APPLIED THERMODYNAMICS

MCE 4705

PROJECT 1

On

Design and thermodynamic analysis of a thermal power plant

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Executive summery

This report details an engineering design project undertaken by a Mechanical Engineering undergraduate student at the Islamic University of technology, OIC. The goal of the project is to design a thermal power plant.

The design is very much open ended as not much requirements and background were provided. So, the final design depends on the decision of the author which will be made through rational and engineering reasoning.

This report serves to document the entire design process in detail. The design has evolved throughout the process and the final design is the result of repeated engineering analysis.

This report documents the methods that were followed to ensure that all requirements and engineering specifications were satisfied.

At the end, the author made some recommendations concerning the improvement of the final design and suggested future work.

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Design Problem & Objectives

The project is about designing a thermal power plant with some requirements. The only three requirements of the design problem are as follows:

- A 250 MVA electrical generator supplying electricity to the regional grid need to be used.
- The first law efficiency of the plant should be more than 30%.
- The cost of electricity should be similar to the existing power plants on same design principle.

No other background, requirements or budget limits were provided. So, Realistic and reasonable assumptions will be taken to complete the design. The decisions regarding design will be made through engineering analysis and fundamental laws of thermodynamics.

The final design will be evaluated from both first law and second law perspective. The detailed calculations for every aspects of thermodynamic analysis will be presented. The specification of the components required for the final designed will be mentioned and also, the economic analysis of the designed power plant will be done to validate the third requirement.

Design Operating Cycle

The requirement on the capacity of the power plant plays a major role determining the underlying cycle that the power plant design will be based upon.

As in requirement 1, The maximum output capacity of the generator can be 250 MW in case of power factor being 1. Generally, power factor depends on the load type (resistive or inductive). For resistive load, power factor is 1 but for inductive load power factor is less than 1. So, depending on the load condition, overall power factor may lie within 0.8 – 1.0

Taking a conservative approach (as close to 1 as possible), power factor is assumed 0.95. So, the max capacity of the generator will be

$$250 \text{ X } 0.95 = 237.5 \text{ MW}$$

In recent times, the efficiency of large generator can be very high as just short of 100%. This high efficiency is the result of decades of refinement. So, in conservative manner, the efficiency was assumed to be 95%. So, when the generator is at max capacity (i.e. producing 237.5 MW), the mechanical input from the prime mover needs to be

$$237.5 / 0.95 = 250 \text{ MW}$$

So, the design capacity of the prime mover will be 250 MW.

The available fuel and power plant location are also big factors. Assuming the location of the power plant will be in the Host country, Bangladesh, both natural gas and coal could be a potential fuel for the plant. The operating cycle could be Brayton cycle for natural gas or, Rankine cycle for coal or, Even, Combined cycle for both fuel.

Now, 237.5 MW plant does not justify the complexity and initial high cost of Combined power plant is search of higher efficiency. Because, generally more than one gas turbine is needed to supply sufficient heat to the bottom (steam) cycle. A steam cycle seems to be the perfect candidate for 237.5 MW base load power plant. Regarding the efficiency of steam cycle, it can be increased with reheat and regeneration. Efficiency can be further increased by operating in super or ultra-super critical pressure. But the complexity and economic perspective need to be justified first before supercritical pressure can be included in design.

Regarding reheat stages, one stage reheat will be in design as double reheat could not be considered. Since, operating pressure is decided to be sub-critical, double reheat would result in superheated exhaust as the turbine inlet pressure was not high enough, which would reduce the efficiency of the cycle.

Regarding regeneration, one open feedwater heater is decided to be included in the design cycle. The choice of number of FWH is dependent upon the economic viability. The author has decided that optimum number of FWH for this project would be 1. Also, the author couldn’t determine if turbines in 250 MW range have stream extraction point. So, the steam will be extracted at the intermediate pressure before steam is sent to the reheater. The choice between close or open FWH is open FWH as it provides efficient heat transfer.

So, the operating cycle of the power plant will be based on ideal single stage reheat Rankine cycle with one open FWH at high pressure turbine outlet.

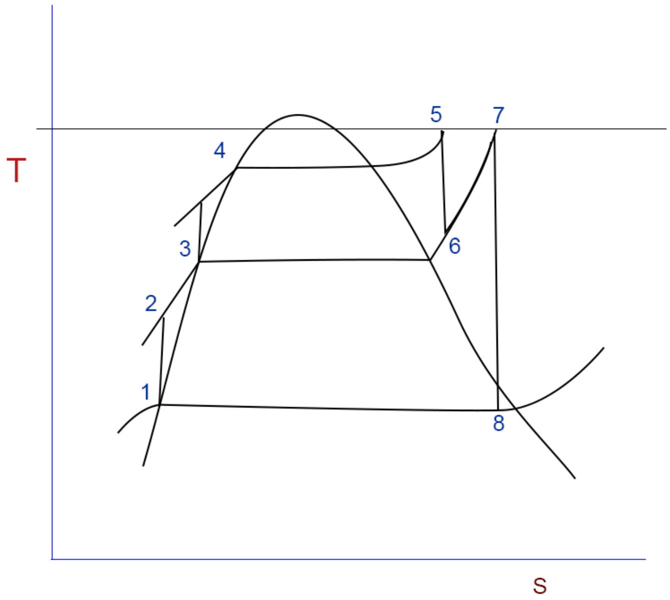

Figure 1 Operating Cycle of the Plant

Design Ideal Cycle Parameters

In previous section, the underlying cycle was determined. In this section, the optimum operating parameters will be determined.

It Is evident from the basic laws of thermodynamics that the higher the operating temperature and pressure in ranking cycle, The higher the efficiency. So, it is desirable to choose these two parameters as high as possible. But, before taking this design decision, the limiting factors from metallurgical and economical side need to be thought of too.

The highest operating temperature and pressure can be determined from available components in market. So, in this stage of design, the author would like to focus on the available turbines in the market for 250 MW output range. Three turbines from General Electric for this output range as follows: [1]

Table 1 Turbine Specifications

Model	Max Temperature Inlet/ Reheat (°C)	Max Pressure (bar)	Capacity (MW)
STF A650	585/585	190	100-300
STF A850	585/585	245	150-300
STF A1050	600/600	300	150-300

From the maximum pressure rating, it is evident that, first one is for sub-critical operation and later two are for super and ultra-supercritical operation.

So, the inlet and reheat temperature is selected to be 855 K. And, the boiler pressure is selected to be 19 MPa.

The condenser pressure is limited by the temperature of the cooling medium. Moreover, the specifications of above turbines states that maximum back pressure should be less than 70 kPa. As assumed in previous section, the location of the power plant is in South Asian country, Bangladesh, where the average temperature can be assumed to be 30 °C. For effective heat transfer, if 10 °C temperature difference is assumed, the condenser temperature will be 40 °C. The saturation pressure is 7.3851 kPa at 40 °C.

So, the condenser temperature is selected to be 313 K and (thus, pressure to be 7.3851 kPa). And as in ideal cycle, the steam leaves the condenser as saturated liquid (x=0).

Now, the specification of the turbine doesn't reference any maximum allowable moisture content. So, it is assumed that maximum allowable moisture is 15%. The optimum moisture content is yet to be determined. Moisture content determines the intermediate/ reheat pressure.

Now, the energy balanced equations for the ideal cycle will be derived to calculate the efficiency of the cycle.

The state points will be calculated in the following way:

Table 2 Methods for determining States of ideal cycle

State identifier	State Specifying method
5	Temperature = Turbine inlet temperature Pressure = Turbine inlet Pressure
8	Pressure = Condenser pressure Quality = 1 - moisture content
7	Temperature = Turbine reheat temperature Entropy = same as state 8
6	Pressure = same as state 7 = reheat pressure Entropy = same as state 5
1	Pressure = Condenser Pressure Quality = 0 (saturated liquid)
2	Pressure = reheat pressure Entropy = same as state 1
3	Pressure = reheat pressure Quality = 0 (saturated liquid)
4	Pressure = Turbine inlet Pressure Entropy = same as state 3

After, determining the state points, following Equations will be used for calculation. Subscript denotes the state identifier

Table 3 Equations for calculations

Fraction of extracted steam	$y = (h_3 - h_2) / (h_6 - h_2)$
Boiler Heat input	$q_{45} = h_5 - h_4$
Reheater Heat input	$q_{67} = (1 - y) \times (h_7 - h_6)$
Total Heat input	$q_{in} = q_{45} + q_{67}$
HP Turbine work output	$W_{HP} = h_5 - h_6$
LP Turbine work output	$W_{LP} = (1 - y) \times (h_7 - h_8)$
Turbine work output	$W_{turbine} = W_{HP} + W_{LP}$
Pump work input	$W_{pump} = W_{pump\ 1} + W_{pump\ 2} = (1 - y) \times (h_2 - h_1) + h_4 - h_3$
Net Work	$W_{net} = W_{turbine} - W_{pump}$
Efficiency	$\eta = \frac{W_{net}}{q_{in}}$

At this point, the above equations for ideal cycle will be implemented on python using CoolProp[2] library. The *ideal()* function takes 4 parameters (T_5, P_5, T_8, x_8) and outputs the efficiency of the cycle.

Now, using above program, a plot of moisture content vs efficiency was generated for T5= 855K, P5 = 19 MPa, T8 = 313K

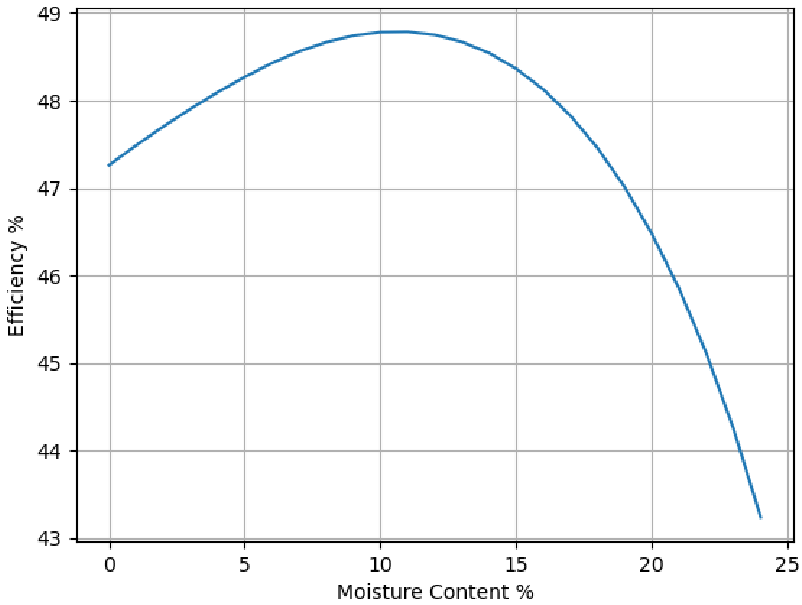

Figure 2 Effect of Moisture Content on Cycle Efficiency

The efficiency becomes maximum 48.79% for moisture content of 11% which below maximum allowable quantity (15%). The reheat pressure for 11% moisture content is 3.325 MPa which is reasonable as optimum reheat pressure is generally about one fourth of maximum cycle pressure. [3]

Design Actual Cycle Parameters

Actual cycle will differ from the ideal cycle discussed in previous section due to the irreversibilities in various components.

Pressure loss due to Fluid friction and heat loss to the surroundings are the two common sources of irreversibilities.

The pressure at turbine inlet will not be same as boiler inlet due to the pressure losses in boiler. So, to maintain the design pressure at the turbine inlet, fluid must be pumped to higher pressure at boiler inlet.

The boiler pressure losses can be assumed 0.7 MPa. The reheater pressure losses is assumed to be 0.3 MPa

Due to the pressure losses is condenser, the pump inlet pressure will be lower than the turbine exhaust pressure. The pressure loss in condenser is generally negligible. In this case, the pressure loss is assumed 1 kPa.

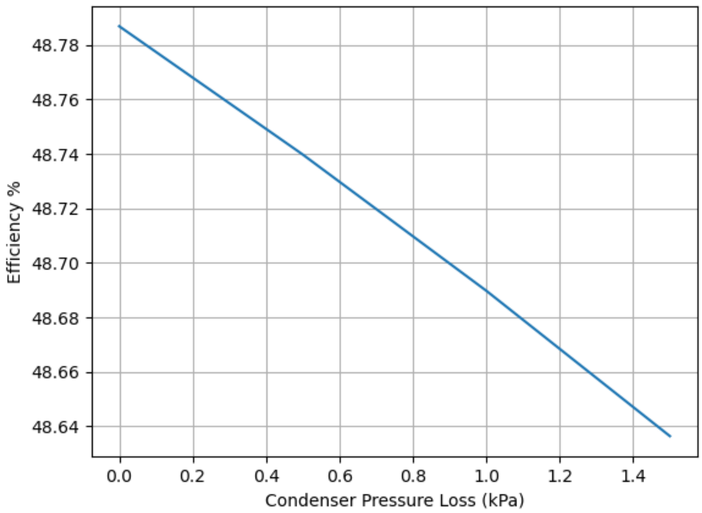

Figure 3 Effect of Pressure loss on the Cycle Efficiency

To prevent cavitation, fluid is generally subcooled before it is sent to the pump. In this case, the degree of subcooling is selected to be 5˚C for both pump.

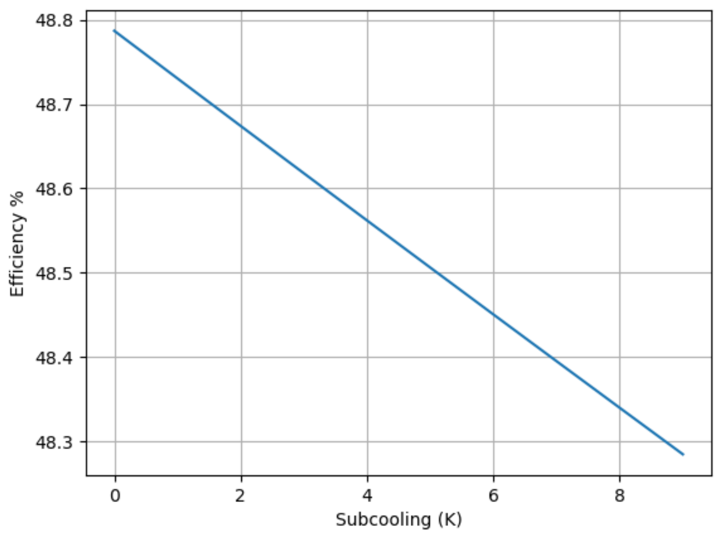

Figure 4 Effect of degree of subcooling on the Cycle Efficiency

There are also heat losses from the piping. Now the amount of heat losses from the pipe would be difficult to predict without considering all the parameters heat transfer is dependent upon. So, a reasonable amount of 100kJ/kg heat losses will be assumed. To maintain the same design inlet and reheat temperate, this additional amount of heat needs to be added in the superheater.

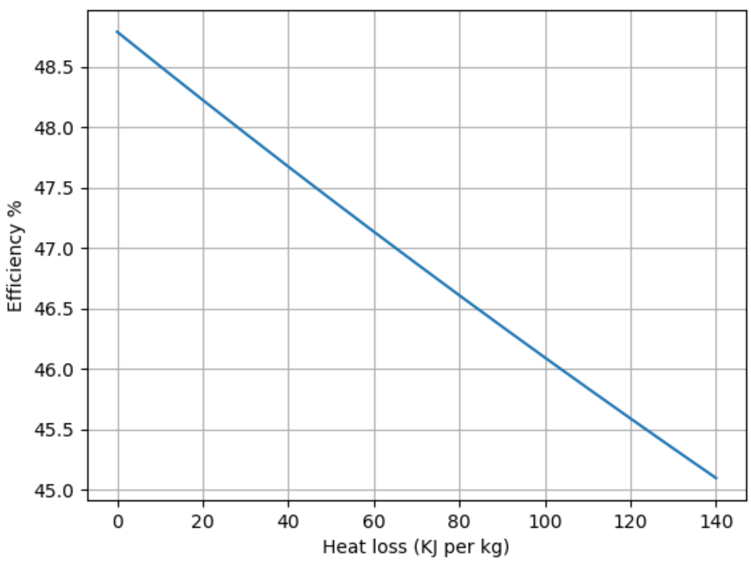

Figure 5 Effect of Heat loss on the Cycle Efficiency

The pressure losses in piping is not negligible. The pressure losses will be assumed 0.7 MPa for piping from boiler to turbine for both main and reheat lines. To maintain same design main and reheat pressure, fluids must be pumped to higher pressure in first case and the outlet pressure of high-pressure turbine need to be increased in second case. The pressure losses in FWH is ignored.

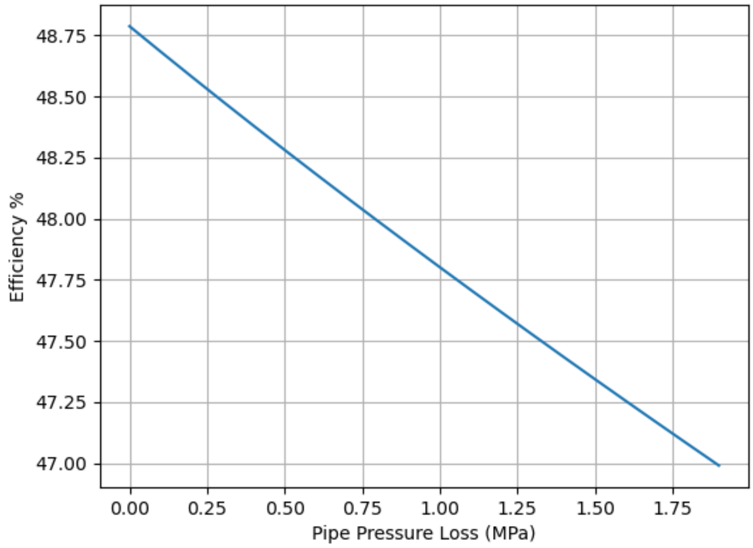

Figure 6 Effect of Pipe Pressure loss on the Cycle Efficiency

The isentropic efficiency of the turbine could not be found in the specification. Isentropic efficiency of 95% is assumed for both HP and LP turbine. For both pump, it is assumed to be 90%

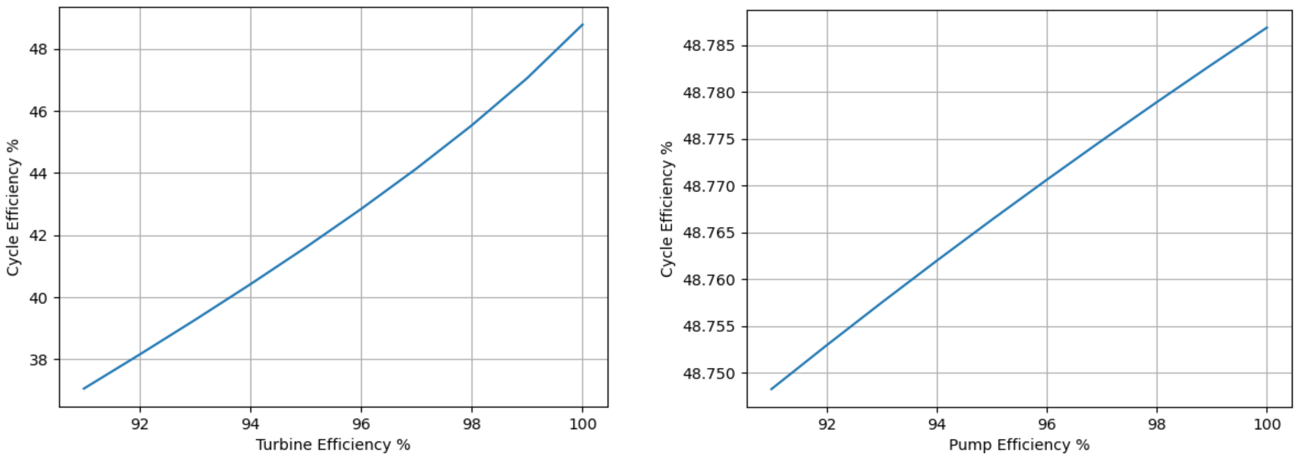

Figure 7 Effect of Pump & Turbine efficiency on the Cycle Efficiency

The turbine used in the design is mentioned before. Generally, power plant components manufacturer designs and manufacture based on the end requirement of the customer. So, specific models for all the components couldn't be mentioned here. But the specification required for any component can be understood from *Design Evaluation Section*

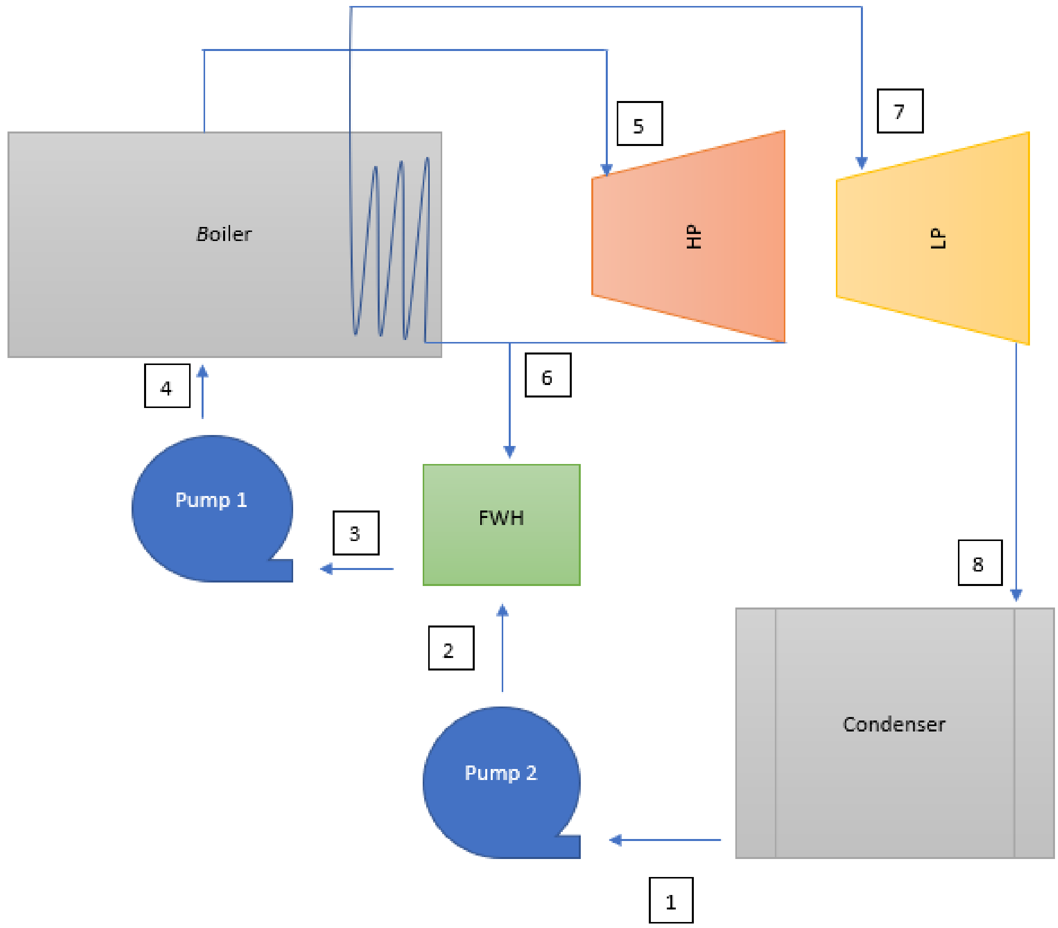

Figure 8 Component arrangements of the Plant

The states of the actual cycle can be determined in following way:

Table 4 Methods for determining States of actual cycle

State identifier	State Specifying method
5a	$T_{5a} = T_5$ $P_{5a} = P_5$
8a	$P_{8a} = P_8$ $x_{8a} = x_8$
7a	$T_{7a} = T_7$ $h_{7a} = \eta_T \times (h_7 - h_8) + h_8$
6a	$P_{6a} = P_7 +$ Pipe pressure loss + reheater pressure loss $h_{6a} = h_5 - \eta_T \times (h_5 - h_6)$ {6\` would be the point if there was isentropic expansion from P_5 to P_{6a}}
1a	$P_{1a} = P_{6a} -$ Condenser Pressure Loss $T_{1a} = T_{sat}@P_{1a} -$ subcooling
2a	$P_{2a} = P_{6a}$ $h_{2a} = \frac{(h_2 - h_1)}{\eta_P} + h_1$
3a	$P_{3a} = P_{6a}$ $T_{3a} = T_{sat}@P_{3a} -$ subcooling
4a	$P_{4a} = P_5 +$ Boiler Pressure loss + Pipe pressure loss $h_{4a} = \frac{h_{4`} - h_{3a}}{\eta_P} + h_{3a}$ {$h_{4`}$ would be the point if there was isentropic compression from P_{3a} to P_{4a}}

The Equation for calculating net Work, Efficiency are same as table 3. Only heat losses will be added in total heat input.

The above actual cycle was modelled in python. The model contains some input combination (state 7a) that CoolProp does not support yet in HEOS Backend. So New model was built from REFPROP backend. If reader wish to run the code, he has to set the REFPROP library directory in code. The code will be in this GitHub Repository. [4]

Design Evaluation

Results of the ideal cycle

Table 5 property values at the state points of ideal cycle

State point	Properties	State point	Properties
1	P = 7.326 kPa T = 313.0 K h = 166.91 kJ/kg s = 0. 5704 kJ/kg	5	P = 19.0 MPa T = 855.0 K h = 3497.74 kJ/kg s = 6.48 kJ/kg
2	P = 3.33 MPa T = 313.1 K h = 170.247 kJ/kg s = 0.5704 kJ/kg	6	P = 3.33 MPa T = 573.43 K h = 2984.79 kJ/kg s = 6.481 kJ/kg
3	P = 3.33 MPa T = 512.8 K h = 1035.78 kJ/kg s = 2.6985 kJ/kg	7	P = 3.33 MPa T = 855.0 K h = 3638.93 kJ/kg s = 7.413 kJ/kg
4	P = 19 MPa T = 516.37 K h = 1054.92 kJ/kg s = 2.6985 kJ/kg	8	P = 7.326 kPa T = 313.0 K h = 2308.54 kJ/kg s = 7.412 kJ/kg

Table 6 Results for ideal cycle

Result	**kJ/kg**
Heat Input 4-5	2442.83
Heat input 6-7	452.98
Total Heat input	2895.81
Heat output 8-1	1483.04
Work Input 1-2	2.31
Work input 3-4	19.13
Work output 5-6	512.96
Work output 7-8	921.26
Turbine Work	1434.22
Pump Work	21.45
Net work	1412.78
Efficiency	48.79%
Steam extracted for FWH	30.75%

Results of the actual cycle

Table 7 Actual Cycle Parameters

Turbine inlet and reheat temperature	855 K
Turbine inlet pressure	19 MPa
Condenser Temperature	313 K
Turbine exit steam Quality	89%
Turbine efficiency	95%
Pump efficiency	90%
Subcooling	5 K
Pressure loss in Boiler	0.7 MPa
Pressure loss in piping	0.5 MPa
Pressure loss in condenser	1 kPa
Heat loss in Pipe	100 kJ/kg
Pressure loss in Reheater	0.3 MPa

Table 8 property values at the state points of actual cycle

State point	Properties	State point	Properties
1a	P = 6.326 kPa T = 305.27 K h = 134.63 kJ/kg s = 0.465 kJ/kg	5a	P = 19.0 MPa T = 855.0 K h = 3497.74 kJ/kg s = 6.48 kJ/kg
2a	P = 11.76 MPa T = 305.8 K h = 147.72 kJ/kg s = 0.470 kJ/kg	6a	P = 11.76 MPa T = 770.0 K h = 3344.56 kJ/kg s = 6.491 kJ/kg
3a	P = 11.76 MPa T = 591.3 K h = 1449.22 kJ/kg s = 3.426 kJ/kg	7a	P = 10.96 MPa T = 855.0 K h = 3572.41 kJ/kg s = 6.802 kJ/kg
4a	P = 20.2 MPa T = 596.2 K h = 1463.00 kJ/kg s = 3.428 kJ/kg	8a	P = 7.326 kPa T = 313.0 K h = 2308.54 kJ/kg s = 7.412 kJ/kg

Table 9 Results for actual cycle

Result	kJ/kg
Heat Input 4-5	2134.74
Heat input 6-7	194.37
Total Heat input	2329.11
Work Input 1-2	7.77
Work input 3-4	13.78
Work output 5-6	153.18
Work output 7-8	902.5
Turbine Work	880.95
Pump Work	21.55
Net work	880.95
Efficiency	37.82%
Steam extracted for FWH	40%

Cost Analysis

Since, turbine output is 880.96 kJ/kg, the mass flow required to get 250 MW output from the turbine, is 283.79 kg/s. The electric output is 237.5 MW

Again, Total Heat input is 2329.11 kJ/kg. So, heat input rate = 660.98 MW.

Assuming, boiler efficiency is 92%, so, Fuel need to burned at a rate of 718.5 MW. Assuming, Heating value of coal being 26 MJ/kg, amount of coal required is 28 kg/s.

So, 28 kg coal required for 237.5 MJ electric output. Then, for 1 kWh or 3600000 J electric output, 0.424 kg of coal is required. Assuming Coal price $150/Metric Ton, cost of 1-unit electric power becomes $ 0.0636 which is equivalent of 5.4 BDT. This fulfils the third requirement of the project.

Demining CAPEX and OPEX will require to have a detailed cost analysis of the project which is out of scope of this course. So, a reasonable guess ,in the perspective of Bangladesh, for the cost of coal power plant can be $2000 /kW, and annual operating cost can be $40/ kW [5]

So, for 237.5 MW coal power plant, expected capital cost is 475 million dollars and annual cost is 9.5 million dollars

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Conclusion & Recommendation

The designed power plant met all the requirements of the project. The efficiency is 37.82% which is more than 30% requirement. Due to the generator requirement, it was assumed to be low budget power plant. Therefore, the plant was designed for subcritical pressure operation.

Although, cost analysis was demanded in the project, Author could not conduct it properly due to lack of data of the price of the components. The author suggests to further improve the work on economical perspective.

Acknowledgements

The authors would like to gratefully acknowledge that, the knowledge acquired from Applied Thermodynamics Course greatly helped the author to accomplish this report.

The following software were used for the completion of the project.

CoolProp and Matplotlib Library in PyCharm [a python IDE] for the calculation and modelling of the cycle. ThermoState [6] for hand calculation. MS Word for Word Processing and Adobe photoshop for some drawing and image editing.

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