Problem 1: Steam (dry and saturated) is supplied by the boiler at 15 bar and the condenser pressure is 0.4 bar. Calculate the Carnot and Rankine efficiencies of the cycle. Neglect the pump work.

Problem 2: Water is the working fluid in an ideal Rankine cycle. Dry saturated vapour enters the turbine at 16 MPa, and the condenser pressure is 8 kPa. The mass flow rate of steam entering in the turbine is 120 kg/s. Calculate:

(a) the net power developed (in MW);

(b) rate of heat transfer to the steam passing through the boiler (in MW);

(c) thermal efficiency;

(d) mass flow rate of the condenser cooling water (in kg/s), if the cooling water undergoes a temperature increase of 18^{o}C with negligible pressure change in passing through the condenser. Assume that the heat capacity at constant pressure $\left(C_p\right)$ of the cooling water is 4.18 $\frac{kJ}{kg.^{\circ}C}$.

Problem 3: The table below represents the steps of an idealised steam power plant:

Step	**Location**	**Pressure (bar)**	**Temperature (°C)**	**Quality / State**	**Velocity m/s**
1	Inlet to turbine	60	380	–	–
2	Exit from turbine and inlet to condenser	0.1	–	0.9	200
3	Exit from condenser and inlet to pump	0.09	–	Saturated Liquid	– –
4	Exit from pump and inlet to boiler	100	–	–	–
5	Exit from boiler	80	440	–	–

Assume that the steam mass flow rate leaving the boiler is 10^4 kg.h^{-1}. Sketch the cycle numbering each stage. Calculate:

(a) Specific enthalpies of all streams;

(b) Power output of the turbine;

(c) Heat transfer per hour in the boiler and condenser;

(d) Mass rate of cooling water circulated (kg/h) in the condenser assuming inlet and outlet fluid temperatures from the condenser of 20°C and 30°C. Assume the heat capacity at constant pressure of the cooling water (C_p) is 4.18 $\dfrac{\text{kJ}}{\text{kg.}^{\circ}\text{C}}$;

(e) Diameter of the pipe connecting the turbine with the condenser;

(f) Sketch the Ts diagram, indicating each step of the cycle.

Problem 4: In the secondary cooling circuit of a nuclear power plant, the steam generator (boiler / reheater) produces superheated steam (SHS, Fig. 1) and is connected to two turbines operating as a reheat Rankine cycle. Isentropic efficiencies of the first (η_{T1}) and second (η_{T2}) turbines are 84%, 80%, respectively. The mass flow rate of water in the system is 1000 kg.s^{-1}.

Stage	P (bar)	T (°C)	State	Quality	h (kJ.kg^{-1})	s (kJ.(kg.K)$^{-1}$)
1	40	320	SHS	–	**(a)**	**(b)**
2	–	**(c)**	**(d)**	**(e)**	**(f)**	**(g)**
3	7	370	SHS	–	**(h)**	**(i)**
4	0.10	**(j)**	**(k)**	**(l)**	**(m)**	**(n)**
5	0.10	**(o)**	**(p)**	–	**(q)**	**(r)**
6	40	–	**(s)**	–	**(t)**	–

Table 1: **Problem 4:**.

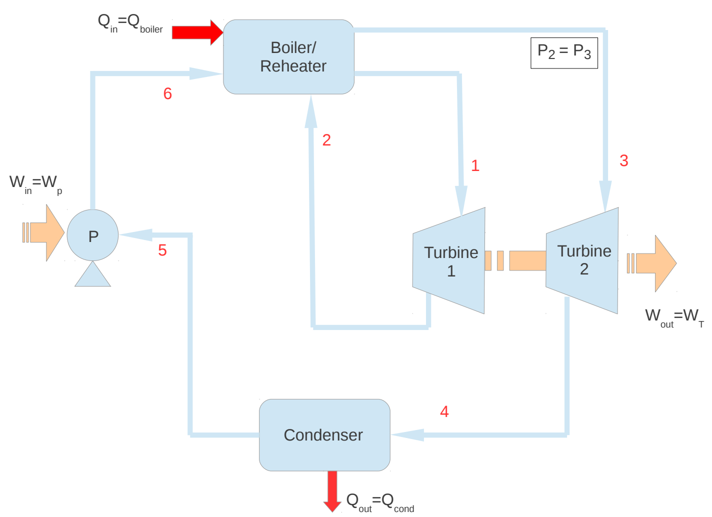


Figure 1: **Problem 4:**

- Determine *(a)-(t)* in Table 1;
- Calculate the power produced by the turbines;
- Calculate the heat supplied by the boiler;

- Calculate the heat extracted from the condenser. Assume that the heat capacity at constant pressure $\left(\mathrm{C_p}\right)$ is 4.18 $\frac{\mathrm{kJ}}{\mathrm{kg.^{o}C}}$;
- Sketch the Ts diagram of the cycle.

Problem 5: R-22 is the refrigerant fluid in a geothermal heat pump system for a house (Fig. 2). The heat pump uses underground water from a well $\left(T_{\mathrm{w}}^{\mathrm{in}} = 13^{\mathrm{o}}\mathrm{C}; T_{\mathrm{w}}^{\mathrm{out}} = 7^{\mathrm{o}}\mathrm{C}\right)$ to produce a heating capacity of 4.2 tons. Determine:

(a) Volumetric flow rate of heated air to the house $\left(m^3/s\right)$;

(b) Isentropic efficiency $\left(\eta_c\right)$ and power $\left(\dot{W}_c\right)$ of the compressor;

(c) Coefficient of Performance;

(d) Volumetric flow rate of water from the geothermal well $\left(l/h\right)$;

(e) Sketch the TS diagram.

Given the heat capacity $\left(C_p^{\mathrm{air}} = 1.004\frac{kJ}{kg.K}\right)$ and molecular weight $\left(MW^{\mathrm{air}} = 28.97\frac{kg}{kgmol}\right)$ of air and heat capacity of water $\left(C_p^{\mathrm{water}} = 4.1813\frac{kJ}{kg.K}\right)$.

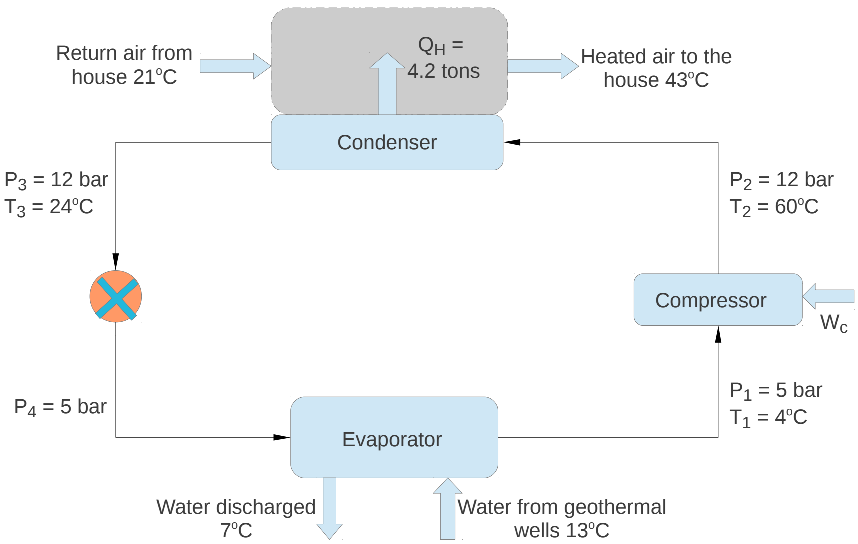
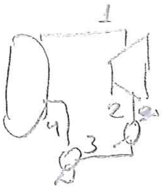


Figure 2: Heat pump cycle: Problem **Problem 5:**.

P1 Water-Steam system



$\left\{\begin{array}{l} P_1 = 15 \text{ bar} \\ x_1 = 1 \text{ (dry \& saturated)} \end{array}\right.$ $\Rightarrow$ from saturated table $\left\{\begin{array}{l} T_1 = T_{sat} = 198.3°C \\ h_1 = h_g = 2792.2 \text{ KJ/Kg} \\ S_1 = S_g = 6.4448 \text{ KJ/Kg.K} \end{array}\right.$

$P_2 = 0.4 \text{ bar}$ $\left\{\begin{array}{ll} T_2 = T_{sat} = 75.87°C & \\ h_g = 2636.8 \text{ KJ/Kg} & ; S_g = 7.6700 \text{ KJ/Kg.K} \\ h_f = 317.58 \text{ KJ/Kg} & ; S_f = 1.0259 \text{ KJ/Kg.K} \end{array}\right.$

$$\eta_{Carnot} = 1 - \frac{T_2}{T_1} = 1 - \frac{(75.87 + 273.15)K}{(198.3 + 273.15)K} = 0.2597 \therefore \underline{\underline{25.97\%}}$$

$$\eta_{Rankine} = \frac{\text{Adiabatic or Isentropic Heat Drop}}{\text{Heat Supplied}} = \frac{h_1 - h_2}{h_1 - h_{f4}}$$

assume $h_{f2} = h_{f4}$

$h_2 ? \Rightarrow h_2 = h_{f2} + x_2(h_{g2} - h_{f2})$

$x_2 : ?$

Steam expands isentropically : $S_1 = S_2$

$$S_2 = S_{f2} + x_2(S_{g2} - S_{f2}) = S_1 = 6.4448$$

$$x_2 = 0.8156 \therefore 81.56\%$$

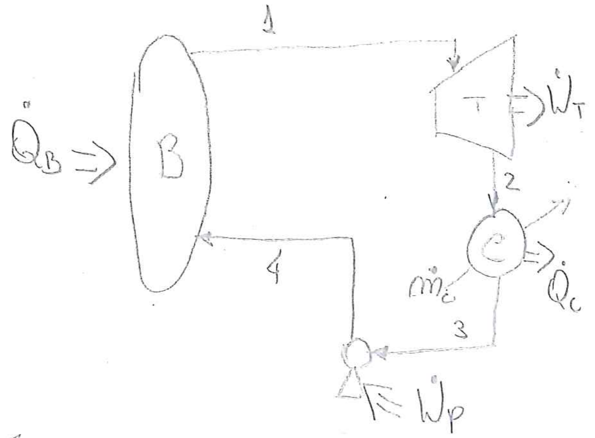
$h_2 = 2209.14 \ KJ/Kg$

$\eta_{Rankine} = 0.2356 \therefore 23.56\%$

P2: Ideal Rankine

10/03

$P_1 = 16\ MPa = 160\ bar$ (dry saturated vapour)



$P_3 = 8\ KPa = 0.08\ bar$

$\dot{m}_w = 120\ Kg/s$

$\dot{W}_{cycle} = ?$ $\dot{Q}_B = ?$ $\eta = ?$

$\dot{m}_c\ (\Delta T_c = 18°C) = ?$

First, let's obtain specific enthalpies of all fluid streams:

(1) dry saturated vapour @ 160 bar
$$\begin{cases} h_1 = 2580.6\ KJ/Kg \\ S_1 = 5.2455 \\ T_1 = 347.4\ °C \end{cases}$$

(2) Isentropic expansion $(S_2 = S_1)$ with quality,

$$S_2 = S_{f_2} + x_2 (S_{g_2} - S_{f_2}) \quad @\ P_3 = P_2$$

$$5.2455 = 0.5926 + x_2 (8.2287 - 0.5926)$$

$$x_2 = 0.6093$$

$$h_2 = h_{f_2} + x_2 (h_{g_2} - h_{f_2})$$

$$h_2 = 1638.10\ KJ/Kg$$

(3) $P_3 = 0.08\ bar$ (saturated liquid)

$$h_3 = h_{f_3}(P = 0.08\ bar) = 173.88\ KJ/Kg$$

(4) $P_4 = 160\ bar$ (isentropic compression of a incompressible fluid):

$$dh = v\,dP \therefore h_4 = h_3 + v_3 (P_4 - P_3)$$

$$h_4 = 173.88\ \frac{KJ}{Kg} + 1.0084 \times 10^{-3}\ \frac{m^3}{Kg} (160 - 0.08)\ bar = 190.01\ KJ/Kg$$

(a) $\dot{W}_{cycle} = ?$

$$\dot{W}_{cycle} = \sum \dot{W}_i = \dot{m}_\omega \left[\underbrace{(h_2 - h_1)}_{-942.50} + \underbrace{(h_4 - h_3)}_{16.13} \right]$$

$$\dot{W}_{cycle} = -111164.4 \ \text{KJ/s} = -1.11 \times 10^{-5} \ \text{KJ/s}$$

↳ Net Power of <u>111 MW</u>

(b) $\dot{Q}_B : ?$

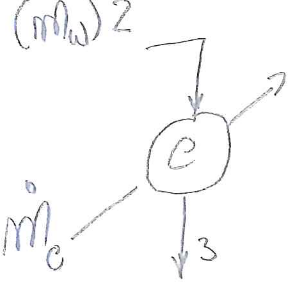
$$\dot{Q}_B = \dot{m}_\omega \, (h_1 - h_4) = 286860 \ \text{KJ/s} = 2.87 \times 10^5 \ \text{KJ/s}$$

↳ <u>287 MW</u> of heat is supplied by the boiler

(c) $$\eta = \frac{|\dot{W}_{cycle}|}{\dot{Q}_B} = \frac{111164.4}{286860} = 0.3875$$

↳ 38.75 %

(d) Defining a control volume around the condenser, and assuming <u>no</u> heat losses, the energy balance leads to:

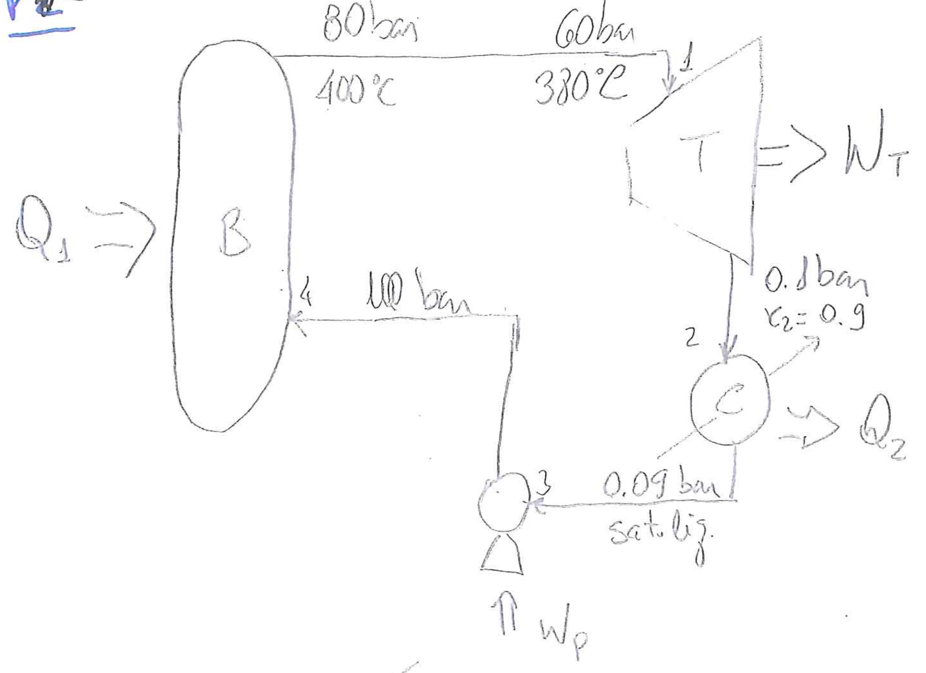


$$\dot{m}_\omega (h_3 - h_2) + \dot{m}_c \underbrace{\Delta h_c}_{= C_{p_c} \Delta T_c} = 0$$

$$120 \ \text{Kg/s} \, (173.88 - 1638.10) + \dot{m}_c \times 4.18 \ \frac{\text{KJ}}{\text{Kg} \cdot {}^\circ\text{C}} \times 18^\circ\text{C} = 0$$

$$\dot{m}_c = 2335.28 \ \text{Kg/s}$$

P3



(a) Calculating $\underline{h}$ and $\underline{s}$ of all streams.

(1) Fluid entering the turbine $\left\{ \begin{array}{l} P_1 = 60 \text{ bar} \\ T_1 = 380 °C \end{array} \right.$

- superheated steam: $T_1 > T_{sat}$ (275.6 °C)
- through linear interpolation: $h_1 = 3124.15$ KJ/Kg, $S_1 = 6.4595$ KJ/Kg.K

(2) Fluid leaving the turbine (isentropic expansion)

$$S_2 = S_1$$

and the fluid is 90% dry ($x_2 = 0.90$) at 0.1 bar

$$h_2 = h_{f_2} + x_2 (h_{g_2} - h_{f_2})$$

$$h_2 = 191.83 + 0.90 (2584.7 - 191.83) = 2345.41 \text{ KJ/Kg}$$

(3) Saturated liquid leaving the condenser:

$$h_3 = h_f (P = 0.09 \text{ bar}) = 182.86 \text{ KJ/Kg}$$

$S_3 = S_f(P = 0.09\,bar) = 0.6210\ KJ/Kg.K$

(4) $P_4 = 75\,bar$ (incompressible fluid)

$$\int_{h_3}^{h_4} dh = v\int_{P_3}^{P_4} dP$$ (isentropic compression)

$S_4 = S_3$

$$h_4 = h_3 + v_3\,(P_4 - P_3)$$

$$v_3 = v_f(P_3 = 0.09\,bar) = 1.0093\times10^{-3}\,m^3/Kg$$

$$h_4 = 182.86\,\frac{KJ}{Kg} + 1.0093\times10^{-3}\,\frac{m^3}{Kg}\,(100 - 0.09)\,bar$$

$$h_4 = 192.94\ KJ/Kg$$

(b) Power from turbine

$$\dot{W}_T = \dot{m}_\omega\,(h_2 - h_1) = 10^4\,\frac{Kg}{h}\,(2345.41 - 3124.15)\,\frac{KJ}{Kg}$$

$$\dot{W}_T = -2163.17\ KJ/s = -2163.17\ KW$$

(c) HT (per hour)

- Boiler: $\left\{\begin{array}{l} 80\ bar \\ 440°C \end{array}\right\}$ – superheated steam:

$440°C > T_{sat}(= 295.06°C)$

$h = 3246.1\ KJ/Kg$

$S = 6.5190\ KJ/Kg.K$

$$\dot{Q}_1 = \dot{m}_\omega \left[h(P = 80\,bar) - h_4 \right]$$

$$\dot{Q}_1 = 10^4 \frac{Kg}{h} \left[3246.1 - 192.94 \right] KJ/Kg = 8481\ KJ/s$$

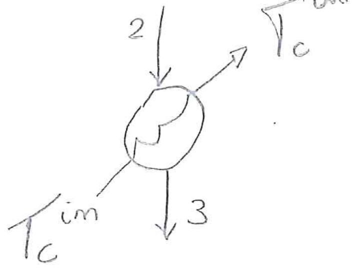
$$\dot{Q}_1 = 8481\ KW = 3.05 \times 10^7\ KJ/h$$

- Condenser:

$$\dot{Q}_2 = \dot{m}_\omega \left[h_3 - h_2 \right] = 10^4 \frac{Kg}{h} (182.86 - 2345.41) \frac{KJ}{Kg}$$

$$\dot{Q}_2 = -2.16 \times 10^7\ KJ/h = -6007.08\ KW$$

(d) $\dot{m}_c$: ? Heat lost from the steam is fully transferred to the cooling water



$$\dot{Q}_c = -\dot{Q}_2$$

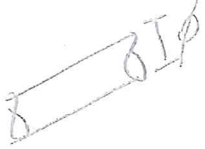
$$\dot{Q}_c = -2.16 \times 10^7 \frac{KJ}{h}$$

$$\dot{Q}_c = \dot{m}_c C_{p_c} (T_c^{out} - T_c^{in})$$

$$\dot{m}_c \times 4.18 \frac{KJ}{Kg.°C} (30 - 20)°C = 2.16 \times 10^7 \frac{KJ}{h}$$

$$\dot{m}_c = 5.17 \times 10^5\ Kg/h$$

(e) diameter of the pipe : ϕ



$$\underbrace{\frac{\pi}{4}\phi^2}_{\text{cross-section area of the pipe}} \; u_2 = \dot{m}_w \, v_2$$

u_2 = 200 m/s

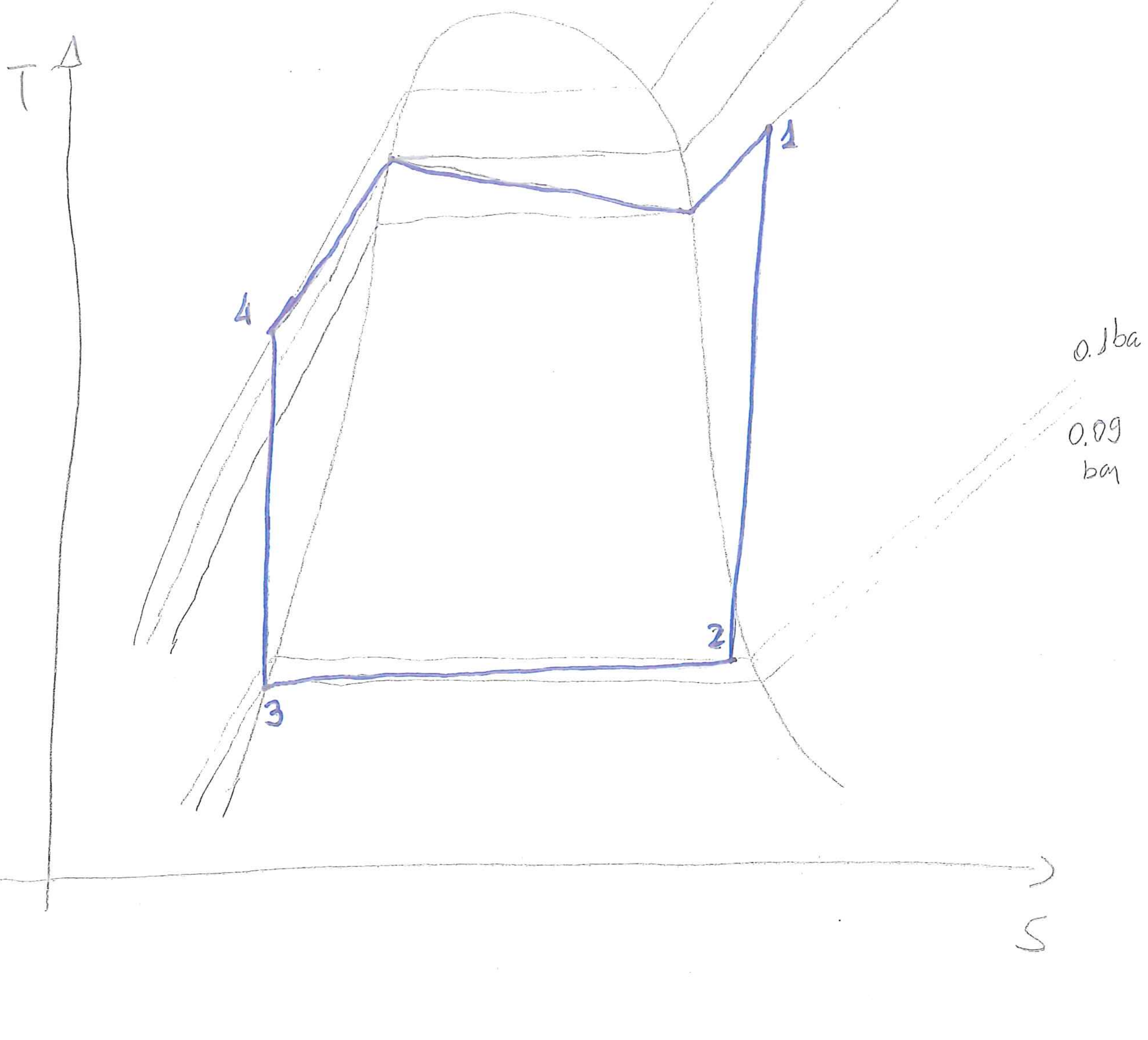
$P_2 = 0.1$ bar

$$v_2 = v_{f_2} + x_2\,(v_{g_2} - v_{f_2})$$

$$v_2 = 13.21 \ m^3/kg$$

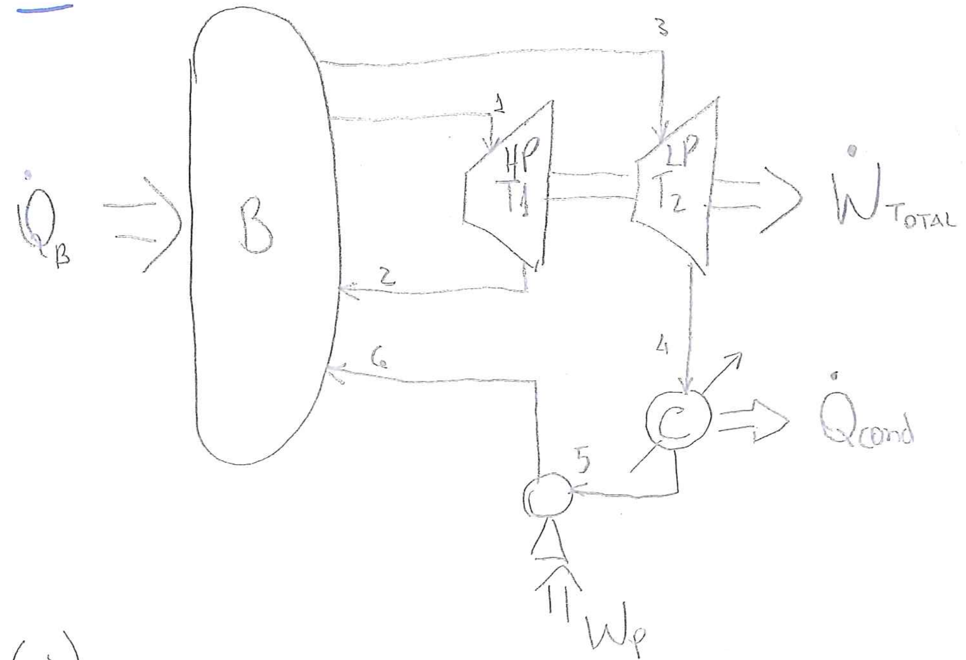
$$\boxed{\phi = 0.483 \, m}$$

(g)



P[redacted]4

$P_2 = P_3$



$\eta_{T_1} = 84\%$

$\eta_{T_2} = 80\%$

$\eta_P = 61\%$

(i)

(1) $P_1 = 40$ bar $\left\{ h_1 = 3015.4 \text{ kJ/kg} \right.$ (a)

$T_1 = 320°C$ $\quad s_1 = 6.4553$ kJ/kg.K (b)

(2) Ideal ; $s_{2s} = s_1$

$P_2 = P_3 = 7$ bar

$\hookrightarrow s_g(P = 7 \text{ bar}) = 6.7080$ kJ/kg.K $> s_{2s}$

(d)

(c) $\boxed{T_2 = T_{sat} = 165°C}$ $\Leftarrow$ $\boxed{\text{Saturated vapour}}$ (2 phase region)

(wet vapour)

Calculating how vaporised the water-steam system is:

$$s_{2s} = s_{f_2} + x_{2s}(s_{g_2} - s_{f_2})$$

$$6.4553 = 1.9922 + x_{2s}(6.7080 - 1.9922)$$

$$x_{2s} = 0.9464$$

$$h_{2s} = h_{f2} + x_{2s}(h_{g2} - h_{f2})$$

$$h_{2s} = 2652.75 \text{ KJ/Kg}$$

with the efficiency of the first turbine:

$$\eta_{T_1} = \frac{h_2 - h_1}{h_{2s} - h_1} = 0.84 \therefore \boxed{h_2 = 2710.77 \text{ KJ/Kg}} \quad (f)$$

actual enthalpy

now calculating the actual quality:

$$h_2 = h_{f2} + x_2(h_{g2} - h_{f2})$$

$$\boxed{x_2 = 0.9745} \quad (e)$$

and the actual entropy:

$$S_2 = S_{f2} + x_2(S_{g2} - S_{f2})$$

$$\boxed{S_2 = 6.5877 \text{ KJ/Kg.K}} \quad (g)$$

(3) $P_3 = 7$ bar, $T_3 = 240°C$ $\left\{ \begin{array}{ll} h_3 = 2932.2 \text{ KJ/Kg} & (h) \\ S_3 = 7.0641 \text{ KJ/Kg.K} & (i) \end{array} \right.$

(4) Ideal: $S_{4s} = S_3$

$P_4 = 0.1$ bar $\therefore S_g(P = 0.1 \text{ bar}) = 8.1502 \text{ KJ/Kg.K} > S_{4s}$

$\Downarrow$

(j) $\boxed{T_4 = T_{sat} = 45.81°C} \Leftarrow$ $\boxed{\text{Saturated vapour}}$ (K)

(wet vapour)

Calculating ideal quality:

$$S_{4s} = S_{f4} + x_{4s}(S_{g4} - S_{f4})$$

$$7.0641 = 0.6493 + x_{4s}(8.1502 - 0.6493)$$

$$x_{4s} = 0.8552$$

and ideal enthalpy

$$h_{4s} = h_{f4} + x_{4s}(h_{g4} - h_{f4})$$

$$h_{4s} = 2238.21 \text{ KJ/Kg}$$

With the efficiency of the second turbine:

$$\eta_{T_2} = \frac{h_4 - h_3}{h_{4s} - h_3} = 0.80 \therefore \boxed{h_4 = 2377.01 \text{ KJ/Kg}} \text{ (m)}$$

now calculating actual quality:

$$h_4 = h_{f4} + x_4(h_{g4} - h_{f4}) \therefore \boxed{x_4 = 0.9132} \text{ (l)}$$

and the actual entropy:

$$S_4 = S_{f4} + x_4(S_{g4} - S_{f4}) \therefore \boxed{S_4 = 7.4991 \text{ KJ/Kg.K}} \text{ (m)}$$

(5) Fluid leaving the condenser is $\boxed{\text{saturated liquid}}$ (p)

with $\boxed{T_5 = T_{sat}(P_5 = 0.1 \text{ bar}) = 45.81 °C}$ (o)

$$\begin{cases} h_5 = h_f(P_5 = 0.1 \text{ bar}) = 191.83 \text{ KJ/Kg} & \text{(q)} \\ S_5 = S_f(P_5 = 0.1 \text{ bar}) = 0.6493 \text{ KJ/Kg.K} & \text{(r)} \end{cases}$$

(6) Fluid leaving the pump have undertaken an isentropic compression to $P_6 = 40$ bar. We can assume the fluid is incompressible; thus the fundamental thermodynamic relation

$$T ds = dh - v dP$$

is reduced ($ds = 0$) to $dh = v dP$. Integrating from state 5 to 6 and assuming incompressibility ($v_5 = v_6$):

$$h_6 = h_5 + v_5 (P_6 - P_5)$$

$$h_6 = 191.83 \frac{KJ}{Kg} + 1.0102 \times 10^{-3} \frac{m^3}{Kg} (40 - 0.10) \text{ bar}$$

$$\boxed{h_6 = 195.86 \text{ KJ/Kg}}$$

$$h_6 < \left[h_g (P = 40 \text{ bar}) = 1087.3 \text{ KJ/Kg} \right]$$

$$\Downarrow$$

$$\boxed{\text{subcooled liquid}} \qquad (S)$$

(ii) Total Power produced:

$$\dot{W}_{T1} = \dot{m}_w (h_2 - h_1) = 10^3 \frac{Kg}{s} (2710.77 - 3015.4) \frac{KJ}{Kg} = -3.05 \times 10^5 \text{ KJ/s}$$

$$\dot{W}_{T_2} = \dot{m}_\omega (h_4 - h_3) = 10^3 \frac{Kg}{s} (2377.03 - 2932.2) \; KJ/Kg$$

$$\dot{W}_{T_2} = -5.55 \times 10^5 \; KJ/s$$

$$\dot{W}_{Total} = \dot{W}_{T_1} + \dot{W}_{T_2} = -8.60 \times 10^5 \frac{KJ}{s}$$

↳ The turbines produced 860 MW of power.

(iii) Heat supplied by the boiler:

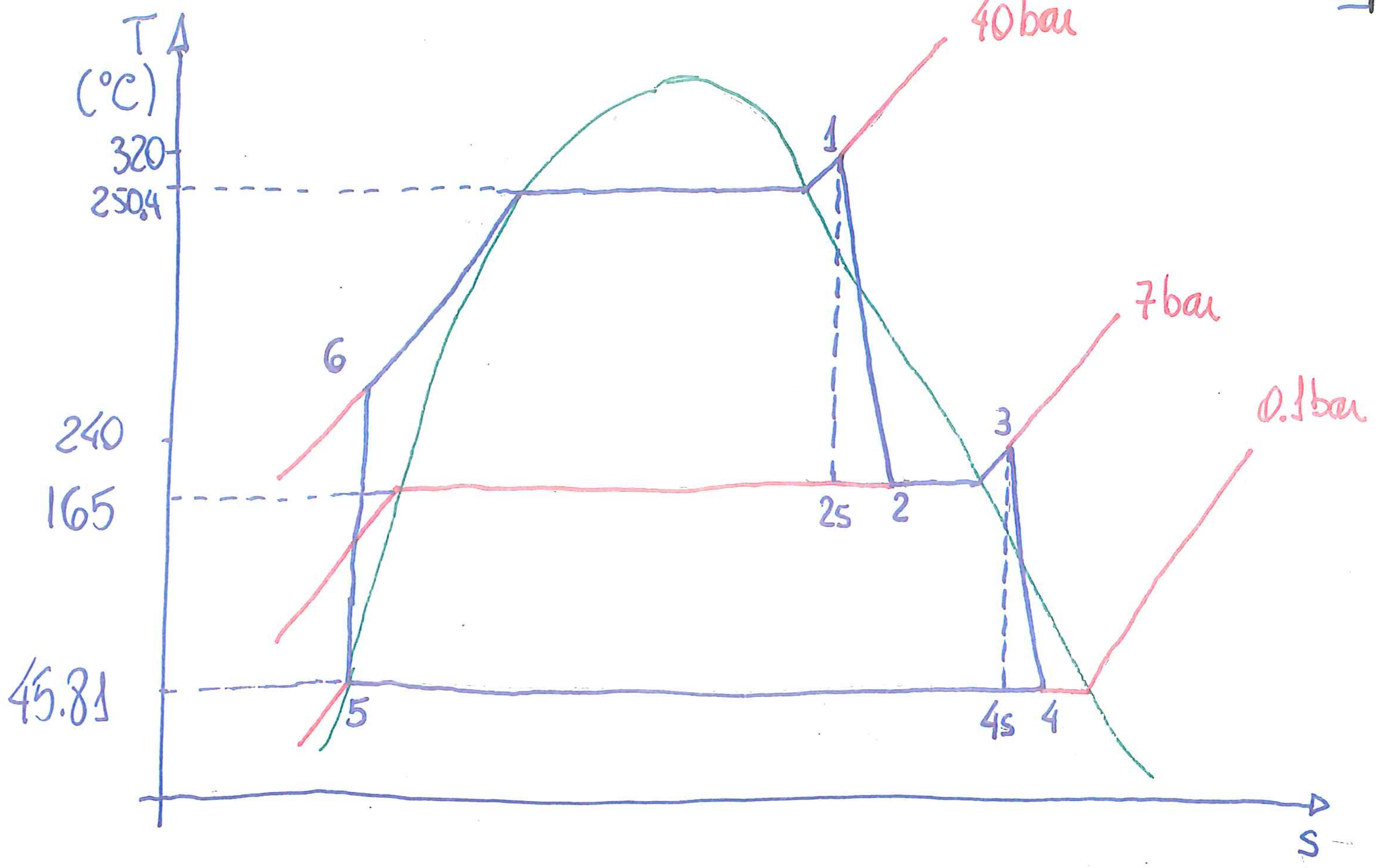
$$\dot{Q}_B = \dot{m}_\omega [(h_1 + h_3) - (h_2 + h_6)] = 3.04 \times 10^6 \; KJ/s$$

↳ The boiler supplied 3041 MW of heat to the system.

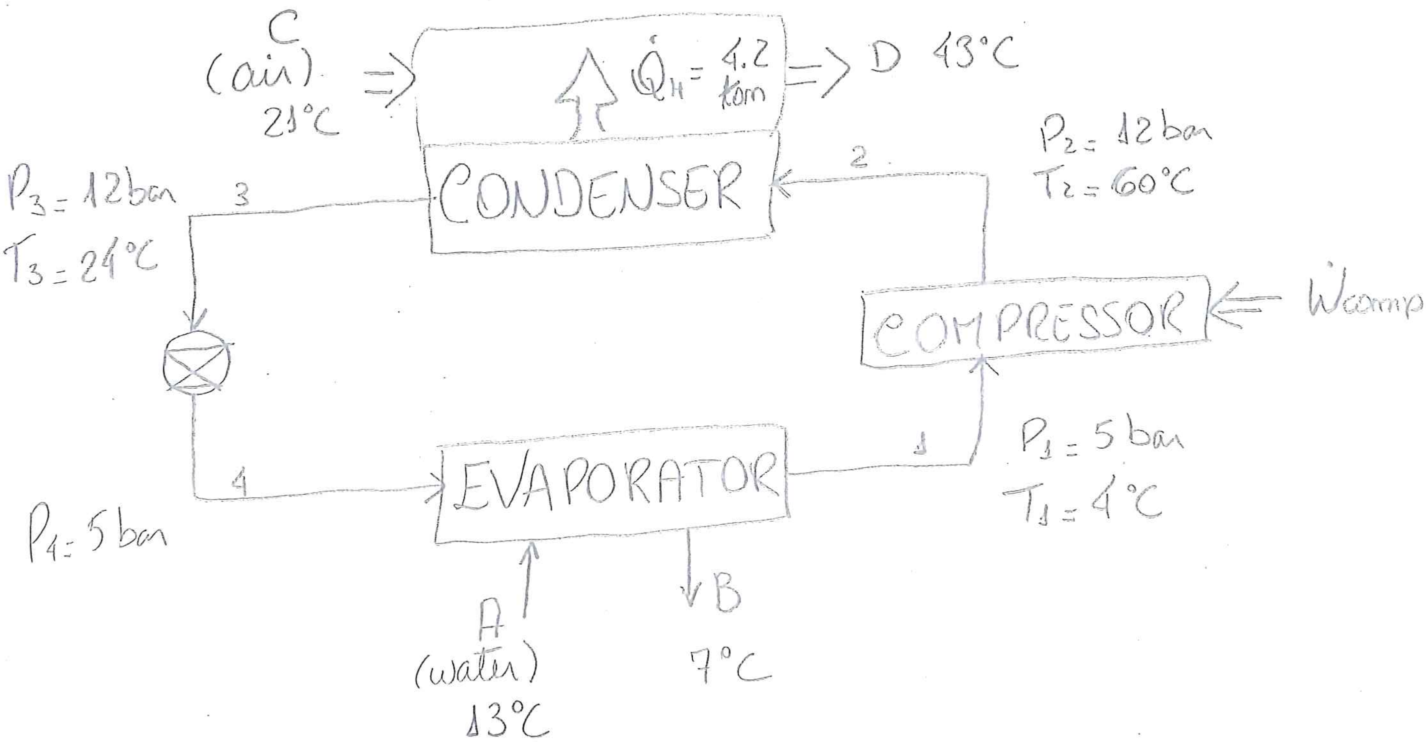
(iv) Heat extracted from the condenser

$$\dot{Q}_{cond} = \dot{m}_\omega (H_5 - H_4) = -2.19 \times 10^6 \frac{KJ}{s}$$

↳ 2185 MW of heat are extracted from the cycle.


T
(°C)
320
250.4
240
165
45.81
40 bar
7 bar
0.1 bar
1
2s
2
3
4s
4
5
6
s

P5: Geothermal heat pump - R22



Calculating enthalpies:

• 1: $P_1 = 5$ bar, $T_1 = 4°C$ $\Big\{$ $T_1 \gg T_{sat}$ (= 0.12°C) ⇒ superheated vapour

↓ linear interpolation

$h_1 = 252.83$ KJ/Kg

$S_1 = 0.9372$ KJ/Kg.K

• 2: $P_2 = 12$ bar, $T_2 = 60°C$ $\Big\{$ $T_2 \gg T_{sat}$ (= 30.25°C) ⇒ superheat vapour

$h_2 = 284.43$ KJ/Kg

$S_2 = 0.9666$ KJ/Kg.K

• **3**: $P_3 = 12$ bar, $T_3 = 24°C$ $\Big\}$ $T_3 \ll T_{sat}$ $(=30.25°C)$ $\rightsquigarrow$ subcooled liquid

$$h_3 = h_f(@24°C) = 74.04 \text{ KJ/Kg}$$

$$s_3 = s_f(@24°C) = 0.2772 \text{ KJ/Kg.K}$$

• **4**: Isenthalpic expansion $(P_4 = 5 \text{ bar})$

$$h_4 = h_3 = 74.04 \text{ KJ/Kg} > h_f(@5 \text{ bar})$$

$$x_4 = \frac{h_4 - h_f}{h_g - h_f} = 0.1406$$

$$x_4 = \frac{s_4 - s_f}{s_g - s_f} \therefore s_4 = 0.2831 \text{ KJ/Kg K}$$

(a) $\dot{V}_{air}^{(out)}$? (m^3/s)

$$\dot{Q}_H = \dot{m}_{air}\left(h_{out}^{air} - h_{in}^{air}\right) = \dot{m}_{air} C_{p_{air}}\left(T_{out}^{air} - T_{in}^{air}\right)$$

$$4.2 \text{ ton} \times \frac{210 \text{ KJ/min}}{1 \text{ ton}} \times \frac{1 \text{ min}}{60 \text{ s}} = \dot{m}_{air} \times 1.005 \frac{\text{KJ}}{\text{Kg.K}} \times (43-21) \text{ K}$$

$$\dot{m}_{air} = 0.6649 \text{ Kg/s}$$

ideal gas

$$P_{air}^{(out)} \dot{V}_{air}^{(out)} = m_{air} R T_{air}^{(out)} = \frac{\dot{m}_{air}}{MW_{air}} R T_{air}^{(out)}$$

$$\dot{V}_{air}^{(out)} = \frac{\dot{m}_{air}}{MW_{air}} \frac{R T_{air}^{(out)}}{P_{air}^{(out)}}$$

$P_{air}^{(out)} = P_{atm} = 1.01325$ bar $\quad T_{air}^{(out)} = 43°C = 316.15 \text{ K}$

$$\dot{V}_{air}^{(out)} = 0.6649\ \frac{\cancel{kg}}{s}\ \frac{\cancel{gmol}}{29\,g} \times 0.08314\ \frac{bar.m^3}{\cancel{Kgmol}.K} \times 316.15\,K \times \frac{1}{1.01325\,bar}$$

$$\times \frac{1000\,\cancel{g}}{1\,\cancel{Kg}} \times \frac{1\,Kgmol}{1000\,gmol}$$

$$\boxed{\dot{V}_{air}^{(out)} = 0.5948\ \frac{m^3}{s}}$$

(b) η_c ? $\dot{W}_{comp}$?

$$\eta_c = \frac{h_{2s} - h_1}{h_2 - h_1}$$

$P_2 = 12\ bar$

$S_{2s} = S_1 = 0.9372\ KJ/Kg.K$

$\downarrow$ linear interpolation

$h_{2s} = 274.83\ KJ/Kg$

$T_{2s} = 48.46\,°C$

$$\boxed{\eta_c = 0.6962 \therefore 69.62\%}$$

$$\dot{W}_{comp} = \dot{m}_R\,(h_2 - h_1)$$

$\dot{m}_R$?

The heating to house:

$$\dot{Q}_H = \dot{m}_R (h_3 - h_2)$$

$$-4.2\ \text{ton} \times \frac{210\ kJ/min}{1\ \text{ton}} \times \frac{1\ min}{60 s} = \dot{m}_R (74.04 - 284.43) \frac{kJ}{kg}$$

heat leaving the system

$$\dot{m}_R = 0.0699\ kg/s$$

$$\boxed{\dot{W}_{comp} = 2.21\ kW}$$

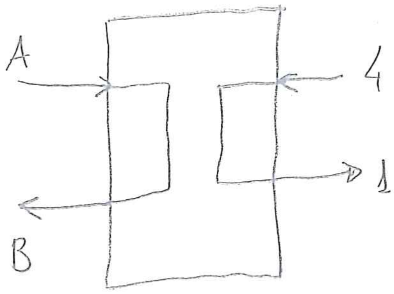
(c) $\boxed{COP = \dfrac{|\dot{Q}_H|}{\dot{W}_{comp}} = 6.65}$

(d) $\dot{V}_w$ (l/h)

Energy balance across the evaporator:

$$Q_R + Q_w = 0 \therefore Q_R = -Q_w$$

$$\dot{m}_R (h_1 - h_4) = -\dot{m}_w (h_B - h_A)$$



$$\dot{m}_R (h_1 - h_4) = -\dot{m}_w C_{p_w} (T_B - T_A)$$

$$\dot{m}_w = 0.4983\ kg/s$$

At 13°C, the specific volume of the water (saturated liquid) is $1.0007 \times 10^{-3} m^3/kg$ (v_f). Thus the volumetric flow rate of water is

$$\dot{V}_\omega = \dot{m}_\omega v_f = 0.4983 \frac{kg}{s} \times 1.0007 \times 10^{-3} \frac{m^3}{kg}$$

$$\dot{V}_\omega = 0.0004986 \frac{m^3}{s} \times \frac{3600 s}{1 h} \times \frac{1 l}{10^{-3} m^3}$$

$$\boxed{\dot{V}_\omega = 1794.96 \ l/h}$$