Q2 The diagram shows an idealised regenerative steam cycle. In the turbine, heat is transferred from the steam to the feed-water and no heat is lost to the surroundings. The water at point (3) is saturated at 0.05 bar pressure. The water at point (5) is saturated at 200 bar pressure. The steam at point (3) is at 600°C. The feed pump process is adiabatic and reversible. The expansion in the turbine from point (6) to point (2) is isentropic.

(a) Draw the T – s diagram for the cycle indicating the heat gained by the feed-water from (4) to (5) and the heat lost by the steam from(1) to (6).

(b) Assuming a cycle efficiency of 40%, determine the dryness fraction at point (2) and the work output of the cycle.

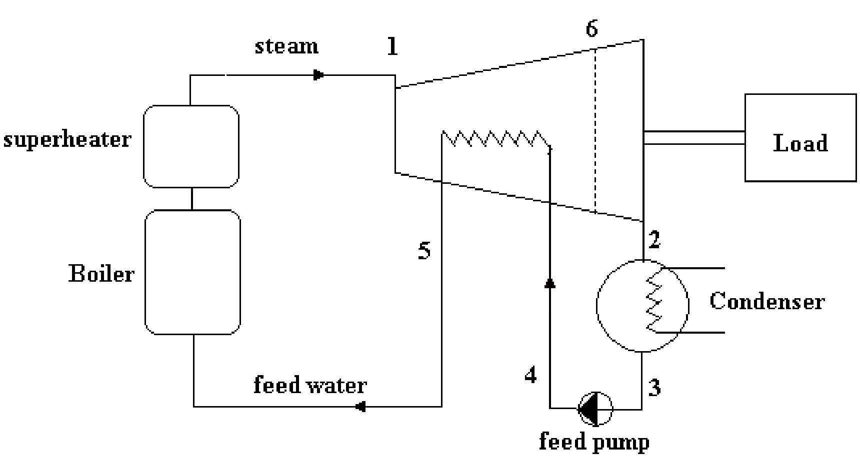
(c) Determine the temperature of the steam at (6), the dryness fraction and enthalpy.

(d) Comment on the distribution between work output and heat transfer within the turbine.

Assume the specific heat capacity of water is 4.187 kJ/kg K. Also assume straight condition lines for the steam and feed-water in the regenerative section of the turbine.

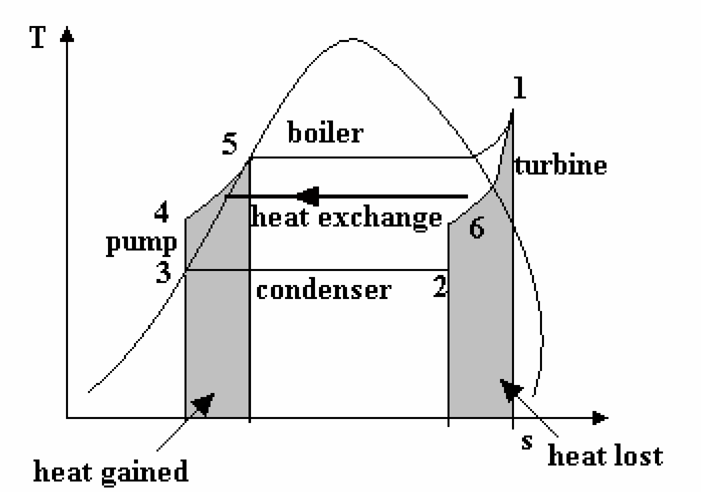
COMMENT

As will be seen below, I cannot obtain sensible answers to this question and suspect the 40% efficiency is the cause of the problem but if anyone can point out an error in my method, please let me know.



SOLUTION

a) The shaded areas represents the heat transfer inside the turbine from the steam into the feed water so the areas should be equal.



(b)

Point (1) 200 bar 600°C h_1 = 3537 kJ/kg s_1 = 6.505 kJ/kg K

Point (2) 0.05 bar t_s = 32.9°C

Point (3) saturated water @ 0.05 bar h_3 = 138 kJ/kg s_3 = 0.476 kJ/kg K t_s = 32.9°C

Point (4) s_4 = 0.476 (rev adiabatic 3 to 4)

Point (5) saturated water @ 200 bar h_6 = 1827 kJ/kg s_6 = 4.01 kJ/kg K t_s = 365.7°C

BOILER

Q(in) = $h_1 - h_5$ = 3537 – 1827 = 1710 kJ/kg

η = 40% = W(nett)/Q(in)

NETT WORK

W(nett) = 0.4 x 1710 = 684 kJ/kg This is the work output of the cycle.

PUMP

Work input = volume x Δp = 0.001 m^3/kg x (200 – 0.05) x 10^5 = 19995 J/kg or 20 kJ/kg

Pump work = 20 kJ/kg = c Δθ Δθ = 20/4.187 = 4.8 K

θ_3 = t_s @ 0.05 bar = 32.9 °C

Work out of turbine = W (out) = 684 + 20 = 704 kJ/kg

CONDENSER

Heat Loss from cycle = Q(out) = Q(in) – W(nett) = 1710 – 684 =1026 kJ/kg

Check η = 1 - Q(out)/ Q(in) = 1 - 1026/1710 = 40%

$h_2 = h_3$ + Q(out) = 138 + 1026 = 1164 kJ/kg

h_2 = 1164 = $h_f + x\, h_{fg}$ at 0.05 bar = 138 + 2423 x

x_2 = 0.423

$s_2 = s_f + x\, s_{fg}$ at 0.05 bar = 0.476 + .423 (7.918) = 3.825 kJ/kg K = s_6

(c) **HEAT TRANSFER**

Heat received from (4) to (5) Q = shaded area under process line.

θ_4 = 32.9 + 4.8 = 37.7 °C

Q_T =($s_5 - s_4$) (37.7 + 365.7)/2 = (4.014 – 0.476) (37.7+ 365.7)/2 = 713.6 kJ/kg

Q_T = 713.6 kJ/kg This is almost equal to the work output of the turbine.

This is the same for process 1 to 6 and can be used to find T_6

Q_T =(s_1 - s_6) (600 + T_6)/2

Q_T = (6.505 – 3.825) (600 + T_6)/2 = 713.6 kJ/kg

(2.68) (600 + T_6)/2 = 713.6

(600 + T_6) = 532.5

T_6 = -67.5 silly ??????

Another approach is as follows.

$h_1 - h_2$ = W(out) + Q_T

3537 – h_2 = 704 + 713.6 = 1417.6 h_2 = 3537 - 1417.6

h_2 = 2119.4 kJ/kg and this does not agree with the other method

h_2 = 2119.4 = $h_f + x\, h_{fg}$ at 0.05 bar = 138 + 2423 x

x_2 = 0.818

$s_2 = s_f + x\, s_{fg}$ at 0.05 bar = 0.476 + .818 (7.918) = 6.951 kJ/kg K This is larger than s_1 so this is also a silly answer. No sensible answer to this question.

A third approach

Ideal conditions suggest that $T_6 = T_4$ so that there is isothermal heat transfer all through the heat exchanger.

In this case T_6 = 37.7°C and p_s = 0.065 bar

$s_6 = s_2 = s_f + x\, s_{fg}$ at 0.065 bar but there are two possible values from above.