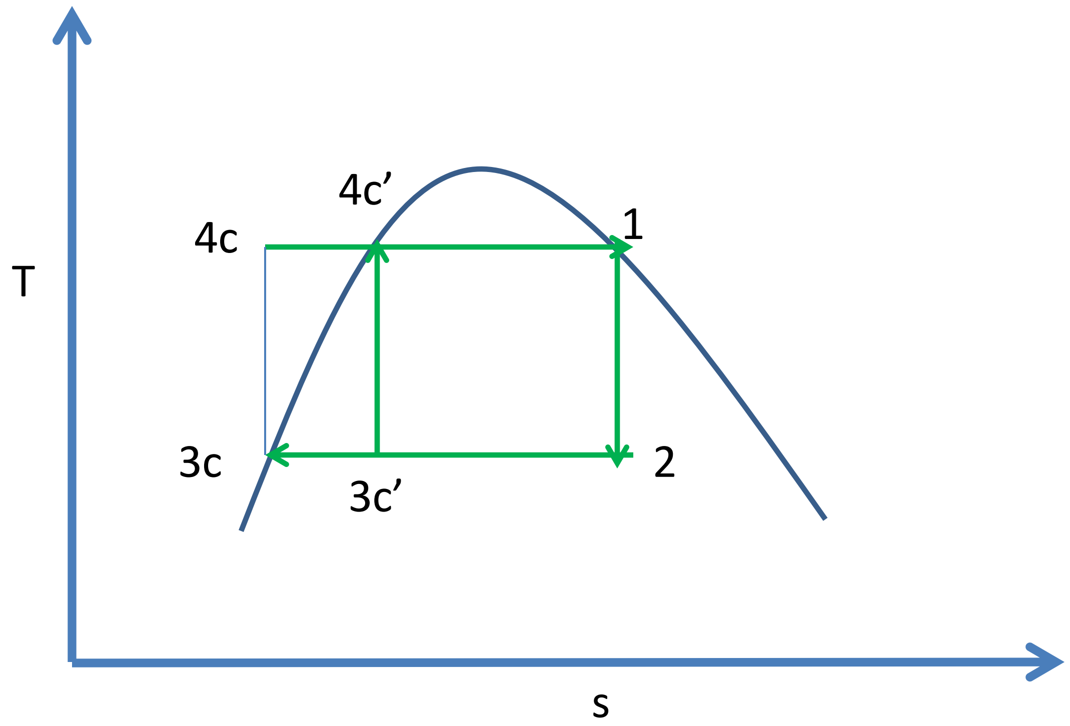
Vapour Power Cycles

- Water is the most suitable working fluid to use for power plant cycles
- While working fluid advantages and disadvantages exist, the most overpowering reason to use water appears to be its easy availability and non-toxic nature
- A Carnot cycle can be thought about for the use of vapour as working fluid for power generation

Vapour Power Cycle- Carnot



Problems with Vapour Carnot Cycle

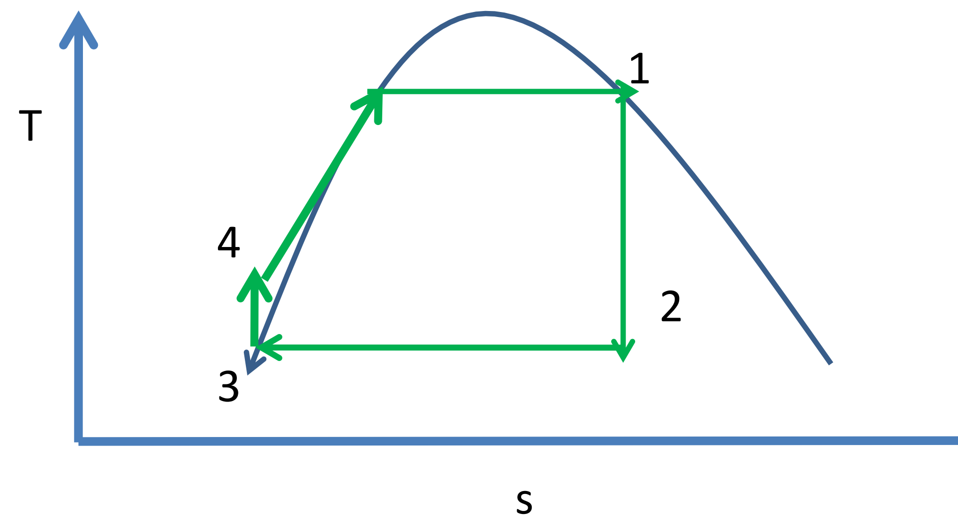
1. Isothermal process 4c-4c’ difficult
2. Pump work from 3c-4c very large
3. Very difficult to control quality at 3c’.
4. Multiphase fluid pumping very difficult.
5. Moisture levels at turbine exit high and therefore unacceptable.
6. Rankine cycle is an engineering compromise

Rankine cycle-1

- Main contribution is moving of two phase pumping into the single phase zone – big engineering advantage
- Now heating is both in single and two phase zone – isothermal heat addition condition is compromised
- Exit quality at turbine may be a problem but can be tackled.
- Problems are reduced but so is the efficiency

Rankine cycle-2

- 1-2: Expansion in turbine
- 2-3: Condenser
- 3-4: Pump
- 4-1: Boiler



Superheat Rankine Cycle

- Super heat the steam before turbine entry. This reduces exit quality and increases specific output. Efficiency increases if condenser temperature is constant.
- Superheating has disadvantages also:
  - Gases on both sides of heat exchanger lower heat transfer coeffs. result in longer piplines, higher wall temperatures, expensive materials

Superheat Rankine cycle

- Efficiency can be further increased by reducing the exit pressure. Align the exit pressure such that temperature is nearly atmospheric. This way condenser pressure vacuum but need to worry only about air ingress.
- Efficiency also increases by working pressure increase – mean temperature of heat addition increases. Increased quality at turbine exit is taken care by further superheating.

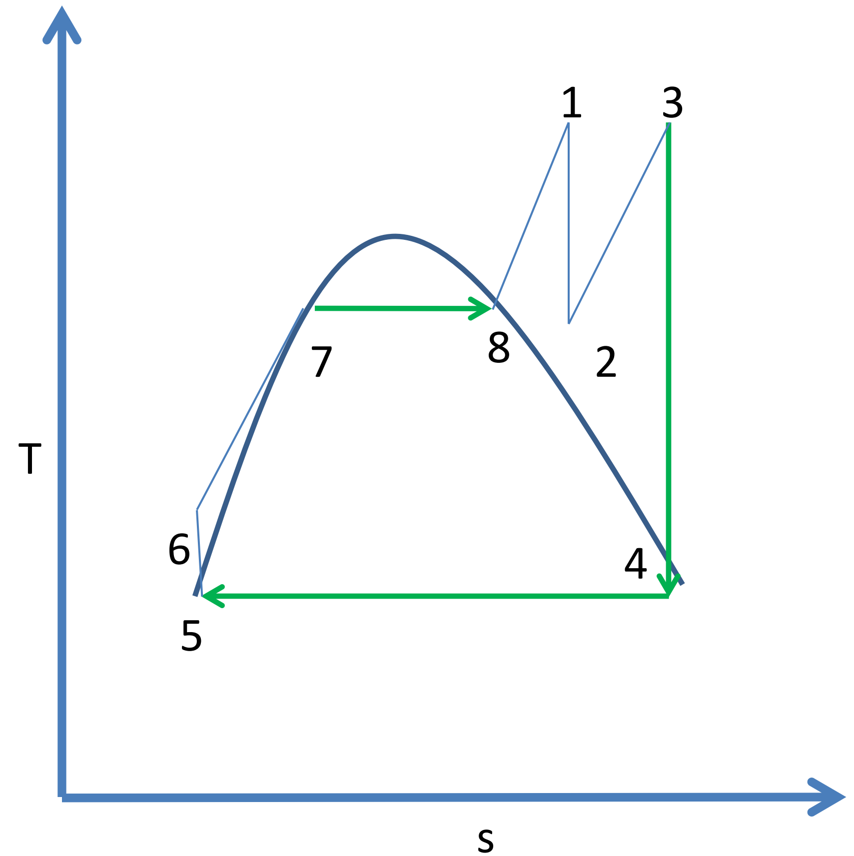
Reheat Cycle

- Superheat alone may not be enough at very high pressures to limit turbine exit quality.
- Reheat is the solution. Additional heat addition in the 2-3 region.
- Efficiency may or may not improve but proper choice of reheat pressures may bring about slight improvements in efficiency by increasing mean temp. of heat addition.

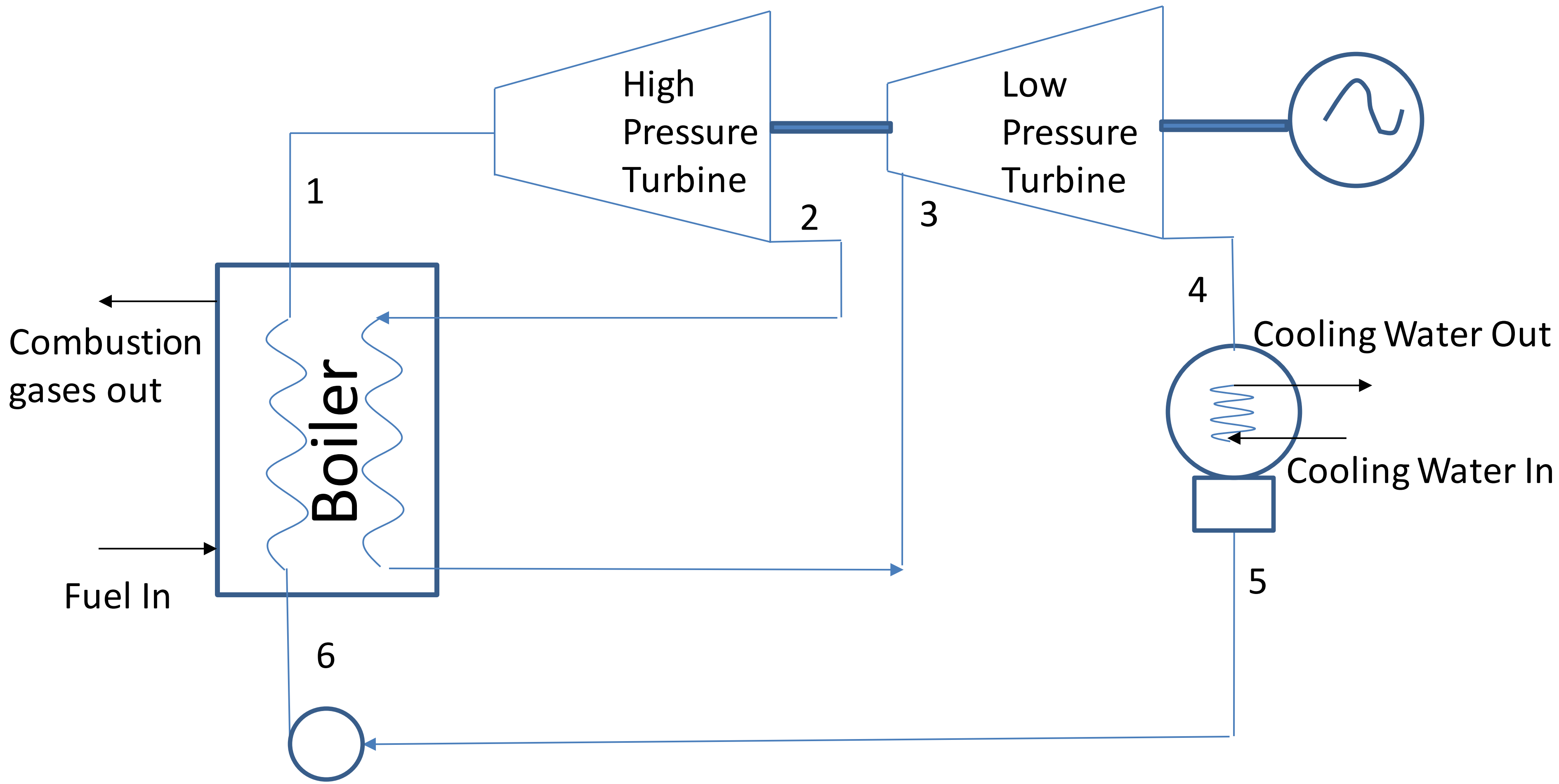
$$T_m = \frac{(h_1 - h_4) + (h_3 - h_2)}{s_4 - s_5}$$

Reheat Cycle

- 1-2: High Pressure Turbine
- 2-3: Reheat
- 3-4: Low Pressure Turbine
- 4-5: Condenser
- 5-6: Pump
- 6-1: Heat input



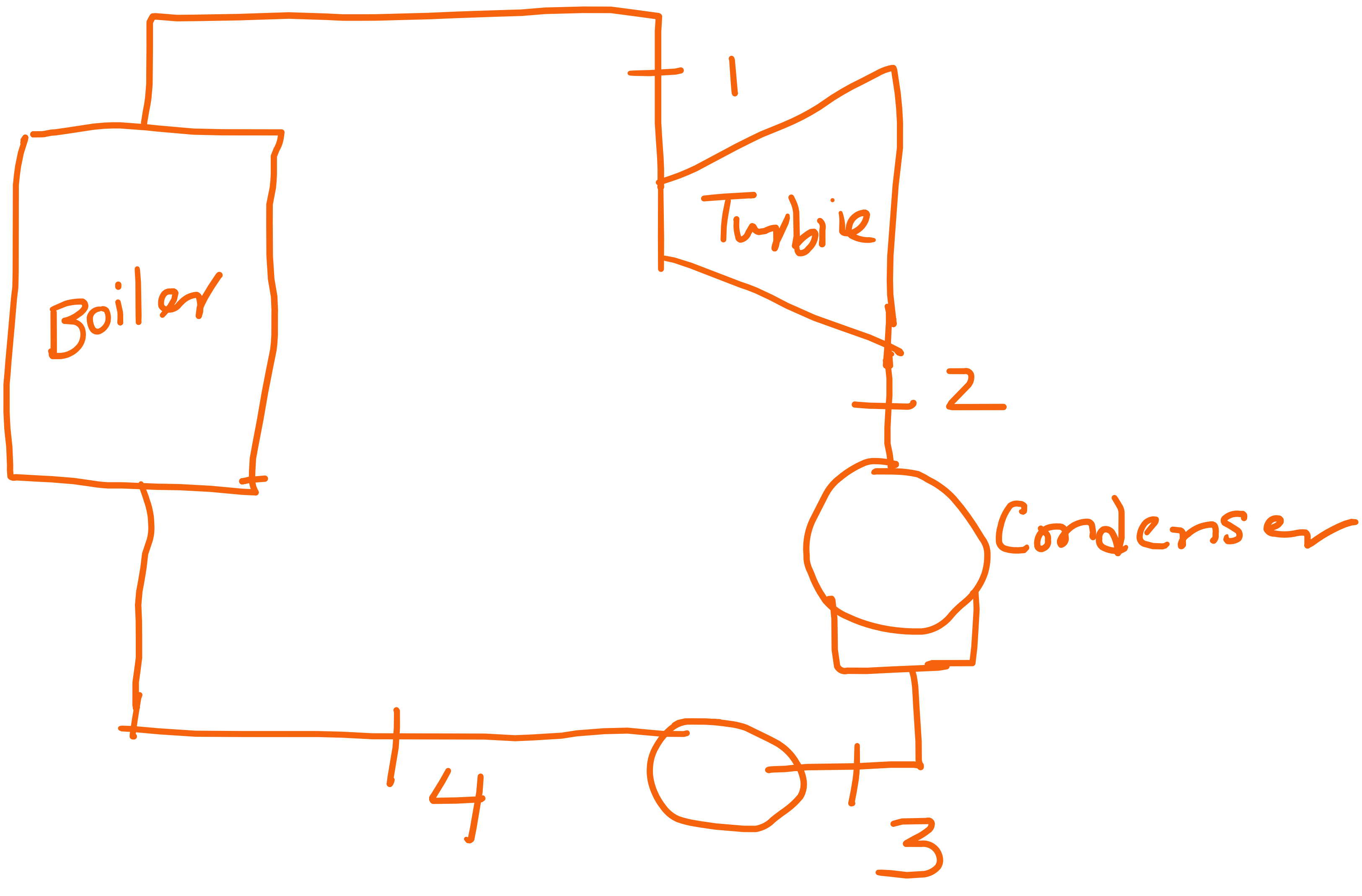
Steam Power Generation Unit



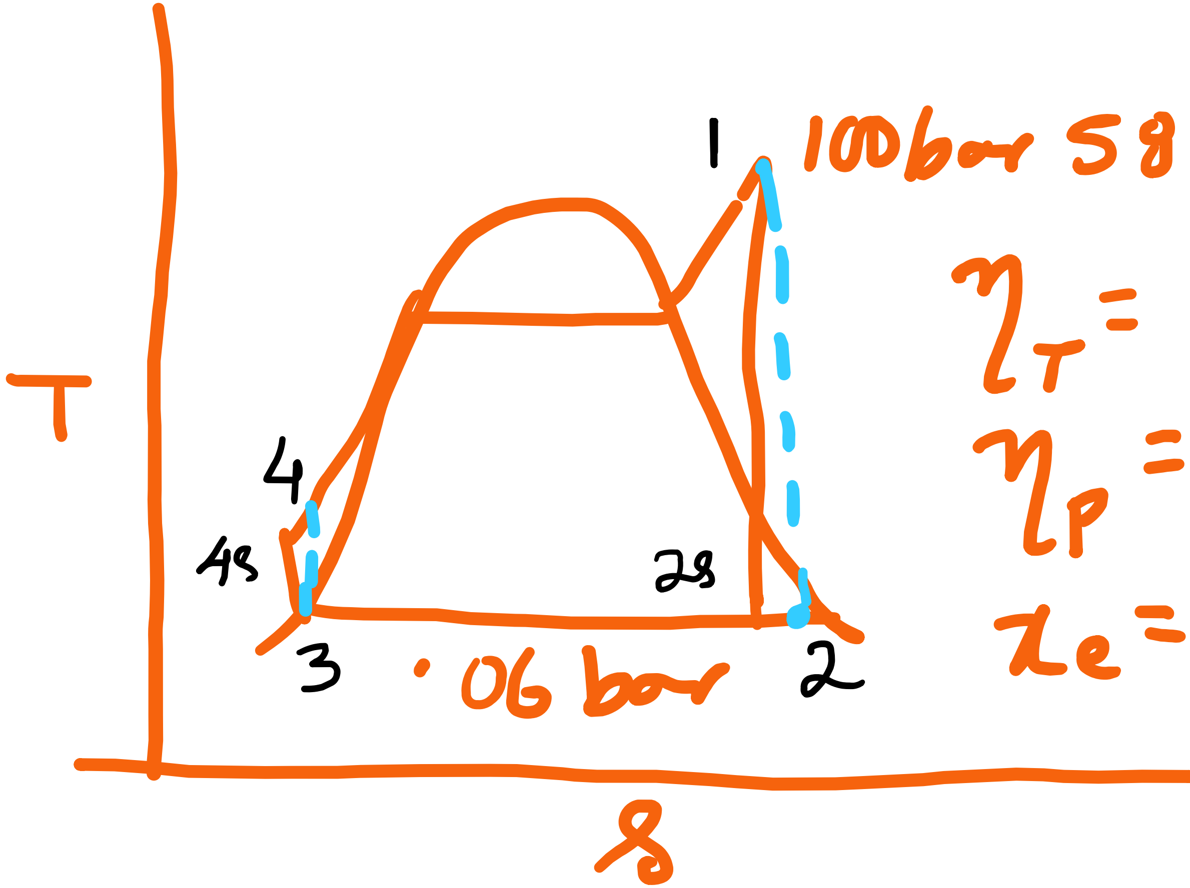
- Heat rate: Heat input per unit work output
- Back work ratio: Ratio of pump work to turbine work
- Pump work = $\int v\,dp \approx v\Delta p$

Example

- Steam enters the turbine of a simple vapor power plant with a pressure of 10 MPa and temperature T, and expands adiabatically to 6 kPa. The isentropic turbine efficiency is 85%. Saturated liquid exits the condenser at 6 kPa and the isentropic pump efficiency is 82%. (a) For T = 580°C, determine the turbine exit quality and the cycle thermal efficiency. (b) Plot the quantities of part (a) versus T ranging from 580 to 700°C.



Boiler
Turbine
Condenser
1
2
3
4



$\eta_T = 0.95$

$\eta_P = 0.82$

$x_e = ?\ ;\ \eta = ?$

$h_1 = 3576.72$ (h at $P = 100$ bar, $T = 580°C$)

$s_1 = 6.847367$ kJ/kg K

$s_{2s} = s_1$

$$h_{2s} = 2108.339\ \text{kJ/kg} \left[\begin{matrix} P = .06\text{bar} \\ s = 6.847367 \end{matrix}\right]$$

$$\eta_T = \frac{h_1 - h_2}{h_1 - h_{2s}} = \frac{3576.72 - h_2}{3576.72 - 2108.339}$$

$$h_2 = 2328.6\ \text{kJ/kg}$$

$x_e = x(h_2, p_2)$ get x correspond. to h_2 and p_2

$= 0.9014$ (Xsteam)

$$h_2 = h_f + x\,h_{fg} = 151.49 + x\left[2566.67 - 151.49\right]$$

$\Rightarrow x_e = 0.9014$ (Tables)

Pressure rise in pump = .06 bar to 100 bar

Pump work $= \int v\,dp$

$= v\Delta p$

$v_f = 0.001006$ at 0.06 bar

$v \approx$ constant since liquid is incomp.

$h_{4s} - h_3 = v\Delta p \qquad h_3 = 151.49\ \text{kJ/kg}$

$$h_{4s} = 151.49 + \frac{.001006\,[100 - .06] \times 10^5}{1000}$$

$= 161.54\ \text{kJ/kg}$

$$\eta_p = 0.82 = \frac{h_{4s} - h_1}{h_4 - h_1}$$

$$0.82 = \frac{161.54 - 151.49}{h_4 - 151.49} \Rightarrow h_4 = 163.75 \text{ kJ/kg}$$

$$W_p = 163.75 - 151.49 = 12.26 \text{ kJ/kg}$$

$$W_T = h_1 - h_2 = 1248.12 \text{ kJ/kg}$$

$$W_{net} = 1248.12 - 12.26 = 1235.86 \text{ kJ/kg}$$

$$\eta = \frac{W_{net}}{Q_{in}} = \frac{1235.86}{h_1 - h_4} = \frac{1235.86}{3576.72 - 163.75}$$

$$= 0.362$$

$W_T = 1248.12 \text{ kJ/kg}$

Flow through system $= \dot{m}$ kg/s

Total power $= 1248.12\,\dot{m}$ kJ/s

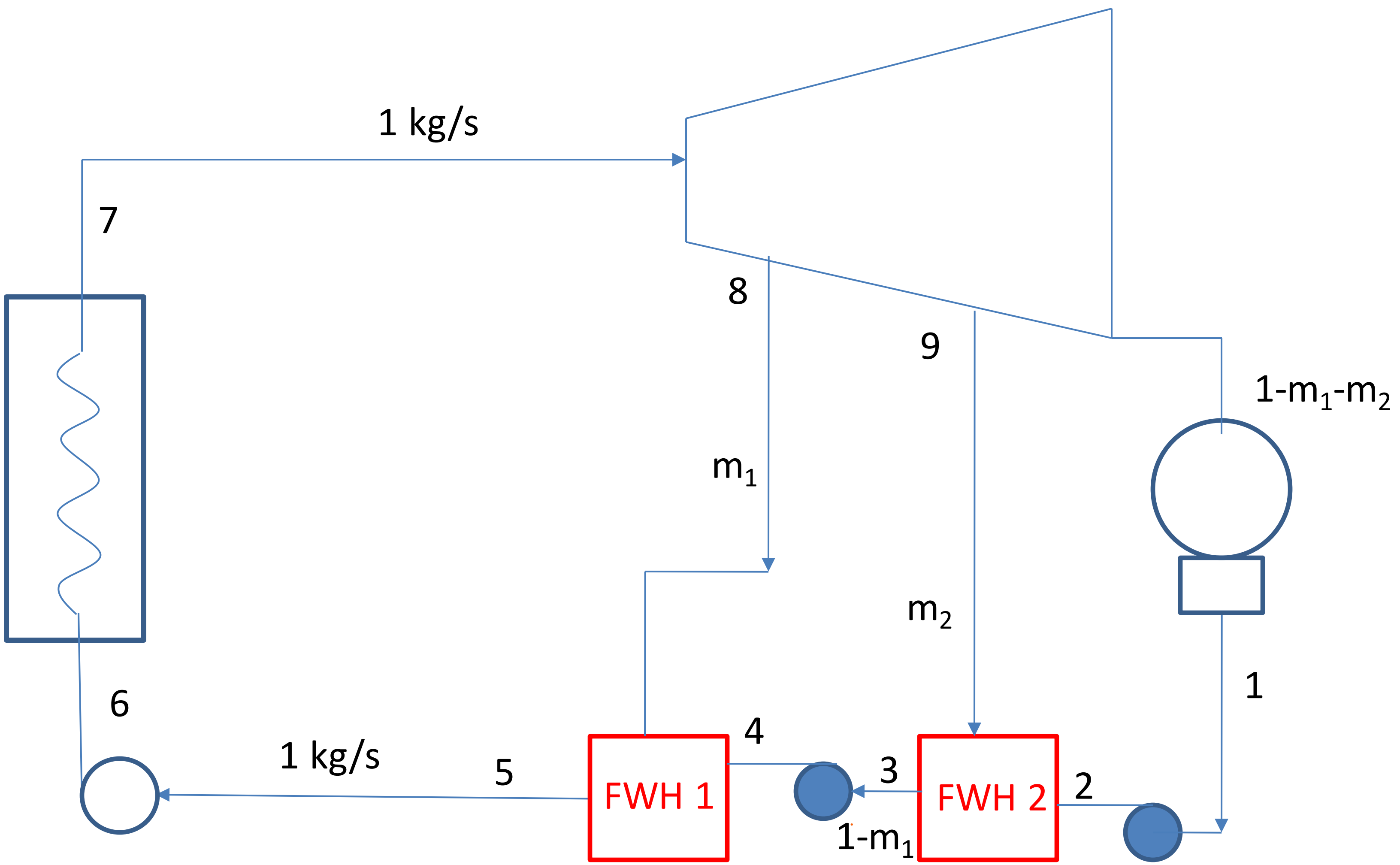
$\dot{m} = 1$ kg/s $\quad \underline{W_T = 1.25 \text{ MW}}$

Kinetic energy and potential energy changes are much smaller than enthalpy changes and their influence is ignored.

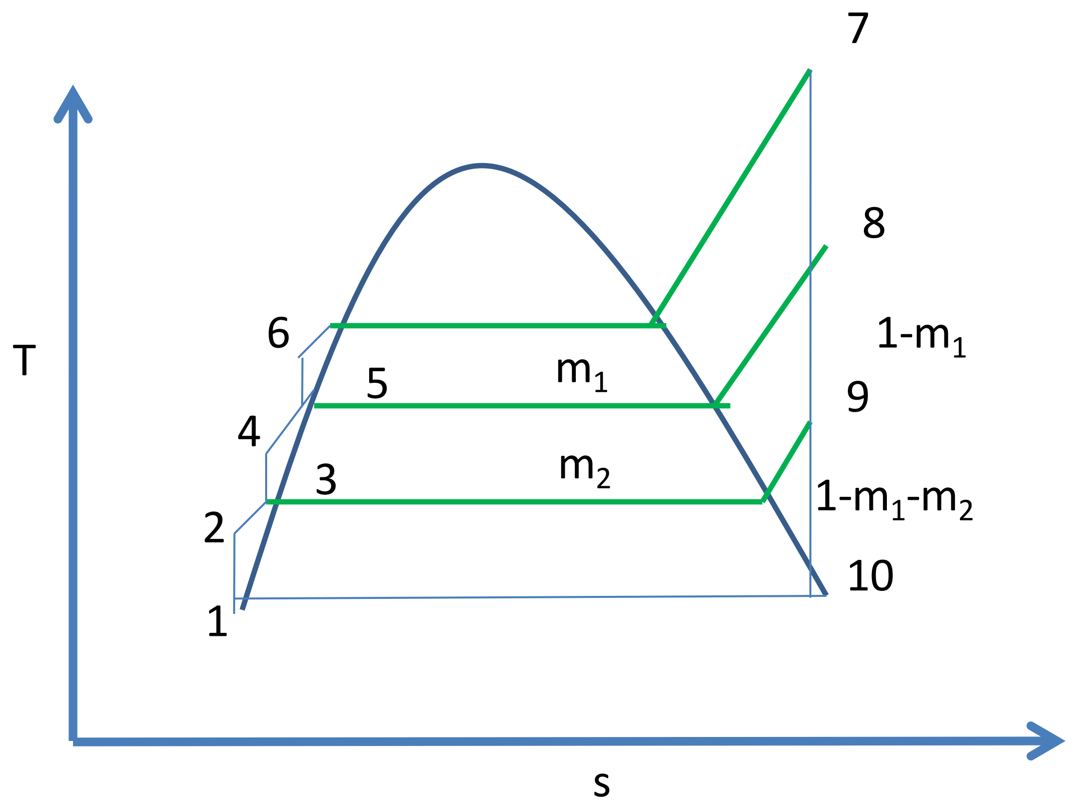
Regeneration

- So far upper temperature modifications have been considered
- Efficiency improvements by heating in the single phase liquid zone are also possible. Cannot use combustion gases since typically they must exit from the stack at about 120^0C or so – buoyancy and atmospheric moisture absorption and acid rain creation view point
- Use internal heat itself for the purpose

Open Feedwater Heater(Water and Steam Mix with each other)



Open Feedwater Heater



Feed water heater energy balance

- Perform energy balance to obtain flow rates that are bled from the turbine.
- Energy balance for the first feedwater heater:

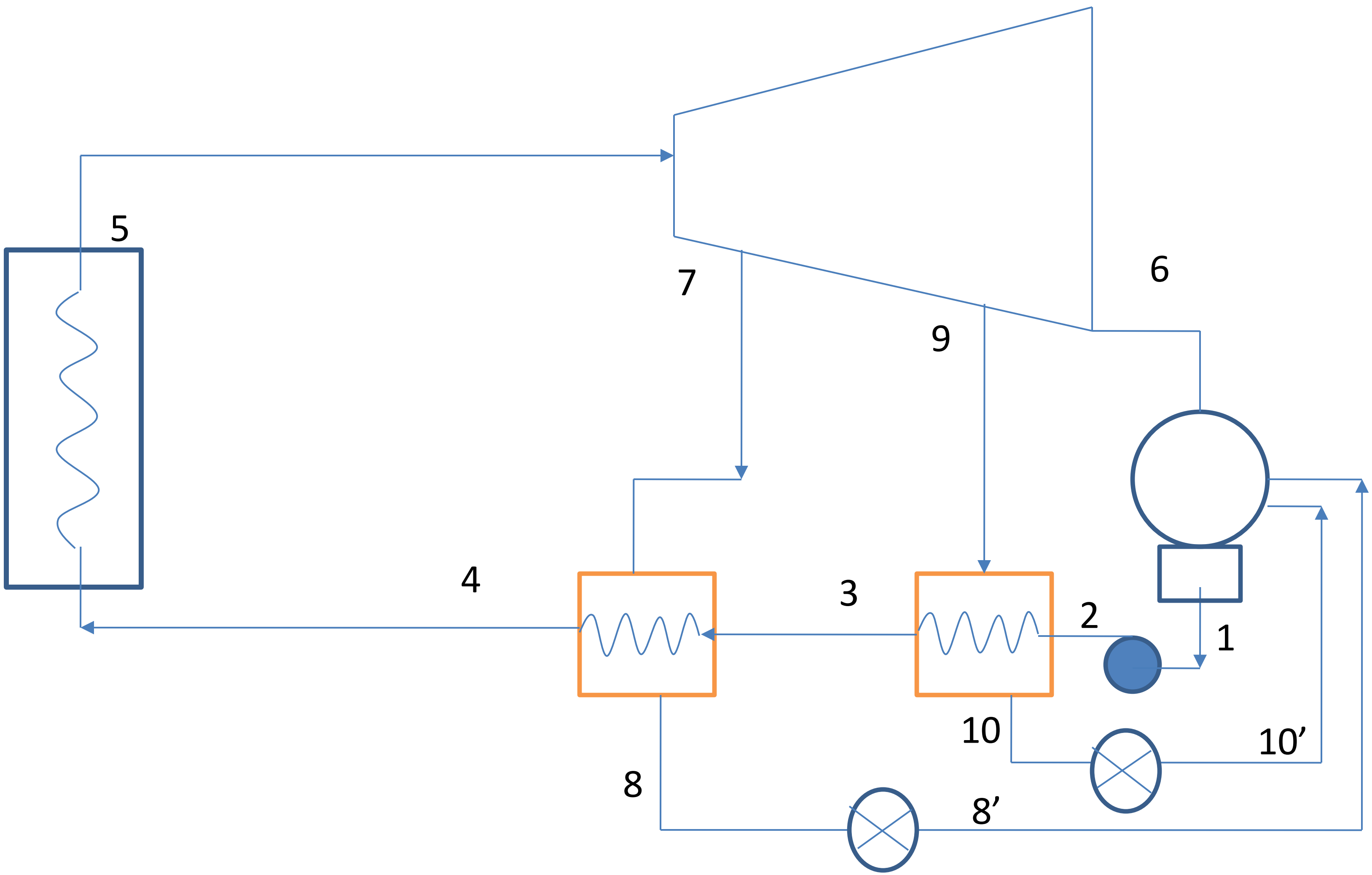
$$m_1 h_8 + (1 - m_1) h_4 = h_5$$

- Energy balance for second feedwater heater:

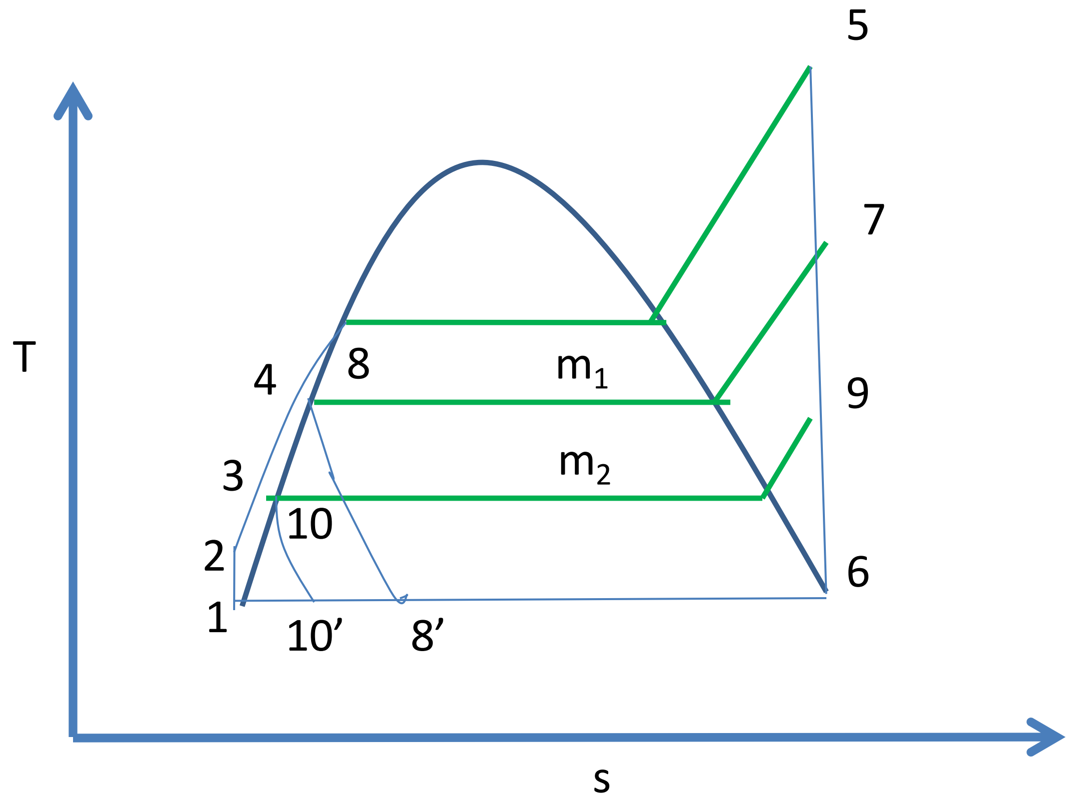
$$m_2 h_9 + (1 - m_1 - m_2) h_2 = (1 - m_1) h_3$$

- First solve for m_1 and then for m_2

Closed Feedwater Heater



Regenerative Heating



Closed Feed water heater Analysis

- Perform energy balance to obtain flow rates that are bled from the turbine.
- Energy balance for the two feedwater heaters is used to obtain the bleed mass flow rates:

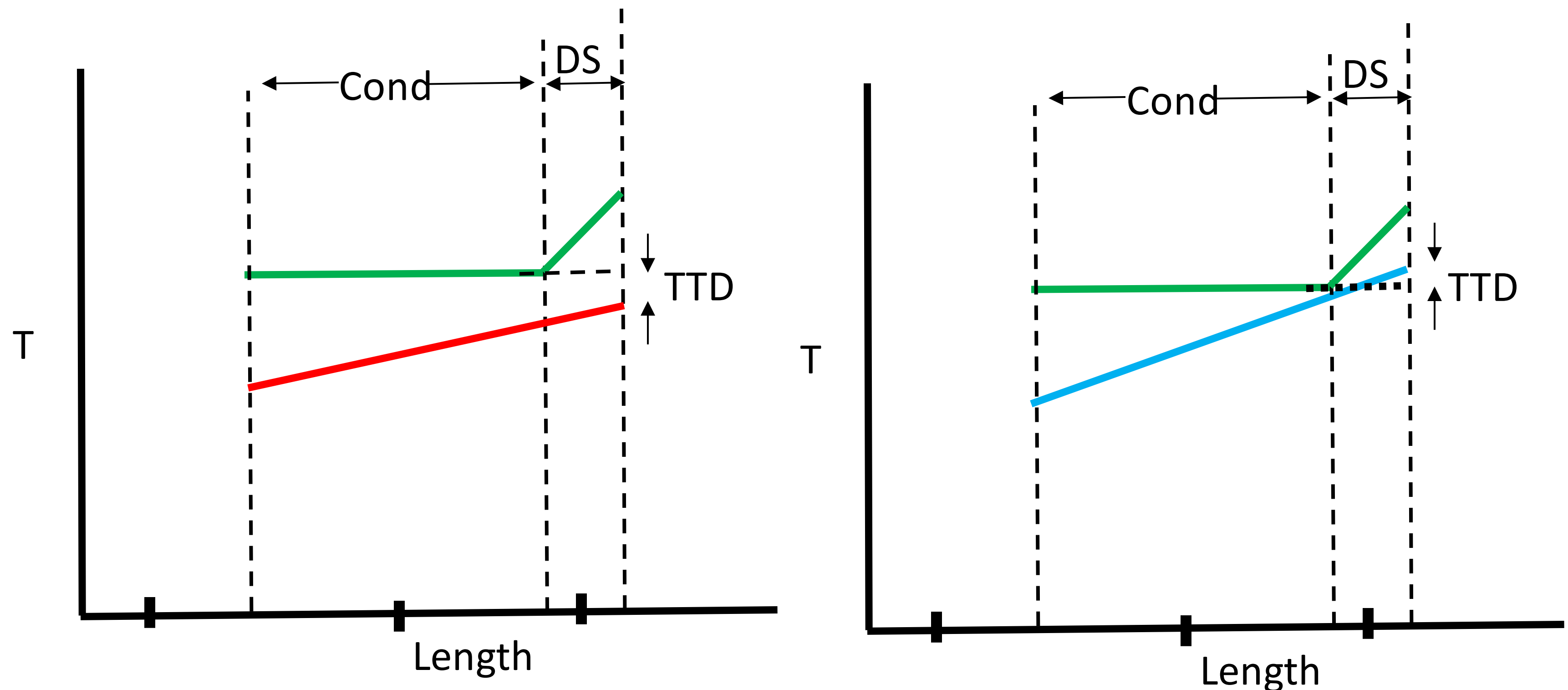
$$m_1 h_7 - m_1 h_8 = h_4 - h_3 ; m_2 h_9 - m_2 h_{10} = h_3 - h_2$$

- Heat rejected from condenser can be obtained after m_1 and m_2 are calculated:

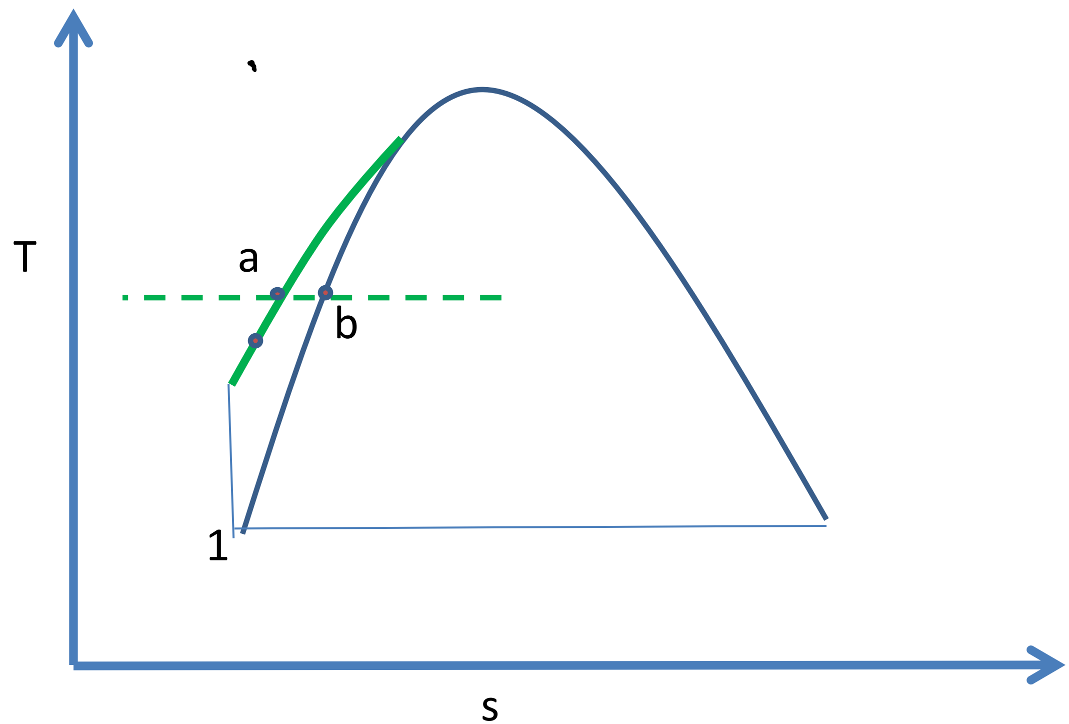
$$(1 - m_1 - m_2)(h_6 - h_1) + m_1 (h_8 - h_1) + m_2 (h_{10} - h_1)$$

- FWH temperature profile. A DeSuperheat region and a condensation region for the extraction(Green). Two possibilities for Feed water(Blue/Red). Terminal Temperature Difference(TTD is specified.)

$$TTD=T_{sat(extraction)}-T_{exit(feedwater)}$$



- Enthalpy at point ‘a’ at which temperature ‘Ta’ is known is required. Point ‘b’ is on the saturation line corresponding to Ta. Pressure at ‘a’ is Pa and at ‘b’ is Psat corresponding to Ta. Ta=Tb;



- $h_a(T_a, p_a) = u_a\,(T_a) + p_a v_a(T_a)$
- For an incompressible liquid the internal energy and specific volume are functions of temperature only
- Saturation enthalpy at Ta is therefore

$$h_b(T_a, p_b) = h_b\,(T_{sat})\; = u_a\,(T_a) + p_b v_a(T_a)$$

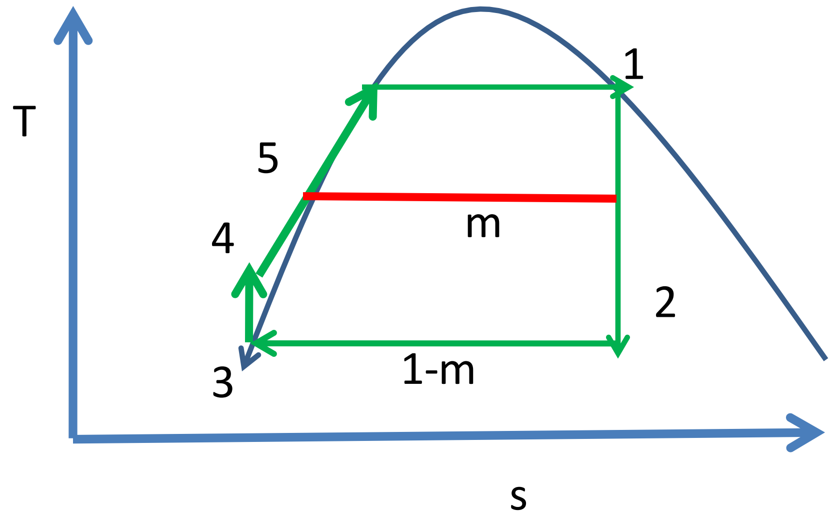
- Therefore:

$$h_a(T_a, p_a) = h_{sat}(T_a) - p_{sat} v_a + p_a v_a$$

Closed Feed water heater Analysis

- Work per unit mass decreases due to extractions. Lesser steam through later stages of turbine and condenser
- Larger the no. of heaters larger is the gain in efficiency. Ideally should have infinitely large number of heaters
- Balance between economics and thermo

- Definition of mean temperature of mean addition is slightly altered due to regeneration



Mean Temperature of Heat Addition

- Heat is added between state 1 and state 5
- The temperature at which heat is being added is variable but define a mean temperature at which heat is added as:

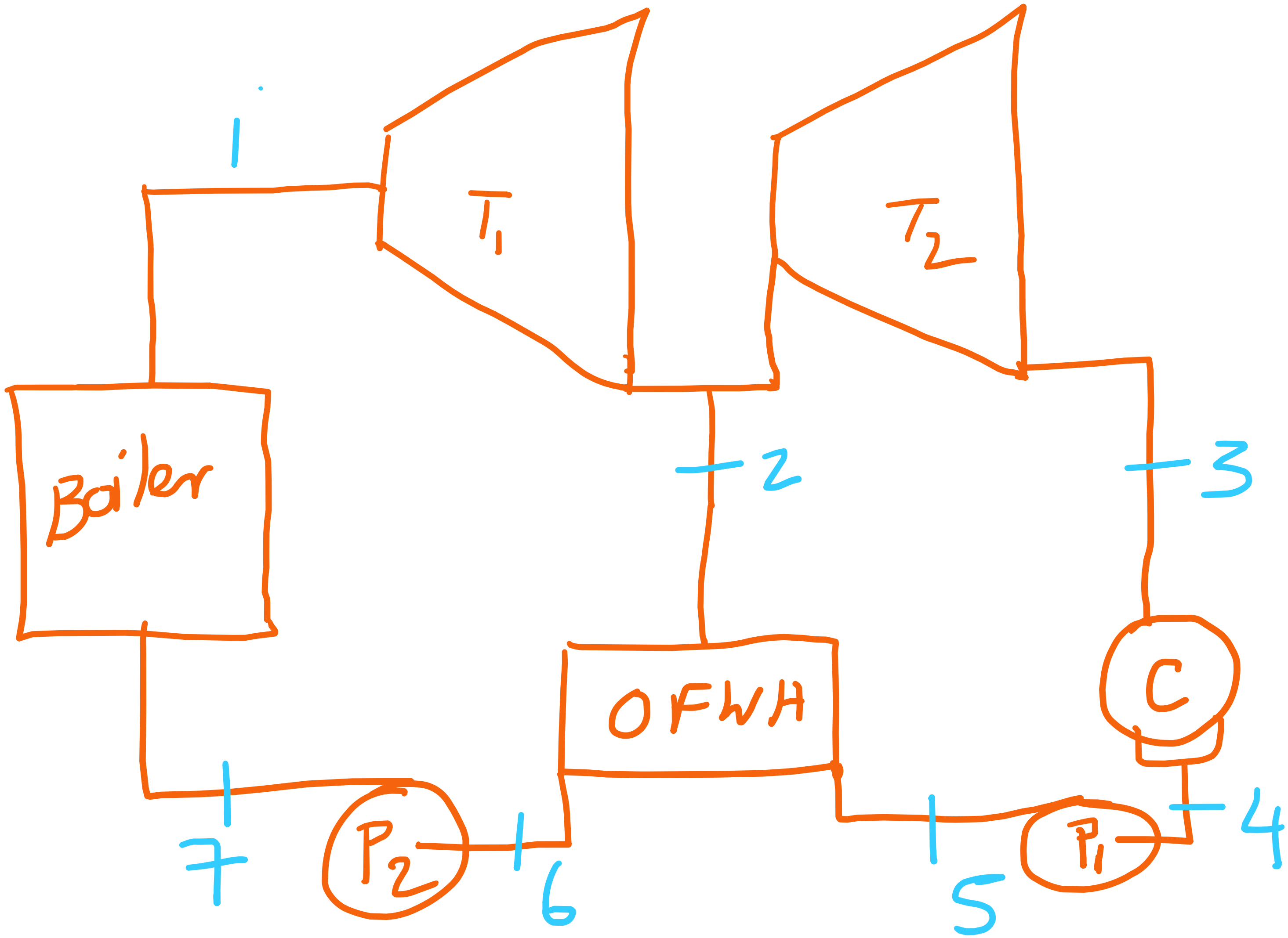
$$h_1 - h_5 = T_{m,add}(s_1 - s_5)$$

- Similarly a mean temp. of heat rejection can be defined. Using these, cycle efficiency becomes:

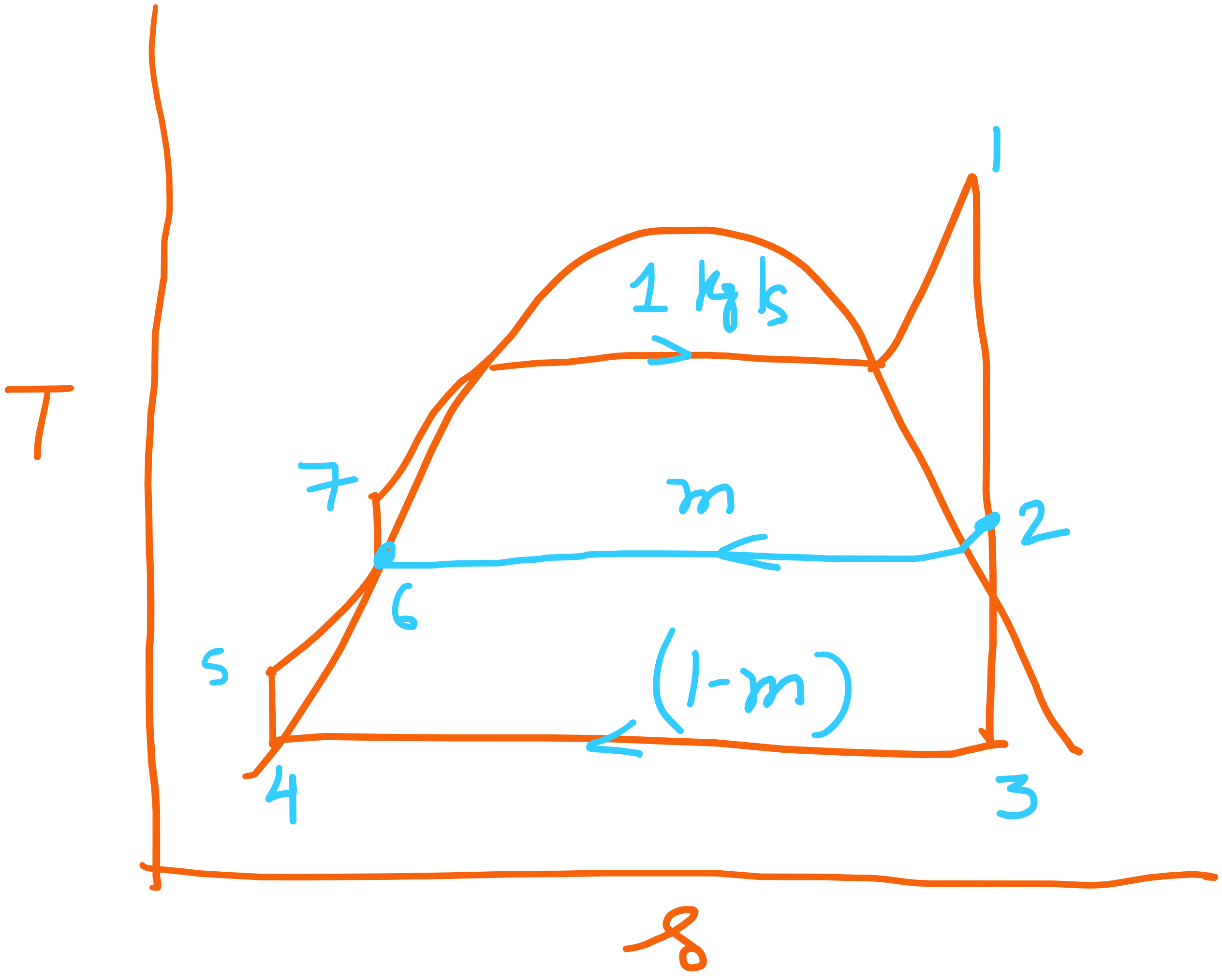
$$\eta = 1 - \frac{Q_{rej}}{Q_{add}} = 1 - \frac{(1-m)(s_2 - s_4)T_{m,rej}}{(s_1 - s_5)Tm, add}$$

- When using finite no. of heaters, it is required to know where to tap steam
- There are 2 opposing effects
  - Better efficiency due to more internal heat regeneration
  - Irreversibility due to heat transfer across large ΔT
- Need more involved thermodynamic analysis to obtain a balance.
- Calculations show that as no. of FWHs increases the efficiency increases first and then levels off, so some optimum is chosen based on engineering considerations(including cost).

- A power plant operates on a regenerative vapor power cycle with one open feedwater heater. Steam enters the first turbine stage at 12 MPa, 520C and expands to 1 MPa, where some of the steam is extracted and diverted to the open feedwater heater operating at 1 MPa. The remaining steam expands through the second turbine stage to the condenser pressure of 6 kPa. Saturated liquid exits the open feedwater heater at 1 MPa. For isentropic processes in the turbines and pumps, determine for the cycle (a) the thermal efficiency and (b) the mass flow rate into the first turbine stage, in kg/h, for a net power output of 330 MW.



1
T1
T2
Boiler
2
3
OFWH
C
P2
P1
7
6
5
4


T
s
1 kg/s
m
(1-m)
1
2
3
4
5
6
7

State 1: 520°C 12 MPa (120 bar)

$h = 3403.4$ kJ/kg

$s = 6.5584$ kJ/kgK

State 2: 10 bar, $s = 6.5584$

$h_2 = 2765.1$ kJ/kg

$(x_2 = 0.994)$
fig needs modification

State 3: 6 kPa, $s = 6.5584$

$h_3 = 2018.96$ kJ/kg

State 4: Saturated liquid at 6 kPa

$h_4 = 151.5$ kJ/kg

State 5: Pump raises pressure to P_2 (10 bar)

$$h = h_4 + v_4 \Delta P$$

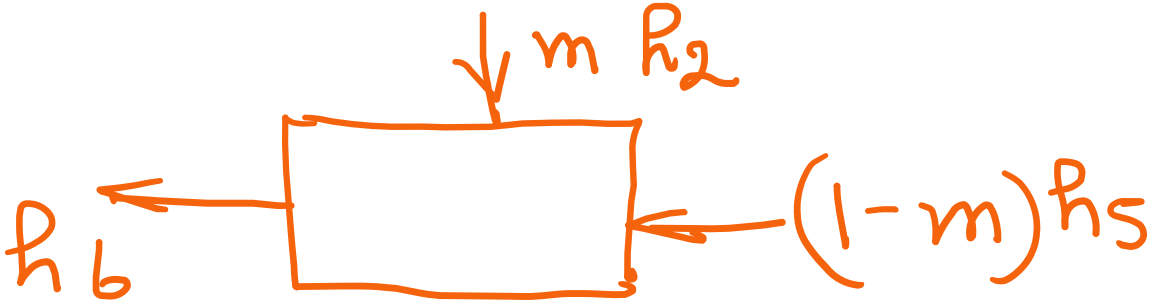
$$= 151.5 + .001006(10 - .06)\frac{10^5}{10^3}$$

$$= 152.5 \text{ kJ/kg}$$

Exit state of FWH is assumed saturated liquid if nothing is mentioned.

State 6: $h_6 = 762.68$ kJ/kg (sat. liq. at 10 bar)

All states available for energy balance over FWH



m = mass rate extracted from turbine

$1-m$ mass rate through second turbine stage and condenser

$$m[2765.1] + (1-m)152.5 = 1(153.5)$$

$$\Rightarrow m = 0.234 \text{ kg/s}$$

m is always < 1 since total flow rate $= 1$

State 7: $h_7 = h_6 + v_6 \Delta P$

$$= 762.68 + [.001127][120-19]\frac{10^5}{10^3}$$

$h_7 = 775.2 \text{ kJ/kg}$

All states are known. Compute η

$$\eta = \frac{W_{out\,net}}{Q_{in}}$$

$$= \frac{(h_1 - h_2) + (1-m)(h_2 - h_3) - (1-m)(h_5 - h_4) - (h_7 - h_6)}{(h_1 - h_7)}$$

$W_{in} = (1 - .234)(152.5 - 151.5) + (775.2 - 762.7)$

$= 13.27$

$W_{out} = (3403.4 - 2765.1) + (1 - .234)(2765.1 - 2018.96)$

$W_{out} = 1209.8 \text{ kJ/kg}$

$Q_{in} = h_4 - h_7 = 3403.4 - 775.2 = 2628.3$

$$\eta = \frac{1209.8 - 13.27}{2628.3} = 0.455.$$

$\text{Net Power} = 330 \text{ MW} = m\left(1196.5 \text{ kJ/kg}\right)$

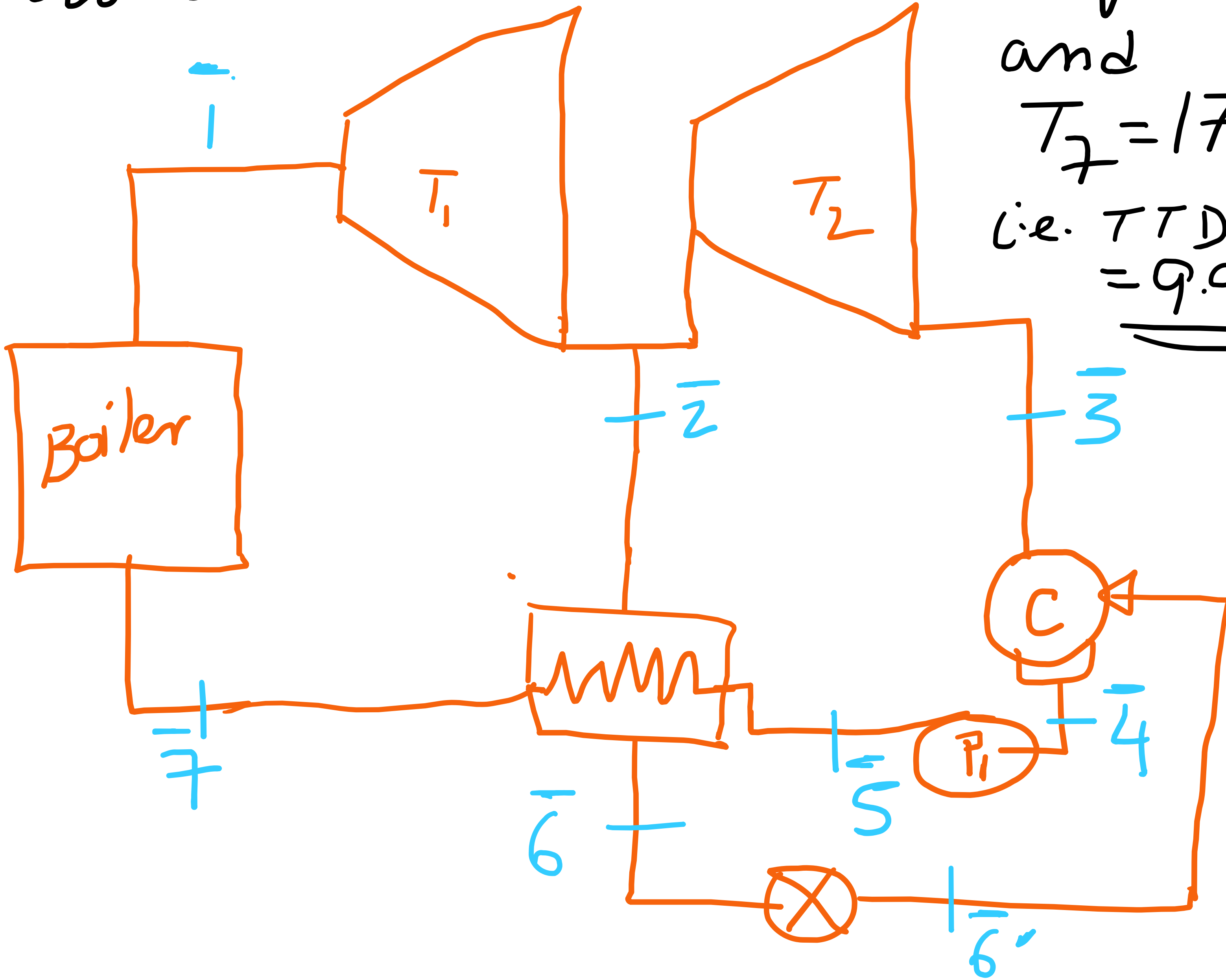
$m = 275.8 \text{ kg/s}$

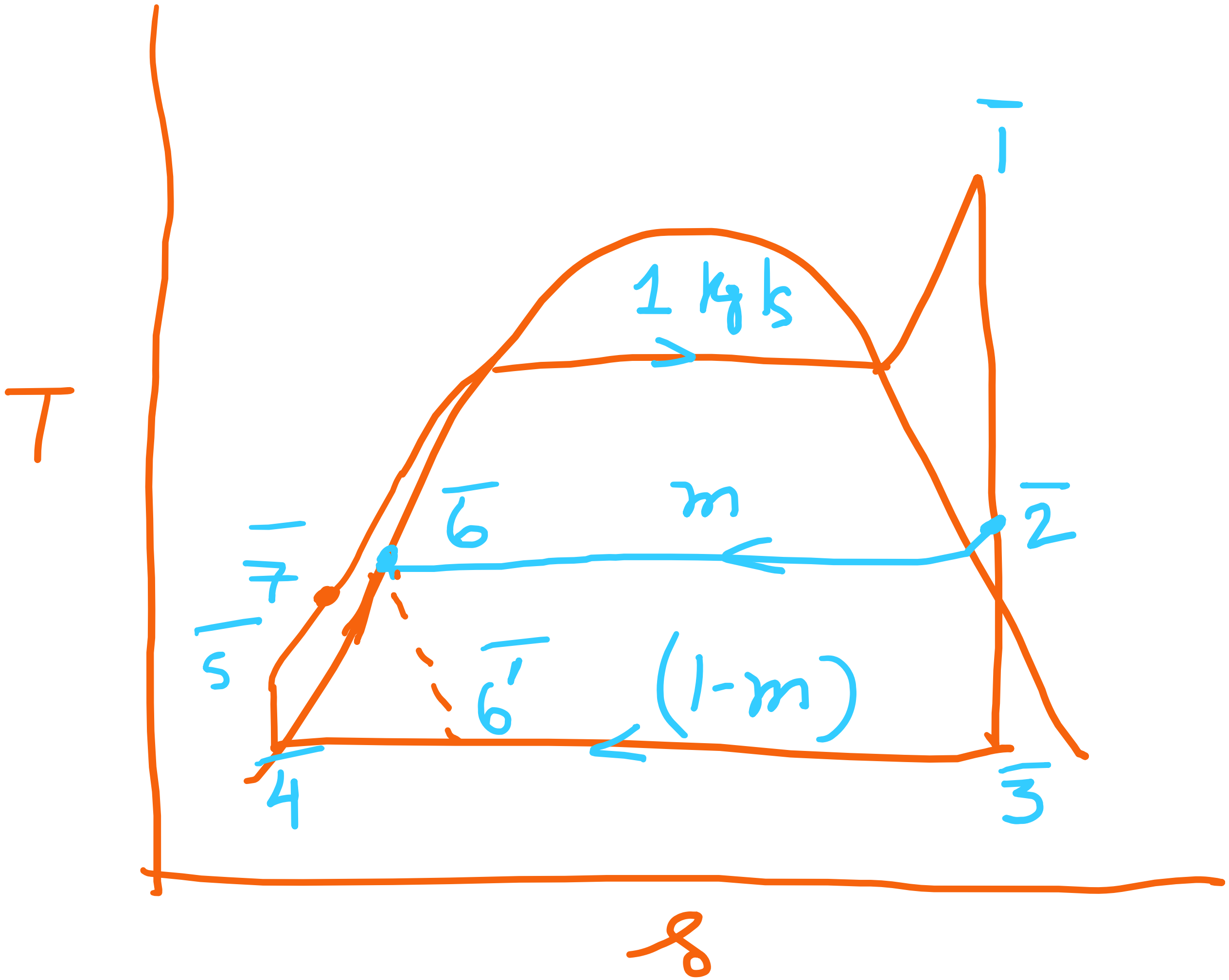
Now assume FWH is closed type

and

$T_7 = 170°C$

i.e. TTD $= 9.9°C$





T
s
1
2
3
4
5
6
6'
7
1 kg/s
m
(1-m)

State points $\bar{1}, \bar{2}, \bar{3}, \bar{4}, \bar{6}$ are all same as those for the previous example

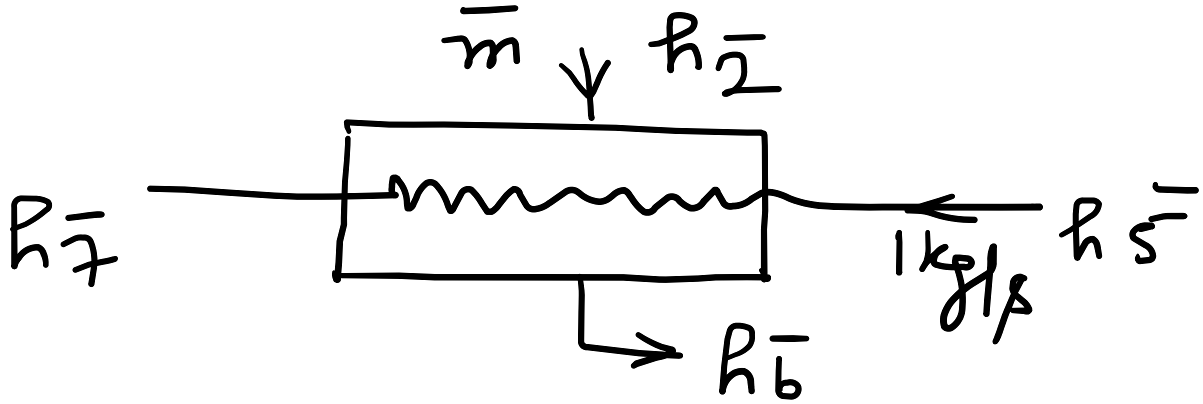
Water at exit of condenser is pumped to boiler pressure

$$h_{\bar{5}} = h_{\bar{4}} + v_{\bar{4}} \Delta p$$

$$= 151.5 + .001006(120 - .06)\frac{\times 10^5}{10^3}$$

$$= 163.6 \text{ kJ/kg}$$

Perform energy balance across FWH



$$1(h_5) + m\, h_2 = 1\, h_7 + m\, h_6$$

$T_7 = 170°c$ given

$$h_7 = h_{sat\,170} + v_{sat\,170}\left(P_7 - P_{sat\,170}\right)$$

$$= 719.21 + .00111\left[120 - 7.92\right] 10^5/10^3$$

$$= 731.65\ kJ/Kg$$

$$1(163.6) + m(2765.1) = 1(731.65) + m(762.68)$$

$$m = 0.284 \text{ kg/s}$$

Liquid leaving FWH is throttled to pressure in condenser and added in condenser

$$h_{\bar{6}} = h_{\bar{6}'}$$

All states are now known.

$$Q_{rej} = (1-\bar{m})\left[h_{\bar{3}} - h_{\bar{4}}\right] + \bar{m}\left[h_{\bar{6}'} - h_{\bar{4}}\right]$$

$$= .716\left[2018.96 - 151.5\right] + \bar{m}\left[762.68 - 151.5\right]$$

$$= 1510.68 \text{ kJ/kg/kg/s}$$

$$Q_{in} = h_1 - h_7 = 3403.4 - 731.65$$

$$= 2671.8 \text{ kJ/kg} \mid \text{kg/s}$$

$$W_{out} = (h_1 - h_2) + (1-m)(h_2 - h_3)$$

$$= (3403.4 - 2765.1) + (.716)(2765.1 - 2018.91)$$

$$= 1172.53 \text{ kJ/kg} \mid \text{kg/s}$$

$$W_{in} = 12.1 \text{ kJ/kg} \mid \text{kg/s}$$

$$W_{net} = W_T - W_p = 1172.53 - 12.1 = 1160.43 \text{ kJ/kg}$$

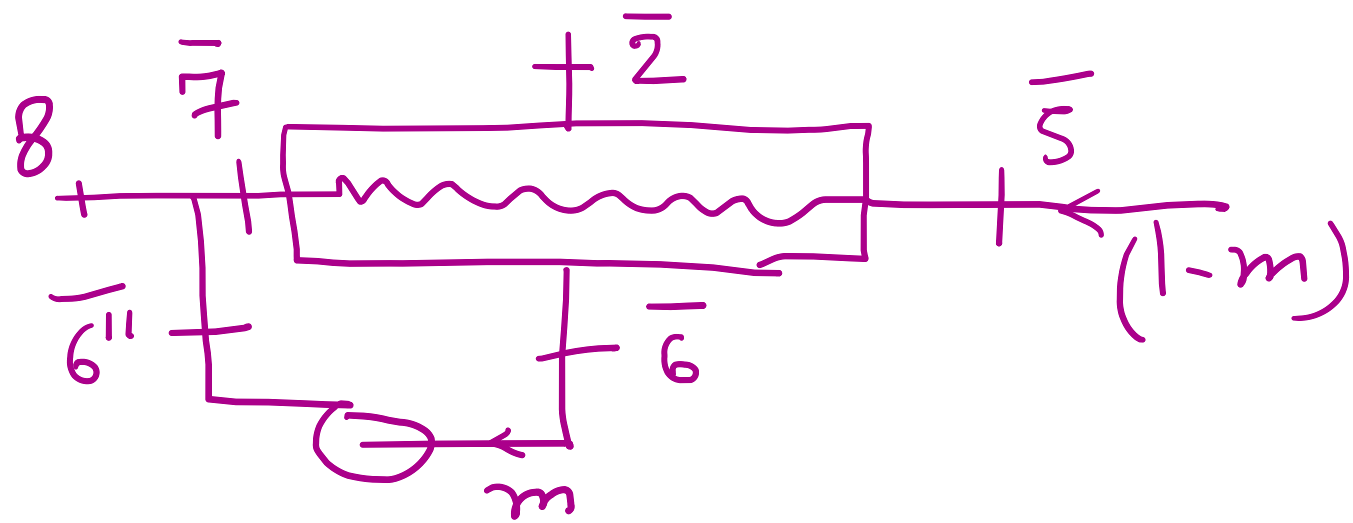
$$W_{net} = Q_{in} - Q_{rej} = 2671.8 - 1510.68 = 1161.1 \text{ kJ/kg.}$$

$$\eta = \frac{W_{net}}{Q_{in}} = \frac{1160.43}{2671.8} = 0.43$$

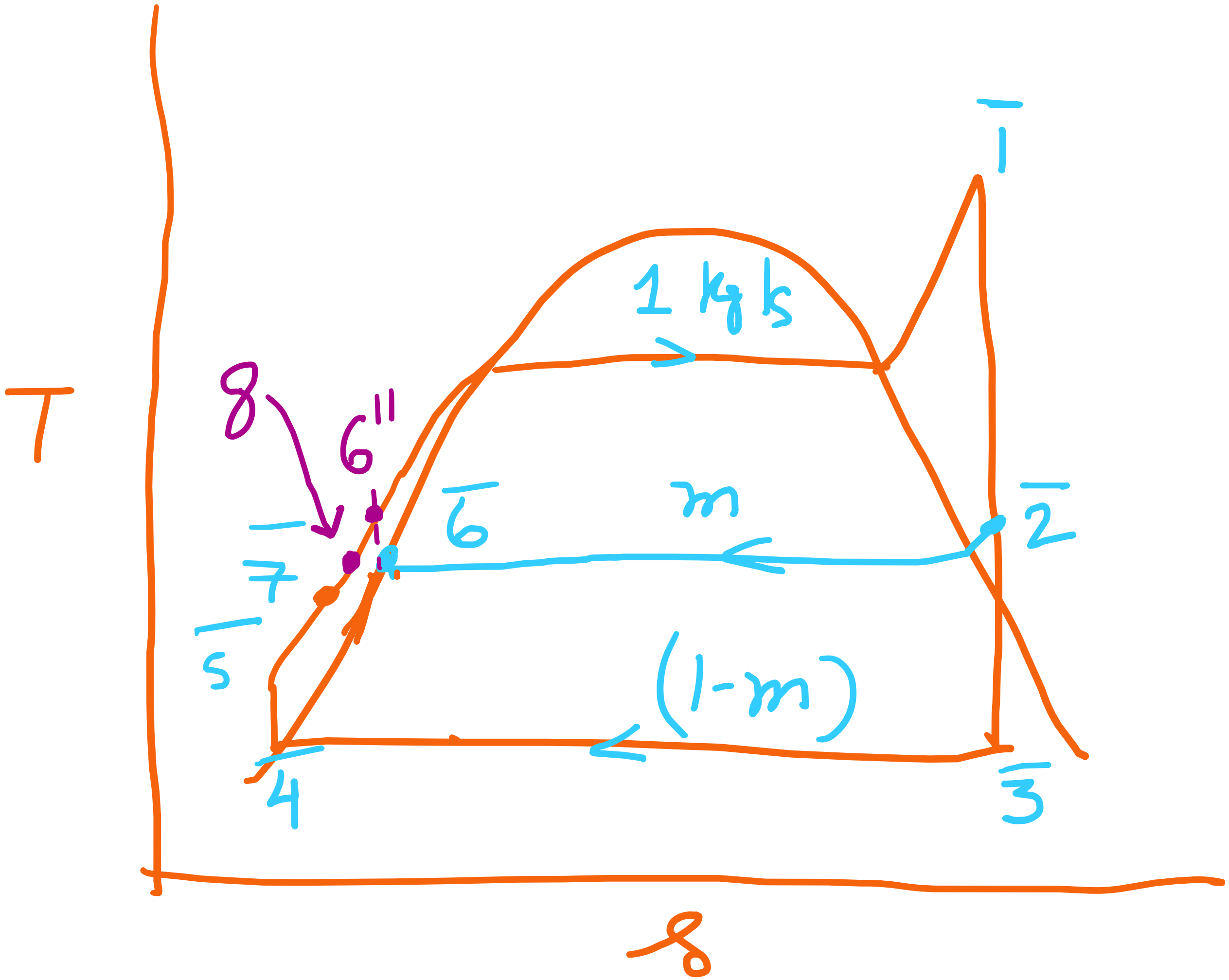
$$W_{net} = 330 \text{ MW} = m\,(1160.43)$$

$$\rightarrow m = 284.4 \text{ kg/s}$$

- Now instead of reducing pressure in valve and releasing heat in condenser, assume a pump is used to raise the pressure and mix the hot water streams at heater exit.



All States remain same except 6'' & 8



$$h_{6''} = h_6 + v_6 \Delta p$$

$$= 762.68 + 0.00127(120-10)\frac{10^5}{10^3}$$

$$= 776.65$$

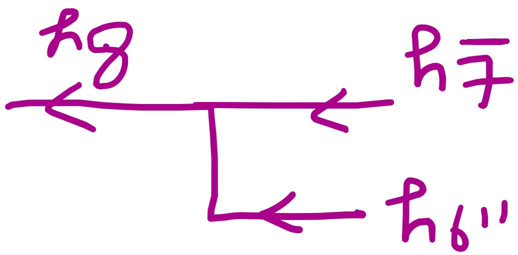
Energy balance for FWH

$$m[h_2 - h_6] = (1-m)[h_7 - h_5]$$

$$m[2765.1 - 762.68] = (1-m)(731.65 - 163.6)$$

$$m = 0.221$$

Now do an energy balance over mixing zone:



$$(1-m)h_7 + m\,h_{6''} = h_8$$

$$h_8 = 731.65(1-.221) + .221(776.65)$$

$$= 741.6 \text{ kJ/kg}$$

$$Q_{rej} = (1-m)(h_3 - h_4)$$

$$= .779(2018.96 - 151.5) = 1454.75 \text{ kJ/kg}$$

$$Q_{in} = h_1 - h_8 = 3403.4 - 741.6$$

$$= 2661.8$$

$$\eta = 1 - \frac{Q_{rej}}{Q_{in}} = 0.454$$

Comparison with simple cycle without regenerative heating.

$$Q_{in} = h_1 - h_5 = 3403.4 - 163.6 = 3239.8$$

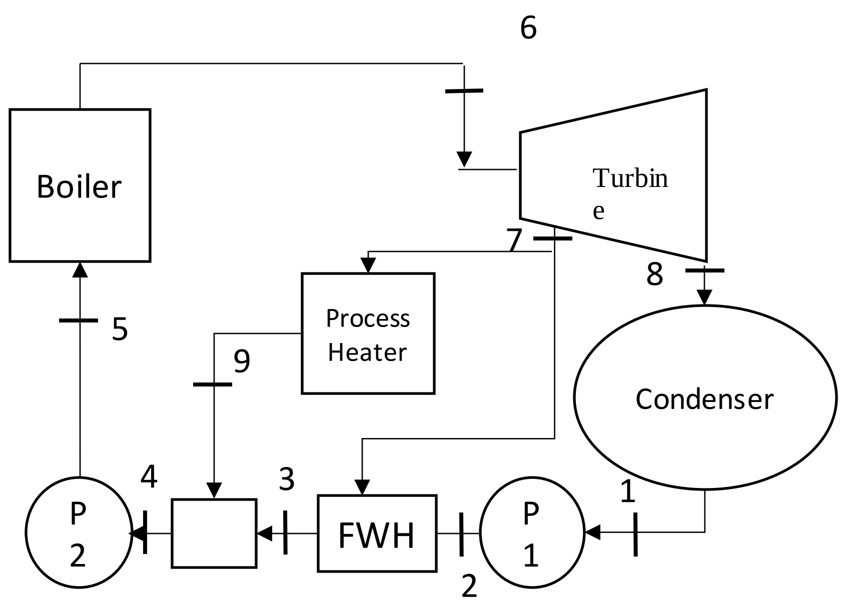
$$Q_{rej} = h_4 - h_3 = 2018.96 - 151.5 = 1867.46$$

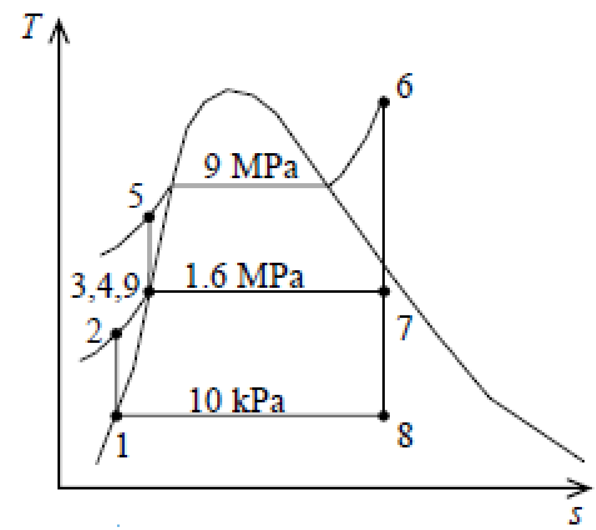
$$\eta = 1 - \frac{1867.46}{3239.8} = 0.42$$

Example 1

Consider a cogeneration power plant modified with regeneration. Steam enters the turbine at 9 MPa and 400°C and expands to a pressure of 1.6 MPa. At this pressure, 35 percent of the steam is extracted from the turbine, and the remainder expands to 10 kPa. Part of the extracted steam is used to heat the feedwater in an open feedwater heater. The rest of the extracted steam is used for process heating and leaves the process heater as a saturated liquid at 1.6 MPa. It is subsequently mixed with the feedwater leaving the feedwater heater, and the mixture is pumped to the boiler pressure.

Assuming the turbines and the pumps to be isentropic, show the cycle on a *T-s* diagram with respect to saturation lines, and determine the mass flow rate of steam through the boiler for a net power output of 25 MW.





- Steady operating conditions exist
- Kinetic and potential energy effects are negligible

From the steam tables

$h_1 = \ h_{f@10\ kPa}$ =191.81 KJ/kg

$v_1 = v_{f@10\ kPa} = 0.00101\, m^3$ /kg

$w_{P1,in} = v_1\ (P_2 - P_1) = (0.00101 m^3$ /kg $) * (1600 kPa -$
$10 kPa) \left(\frac{1\ kJ}{1\ kPa.m^3}\right)$ =1.61 KJ/kg

$h_2 = h_1 + \ w_{P1,in}$ =191.81 +1.61=193.42 KJ/kg

$v_4 = v_{f@1.6\ MPa} = 0.001159\, m^3$/kg

$$w_{P2,in} = v_4\,(P_5 - P_4) = (0.001159 m^3\ /\text{kg})\,(9000 kPa - 1600 kPa)\left(\frac{1\ kJ}{1\ kPa.m^3}\right) = 8.57\text{KJ/kg}$$

$$h_5 = h_4 + w_{P2,in} = 858.44 + 8.57 = 867.02\ \text{KJ/kg}$$

$$\left.\begin{array}{l} P_6 = 9\ MPa \\ T_6 = 400^{o}\text{C} \end{array}\right\} h_6 = 3118.8\ \text{KJ/kg}, S_6 = 6.2876\ \text{KJ/kg.K}$$

$$\left.\begin{array}{l} P_7 = 1.6 MPa \\ S_7 = S_6 \end{array}\right\} \begin{array}{l} x_7 = \frac{S_7 - S_f}{S_{fg}} = \frac{6.2876 - 2.3435}{4.0765} = 0.9675 \\ h_7 = h_f + x_7 h_{fg} = 858.44 + 0.9675 * 1934.4 = 2730\ \text{KJ/kg} \end{array}$$

$$\left.\begin{array}{l} P_8 = 10 kPa \\ S_8 = S_6 \end{array}\right\} \begin{array}{l} x_8 = \frac{S_8 - S_f}{S_{fg}} = \frac{6.2876 - 0.6492}{7.4996} = 0.7518 \\ h_8 = h_f + x_8 h_{fg} = 191.81 + 0.7518 * 2392.1 = 1990.2\ \text{KJ/kg} \end{array}$$

Then, per kg of steam flowing through the boiler, we have

$w_{T,out} = (h_6 - h_7) + (1 - y)\,(h_7 - h_8) = (3118.8 - 2730) + (1 - 0.35)\,(2730 - 1990.2) = 869.7\ \mathrm{KJ/kg}$

$w_{P,in} = w_{P2,in} + (1 - y) w_{P1,in} = 8.57 + (1 - 0.35) * 1.61 = 9.62\ \mathrm{KJ/kg}$

$w_{net} = w_{T,out} - w_{P,in} = 869.7 - 9.62 = 860.1\mathrm{KJ/kg}$

$$\dot{m} = \frac{\dot{W}_{net}}{w_{net}} = \frac{25000\ kJ/s}{860.1\ kJ/kg} = 29.1\ \mathrm{kg/s}$$

Example 2

A large city uses a Rankine steam cycle modified with one closed feedwater heater and a process heater shown below to supply nearby buildings with hot water for heating systems and electrical power. The steam flow rate into the turbine is 100 kg/s. Steam entering the turbine is extracted at 2000 kPa, state 5, for the feedwater heater. Steam entering the turbine is extracted at 700 kPa, state 6, for the process heater and leaves the process heater as a saturated liquid. The states for the boiler feedwater and the condensed steam leaving the feedwater heater are the normally assumed ideal states. Cold process water serves as the coolant for the condenser and receives the heat transferred from the condensing steam in the condenser.

The process water is further heated in the process heater. Use the data provided in the tables given below to determine

(a) the *T-s* diagram for the ideal cycle,

(*b*) the process water flow rate, in kg/s, when 5 percent of the turbine inlet mass flow rate is extracted for the process heater and the temperature rise of the process water is 40°C,and

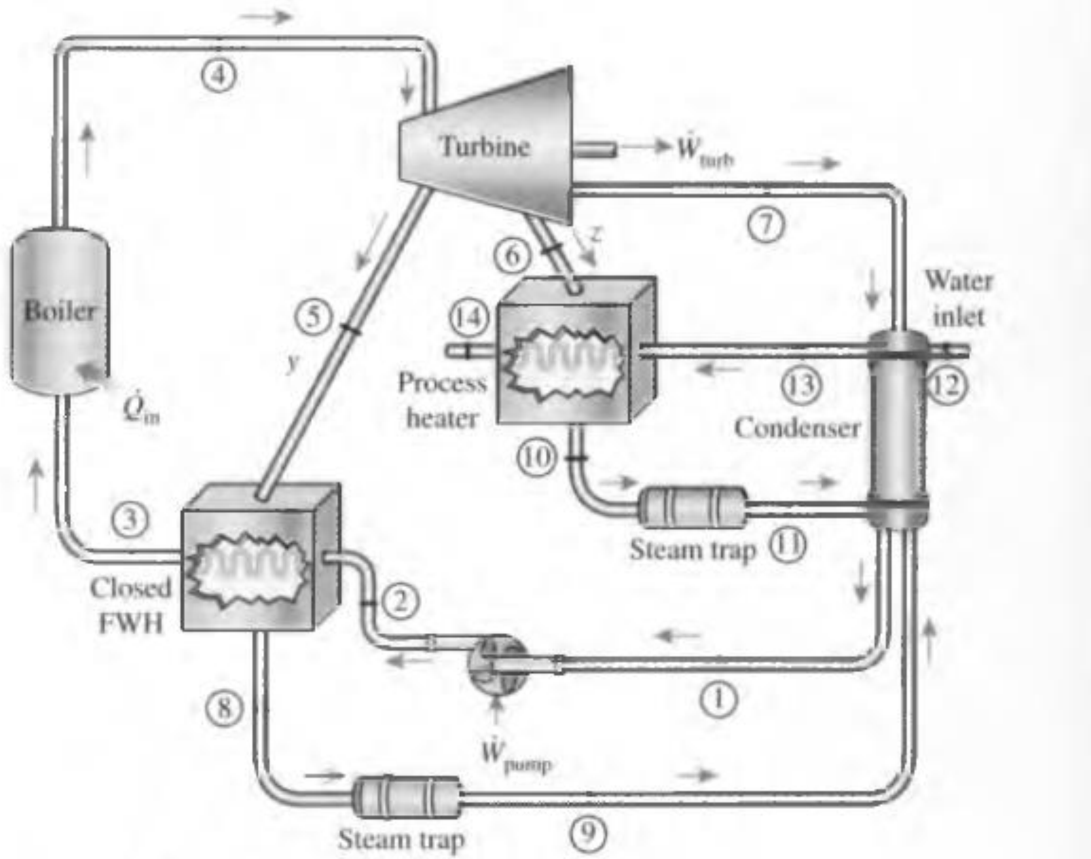
(c) the utilization efficiency of the plant

Process states and selected data

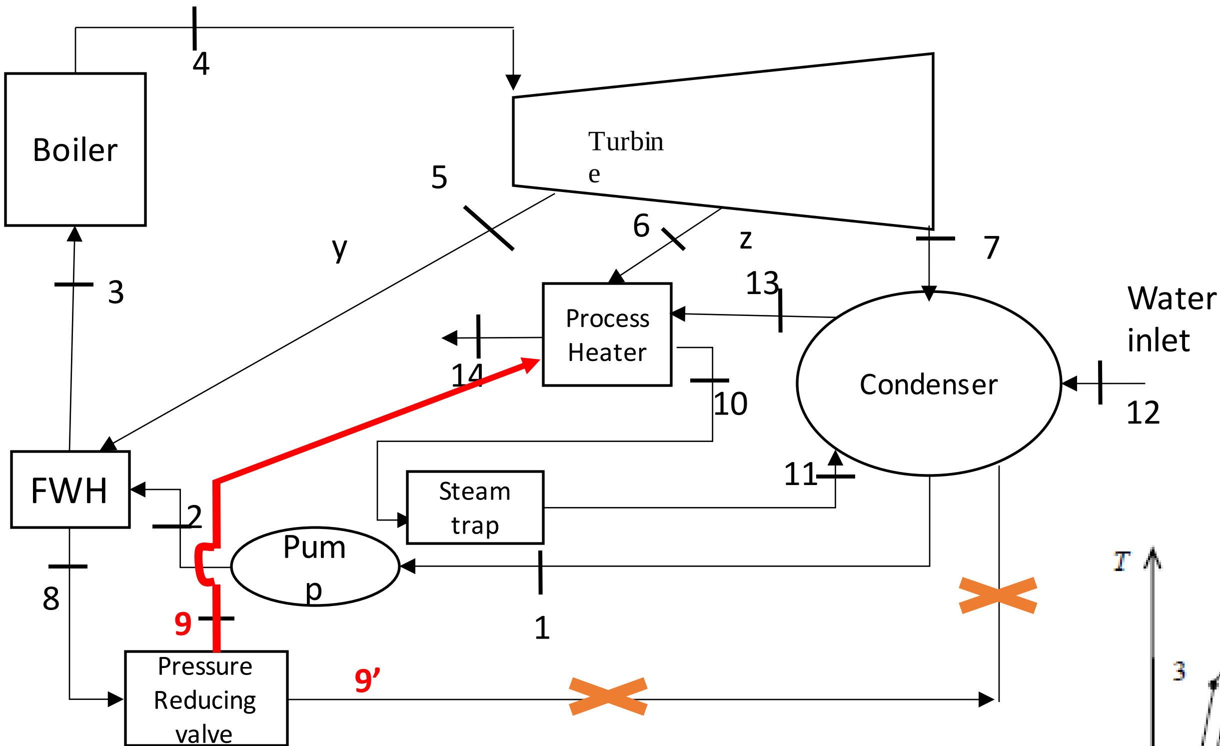
State	P, kPa	T, °C	h, kJ/kg	s, kJ/kg·K
1	10			
2	10000			
3	10000			
4	10000	500	3374	6.597
5	2000		2930	6.597
6	700		2714	6.597
7	10		2089	6.597

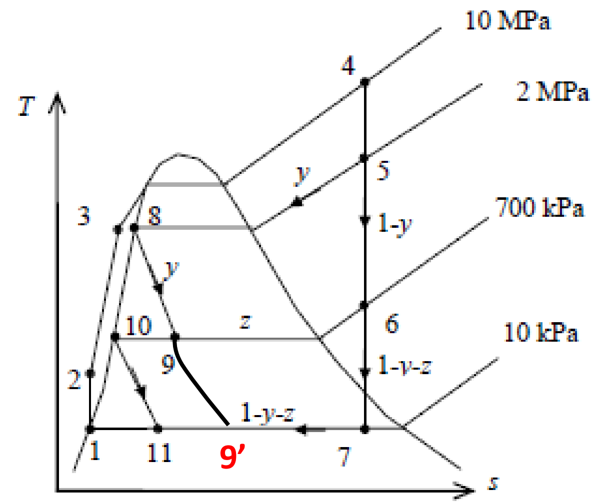
Saturation data

P, kPa	T_{sat}, °C	h_f, kJ/kg	v_f, m^3/kg
10	45.8	191.7	0.00101
700	165	697.3	0.00111
2000	212.4	908.6	0.00118
10000	311	1407.6	0.00145



4
Turbine
$\dot{W}_{turb}$
7
6
z
Water
inlet
Boiler
5
14
y
Process
heater
13
12
$\dot{Q}_{in}$
Condenser
10
3
Steam trap
11
Closed
FWH
2
1
8
$\dot{W}_{pump}$
Steam trap
9





An alternate scheme is shown in red. Red cross represents line removed in this scheme

- Steady operating conditions exist
- Kinetic and potential energy effects are negligible

$$h_1 = h_{f@10\ kPa} = 191.81\ \text{KJ/kg}$$

$$v_1 = v_{f@10\ kPa} = 0.00101\, m^3\ /\text{kg}$$

$$w_{P1,in} = v_1\ (P_2 - P_1)$$

$$= (0.00101 m^3\ /\text{kg}\ )\ (10000 kPa - 10 kPa) \left(\frac{1\ kJ}{1\ kPa.m^3}\right)$$

$$= 10.1\ \text{KJ/kg}$$

$$h_2 = \ h_1 + w_{P1,in} = 191.81 + 10.1 = 201.9\ \text{KJ/kg}$$

$$h_3 = \ h_8 = h_9 = h_{f@2000\ kPa} = 908.47\ \text{KJ/kg}$$

$$h_{10} = h_{11} = h_{f@700\ kPa} = 697\ \text{KJ/kg}$$

An energy balance on the closed feedwater heater gives

$$y(h_5 - h_8) = h_3 - h_2$$

$$y = \frac{h_3 - h_2}{h_5 - h_8} = \frac{908.47 - 201.9}{2930 - 908.47} = 0.3495$$

The process heat is expressed as

$$\dot{Q}_{process} = Z\dot{m}(h_6 - h_{10}) = \dot{m}_w C_p \Delta T_w$$

$$\dot{m}_w = \frac{Z\dot{m}(h_6 - h_{10})}{C_p \Delta T_w} = \frac{0.05 * 100 * (2714 - 697)}{4.18 * 40} = 60.3$$

The net power output is determined from

$$\dot{w}_{net} = \dot{w}_T - \dot{w}_P$$

$$= \dot{m}[y(h_4 - h_5) + z(h_4 - h_6) + (1 - y - z)(h_4 - h_7) - w_P]$$

$$= 100\,[0.3495\,(3374 - 2930) + 0.05(3374 - 2714) + (1 - 0.3495 - 0.05)(3374 - 2089) - 10.1]$$

$$= 94970\ \text{kW}$$

The rate of heat input in the boiler is

$$\dot{Q}_{in} = \dot{m}(h_4 - h_3) = 100 * (3374 - 908.47) = 246553\text{kW}$$

The rate of process heat is

$$\dot{Q}_{process} = 0.05\,\dot{m}\,(h_6 - h_{10}) = 0.05 * 100 * (2714 - 697) = 10085\text{kW}$$

The utilization efficiency of this cogeneration plant is

$$\epsilon_u = \frac{\dot{w}_{net} + \dot{Q}_{process}}{\dot{Q}_{in}} = \frac{94970+10085}{246553} = 0.426$$