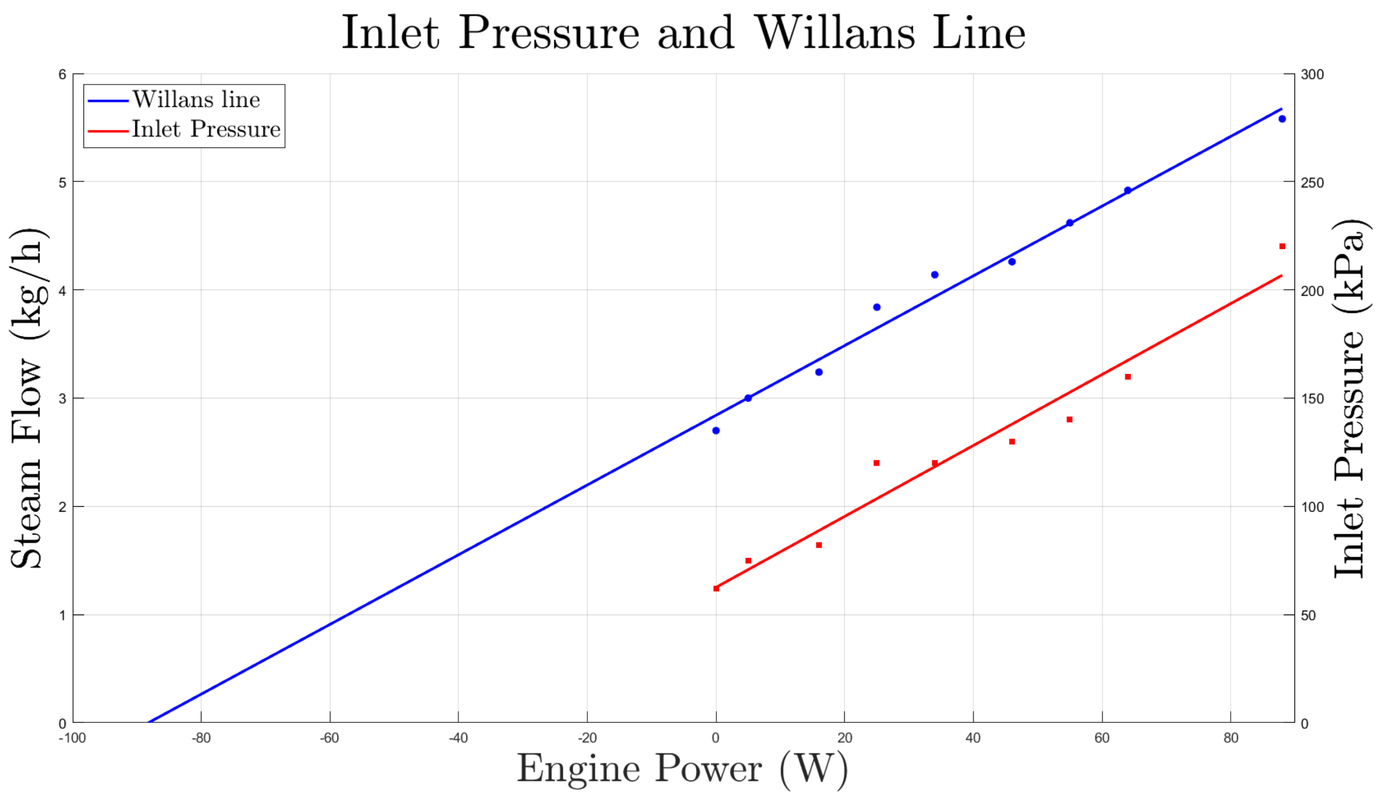

Figure 7 presents a plot of steam flow against engine power on the left y-axis and simultaneously plots a graph of inlet pressure against engine power on the right y-axis.

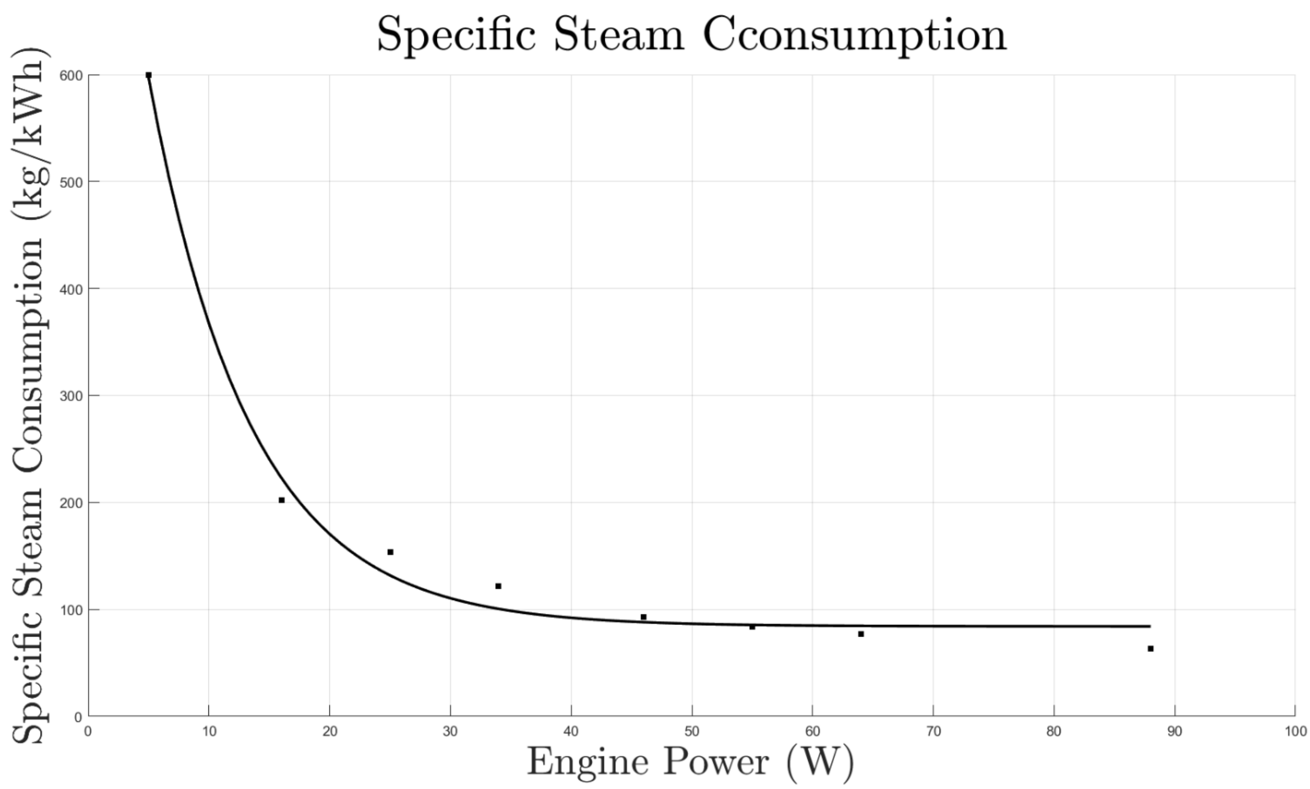

Figure 8 plots the specific steam consumption against the engine power and applies an exponential fit to the data.

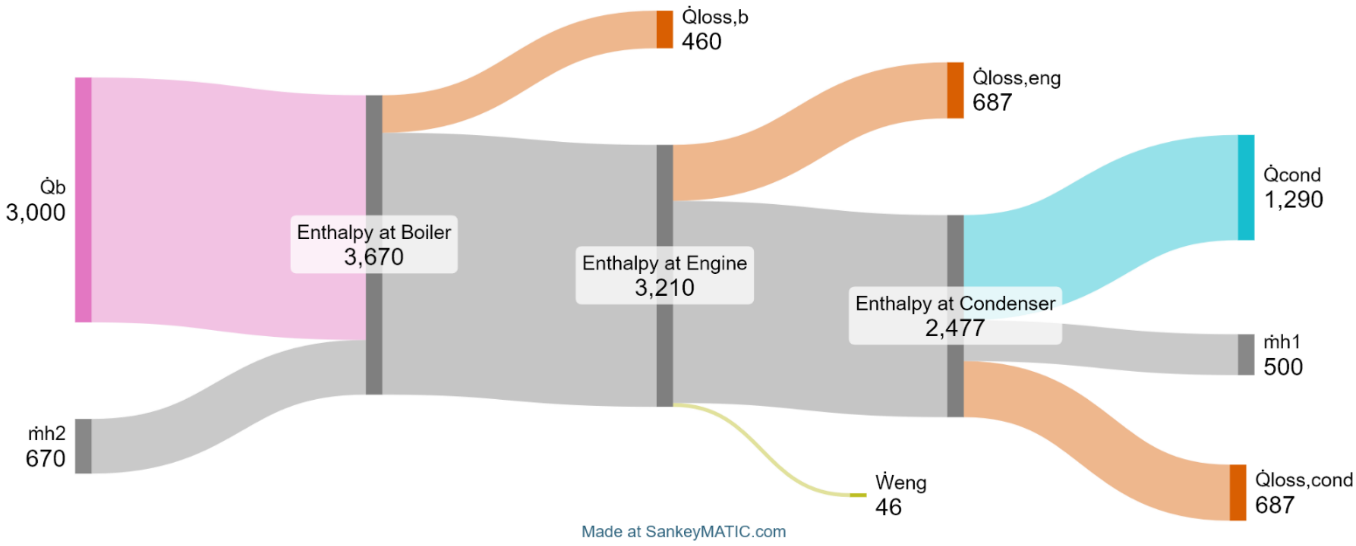

Figure 9 shows the energy balance during the Rankine cycle using a Sankey diagram [7]. All units are in W.

In Figure 9, the following abbreviations stand for:

- $\dot{Q}b$ is the heat supplied to the boiler via electric heaters
- $\dot{Q}loss,b$ is the heat lost via heat transfer to the surroundings from the boiler
- $\dot{Q}loss,eng$ is the heat lost via heat transfer to the ambient, oil and mechanical losses
- $\dot{Q}loss,cond$ is the heat loss to the ambient from the condenser
- $\dot{Q}cond$ is the heat transferred from the steam to the cooling water in the condenser
- $\dot{W}eng$ is the mechanical work output from the engine
- $\dot{m}h2$ is the product of the mass flow rate and enthalpy of the water entering the boiler
- $\dot{m}h1$ is the product of the mass flow rate and enthalpy of the water leaving the condenser

Discussion

Willans Line

The Willans line extrapolates steam flow against engine power to determine the theoretical power required at zero steam flow. As shown in figure 7, the Willans line follows a linear relationship, indicating that an engine power of -88 W is required to operate the steam motor at 0 kg/h of steam flow. This result may seem counterintuitive because it implies that the engine is consuming power when no steam is flowing. However, the Willans line allows us to visualise the mechanical losses experienced by the motor. In practice, this means that the work required to overcome mechanical losses when operational is 88 W. This is only an estimate, as the accuracy of the Willans line depends on the quality of the experimental data and the inherent uncertainties in the measurements.

Figure 7 also includes a plot of inlet pressure against engine power, revealing a linear relationship that helps predict performance trends. This relationship provides insight into how efficiently the engine converts steam energy into mechanical power and allows engineers to determine optimal operating pressures to maximise efficiency.

Specific Steam Consumption Graph

Specific Steam Consumption refers to the amount of steam consumed per unit output of power, as defined in equation 1. Therefore, for any given boiler output, a steam motor's power can be increased by reducing its specific steam consumption - in particular, by increasing its cylinder efficiency and reducing steam leakage [8]. Figure 8 clearly supports the inverse relationship between specific steam consumption and engine power. This trend arises from the constant mechanical losses in the engine, as determined by the Willans line. For low engine powers, the motor must overcome a fixed mechanical loss of 88 W, making it inefficient because a significant portion of the input energy is used to overcome these losses. However, as power output increases, the relative impact of this 88 W loss decreases, leading to improved efficiency. An exponential fit was applied to the data in Figure 8, as this trend is expected to persist.

Energy Distribution

Figure 9 presents a Sankey diagram to visually depict the flow of internal energy entering and leaving the system. While the exact heat losses to the ambient, oil, and mechanical inefficiencies of the engine ($\dot{Q}_{loss,eng}$), as well as the condenser's ambient heat loss ($\dot{Q}_{loss,cond}$), remain unknown, their combined total can be calculated. The total energy lost from both the engine and the condenser was found to be 1.37 kW. For clarity in the Sankey diagram, these losses are represented equally, with each accounting for half of the total loss.

The Ideal Rankine Cycle

The ideal Rankine cycle does not involve any internal irreversibilities and consists of the following four processes [9]:

1-2 Isentropic compression in a pump

2-3 Constant pressure heat addition in a boiler

3-4 Isentropic expansion in a turbine

4-1 Constant pressure heat rejection in a condenser

This process is depicted on figure 2.

Improvements to the ideal Rankine cycle include:

- Lowering the condenser pressure
- Superheating the steam to higher temperatures
- Increasing boiler pressure
- Reheating after isentropic expansion
- Open/closed feedwater heaters

The Real Rankine Cycle

The real Rankine cycle is not the same as the ideal Rankine cycle, as shown in figure 6. These differences stem from irreversibilities and losses, the most common being fluid friction and heat loss to the surroundings, leading to a decrease in cycle efficiency. Under ideal conditions the flow through the pump and turbine is isentropic. The deviation of actual pumps and turbines from the isentropic ones can be accounted for by utilising isentropic efficiencies as stated by

Yunus a Cengel [10]. The isentropic losses account for the deviations observed in processes 1-2 and 3-4; however, they do not explain the discrepancy between the real and ideal cycles through the condenser, as shown in Figure 6. In the TD 1050 system, the condenser consists of a coiled tube submerged in cooling water, where the exhaust steam releases heat and condenses before being collected in the measuring cylinder. Due to viscous forces within the working fluid, pressure losses occur during process 4-1, requiring the steam to fully condense before reaching point 4' to ensure it returns to atmospheric conditions.

Improvements to the real Rankine cycle include:

- Reducing number of pipe bends
- Replacing pipe elbows with curves
- Shortening pipe lengths
- Using a less viscous fluid
- Remove valves
- Use a closed loop system to avoid enthalpy leaks

Improvements mentioned for the ideal Rankine cycle also apply for the real Rankine cycle.

Main Losses and Inefficiencies

Losses mentioned in 'The Real Rankine Cycle' section of this discussion arise due to several reasons:

- Bends in pipes leading to turbulence
- Valves leading to turbulence
- Open loop system leading to (partially) uncooled water leaving the system
- Viscous forces in pipes reducing fluid head
- Heat transfer to the surroundings

Experimental Uncertainties

Experimental uncertainties arise from various sources, including measurement limitations, instrument precision and environmental factors, which can affect the accuracy and reliability of the results. Identifying these uncertainties helps assess the validity of the experimental findings and their potential impact on the conclusions drawn as well how these uncertainties could be avoided in the future. During the steam lab, the following uncertainties were identified:

- 1 decimal place precision for thermometers
- Meniscus of the condenser fluid in the measuring cylinder
- Human starting and stopping the 60 second timer
- Two people adjusting speed and torque to control the power output
- Working fluid collected in the measuring cylinder was not pure (oil)
- Hysteresis of torque application from dynamometer

For future use, reducing the sources of experimental errors is critical and applying the following changes will improve the reliability, repeatability and accuracy of experimental results:

- Using higher precision thermometers as well as measuring cylinder
- Using control systems instead of humans

Conclusion

To conclude this lab report, the ideal and real Rankine cycles were compared and analysed using T-s diagrams. The real Rankine cycle was further analysed for the TD 1050 using the Willans line to understand the mechanical losses in the steam motor which was subsequently used to support the claim that a steam motor's power can be increased by reducing its specific steam consumption. The internal energy of the working fluid was determined at each process of the cycle to draw a Sankey diagram that illustrates the flow of energy throughout the cycle. Where the enthalpy of the fluid was unknown, an isenthalpic device was used to determine the internal energy of the fluid. The efficiency of the ideal Rankine cycle was found to be 8% whereas the efficiency of the real Rankine cycle was 1.5%

References

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Appendix

1. Dryness fraction

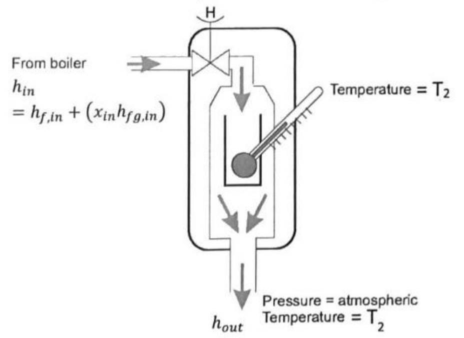
Boiler steam pressure (absolute) (bar):
Calorimeter temperature (°C):

$h_{f,in} = 561 \quad kJ/kg$

$h_{fg,in} = 2164 \; kJ/kg$

$h_{out} = 2676 \; kJ/kg$

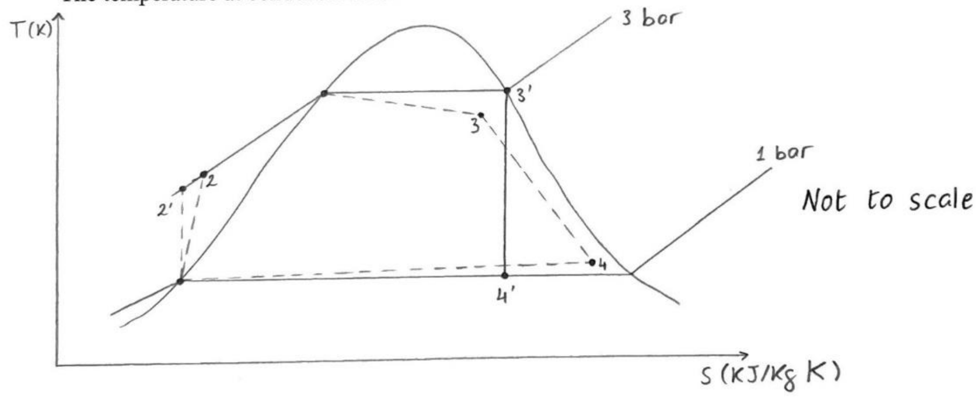
$x_{in} = \dfrac{h_{out} - h_{f,in}}{h_{fg,in}} = 0.977$



2. Ideal Rankine cycle

Sketch the *T-s* diagram of an ideal Rankine cycle (ignore feed pump work) operating with a boiler pressure of 240 kN/m^2 (gauge pressure) and a condenser pressure equal to ambient, and the actual *T-s* diagram of the steam plant including the effects of dryness fraction and engine efficiency. For your report you will need to identify the following:

- Condensate temperature (assume saturated)
- The water temperature at boiler inlet (ignore pump work)
- The boiler steam temperature
- The temperature at condenser inlet



Ideal Rankine cycle	Enthalpy (kJ/kg)	Absolute Pressure (kN/m^2)
Condenser outlet (assume saturated)	417	100
Boiler inlet (ignore pump work)	417	300
Boiler outlet / engine inlet	2725	300
Engine outlet / Condenser inlet	2540	100

$$\eta_{th} = \frac{work\ output}{heat\ supplied} = 8\ \%$$

3. Willans Line and Performance

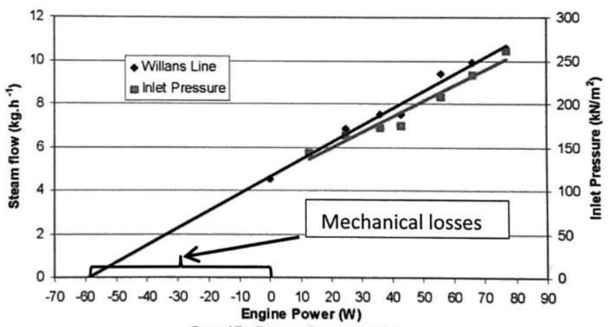
No.	Motor Power	Motor Power	Condensate		SSC	Pressure (kN/m^2)
	(W)	(kW)	Flow (L/min)	Flow (kg/h)	(kg/kWh)	Motor inlet
1	46	0.046	0.071	4.26	~~0.196~~ 92.61	130
2	55	0.055	0.077	4.62	~~0.254~~ 84.00	140
3	64	0.064	0.082	4.92	~~0.315~~ 76.88	160
4	88	0.088	0.093	5.58	~~0.491~~ 63.41	220
5						
6						
7						
8						
9						

Specific Steam Consumption (kg/kWh) = Steam Flow (kg/h)/Power Output (kW)

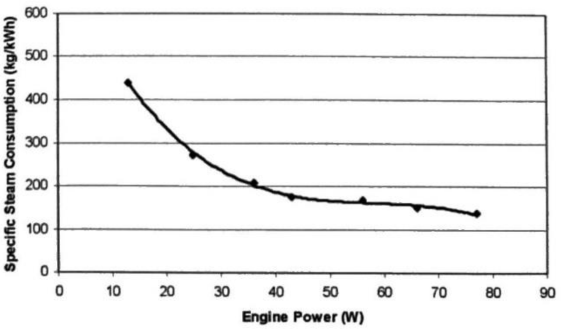
Plot Willans Line (use linear fit), inlet pressure and SSC similar to these two examples to the right.

Estimate mechanical losses for the steam motor from the Willans line.

Steam flow and Willans line



Specific Steam Consumption



4. Steady flow analysis (Energy balance)

Quantity	Symbol	Reading	Absolute values
Pressures			
Atmospheric (absolute)	p_a	1 bar	100 kN/m^2
Boiler	p_1	230 kN/m^2	330 kN/m^2
Flow Rates			
Condensate (Steam)	$\dot{m}$	0.071 LPM	0.0012 kg/s
Cooling water	$\dot{m}_w$	1.000 LPM	0.0167 kg/s
Temperatures			
Boiler steam	T_1	136.2 °C	409.2 K
Cooling water inlet	T_3	16.9 °C	289.9 K
Cooling water outlet	T_4	35.0 °C	308.0 K
Motor Power and Speed			
Motor Power	$\dot{W}_{eng}$	46 Watts (0.046 kW)	
Motor speed	N	1660 rev.min^{-1}	
Boiler Power			
Electrical power input to boiler	$\dot{Q}_b$	3 kW	

For the boiler steam and water (T_1):

Enthalpy of the saturated water (h_f) = 561 KJ/Kg

Enthalpy of evaporation (h_{fg}) = 2164 KJ/Kg

Dryness fraction (x_{in}) = 0.977

Enthalpy of the boiler steam (h_3) = $h_{in} = h_f + x_{in} h_{fg}$ = 2676 KJ/Kg

The heat energy flow rate: $\dot{m}h_3$ = 3.21 KW

Assume: $\dot{m}h_2 = \dot{m}h_f$ = 0.67 KW

So $\dot{Q}_{loss,b} = \dot{Q}_b + \dot{m}h_2 - \dot{m}h_3$ = ~~0.85~~ 0.46 KW

Boiler efficiency: $\eta_b = \frac{\dot{m}h_3 - \dot{m}h_2}{\dot{Q}_b}$ = 0.85

The specific heat of water c_w = 4.18 kJ/(Kg.K):

$\dot{Q}_{cond} = \dot{m}_w \times c_w \times (T_4 - T_3)$ = 1.29 KW

The enthalpy of the condensate leaving the condenser is kJ/kg (assume saturated water at 1 bar):

$\dot{m}h_1$ = 0.0012 × 417 = 0.50 KW

$\dot{Q}_{loss,eng} + \dot{Q}_{loss,cond} = \dot{Q}_b - \dot{Q}_{loss,b} - \dot{Q}_{cond} - \dot{W}_{eng} + \dot{m}h_2 - \dot{m}h_1$

= 3 − 0.46 − 1.29 − 0.046 + 0.67 − 0.5 = 1.37 KW

The overall thermal efficiency: $\eta_{th} = \frac{\dot{W}_{eng}}{\dot{Q}_b}$ = 1.5 %