ELEC6115 Conventional Generation Technologies Assignment 2: Steam Plant Analysis

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I. Steam Plant 1 Design

Considering the design of the plant in Figure 1, several design flaws are apparent. Of primary concern is the steam quality at the exit of the low pressure (LP) turbine. The steam quality is shown as 0.8 in Figure 1 and describes the ratio of vapour to liquid. Alternatively one can view this as a moisture content of 20%. Modern steam turbines are only able to tolerate a steam quality of around 0.9 [2].

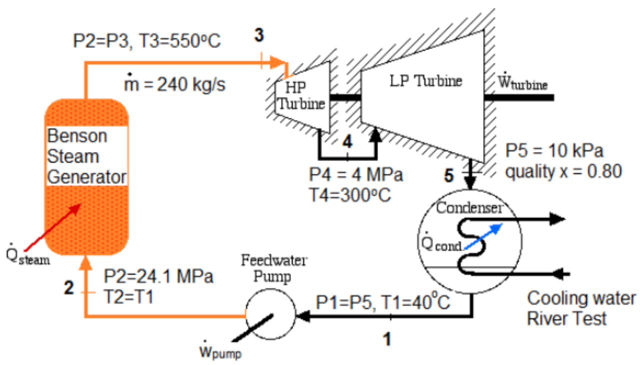

Fig. 1. Steam Plant 1 with Design Flaws (taken from [1])

Excessive moisture at the exhaust of the turbine is often caused by attempts to raise the efficiency of the Rankine cycle. Three common ways to increase the efficiency of the Rankine cycle are to:

- Decrease the condenser pressure,
- Superheat the steam to a higher temperature,
- Increase the boiler pressure.

Decreasing the condenser pressure inherently increases the moisture content of the steam in the final stages of the turbine. Additionally, for a fixed turbine inlet temperature, increasing the boiler pressure shifts the cycle to the left in a T-s diagram, also increasing the moisture content of the steam [2].

The results of excessive moisture in the steam turbine are decreased turbine efficiency and excessive wear on the turbine blades themselves [2]. The expanded volume of the vapour in the LP turbine stage means that much larger blades are used. In turn, the blades have a much higher tip speed, of up to $550ms^{-1}$ [3]. Under continuous operation, even small deformations caused by impacting water particles can accumulate above tolerable levels. The turbine blades shown in Figure 2 shows damage to the tip of the turbine blades. The moisture content of the steam in this example was 12.6% and the blades used had unprotected leading edges [3]. However, it is clear that operating under these conditions is not desireable and an alternate design must be found.

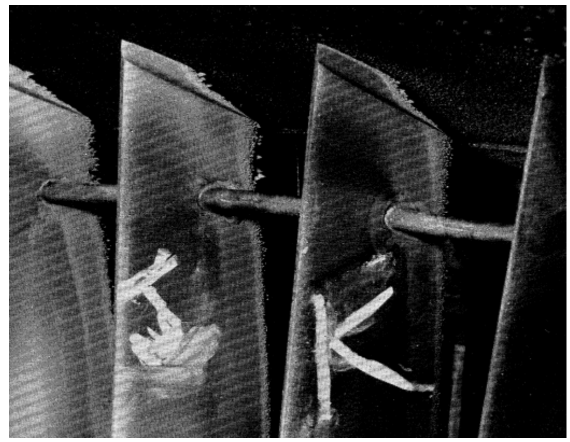

Fig. 2. Water droplet erosion on unshielded stainless iron blade (taken from [3])

An additional issue with the design is the very low pressure in the condenser. 10kPa is much below atmospheric pressure of 101.325kPa [4]. The pressure difference will allow some air to leak into the system, which will deteriorate the performance of the system [2].

II. Steam Plant 2 Design

A. Flaw correction

The flaws have been corrected in figure 3. There are two ways to take advantage of the higher efficiencies at higher boiler pressures without causing the excessive moisture problem. One possibility is to superheat the steam to very high temperatures before it enters the turbine. The average temperature at which heat is added would also increase, increasing the cycle efficiency. However this is not possible due to the temperature of the metals used in the boiler and piping.

The second possibility, which has been implemented in figure 3, is to expand the steam in two stages. The steam is expanded in the high pressure turbine, then reheated at constant pressure before being further expanded in the low pressure turbine. This addresses the excessive moisture issue by heating and expanding in stages, and is widely used in the power industry today [2].

To address the low condenser pressure, a condensate pump has been added to raise the pressure from $10kPa$ in the condenser to $800kPa$. The water is passed through a de-aerator which seperates gases from the feedwater.

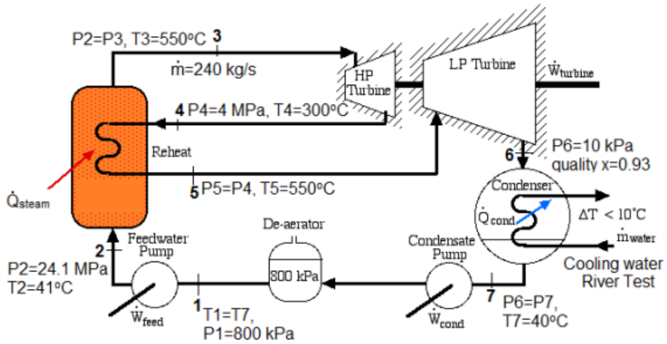

Fig. 3. Steam Plant 2 (taken from [1])

B. Discussion of T-s Diagram

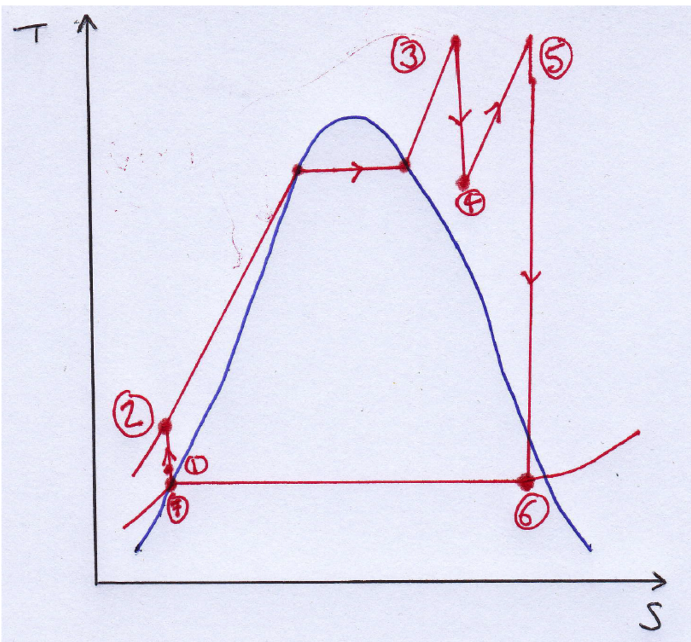

Fig. 4. T-s Diagram for steam plant 2

The T-s diagram for an ideal reheat rankine cycle is shown in figure 4. There are seven defined points on the diagram, which correlate with the numbers shown in figure 3.

The stages 1-3 are not particularly different from the standard Rankine cycle. Between 1 and 2, the feedwater pump compresses the feed water isentropically. In this ideal case, isentropic compression is shown by entropy remaining constant. Between 2 and 3, the system is kept at constant pressure while heat is added in the boiler to reach the turbine inlet temperature.

Between 3 and 4, the now superheated steam undergoes isentropic expansion in the high pressure turbine. Unlike a non-reheat rankine cycle, the steam is expanded to an intermediate pressure. Between 4 and 5, the steam re-enters the boiler in a reheat cycle. The steam is reheated at constant pressure to approximately the same temperature as the first heating stage. Between 5 and 6 the steam enters the low pressure turbine, where the steam is fully expanded isentropically to the condenser pressure.

The final addition in this cycle is the condensate pump. It is reasonable to assume this is approximate to an isentropic compression, much like the feedwater pump, but only compressing to the pressure required for the deaerator. The deaerator solely removes gases from the feedwater, it is assumed that this is well insulated and does no work, and so has no consequence in our calculations.

The steam plant in figure 3 is not an ideal reheat rankine cycle. Many of the processes are adiabatic and lossless, however the turbine and pumps cannot be considered isentropic. This is due to irreversibilities usually caused by internal friction of steam or water. This causes stage 5-6 and 2-7 to stray from vertical, as the entropy now changes. This results in a slanting T-s diagram as shown in figure 5. Additionally there can be further deviations from the ideal cycle if there are leaks in the boiler and condenser. However, in this exercise, the system is assumed to be lossless and so there is no issue here.

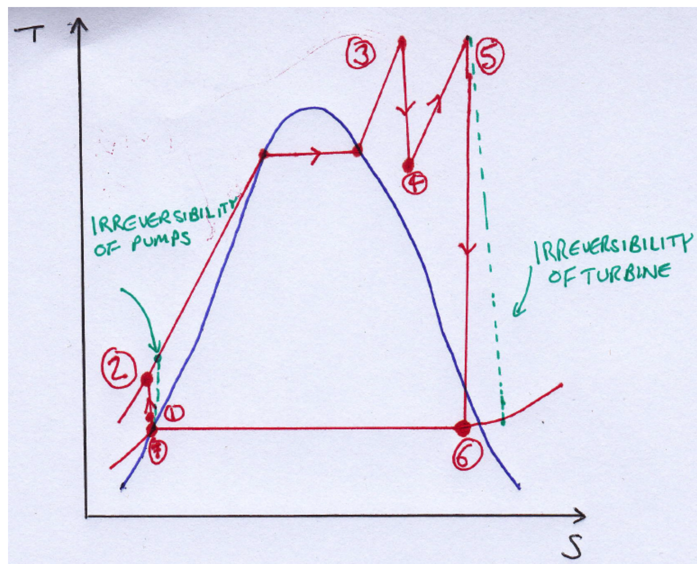

Fig. 5. T-s Diagram for steam plant 2 with irreversibilities in the turbines and pumps

C. Working Assumptions

These assumptions have been made in the following calculations:

- Turbines and pumps are adiabatic and lossless.
- Steady state operating conditions exist.
- Changes in kinetic and potential energy are neglidgible.
- Intervals of the steam/water tables can be found by linear interpolation.

D. Turbine output power calculations

The following calculations assume that the both turbines are adiabatic (that is, no heat is lost) and lossless. This means that all changes in entropy go into the work. However, it does not necessarily mean the turbines are isentropic. Isentropic efficiencies are used to model the irreversibilites in both the turbine and pumps. However, since the pressure and temperature are given at every stage (apart from stage 6) this information can be used to find the actual enthalpy at that point without calculating any isentropic efficiencies.

In order to find the output power of the turbines, we first need to find the output work of the turbines. This can be found using equation 1.

$$W_{out} = W_{hp} + W_{lp} = (h_3 - h_4) + (h_5 - h_6) \tag{1}$$

In order to use equation 1 we need to know the enthalpies at states 3-6. The information given in figure 3 is shown in table I.

TABLE I. INFORMATION GIVEN IN FIGURE 3

Stage	Pressure	Temperature
3	$24.1MPa$	550^oC
4	$4MPa$	300^oC
5	$4MPa$	550^oC
6	$10kPa$	Unknown, but we know steam quality $x = 0.93$

The enthalpy at states 3-5 can easily be found by looking in superheated steam tables. Some tables may have slight differences, depending on how the table has been collated. The values here are taken from the recommended course text, thermodynamics: an engineering approach [2].

The enthalpy is found by looking for the given pressure in the table, then looking for the temperature, which can then be read accross to give enthalpy, h. Where there are not exact matches, a linear interpolation has been used.

The enthalpy at stage 3 is found by taking the given values at $20MPa$ and $25MPa$ for 550^oC then using a linear interpolation to find the required value. The values given in the steam table are shown in table III.

TABLE II. INFORMATION FROM STEAM TABLES [2]

Table Reading	Pressure	Temperature	Enthalpy
A	$20MPa$	550^oC	$3396.2kJ/kg$
B	$25MPa$	550^oC	$3339.2kJ/kg$

$$h_3 = h_a + (h_b - h_{20a})(\frac{P_3 - P_a}{P_b - P_a}) \tag{2}$$

$$h_3 = 3396.2 + (3339.2 - 3396.2)(\frac{24.1 - 20}{25 - 20}) \tag{3}$$

$$h_3 = 3349.46kJ/kg \tag{4}$$

A quick sanity check confirms that the value in equation 4 is correct, as it is correspondingly closer to the enthalpy at $25MPa$.

The enthalpy for h_4 can be simply read from the table. The enthalpy at $4MPa$ and 300^oC is given as $2961.7kJ/kg$.

The enthalpy at h_5 requires another linear interpolation, since the temperature at 550^oC is not given. Instead values are available for the enthalpy at $4MPa$ and 500^oC and at $4MPa$ and 600^oC.

TABLE III. INFORMATION GIVEN IN STEAM TABLES [2]

Table Reading	Pressure	Temperature	Enthalpy
A	$4MPa$	500^oC	$3446.0kJ/kg$
B	$4MPa$	600^oC	$3674.9kJ/kg$

Hence to find the enthalpy at 550^oC:

$$h_5 = h_a + (h_b - h_a)(\frac{T_5 - T_a}{T_b - T_a}) \tag{5}$$

$$h_5 = 3446.0 + (3674.9 - 3446.0)(\frac{550 - 500}{600 - 500}) \tag{6}$$

$$h_5 = 3560.45kJ/kg \tag{7}$$

A quick sanity check on equation 28 shows it to be correct, since it is exactly halfway between the two given enthalpies as expected.

In order to calculate the enthalpy at stage 6, we know only the pressure and the steam quality. We can find the enthalpy at this point by reading from the saturated water tables and taking the values for the mixture of water and vapour as in equation 8.

$$h_6 = h_f + x_6 h_{fg} \tag{8}$$

These values can be read directly from the table, giving $h_f = 191.81kJ/kg$ and $h_{fg} = 2392.1k/kg$. We are able to calculate the enthalpy at stage 6 using equation 8.

$$h_6 = h_f + x_6 h_{fg} = 191.81 + 0.93(2392.1) \tag{9}$$

$$h_6 = 2416.5kJ/kg \tag{10}$$

The enthalpies that we have calculated are summarised in table IV

TABLE IV. CALCULATED ENTHALPIES FOR STAGES 3-6

Stage	Enthalpy
3	$3349.46kJ/kg$
4	$2961.7kJ/kg$
5	$3560.45kJ/kg$
6	$2416.5kJ/kg$

Hence we are able to use equation 1 to calculate the work output of both turbines.

$$W_{out} = (3349.46 - 2961.7) + (3560.45 - 2416.5) \tag{11}$$

$$W_{out} = 1532.01kJ/kg \tag{12}$$

Power is defined as the rate at which work is transferred. 1W is equivalent to 1J/s [2]. In figure 3 we are given the mass flow through the turbine. We can use this information alongside the work out to determine the power of the turbines:

$$P_{out} = W_{out} \times MassFlow \tag{13}$$

$$P_{out} = 1532.01kJ/kg \times 240kg/s \tag{14}$$

$$P_{out} = 373432.8kJ/s = 367.6MW \tag{15}$$

A quick sanity check confirms that the units cancel properly to give the units of power as defined in [2]. Also, 367.6 MW is a perfectly reasonable size power output for a turbine. If one assumes that a generator is 95% efficient, this gives a crude estimate of roughly 350MW electrical output. This is an entirelty plausible size for a UK power station. Hence I am reasonably confident that my calculations are correct.

E. Pump power requirement calculations

This section concerns calculating the power required to drive the condensate and feedwater pumps. The pumps are considered to be adiabatic meaning no heat or matter is transferred. This does not necessarily mean the pumps are isentropic. The normal way to deal with irreversibilities is to use isentropic efficiencies. However, since the temperature and pressure are given at each point of the cycle, it is possible to work out the enthalpies from tables, and no isentropic efficienies need be calculated.

This can be calculated in a similar way to the turbine power output, by first calculating the work input required, and then multiplying by the mass flow of the system. The work required for the pumps are shown in equation 16. The work done by each pump is then defined by the isentropic compression of the water multiplied by the volume of the water, as in equations 17 and 18.

$$W_{in} = W_{FeedwaterPump} + W_{CondensatePump} \tag{16}$$

$$W_{FeedwaterPump} = h_2 - h_1 \tag{17}$$

$$W_{CondensatePump} = h_1 - h_7 \tag{18}$$

It is noted that this can be simplified to equation 19.

$$W_{in} = h_2 - h_1 + h_1 - h_7 = h_2 - h_7 \tag{19}$$

First of all we will calculate the value of h_7. This can be read directly from the steam tables, the enthalpy at $P_7 = 10kPa$ and $T_7 = 40^oC$ is $h_7 = 167.53kJ/kg$.

The value of h_2 can be calculated from the compressed water tables. The values in table V are taken directly from the compressed water table.

At state 2, $P_2 = 24.1MPa$ and $T_2 = 41^oC$ hence we need to perform three interpolations to find h_2. This is performed in the workings below.

Firstly, we will find the enthalpy, h_e, which is at $P_e = 20MPa$ and $T_e = 41^oC$.

$$h_e = h_a + (h_b - h_a)(\frac{T_2 - T_a}{T_b - T_a}) \tag{20}$$

TABLE V. INFORMATION GIVEN IN COMPRESSED WATER TABLE [2]

Table Reading	Pressure	Temperature	Enthalpy
A	$20MPa$	40^oC	$185.16kJ/kg$
B	$20MPa$	60^oC	$267.92kJ/kg$
C	$30MPa$	40^oC	$193.90kJ/kg$
D	$30MPa$	60^oC	$276.26kJ/kg$

$$h_e = 185.16 + (267.92 - 185.16)(\frac{41 - 40}{60 - 40}) \tag{21}$$

$$h_e = 189.298kJ/kg \tag{22}$$

Next, we will find the enthalpy, h_g, which is at $P_e = 30MPa$ and $T_e = 41^oC$.

$$h_g = h_c + (h_d - h_c)(\frac{T_2 - T_c}{T_d - T_c}) \tag{23}$$

$$h_g = 193.90 + (276.26 - 193.90)(\frac{41 - 40}{60 - 40}) \tag{24}$$

$$h_g = 198.018kJ/kg \tag{25}$$

Finally we interpolate between h_e and h_g to find enthalpy, h_2.

$$h_2 = h_e + (h_g - h_e)(\frac{P_2 - P_e}{P_g - T_e}) \tag{26}$$

$$h_e = 189.298 + (198.018 - 189.298)(\frac{24.1 - 20}{30 - 20}) \tag{27}$$

$$h_e = 192.87kJ/kg \tag{28}$$

We now have the values required to calculate the iput work by the pumps, summarised in table VII.

TABLE VI. CALCULATED ENTHALPIES FOR STAGES 2 & 7

Stage	Enthalpy
2	$192.87kJ/kg$
7	$167.53kJ/kg$

The work in can thus be found in equation 29.

$$W_{in} = h_2 - h_7 = 192.87 - 167.53 = 25.34kJ/kg \tag{29}$$

Power is defined as the rate at which work is transferred. 1W is equivalent to 1J/s [2]. In figure 3 we are given the mass flow through the turbine. It is assumed that the system has no leaks, and the massflow remains constant around the system. We can use this information alongside the work out to determine the power of the turbines:

$$P_{in} = W_{in} \times MassFlow \tag{30}$$

$$P_{in} = 25.34kJ/kg \times 240kg/s \tag{31}$$

$$P_{in} = 6081.6kJ/s = 6.08MW \tag{32}$$

A quick sanity check shows that the power put into the system is much less than the power coming out. This makes sense since the extra power out comes from the thermal input from the combustion process. Hence I am fairly confident that these calculations are reasonable.

F. Cycle efficiency calculations

The equation I am going to use for cycle efficiency is equation 42. It was chosen since we have already calculated work in and work out, making the net work simple to find. The equation for the specific heat input is given in 34.

$$\eta_{th} = \frac{W_{net}}{q_{in}} \tag{33}$$

$$q_{in} = q_{primary} + q_{reheat} = (h_3 - h_2) + (h_5 - h_4) \tag{34}$$

To calculate the net work, I used the work in and work out that was found during parts C and D.

$$W_{net} = W_{out} - W_{in} = 1532.01 - 25.34 = 1506.67kJ/kg \tag{35}$$

In order to calculate q_{in} we need to know the enthalpies at stages 2 to 5. We already calculated these throughout parts C & D.

TABLE VII. CALCULATED ENTHALPIES FOR STAGES 2 - 5

Stage	Enthalpy
2	$192.87kJ/kg$
3	$3349.46kJ/kg$
4	$2961.7kJ/kg$
5	$3560.45kJ/kg$

Hence q_{in} can be found using equation 34 as shown in equation 62.

$$q_{in} = (3349.46 - 192.87) + (3560.45 - 2961.7)kJ/kg \tag{36}$$

$$q_{in} = 3755.34kJ/kg \tag{37}$$

The efficiency of the cycle is thus found in equation 38.

$$\eta_{th} = \frac{1506.67}{3755.34} = 40.1\% \tag{38}$$

Sanity check: 40.1% seems to be logical. The example in Boles [2] has an efficiency of 45%. The example in Boles is for an ideal system, where the turbines and pumps are isentropic, so it is expected that a system where there are irreversibilities will be less efficient.

G. Heat rejected at condenser calculations

The heat rejected at the condenser is equivalent to q_out. q_out can be calculated as shown in equation 39.

$$q_{out} = h_6 - h_7 \tag{39}$$

h_6 was calculated in part C using the pressure and steam quality information, and h_7 was calculated when working out the pump power requirement. These are summarised in table VIII.

TABLE VIII. CALCULATED ENTHALPIES FOR STAGES 6 & 7

Stage	Enthalpy
6	$2416.5kJ/kg$
7	$167.53kJ/kg$

q_{out} can now be calculated:

$$q_{out} = 2416.5 - 167.53 = 2248.97kJ/kg \tag{40}$$

ASIDE: This provides an opportunity to check my work. An alternate equation can be used to recalculate the efficiency of the plant.

$$\eta_{th} = 1 - \frac{q_{out}}{q_{in}} \tag{41}$$

Since we have calculated q_{in} we can check our previous answer:

$$\eta_{th} = 1 - \frac{2248.97}{3755.34} = 40.1\% \tag{42}$$

This confirms that the workings so far are likely to be correct.

Back to calculating the heat rejected at the condenser: The heat rejected at the condenser is as follows:

$$\dot{Q}_{cond} = q_{out} \times MassFlow \tag{43}$$

$$\dot{Q}_{cond} = 2248.97kJ/kg \times 240kg/s \tag{44}$$

$$\dot{Q}_{cond} = 539752kJ/s = 539MW \tag{45}$$

H. Cooling water flow calculations

By taking a control volume around the condenser, we can define the energy balance at steady state:

$$0 = \dot{Q}_{cond} - \dot{W}_{cond} + \dot{m}_{cw}(h_{cw,out} - h_{cw,in}) + \dot{m}(h_6 - h_7) \tag{46}$$

If we solve this for the mass of the cooling water required:

$$\dot{m}_{cw} = \frac{\dot{m}(h_6 - h_7)}{(h_{cw,out} - h_{cw,in})} \tag{47}$$

The numerator of this equation has already been calculated, as value $\dot{Q}_{cond}$:

$$\dot{m}_{cw} = \frac{\dot{Q}_{cond}}{(h_{cw,out} - h_{cw,in})} \tag{48}$$

To determine the enthalpies of the cooling water at the entrance and exits of the condenser, we must first logically define the temperatures at these points. It is stated in the instructions that the average temperature of the river Test is 11^oC. If the change in temperature cannot be more than 10^oC, then the output temperature of the cooling water from the condenser must be 21^oC. For simplicity, I have made the approximation to 10^oC and 20^oC respectively. This means the values can be directly read from the saturated water tables. Taking the saturated liquid value for these temperatures we get the values in table IX.

TABLE IX. CALCULATED ENTHALPIES FOR CONDENSER INPUT AND OUTPUT WATER

Stage	Enthalpy	Temperature
$h_{cw,out}$	$83.915kJ/kg$	20^oC
$h_{cw,in}$	$42.022kJ/kg$	10^oC

Hence we can use equation 48 to calculate the cooling water flow rate:

$$\dot{M}_{cw} = \frac{539752\dot{k}J/s}{(83.915kJ/kg - 42.022kJ/kg)} \quad (49)$$

$$\dot{M}_{cw} = 12884kg/s \quad (50)$$

III. STEAM PLANT 3 DESIGN

Another design of steam plant has been included in figure 6. It differs from plant 2 in that a portion of the steam, mass fraction y, is bled from the low pressure turbine and used to pre-heat the feedwater. This is called a regenerative rankine cycle.

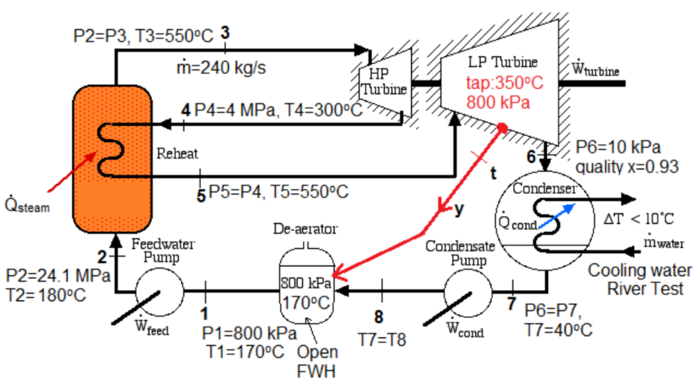

Fig. 6. Steam Plant 3 Design (taken from [1])

Steam plant 3 has both reheat and regeneration stages. Regeneration is an effective way to increase the heat-addition temperature, which in turn increases cycle efficiency [2]. The steam bled from the low pressure turbine could have been used to provide further work from the turbine. However it is instead used to pre-heat the feedwater before it reaches the boiler. This heating happens in an open feedwater heater. In this device, the steam and condensate water mix together. Alternatively a closed feedwater heater could be used, which replaces the mixing chamber with a heat exchanger so the two fluids no longer mix. Regeneration has been widely used since the 1920s [2].

A. Discussion of T-s Diagram

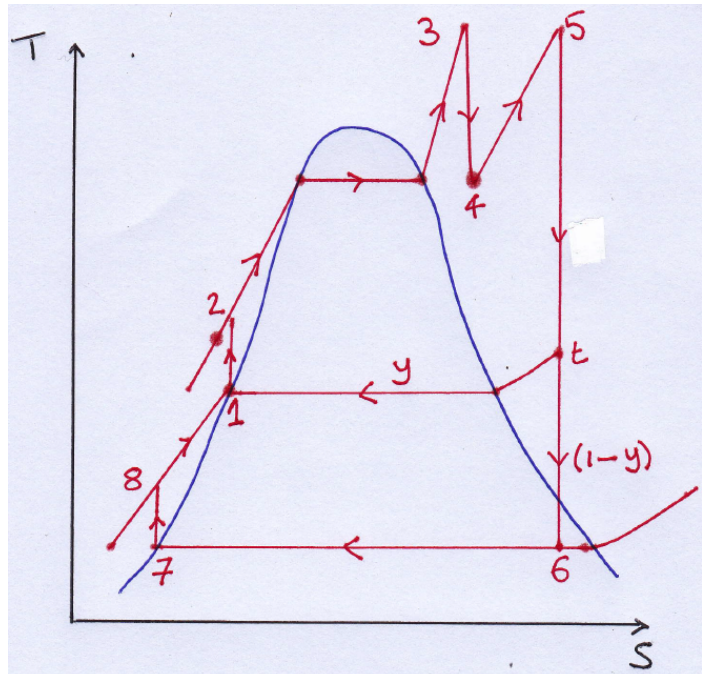

Fig. 7. T-s diagram of steam plant 3 - note that irreversibilities in the system are not pictured on the diagram. It shows an ideal regenerative reheat rankine cycle.

The regenerative reheat rankine cycle is pictured in figure 7. We will discuss it in sequence.

a) Stage 1-2: Isentropic (in the ideal case) compression by the feedwater pump.

b) Stage 2-3: Constant pressure heat addition in the boiler. This is assumed lossless in the ideal case.

c) Stage 3-4: Isentropic (in the ideal case) expansion in the high pressure turbine.

d) Stage 4-5: Constant pressure heat addition in the reheat cycle. Steam is taken from the high pressure turbine and passed through an additional heat exchanger stage in the boiler to reheat the steam to the turbine inlet temperature.

e) Stage 5-t: The whole of the steam is expanded isentropically (in the ideal case) in the low pressure. At point t a portion of the steam is bled from the turbine to be fed into the open feedwater heater.

f) Stage t-6: The remainder of the steam continues to expand isentropically (in the ideal case) in the low pressure turbine to the condenser inlet pressure.

g) Stage 6-7: The remainder of the steam undergoes constant pressure heat rejection, and condensed to a saturated liquid in the condenser.

h) Stage 7-8: The condensed water undergoes isentropic (in the ideal case) compression in the condensate feedwater pump, to the pressure of the open feedwater heater.

i) Stage 1: The steam bled from the low pressure turbine mixes with the pressurised condensate water in the open feedwater heater. The fraction of the steam bled off, y, is such that the mixture leaves the open feedwater heater as a saturated liquid at the operating pressure of the open feedwater heater [2]. The mixture then enters the feedwater pump as described in stage 1-2.

Intuitively, it is clear that the efficiency of the system will increase due to regeneration. By bleeding steam from the low pressure turbine, we can raise the temperature at which the feedwater enters the boiler. Some industrial plants use many regeneration stages, and the number of such arrangements is determined by the cost to install compared to the cost saving by increased efficiency [2]. Irreversibilities in the system are not pictured. While they are important in characterising the system, the enthalpy at all points of the system is defined and hence it is not necessary to deal with isentropic efficiencies in the following calculations.

B. Mass Fraction Calculation

A portion of the steam is bled from a tap in the low pressure turbine and fed into an open feedwater heater. Energy analysis of open feedwater heaters is analagous to that of a mixing chamber [2]. Since it is assumed the open feedwater heater is adiabatic (no heat is lost) and it is a work neutral device (i.e. $\dot{W} = 0$) the energy balance can be calculated as follows. Note that y is the mass fraction of the steam, and h_t is the enthalpy at the tap in the turbine.

$$yh_t + (1-y)h_7 = h_1 \tag{51}$$

Solving for y we get:

$$y = \frac{h_1 - h_7}{h_t - h_7} \tag{52}$$

Enthalpies h_1 and h_t have not yet been found, but this can be found from the steam/water tables directly. Table X shows the required values.

TABLE X. CALCULATED ENTHALPIES FOR STAGES T & 7 & 1

Stage	Enthalpy
1	$719.08kJ/kg$
7	$167.53kJ/kg$
t	$3162.2kJ/kg$

Hence, using equation 54 we can find y:

$$y = \frac{h_1 - h_7}{h_t - h_7} = \frac{719.08 - 167.53}{3162.2 - 167.53} \tag{53}$$

$$y = 0.184 \ (no\ units) \tag{54}$$

This would seem to be a reasonable value consistent with logic and with examples from lectures.

C. Thermal Efficiency

We will use equation 55 to calculate efficiency.

$$\eta_{th} = 1 - \frac{q_{out}}{q_{in}} \tag{55}$$

The open feedwater heater means that only the unbled steam goes into the condenser. Hence q_{out} is reduced. by a scaling factor of $(1-y)$ compared with steam plant 2. Note that we use the calculated value for q_{out} from steam plant 2 here.

$$q_{out} = (1-y)(h_6 - h_7) = (1-y)q_{outsteamplant2} \tag{56}$$

$$q_{out} = (1 - 0.184)(2248.97) = 1835.16kJ/kg \tag{57}$$

The value of q_{in} is also simple to recalculate. Due to the heat added in the open water heater, the temperature of the water at the entrance to the boiler has increased compared to steam cycle 2. Hence recalculating for $P_2 = 24.1MPa$ and $T_2 = 180^oC$ we need to perform a linear interpolation on the values given in the compressed water table, shown in table XI.

TABLE XI. INFORMATION GIVEN IN COMPRESSED WATER TABLE [2]

Table Reading	Pressure	Temperature	Enthalpy
A	$20MPa$	180^oC	$773.02kJ/kg$
B	$30MPa$	180^oC	$778.55kJ/kg$

Hence we can find h_2:

$$h_2 = h_a + (h_b - h_a)(\frac{P_2 - P_a}{P_b - T_a}) \tag{58}$$

$$h_e = 773.02 + (778.55 - 773.02)(\frac{24.1 - 20}{30 - 20}) \tag{59}$$

$$h_e = 775.29kJ/kg \tag{60}$$

The values require to recalculate q_{in} for steam plant 3 are shown in table XII.

TABLE XII. CALCULATED ENTHALPIES FOR STAGES 2 -5

Stage	Enthalpy
2	$775.29kJ/kg$
3	$3349.46kJ/kg$
4	$2961.7kJ/kg$
5	$3560.45kJ/kg$

$$q_{in} = (3349.46 - 775.29) + (3560.45 - 2961.7)kJ/kg \tag{61}$$

$$q_{in} = 3172.92kJ/kg \tag{62}$$

Hence the thermal efficiency can be calculated using equation 55.

$$\eta_{th} = 1 - \frac{1835.16}{3172.92} = 42.1\% \tag{63}$$

This has increased from 40.1% for steam plant 2. This seems like a reasonable increase in efficiency for a regenerative cycle, and is consistent with examples in Boles [2] and lecture notes.

D. Power Output of Turbines

This has changed from steam plant 2, since some of the steam is bled off from the low pressure turbine, leaving less steam to be expanded after the tap. This can be expressed by equation 64.

$$W_{out} = W_{hp} + W_{lp} = (h_3 - h_4) + (h_5 - h_t) + (1-y)(h_t - h_6) \tag{64}$$

All of the values in equation 64 have been calculated previously as shown in table XIII.

TABLE XIII. CALCULATED ENTHALPIES FOR STAGES 3-6 & T

Stage	Enthalpy
3	$3349.46kJ/kg$
4	$2961.7kJ/kg$
5	$3560.45kJ/kg$
6	$2416.5kJ/kg$
t	$3162.2kJ/kg$

Hence we can calculate W_{out} (note that equation 65 and 66 are the same equation, it is just very long!):

$$W_{out} = (3349.46 - 2961.7) + (3560.45 - 3162.2) \tag{65}$$

$$+ (1 - 0.184)(3162.2 - 2416.5) \tag{66}$$

$$W_{out} = 387.76 + 398.25 + 608.49 = 1394.5kJ/kg \tag{67}$$

Similar to the method for steam plant 2, we can calculate combined turbine output power:

$$P_{out} = W_{out} \times MassFlow \tag{68}$$

$$P_{out} = 1394.5kJ/kg \times 240kg/s \tag{69}$$

$$P_{out} = 334680kJ/s = 334.7MW \tag{70}$$

E. Comparison with Steam Plant 2

Before we undertake any verbal comparision, we will consider the numbers side by side in table XIV.

TABLE XIV. COMPARING CHARACTERISTICS OF STEAM PLANTS 2 & 3

Attribute	Plant 2	Plant 3
P_{out}	$367.6MW$	$334.7MW$
Q_{in}	$3755.34kJ/kg$	$3172.92kJ/kg$
η_{th}	40.1%	42.1%

It is clear from table XIV that Plant 3 has a greater efficiency and lower heat input than Plant 2. However the power output has also dropped by approximately 10%. Higher efficiencies are environmentally and economically beneficial for a commercial plant, as less fuel is required for the same output. However, there is a sacrifice in the power output between the two plants. If Plant 3 had the same heat input, then it would produce more power than Plant 2. Apart from the additional engineering complexity, Plant 3 is a superior plant.

F. Calculate Total Heat Transfer to Steam Generator

This is equivalent to calculating the heat input required, $\dot{Q}_{steam}$. We have already calculated q_{in} and we know the mass flow in the system.

$$\dot{Q}_{steam} = q_{in} \times MassFlow \tag{71}$$

$$\dot{Q}_{steam} = 3172.92kJ/kg \times 240kg/s \tag{72}$$

$$\dot{Q}_{cond} = 761500.8kJ/s = 761.5MW \tag{73}$$

G. Heat Rejected to the Condenser

The heat out of the system has already been calculated, $q_{out} = 1835.16kJ/kg$. We can use this in conjunction with the mass flow through the condenser (which is $(1-y)\dot{m}$) to find the heat rejected to the condenser.

The heat rejected at the condenser is as follows:

$$\dot{Q}_{cond} = q_{out} \times MassFlow \tag{74}$$

$$\dot{Q}_{cond} = 1835.16kJ/kg \times 240kg/s \tag{75}$$

$$\dot{Q}_{cond} = 440438.4kJ/s = 440MW \tag{76}$$

This is significantly less than steam plant 2, which is expected since some of the steam is being bled and is not passed through the condenser.

H. Cooling Water Flow Rate

As with steam plant 2, we can use the equation 77 to calculate the mass of the cooling water required (note this is derived in section 2).

$$\dot{m}_{cw} = \frac{\dot{Q}_{cond}}{(h_{cw,out} - h_{cw,in})} \tag{77}$$

To determine the enthalpies of the cooling water at the entrance and exits of the condenser, we must first logically define the temperatures at these points. It is stated in the instructions that the average temperature of the river Test is 11^oC. If the change in temperature cannot be more than 10^oC, then the output temperature of the cooling water from the condenser must be 21^oC. For simplicity, I have made the approximation to 10^oC and 20^oC respectively. This means the values can be directly read from the saturated water tables. Taking the saturated liquid value for these temperatures we get the values in table XV.

TABLE XV. CALCULATED ENTHALPIES FOR CONDENSER INPUT AND OUTPUT WATER

Stage	Enthalpy	Temperature
$h_{cw,out}$	$83.915kJ/kg$	20^oC
$h_{cw,in}$	$42.022kJ/kg$	10^oC

Hence we can use equation 48 to calculate the cooling water flow rate:

$$\dot{m}_{cw} = \frac{440438.4kJ/s}{(83.915kJ/kg - 42.022kJ/kg)} \tag{78}$$

$$\dot{m}_{cw} = 10513.4kg/s \tag{79}$$

This is some $1500kg/s$ less cooling water mass required than for a non-regenerative cycle. This is expected since there is less heat rejected to the environment. This results in less thermal pollution to the environment.

IV. Cooling Towers

Cooling towers are an important part of power plants, allowing vast quantities of rejected heat to be disipated to the atmosphere. There are two major types of cooling tower; induced draft and natural draft.

Fig. 8. Natural draft cooling towers at Didcot A power station, Oxfordshire. Didcot A was a 2GW coal station that entered operation in 1970 and closed in March 2013 under the LCP directive [5].

Fig. 9. Induced draft cooling towers at Didcot B power station, Oxfordshire. Didcot B is a 1.4GW Siemens CCGT Station [5].

Natural draft cooling towers work like a normal chimney. The air inside the cooling tower has a higher water-vapour content and is thus lighter. Hence it rises, and heavy air from outside is drawn into the bottom of the tower. This is a natural process, and therefore requires no fans, but does result in some very large structures. The towers at Didcot A (figure 9) are iconic and can be seen for miles in every direction. Induced draft cooling towers are much smaller in size, and are thus easier to gain planning approval in the UK. Instead of a natural draft, the draft is provided by a large fan. This means there is a power requirement for these towers.

A. *Cooling Tower Sizing Procedure*

Choosing the correct cooling tower is critical to the performance and efficiency of a steam plant. It is not just the type of cooling tower that should be considered, but a host of factors. Firstly, one should consider the required cooling capacity. The amount of heat that is required to be dissipated can be very large and will partially determine the required size. The cooling capacity alone is not sufficient information to size a cooling tower. The range over which the cooling must take place is also important. The specification of a cooling tower normally gives a certain flow rate that is cooled from one temperature to another. Clearly, a tower that must cool over a larger range will be larger [6].

The approach of the tower is also a critical value. This is determined by how close the condenser water in the system approaches wet-bulb temperature. The closer to the wet-bulb temperature, the larger the cooling tower needs to be. The wet bulb temperature is determined by geography, so areas that have particularly high wet-bulb temperatures require larger cooling towers [6]. This information together with the cooling capacity and range allows us to choose the right size tower.

We will now consider a theoretical cooling tower problem. The diagram in figure 10 outlines the thermodynamic problem considered.

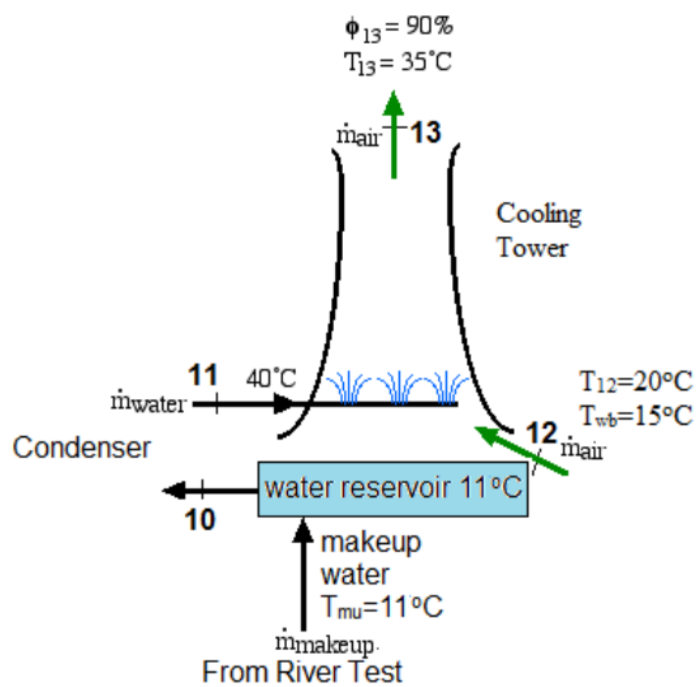

Fig. 10. Cooling Tower Problem Description (taken from [1])

B. Flow Rate of Dry Air

This subsection will calculate the volumetric flow rate of dry air required through this cooling tower as described if figure 10. Firstly we will find the mass balance of the air through the cooling tower. Since air is not lost in the tower, the air entering is equivalent to the air leaving as in equation 80.

$$\dot{m}_{air,in} = \dot{m}_{air,out} = \dot{m}_{air} \tag{80}$$

We can do the same for the mass of water in the system. Water is lost from the system through the evaporation process, as shown in equation 81. Equation 81 can be written as equation 82 using the stages declared in figure 10.

$$\dot{m}_{water,in} + \omega_{in}\dot{m}_{air} = \dot{m}_{water,out} = \omega_{out}\dot{m}_{air} \tag{81}$$

$$\dot{m}_{water,in} + \omega_{12}\dot{m}_{air} = \dot{m}_{water,out} = \omega_{13}\dot{m}_{air} \tag{82}$$

This allows us to calculate the equation for mass of makeup water required:

$$\dot{m}_{water,in} - \dot{m}_{water,out} = \dot{m}_{makeup} = \dot{m}_{air}(\omega_{13} - \omega_{12}) \tag{83}$$

Next we need to find an equation to find the mass of air. We can look at the energy balance of the system to calculate this.

$$\sum_{in} \dot{m}h = \sum_{out} \dot{m}h \tag{84}$$

$$\dot{m}_{water}h_{11} + \dot{m}_{air}h_{12} = \dot{m}_{water,out}h_{10} + \dot{m}_{air}h_{13} \tag{85}$$

$$\dot{m}_{water}h_{11} = (\dot{m}_{water} + \dot{m}_{makeup})h_{10} + \dot{m}_{air}(h_{13} - h_{12}) \tag{86}$$

By rearranging equation 86 the solution for mass of air can be found.

$$\dot{m}_{water}(h_{11} - h_{10}) = \dot{m}_{makeup}h_{10} + \dot{m}_{air}(h_{13} - h_{12}) \tag{87}$$

Substitute the mass of makeup water:

$$\dot{m}_{water}(h_{11} - h_{10}) = \dot{m}_{air}(\omega_{13} - \omega_{12})h_{10} + \dot{m}_{air}(h_{13} - h_{12}) \tag{88}$$

$$\dot{m}_{air} = \frac{\dot{m}_{water}(h_{11} - h_{10})}{(\omega_{13} - \omega_{12})h_{10} + (h_{13} - h_{12})} \tag{89}$$

Note that this equation is slightly different to the equation derived in the example in [2]. This is since state 10 in this example is after the makeup water has been added to the system, hence the addition, whereas in the example in [2] the makeup water is added afterwards.

Now that all of the equations we require have been defined, we can now read the variables required. Firstly, we will gather the variables not in the psychrometric chart.

h_{11} is the enthalpy of the water at the inlet to the cooling tower from the condenser. This water is a saturated liquid at 40^oC, which can be read directly from the saturated water table: $h_{11} = 167.53kJ/kg$.

The enthalpy at the exit of the cooling tower, h_{10}, is a saturated liquid at 11^oC. This is not directly available from

TABLE XVI. INFORMATION GIVEN IN SATURATED WATER TABLE [2]

Table Reading	Temperature	Enthalpy
A	10^oC	$42.022kJ/kg$
B	15^oC	$62.982kJ/kg$

the saturated water table, hence a linear interpolation will be performed from the values given in table XVIII.

$$h_{10} = h_a + (h_b - h_a)(\frac{T_{10} - T_a}{T_b - T_a}) \tag{90}$$

$$h_{10} = 42.022 + (62.982 - 42.022)(\frac{11^oC - 10^oC}{15^oC - 10^oC}) \tag{91}$$

$$h_{10} = 46.214kJ/kg \tag{92}$$

In order to calculate the enthalpy of the air, a psychrometric is used. State 12 is defined by $T_{12} = 20^oC$ and $T_{wb} = 15^oC$. These two temperature readings come from two different types of thermometer used when assessing the characteristics of air. Figure 11 shows the recommended psychrometric chart and the values read from it. The purple cross shows where the vertical $T_{12} = 20^oC$ intersects with the red $T_{wb} = 15^oC$ line. At the point we can read directly horizontally to find the specific humidity, read on the blue curves to find te relative humidity, read along the angled black lines to find the enthalpy of the dry air and read between the angled green lines to find the specific volume of the air.

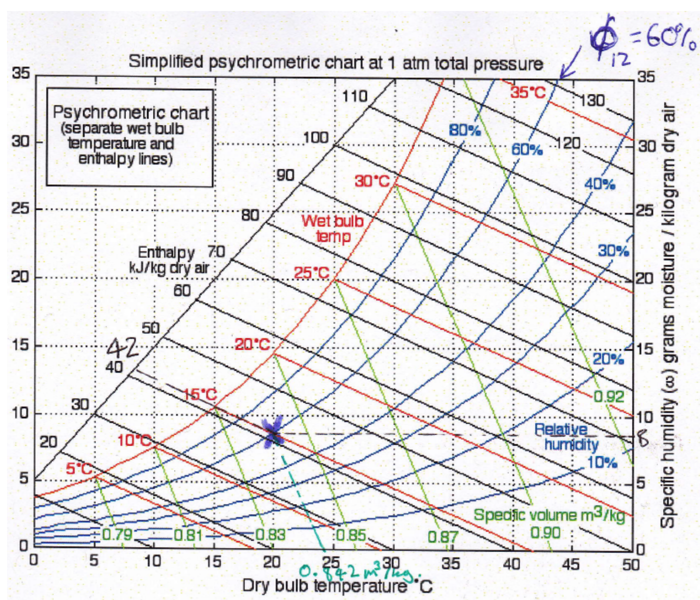

Fig. 11. Psychrometric chart annoted with readings for state 12

State 13 has different values defined, but is read from the chart in a very similar way. Given in the question is that the dry bulb temperature, $T_{13} = 35^oC$ and the relative humidity,

TABLE XVII. STATE 12 READINGS FROM PSYCHROMETRIC CHART

Attribute	Value
Dry bulb temp, T_{12}	20^oC
Wet bulb temp, T_{wb}	15^oC
Enthalpy, h_{12}	$42kJ/kg\ dry\ air$
Specific humidity, ω_{12}	$8gH_2O/kg\ dry\ air$
Specific volume, v_{12}	$0.842m^3/kg$
Relative humidity, ϕ_{12}	60%

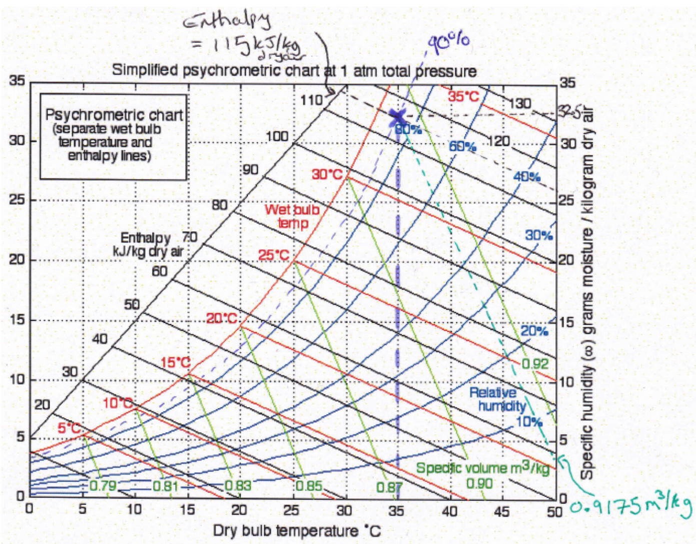

Fig. 12. Psychrometric chart annoted with readings for state 13

TABLE XVIII. STATE 13 READINGS FROM PSYCHROMETRIC CHART

Attribute	Value
Dry bulb temp, T_{13}	35^oC
Relative humidity, ϕ_{13}	90%
Wet bulb temp, T_{wb}	33^oC
Enthalpy, h_{13}	$115kJ/kg\ dry\ air$
Specific humidity, ω_{13}	$32.5gH_2O/kg\ dry\ air$
Specific volume, v_{13}	$0.9175m^3/kg$

$\phi_{13} = 90\%$. The reading was made in the annotated chart in figure 12.

Finally, the last assumption made is that the water flowing in the system is the same as the mass flow of the water calculated for steam plant 3. That is, $\dot{m}_{water} = \dot{m}_{cw} = 10513.4kg/s \approx 10500kg/s$. We now have all the information required to calculate the mass flow of the air through the cooling tower.

Taking equation 94 and substituting the values found gives the following:

$$\dot{m}_{air} = \frac{\dot{m}_{water}(h_{11} - h_{10})}{(\omega_{13} - \omega_{12})h_{10} + (h_{13} - h_{12})} \tag{93}$$

$$\dot{m}_{air} = \frac{10500(167.62 - 46.214)}{(0.0325 - 0.008)46.214 + (115 - 42)} \tag{94}$$

$$\dot{m}_{air} = \frac{10500(121.406)}{1.132 + 73} \tag{95}$$

$$\dot{m}_{air} = 17196kg/s \tag{96}$$

The volume flow rate of the air can be found by multiplying th mass flow by the specific volume of the air entering the cooling tower.

$$\dot{V}_{air} = \dot{m}_{air} \times v_{12} \tag{97}$$

$$\dot{V}_{air} = 17196 \times 0.842 = 14479m^3/s \tag{98}$$

C. Make Up Water Mass Flow

The equation for calculating the make up water required is as follows: This allows us to calculate the equation for mass of makeup water required:

$$\dot{m}_{water,in} - \dot{m}_{water,out} = \dot{m}_{makeup} = \dot{m}_{air}(\omega_{13} - \omega_{12}) \tag{99}$$

$$\dot{m}_{makeup} = 17196(0.0325 - 0.008) \tag{100}$$

$$\dot{m}_{makeup} = 421.3kg/s \tag{101}$$

This would seem to be a sensible amount of makeup water. The water is required due to the water lost in the evaporation process. The makeup water required is far less than the water entering the system from the condenser, in fact the makeup water is about 4% of the mass flow entering the cooling tower. This would seem a sensible value. Further, considering the context of the river test, this is a more than sensible cooling tower. The Environment Agency measure the flow rate of the river Test as between $6 - 37m^3/s$. Considering that $1m^3 = 1000kg$, it is easy to see that the mass flow of the river test is between $6,000 - 47,000kg/s$. Hence it is entirely plausible that a makeup water mass flow in equation 101 is used on the river Test.

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