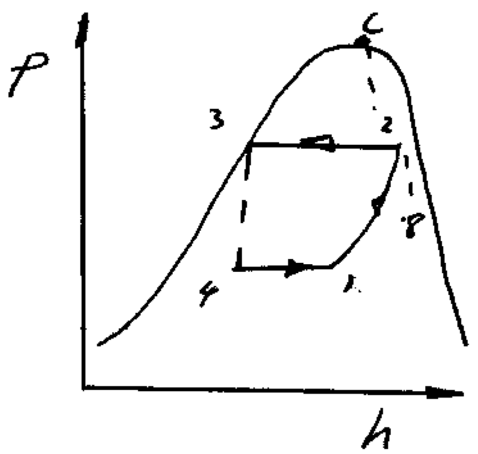
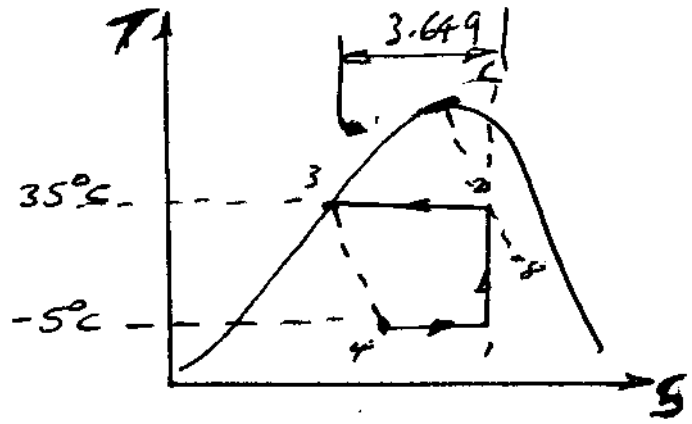
Q1 2000





h_{fg} @ 35°C = 1123.9 kJ/kg

$h_3 = h_f$ @ 35°C = 347.4 kJ/kg

$h_2 = h_f + x\,h_{fg} = 347.4 + 0.8 \times 1123.9 = 1246.52$ kJ/kg

HEAT REJECTED AT CONDENSER

$Q(out) = h_3 - h_3 = 899.12$ kJ/kg

THE SUBSTANCE APPEARS TO BE AMMONIA
(TABLES $h_{fg} = 1123.2$ kJ/kg @ 35°C)

$h_4 = h_3$

$$S_{fg} = \frac{h_{fg}}{T} = \frac{1123.9}{273+35} = 3.649$$

$$S_3 - S_2 = 80\% \times 3.649$$
$$= 2.9192$$

THE EXAMINER SAYS IT REVOLVES ABOUT SETTING h or S TO ZERO AT ANY POINT.

I CANNOT SEE HOW THIS HELPS

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.

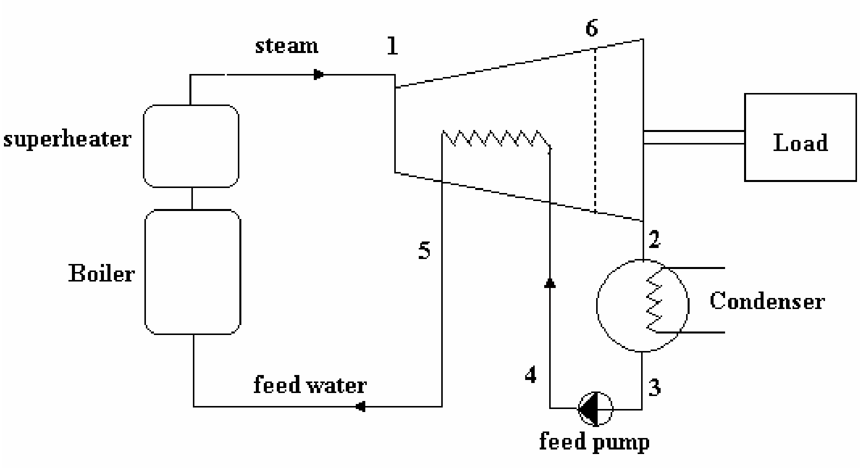
(c) Determine the temperature of the steam at (6), the dryness fraction and enthalpy.

(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.

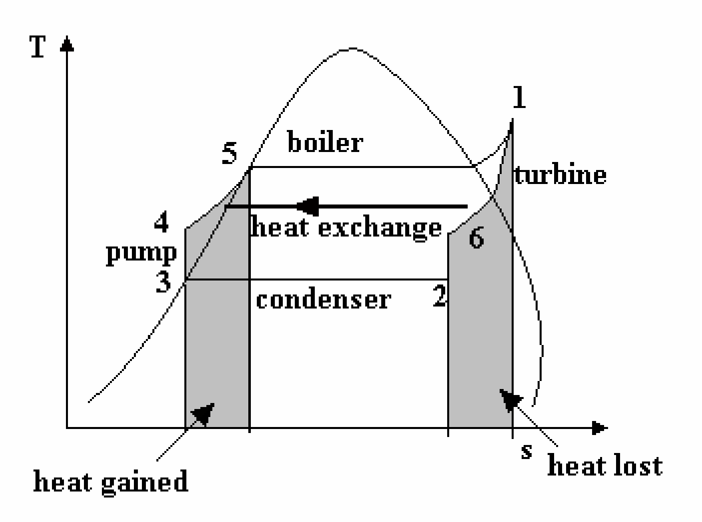
COMMENT

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 $\Delta\theta$ $\Delta\theta$ = 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.

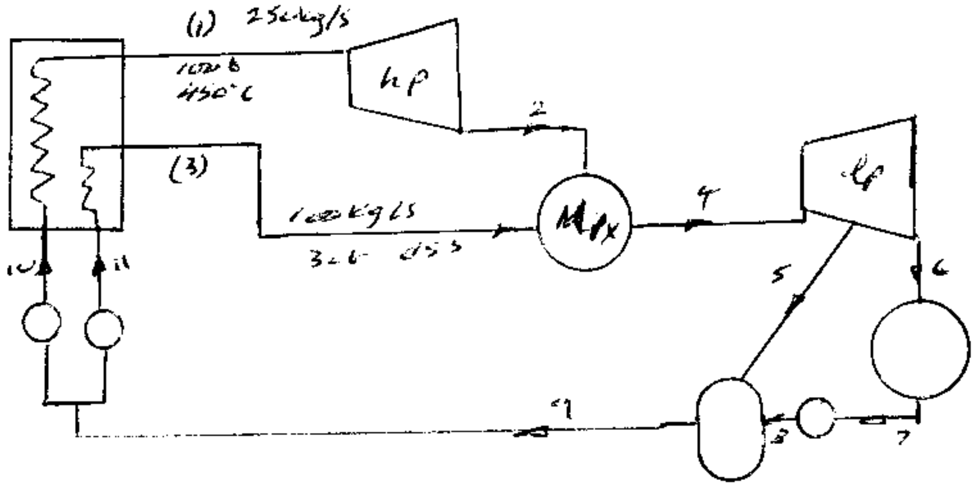
A third approach

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

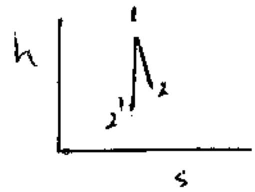
In this case T_6 = 37.7°C and p_s = 0.065 bar

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

Q4 2000



$h_1 = 3241$ kJ/kg
$h_3 = 2803$ kJ/kg



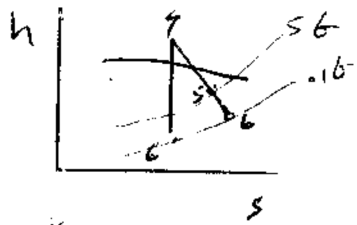
hp TURBINE
From h-s chart $h_2' = 2920$ kJ/kg

$\eta_i = .85 = \frac{3241 - h_2}{3241 - 2920}$ $\quad h_2 = 2968$ kJ/kg

$P_{out} = 250\,(3241 - 2968) = 68.25$ MW

ADIABATIC MIXING $\quad 250\, h_2 + 100\, h_3 = 350\, h_4$
$250 \times 2968 + 100 \times 2803 = 350\, h_4$
$h_4 = 2921$ kJ/kg

lp TURBINE
$p_4 = 20b \quad h_4 = 2921$
From chart $h_6' = 2050$ kJ/kg



$\eta_i = .82 = \frac{2921 - h_6}{2921 - 2050}$ $\quad h_6 = 2190$ kJ/kg

Assume straight line 4 to 6
$h_5 = 2650$ kJ/kg

Q4 2000

P_{4-5} $P = 350(2921 - 2650) = 94.85\text{ MW}$

P_{5-6} $P = 290(2650 - 2190) = 133.4\text{ MW}$

TOTAL POWER OUT $= 68.25 + 94.85 + 133.4 = 296.5\text{ MW}$

PUMPS IDEAL POWER $=$ Vol $\times \Delta p$

7–8 $P = 290 \times .001 \times (5 - .1) \times 10^5 = 142.1\text{ kW}$

9–10 $P = 100 \times .001 (30 - 5) \times 10^5 = 250\text{ kW}$

9–11 $P = 250 \times .001 (100 - 5) \times 10^5 = 2.375\text{ MW}$

NETT POWER OUT $= 296.5 - 0.142 - 0.25 - 2.375$
$P_{nett} = 293.7\text{ MW}$

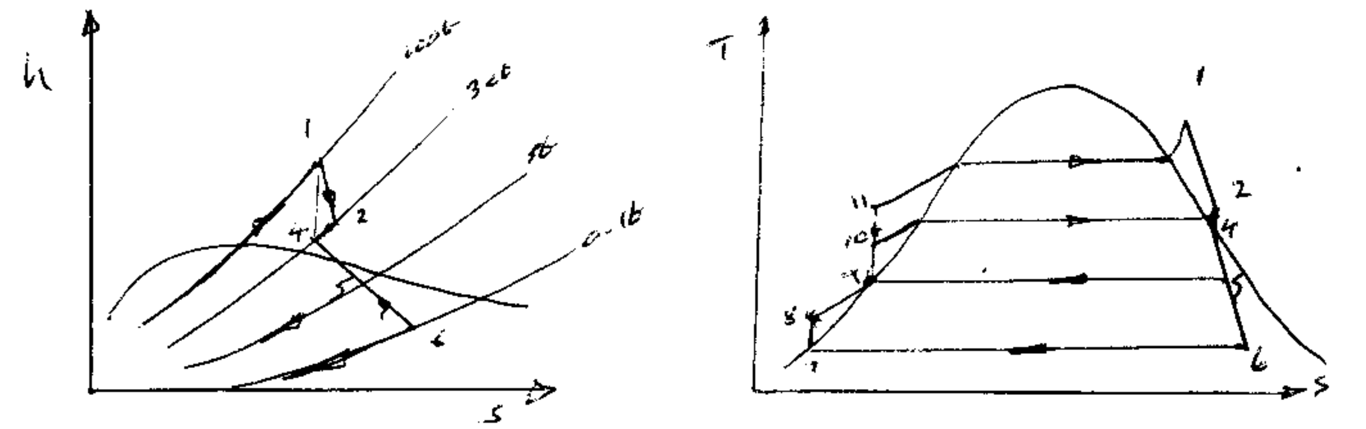
CONDENSER Assume $h_7 = h_f @ .16 = 192\text{ kJ/kg}$

$\dot{Q}_{out} = 290(h_6 - h_7) = 290(2190 - 192)$

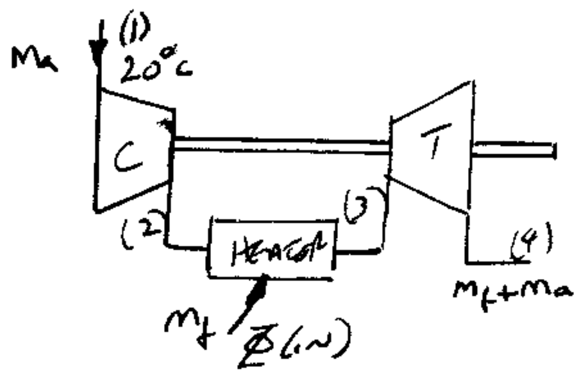
$\dot{Q}_{out} = 579.42\text{ MW}$

$\dot{Q}_{in} = \dot{Q}_{out} + P_{nett} = 876.92\text{ MW}$

$\eta_{th} = P_{nett} / \dot{Q}_{in} = 293.7/876.42 = 33.5\%$



Q6 2000



$r_p = 10 \qquad T_1 = 293\,K$

$T_3 = 1173\,K$

GAS CONSTANTS AIR

$R_a = R_0/\tilde{m} = 287\ J/kgK$

$C_{pa} = 1.004\ kJ/kgK$

$\gamma_a = 1.4$

GAS CONSTANTS (TURBINE)

$R_g = R_0/32 = 259.81\ J/kgK$

$C_{pg} = 1.2\ kJ/kg\,K$

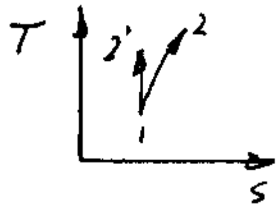
$C_{vg} = C_{pg} - R_g = 1200 - 259.81 = 940.2\ J/kgK$

$\gamma_g = C_{pg}/C_{vg} = 1200/940.2 = 1.276$

<u>COMPRESSOR</u> $T_2' = 293 \times 10^{\frac{\gamma}{\gamma-1}}$

$T_2 = 293 \times 10^{.286} = 566\,K$

<u>ISENTROPIC EFFICIENCY</u>



$\eta_{is} = \dfrac{T_2' - T_1}{T_2 - T_1} \qquad 0.88 = \dfrac{566 - 293}{T_2 - 293}$

$T_2 = 603.2\,K$

$\Phi(in) = M_g C_{pg} T_3 - M_a C_{pa} T_2$

USING FUEL CALORIFIC VALUE

$\Phi(in) = M_f \times 45000 \qquad M_g = M_a + M_f$

$45000 M_f = (M_a + M_f) \times 1.2 \times 1173 - M_a \times 1.004 \times 603.2$

$45000 = \left(\dfrac{M_a}{M_f} + 1\right) \times 1.2 \times 1173 - \dfrac{M_a}{M_f} \times 1.004 \times 603.2$

$45000 = 1407.6\ M_a/M_f + 1407.6 - 605.5\ M_a/M_f$

$43592 = 802\ M_a/M_f \qquad M_a/M_f = \underline{\underline{54.35}}$

Q6 2000

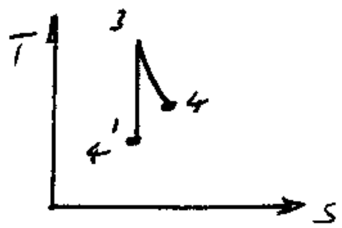
$p_2 = 10 \times 1 = 10\,\text{bar}$

TURBINE $\quad p_3 = 90\% \times p_2 = 9\,\text{bar} \quad r_p = 9$

$$T_4' = T_3 \left(\frac{1}{r_p}\right)^{\frac{\gamma-1}{\gamma}} = 1173 \left(\frac{1}{9}\right)^{.276/1.276} = 729\,K$$

ISENTROPIC EFFICIENCY

$$\eta_{is} = \frac{T_3 - T_4}{T_3 - T_4'} \qquad 0.9 = \frac{1173 - T_4}{1173 - 729}$$



$$T_4 = 773.6\,K$$

Power out $= M_g\, C_{pg} (T_3 - T_4)$

$$P(\text{out}) = M_g \times 1.2 \times (1173 - 773.6) = 479.2\, M_g$$

COMPRESSOR

$$P(\text{in}) = M_a\, C_{pa} (T_2 - T_1) =$$
$$= M_a \times 1.004 (603.2 - 293)$$
$$= 311.4\, M_a$$

NETT POWER

$$P_{nett} = 479.2\, M_g - 311.4\, M_a$$

$$\dot{Q}(\text{in}) = 45000\, M_f$$

EFFICIENCY

$$\eta_{th} = \frac{P_{nett}}{\dot{Q}_{in}} = \frac{499.\cancel{2}\, M_g - 311.4\, M_a}{45000\, M_f}$$

$M_g = M_a + M_f \quad m_a = 54.35\, m_f \quad m_g = 55.35\, M_f$

$$\eta_{th} = \frac{479.2 \times 55.35 M_f - 311.44 \times 54.35\, m_f}{45000\, M_f}$$

$$\eta_{th} = \frac{9612}{45000} = 0.214 \text{ or } 21.4\%$$

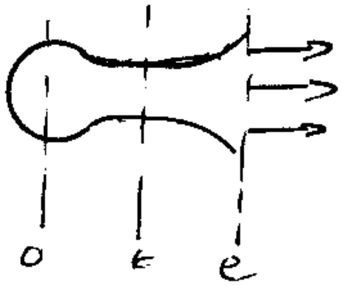
Q6 2000

CARNOT EFFICIENCY

$$\eta_c = 1 - \frac{T_{COLD}}{T_{HOT}}$$

$$= 1 - \frac{293}{1173} = 0.75 \text{ or } 75\%$$

Q8 2000



$\gamma = 1.25$

$\tilde{m} = 34 \text{ kg/kmol}$

$T_o = 2800 \quad P_o = 25b$

$R = R_o / \tilde{m}$

$R = \frac{8314}{34} = 244.53 \text{ J/kgK}$

$$C_p = \frac{R\gamma}{\gamma-1} = \frac{244.53 \times 1.25}{0.25} = 1222.6 \text{ J/kgK}$$

FLOW IS CHOKED so $\frac{T_o}{T_t} = 1 + \frac{(\gamma-1)}{2} = 1.125$

$$T_t = 2800/1.125 = 2489 \text{ K}$$

$$\frac{P_t}{P_o} = \left(\frac{2}{\gamma+1}\right)^{\frac{\gamma}{\gamma-1}} = 0.5549$$

$$P_t = 25 \times 0.5549 = 13.873 \text{ b}$$

SONIC VELOCITY $a_t = \sqrt{\gamma R T_t}$

$$a_t = V_t = \sqrt{1.25 \times 244.53 \times 2489} = 872.2 \text{ m/s}$$

DENSITY $\rho_t = \frac{M}{V} = \frac{P}{RT} = \frac{13.873 \times 10^5}{244.53 \times 2489}$

$$\rho_t = 2.279 \text{ kg/m}^3$$

EXIT $T_e = T_o \, r^{\gamma-1/\gamma} = 2800 \left(1/25\right)^{\frac{.25}{1.25}} = 1470.8 \text{K}$

$$C_p T_o = C_p T_e + \frac{V^2}{2}$$

Q82000

$$2C_p(T_0 - T_e) = V_e^2 \quad \text{(Velocity)}$$

$$2 \times 1222.6(2800 - 1470.8) = V_e^2$$

$$V_e = 1803 \text{ m/s}$$

DENSITY AT EXIT $\rho_e = \frac{M}{V} = \frac{P_e}{RT_e}$

$$\rho_e = \frac{1 \times 10^5}{244.53 \times 1470.8} = 0.278 \text{ kg/m}^3$$

MASS FLOW $= \rho A V$

$$\rho_t A_t V_t = \rho_e A_e V_e \qquad \frac{A_e}{A_t} = \frac{\rho_t V_t}{\rho_e V_e}$$

$$\frac{A_e}{A_t} = \frac{2.279}{0.278} \times \frac{872.2}{1803} = \underline{\underline{3.965}} \quad \text{ANSWER}$$

<u>THRUST</u>

$$F = m\Delta V + A_e \Delta p$$

For $A_t = 1 m^2$

$A_e = 3.965 m^2$

$m = 2.279 \text{ kg} \times AV$

$m = 2.279 \times 1 \times 872.2$

$m = 1988 \text{ kg/m}^2$

$$F = 1988 \times 1803 + 3.965 \times 1 \times 10^5$$

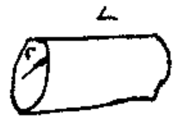
$$F = 3.585 MN + 0.397 MN$$

$$F = \underline{\underline{3.982 MN}} \quad \text{ANSWER}$$

Q9 2000

CONSIDER A THIN CYLINDRICAL LAYER RADIAL THICKNESS dr AND TEMP. DIFFERENCE dT

FOURIER'S LAW $\dot{\varphi} = -kA\dfrac{dT}{dr}$ $A = 2\pi r \times$ Length L



$$\dot{\varphi} = -k \times 2\pi r L \, \frac{dT}{dr}$$

$$2\pi k L \, dT = -\varphi \frac{dr}{r}$$

$$2\pi k L \int_{T_1}^{T_2} dT = -\dot{\varphi} \int_{R_1}^{R_2} \frac{dr}{r}$$

φ SAME AT ALL RADII

$$2\pi k L (T_1 - T_2) = -\dot{\varphi} \ln R_1/R_2 = +\dot{\varphi} \ln R_2/R_1$$

$$\dot{\varphi} = \frac{2\pi k L (T_1 - T_2)}{\ln R_2/R_1} = \text{Constant } (T_1 - T_2)$$

THERMAL RESISTANCE $R = \dfrac{T_1 - T_2}{\dot{\varphi}}$

$$R_1 = \frac{\ln R_2/R_1}{2\pi k L}$$

TAKE $L = 1m$

$K = 1.2$ W/mK For Inner Layer

$$R_1 = \frac{\ln 70/50}{2\pi \times 1.2} = 0.044626 \text{ k/W}$$

$$R_2 = \frac{\ln 90/70}{2\pi \times 2.4} = 0.016666 \text{ k/W}$$ FOR OUTER LAYER

Q9 2000

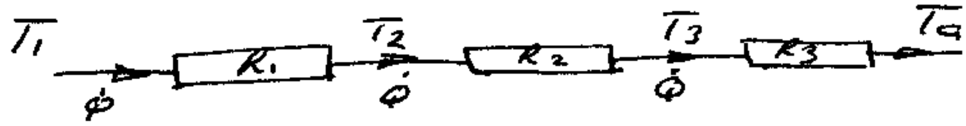
OUTER LAYER CONVECTION

$$\dot{\Phi} = hA(T_a - T_2) = h \times 2\pi r L (T_a - T_2)$$

$$\dot{\Phi} = 30 \times 2\pi \times .09 \times 1 \; (T_a - T_2) \qquad r = 90 \text{ mm}$$
$$= 16.96 (T_a - T_2) \qquad h = 30 \text{ W/m}^2\text{K}$$

THERMAL RESISTANCE $R_3 = \dfrac{T_a - T_2}{\dot{\Phi}} = 0.0589 \dfrac{K}{W}$

ANALOGY 3 RESISTANCES IN SERIES



$$R(\text{TOTAL}) = R_1 + R_2 + R_3 = 0.044626 + 0.016666$$
$$+ 0.0589$$

$$R_T = 0.12024 \text{ K/W}$$

$$\dot{\Phi} = \frac{T_1 - T_a}{R_T} = 8.317 (T_1 - T_a)$$

REVERSING LAYERS

$$R_1 = \frac{\ln 70/50}{2\pi \times 2.4} = 0.022313 \text{ K/W}$$

$$R_2 = \frac{\ln 90/70}{2\pi \times 1.2} = 0.03333 \text{ K/W}$$

$$R_3 = 0.0589 \text{ K/W}$$

$$R_T = 0.1145 \text{ K/W}$$

$$\dot{\Phi} = \frac{T_1 - T_a}{.1145} = 8.73 (T_1 - T_a)$$

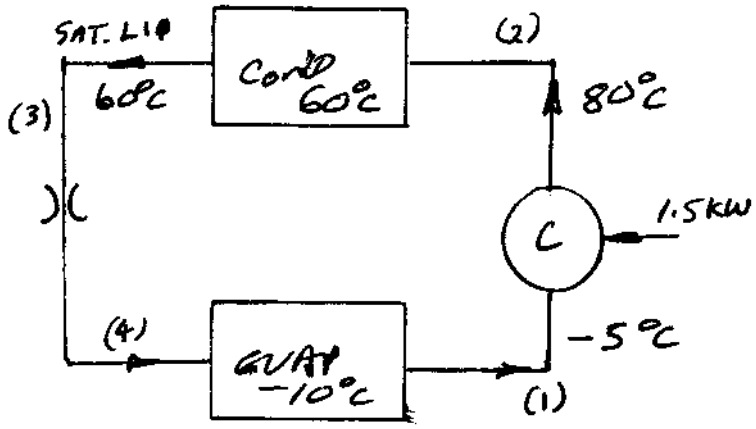
9 9 2000

DIFFERENCE $= 8.73 - 8.317$
$= 0.413$

% OF 8.73 $\frac{.413}{8.73} \times 100 = 4.7\%$

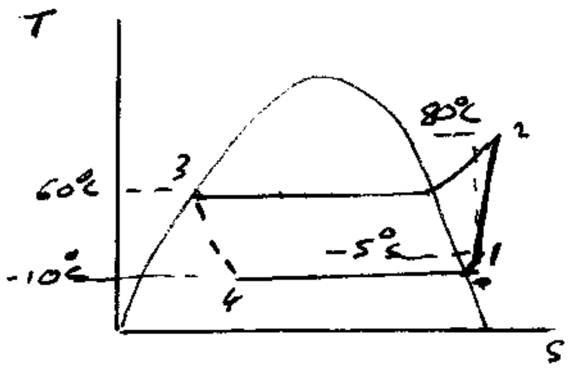
THIS IS 1/2 THE EXPECTED ANSWER.

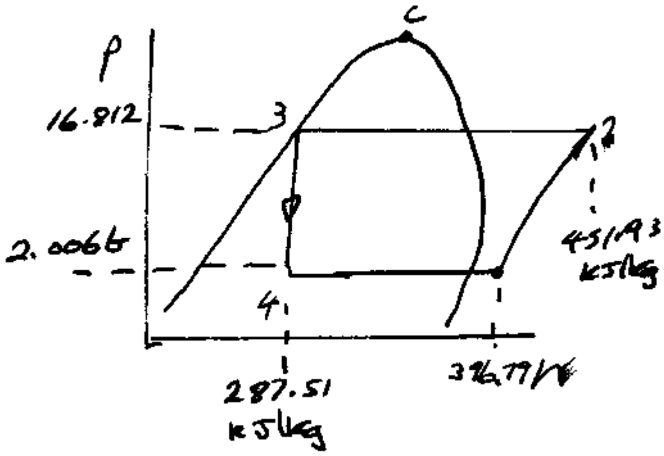
Q1 2001



R134a

$M_r = 0.025$ kg/s





To find h_1

AT 2.006 bar   -10°C   h = 392.51

-5°C   Mid Point h = 396.79

10k Superheat   0°C   h = 401.07

h_2 @ 16.812 bar $t_s = 60°C$

$\therefore$ 80°C is 20k superheat

h_2 @ 16.812 bar 20k S.h. is 451.93 kJ/kg

$h_3 = h_f$ @ 16.812 bar = 287.51 kJ/kg

HEAT PUMP $\quad C \text{ of } P = \dfrac{\Phi(out)}{P(in)}$

$$\Phi(out) = m_r (h_2 - h_3)$$
$$= 0.025 (451.93 - 287.51)$$
$$= 4.11 \text{ kW}$$

$$C \text{ of } P = 4.11/1.5 = \underline{\underline{2.74}}$$

POWER PASSED INTO THE REFRIGERANT IS $m_r (h_2 - h_1)$

$$= 0.025 (451.93 - 396.79)$$
$$= 1.3785 \text{ kW}$$

POWER LOSS FROM CASING $= 1.5 - 1.3785$
$$= \underline{\underline{0.01215 \text{ kW}}}$$

IF COOLED TO 55°C AT POINT (3)

$h_3 \simeq h_f$ @ 55°C (NEAREST WE CAN GET)

$h_3 \simeq 279.46$ kJ/kg

$$\Phi(out) = 0.025 (451.93 - 279.46) = 4.311 \text{ kW}$$

$$C \text{ of } P = 4.311/1.5 = 2.874$$

AN IMPROVEMENT AS EXPECTED BUT WE WOULD NEED MORE EVAPORATION TO MAINTAIN STATED CONDITIONS.

Q4 2001

1) $\eta = 1 - \frac{Q_{out}}{Q_{in}} = 1 - \dfrac{mC_v(T_5 - T_1)}{mC_v(T_3 - T_2) + mC_p(T_4 - T_3)}$

$$\eta = 1 - \frac{C_v(T_5 - T_1)}{C_v(T_3 - T_2) + C_p(T_4 - T_3)}$$

NB HEAT REJECTED AT CONST. VOL (5−1)
HEAT INPUT AT CONST VOL (2−3)
AND CONST. PRESSURE (3−4)

11) $r = 10$

$p_1 = 1\text{bar}$ $T_1 = 290\text{K}$ $m_a = 0.05\text{kg/s}$
$\dot{Q}_{in} = 50\text{kW}$

$\dot{Q}_{in}$ 2−3 = 25kW
$\dot{Q}_{in}$ 3−4 = 25kW

Highest p and T is at point (4)

$\gamma = C_p/C_v = 1.004/0.717 = 1.4$

$T_2 = T_1(r)^{\gamma - 1} = 290 \times 10^{0.4}$

$\cancel{= 290 \times 728.4\,K}$

$290 \times 2.512 = 728.4K$

Const Vol Heating
$25\text{kW} = mC_v(T_3 - T_2)$

$25 = 0.05 \times 0.717(T_3 - 728.4)$
$T_3 = 697.35 + 728.4 = 1425.8K$

Constant Pressure Heating

$$25 = 0.05 \times 1.004\,(T_4 - 1425.8)$$

$$T_4 = 498 + 1425.8 = \underline{\underline{1923.8\ K}}$$

$P_3 = P_4 =$ HIGHEST PRESS

$$P_{\cancel{3}} = \frac{P_1 V_1}{T_1} \times \frac{T_{\cancel{3}}}{V_{\cancel{3}}} = \frac{1}{290} \times \frac{10}{\cancel{1}} \times 1425.8$$

$$P_3 = P_4 = 49.16 \text{ bar}$$

~~Φ = ...~~
~~W~~ Pnett $= \Phi_{in} - \Phi_{out}$

$$\Phi_{out} = m\,c_v\,(T_5 - T_1)$$

$$\frac{P_4 V_4}{T_4} = \frac{P_3 V_3}{T_3}$$

$$\frac{V_3}{V_4} = \frac{P_4}{P_3} \times \frac{T_3}{T_4} = 1 \times \frac{1425.8}{1923.8} = 0.74$$

$$\frac{V_4}{V_3} = 1.35$$

$$T_5 = T_4 \left(r \right)^{0.4}_{V_4/V_1}$$

$$\frac{V_5}{V_4} = \frac{V_5}{V_3} \times \frac{V_3}{V_4} = 10 \times .74 = 7.4$$

$$T_5 = 1923.8 \times \left(\frac{1}{7.4}\right)^{.4} = 863.9\,K$$

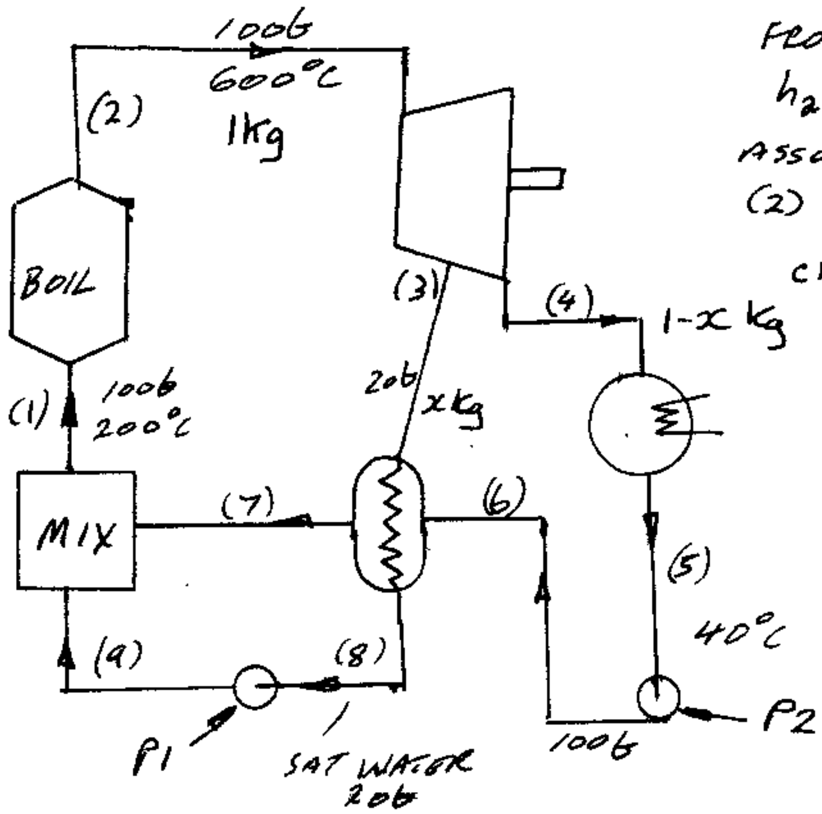
$$\dot{Q}_{out} = 0.05 \times .717 \,(863.9 - 290)$$

$$= 20.57\ kw$$

$$P(nett) = 50 - 20.57 = 29.43\ kw$$

$$\eta = \frac{29.43}{50} = 58.9\%$$

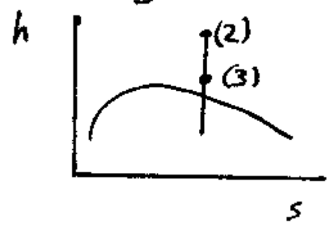
QS 2001



FROM TABLES

$h_2 = 3624$ kJ/kg

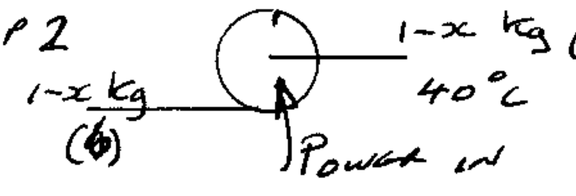
ASSUME IDEAL EXPANSION (2) to (3) FROM CHART $h_3 = 3100$ kJ/kg



KNOWN POINT AT (1) WATER 100b 200°C IDEALLY WE NEED WATER TABLES BUT AS THEY ARE NOT SUPPLIED $h_1 \simeq h_f$ @ 200°C

$h_1 \simeq 855$ kJ/kg

PUMP 2



SATURATED WATER $P_5 = P_s$ @ 40°C

$P_5 = 0.07375$ bar

$P_6 = 100$ bar

POWER INPUT $\simeq$ Vol $\times \Delta p$ Nominally $v = .001$ m³/kg

POWER INPUT $= 0.001 \times (100 - 0.07375) \times 10^5$

$= 10000$ J/kg or 10 kJ/kg

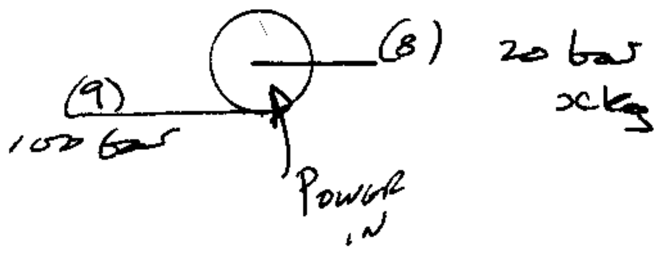
ENERGY BALANCE

$h_6 = h_5 + 10$ kJ/kg

$h_5 = h_f$ @ 40°C $= 167.5$ kJ/kg

$h_6 = 177.5$ kJ/kg

PUMP 1



POWER INPUT $= v\,\Delta p = 0.001\,(100-20)\times 10^5$

$= 8000\ \mathrm{J/kg}$

$= 8\ \mathrm{kJ/kg}$

ENERGY BALANCE $\quad h_9 = h_8 + 8$

$h_8 = h_f\ @\ 20\,\mathrm{bar} = 909\ \mathrm{kJ/kg}$

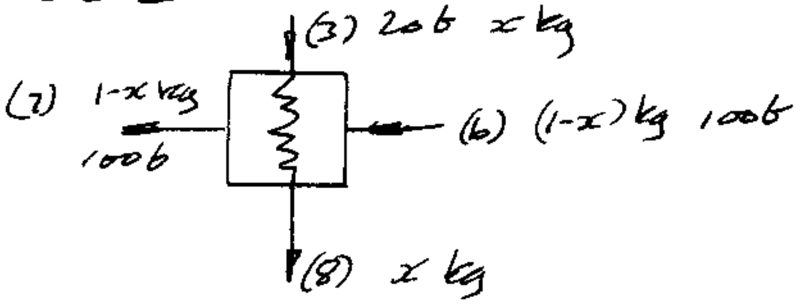
$h_9 = 909 + 8 = 917\ \mathrm{kJ/kg}$

FEED HEATER

$h_3 = 3100\ \mathrm{kJ/kg}$

$h_8 = h_f\ @\ 20\,\mathrm{b}$

$= 167.5\ \mathrm{kJ/kg}$



ENERGY BALANCE

$(1-x)\,h_6 + x\,h_3 = (1-x)\,h_7 + x\,h_8$

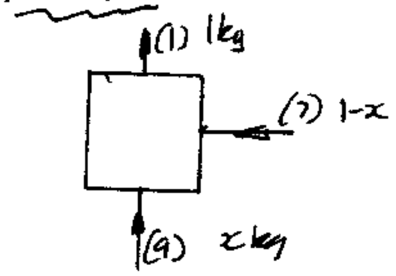
$(1-x)\,177.5 + 3100x = (1-x)\,h_7 + x\,(167.5)$

$177.5 - 177.5x + 3100x = (1-x)\,h_7 + 167.5x$

$177.5 + 2755x = (1-x)\,h_7$

$h_7 = \dfrac{177.5 + 2755x}{(1-x)}$

MIXER



$1\,h_1 = (1-x)\,h_7 + x\,h_9$

$855 = (1-x)\,h_7 + x\,h_9$

$855 = (1-x)\left\{\dfrac{177.5 + 2755x}{(1-x)}\right\} + 917x$

$855 = 177.5 + 2755x + 917x$

$677.5 = 3672x \qquad x = 0.184\ \mathrm{kg}$

PUMP 1 $\quad P = 8 \text{ kJ/kg}$

$= 8 \times 0.184 \text{ kW}$

$= 1.6 \text{ kW}$

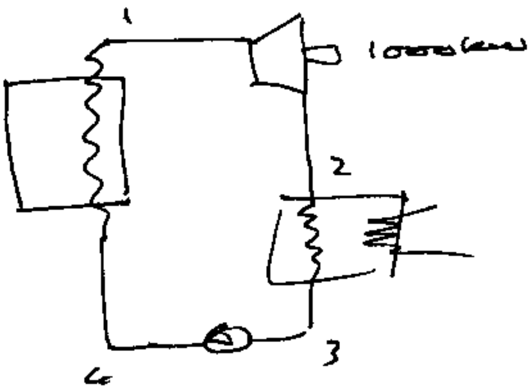
PUMP 2 $\quad P = 10 \text{ kJ/kg}$

$= 10 \times (1 - .184) \text{ kW}$

$= 8.16 \text{ kW}$

Based on 1 kg/s TOTAL FLOW

Q6 2001



$h_1 = h_g @ 27°C = 412.23 + \frac{2}{5}(414.74 - 412.23) = 413.23$
$p_1 = p_s @ 27°C = 6.525 + \frac{2}{5}(7.7 - 6.6525) = \underline{7.076}$

$p_3 = p_s @ 8°C = 3.4966 + \frac{3}{5}(4.1459 - 3.4966) = \underline{3.886 \text{ bar}}$

$h_3 = h_f @ 8°C = 206.75 + \frac{3}{5}(213.57 - 206.75) = \underline{210.84 \text{ kJ/kg}}$

$s_1 = s_g @ 27°C = 1.7158 + \frac{2}{5}(1.7142 - 1.7158) = \underline{1.715 \text{ kJ/kgK}}$

ISENTROPIC EXPANSION
$s_2 = s_1 = 1.7152 = s_f + x s_{fg} @ 8°C$

$s_f = 1.0243 + 3/5(1.0243 - 1.0486) = \underline{1.038} \text{ kJ/kgK}$
$s_g = 1.7238 + 3/5(1.7215 - 1.7238) = \underline{1.7224} \text{ kJ/kgK}$
$s_{fg} = 1.7224 - 1.038 = \underline{0.6837} \text{ kJ/kgK}$

$1.7152 = 1.038 + x \times .6837 \qquad x = \underline{0.989}$

$h_f = 206.75 + 3/5(213.57 - 206.75) = \underline{210.84} \text{ kJ/kg}$
$h_g = 401.33 + 3/5(404.16 - 401.33) = \underline{403.03} \text{ kJ/kg}$
$h_{fg} = 403.03 - 210.84 = \underline{192.19} \text{ kJ/kg}$

$h_2' = 210.84 + .989 \times 192.19 = \underline{400.91 \text{ kJ/kg}}$

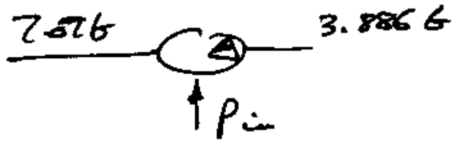
$h_2 = 413.23 + .9(413.23 - 400.91) = \underline{402.1 \text{ kJ/kg}}$

Q6 2001

$P_{out} = 1000\ kW = M_r(h_1 - h_2)$

$1000 = M_r(413.23 - 402.1) \qquad M_r = \underline{\underline{89.85\ kg/s}}$

Pump



$$P_{in} = \frac{\dot{m}\, v\, \Delta p}{\eta_{is}} = \frac{89.85 \times .0075\,(7.07 - 3.886) \times 10^5}{0.85}$$

$P_{in} = \underline{\underline{252.2}}\ kW$

$$h_4 = h_3 + \frac{252.2}{89.85} = 210.84 + 2.81 = \underline{\underline{213.6\ kJ/kg}}$$

$P_{nett} = 1000 - 252.2 = \underline{\underline{747.8}}\ kW$

$\Phi_{in} = M_r(h_1 - h_4) = 89.85(413.23 - 213.6)$

$\Phi_{in} = 17\,937\ kW$

$\eta_{th} = P_{nett} / \Phi_{in} = 5.57\ \%$

$\Phi_{out} = M_r(h_2 - h_3) = 89.85(402.1 - 210.84)$

$\Phi_{out} = 17185\ kW$

$\Phi_{nett} = 17\,937 - 17185 = 752\ kW$

i.e. ~~[illegible]~~

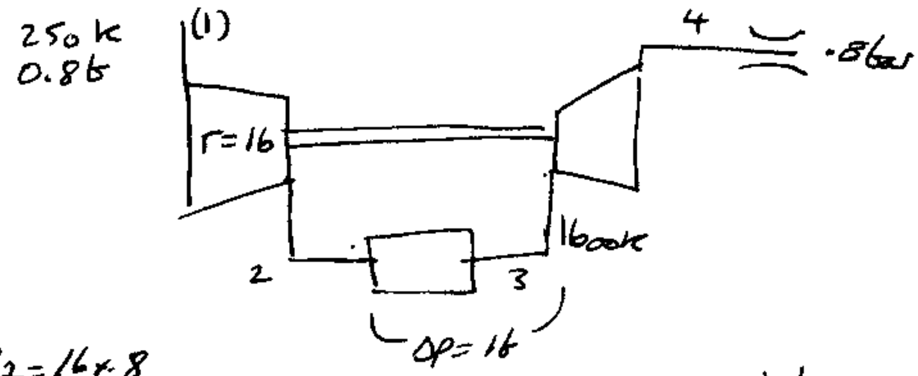
WATER
BOILER $17\,937 = M_w \times 4.2 \times (28 - 26)$

$M_w = \underline{\underline{2135\ kg/s}}$

COND $17185 = M_w \times 4.2 \times (9 - 5)$

$M_w = \underline{\underline{1023\ kg/s}}$

Q7 2001



$p_2 = 16 \times .8$

$p_2 = 12.8\text{bar}$ $\quad T_2' = 250 \times 16^{\frac{\gamma-1}{\gamma}}$ $\quad \gamma = 1.39$

$p_3 = 11.8\text{bar}$

$T_2' = 250 \times 16^{.280} = 544.2\text{K}$

$\eta_{is} = 0.88 = \dfrac{544.2 - 250}{T_2 - 250}$ $\quad T_2 = 584.3\text{K}$

$P(in) = M C_p \Delta T = 1 \times 1.03 (584.3 - 250)$

$P(in) = 344.3 \text{ kJ/kg}$

TURBINE $\quad P(out) = m C_p \Delta T = P_{in}$

$344.3 = 1 \times 1.19 \times (1600 - T_4)$ $\quad \underline{\underline{T_4 = 1310.67\text{K}}}$

THIS IS THE ACTUAL TEMP

$\eta_{is} = 0.9 = \dfrac{1600 - 1310.7}{1600 - T_4'}$ $\quad T_4' = 1278.6\text{K}$

$\dfrac{T_4'}{T_3} = \left(\dfrac{P_4}{11.8}\right)^{\frac{\gamma-1}{\gamma}} = \dfrac{1278.6}{1600} = 0.8 = \left(\dfrac{P_4}{11.8}\right)^{0.242}$

$P_4 = 11.8 \times 0.8^{1/.242} = 11.8 \times .398 = \underline{\underline{4.7\text{bar}}}$

NOZZLE $r = \frac{P_4}{P_1} = \frac{4.7}{0.8} = 5.875 = 0.17$

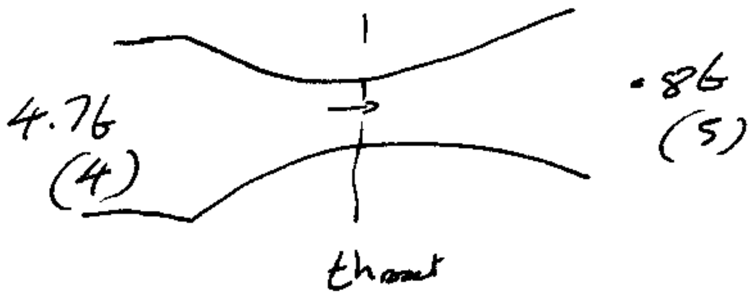
If the nozzle is chocked

$$r_c = \left(\frac{2}{\gamma+1}\right)^{\frac{\gamma}{\gamma-1}} = \left(\frac{2}{1.34+1}\right)^{\frac{1.34}{0.34}} = (.855)^{3.94}$$

$r_c = 0.5386 \qquad p = .5386 \times 4.7 =$

Hence nozzle is chocked

Exit velocity is sonic. Looks like a conv/div nozzle



$$C_p T_4 = C_p T_5 + \frac{u^2}{2}$$

$$T_5' = T_4 \left(\frac{4.7}{.8}\right)^{\frac{\gamma-1}{\gamma}} = 1310.67 \left(\frac{1}{5.875}\right)^{.254}$$

$$= 836.3\text{K}$$

$$\eta_{is} = 0.92 = \frac{1310.67 - T_5}{1310.67 - 836.3}$$

$T_5 = 874.3\text{K}$

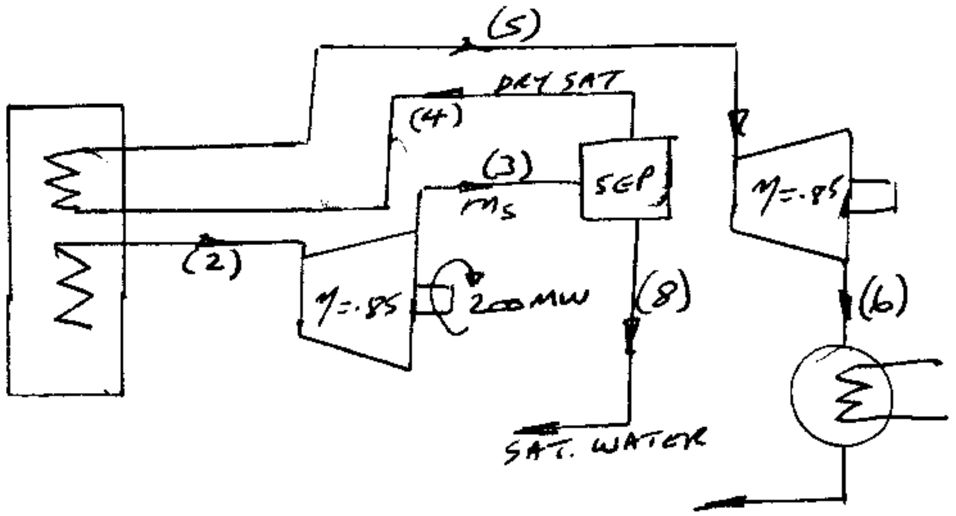
$$1.14 \times 10^3 \times 1310.67 = 1.14 \times 10^3 \times 874.3 + \frac{u^2}{2}$$

$u = 997.5$ m/s

Momentum Thrust

$$F = \dot{m}\,\Delta V$$
$$= 100 \times (997.5 - 250)$$
$$= 74.75\,\text{kN}$$

Q1 2002



FROM TABLES

POINT (2) $\quad h = h_g @ 55b = 2790 \text{ kJ/kg}$

<u>TURBINE</u> $\quad s = s_g @ 55b = 5.931 \text{ kJ/kg K}$

IDEAL EXPANSION 2→3'

$$S_2 = S_3' = s_f + x_{3'} s_{fg} @ 15\text{bar}$$

$$5.931 = 2.315 + x_3' \times 4.130 \qquad x_3' = 0.8755$$

$$h_3' = h_f + x h_{fg} @ 15\text{bar}$$

$$h_3' = 845 + 0.8755 \times 1947 = 2549.7 \text{ kJ/kg}$$

$$\text{ISENTROPIC EFFICIENCY} = 0.85 = \frac{2790 - \overset{h_3}{\cancel{2549.7}}}{2790 - 2549.7}$$

$$h_3 = \overset{2585.7}{\cancel{}} \text{ kJ/kg}$$

$$2585.7 = 845 + x_3 \times 1947 \qquad x_3 = \underline{0.894}$$

$$\text{POWER} = 200\,000 \text{ kW} = m_s (2790 - 2585.7)$$

$$M_s = 978.95 \text{ kg/s}$$

<u>SEPARATOR</u>

$$h_8 = h_f @ 15\text{bar} = 845 \text{ kJ/kg}$$

$$h_4 = h_g @ 15\text{ bar} = 1947 \text{ kJ/kg}$$

$$M_4 = 978.95 \times 0.894 = 875.2 \text{ kg/s}$$

$$M_8 = 103.8 \text{ kg/s}$$

2nd TURBINE

$h_5 = 3039$ kJ/kg $s_5 = 6.919$ kJ/kgK

$s_5 = s_6' = 0.832 + x_6' \, 7.075 = 6.919$ kJ/kgK

$x_6' = \dfrac{6.919 - 0.832}{7.075} = 0.860$

$h_6' = 251 + 0.860 \times 2358 = 2279.7$ kJ/kg

$\eta_{is} = 0.85 = \dfrac{3039 - h_6}{3039 - 2279.7}$ $h_6 = 2394$ kJ/kg

POWER

$P = \dot{m} \Delta h = 875.2 (3039 - 2394)$

$P = 564{,}848$ kW

$P = 564.8$ MW

TOTAL TURBINE POWER $= 764.8$ MW

CONDENSER

$\Phi_{out} = \dot{m} \Delta h = 875.2 (2394 - 251)$

$= 1875.6 \times 10^3$ kW

$= 1875.6$ MW

BOILER

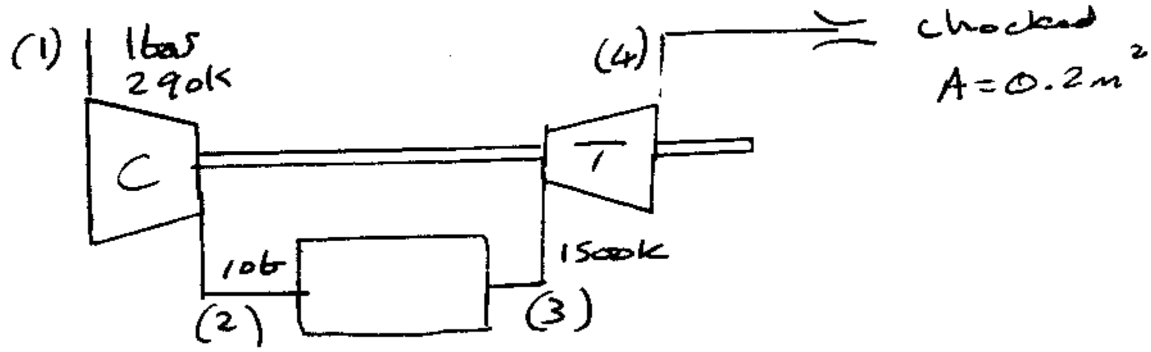
$\Phi_{in} = P + \Phi_{out} = 764.8 + 1875.6$

$\Phi_{in} = 2640.4$ MW

EFFICIENCY $\eta = \dfrac{P}{\Phi_{in}} = \dfrac{764.8}{2640.4} = 0.29$

$\eta = 29\%$

Q2 2002



$T_2' = T_1 \, r_p^{\frac{\gamma-1}{\gamma}} = 290 (10)^{0.286} = 560.3\,K$

$\eta_{is} = 0.9 = \dfrac{560.3 - 290}{T_2 - 290}$ $\quad T_2 = 590.3\,K$

<u>COMPRESSOR</u>

Power $= \dot m C_p \Delta T = \dot m C_p (590.3 - 290)$

$= 300.3\, \dot m C_p$

<u>TURBINE</u>

Power $= \dot m C_p \Delta T = \dot m C_p (1500 - T_4)$

EQUATE $\quad 300.3\, \cancel{\dot m}\cancel{C_p} = \cancel{\dot m}\cancel{C_p}(1500 - T_4)$

$T_4 = 1199.7\,K$

$\eta_{is} = 0.92 = \dfrac{1500 - 1199.7}{1500 - T_4'}$ $\quad T_4' = 1174\,K$

$T_4 = T_3 \, r_p^{-\left(\frac{\gamma-1}{\gamma}\right)}$ $\quad 1174 = 1500\, r_p^{-.286}$

$r_p^{-.286} = \dfrac{1174}{1500} = 0.782$ $\quad r_p = 2.35$

$2.35 = p_3/p_4 = 10/p_4$ $\quad p_4 = 10/2.35$

$p_4 = 4.245$ bar

CHOCKED NOZZLE

$T_5/T_4 = 2/(\gamma+1) = 0.833$ $\quad T_5 = 1000\,K$

IF CHOCKED, EXIT VELOCITY IS SONIC

$$\text{Velocity} = a = \sqrt{\gamma R T_s}$$

$$a = \sqrt{1.4 \times 287 \times 1000} = 633.8 \text{ m/s}$$

VOLUME FLOW RATE = AREA × VELOCITY

$$V = A a = 0.2 \text{m}^2 \times 633.8 \text{ m/s}$$

$$V = 126.77 \text{ m}^3/\text{s}$$

CRITICAL PRESSURE RATIO $r_c = \left(\frac{2}{\gamma+1}\right)^{\frac{\gamma}{\gamma-1}} = 0.528$

$$p_s = 0.528 \times 4.245$$

$$= 2.242 \text{ bar}$$

$$\text{MASS} = \frac{pV}{RT} = \frac{2.242 \times 10^5 \times 126.77}{287 \times 1000}$$

$$\text{MASS} = 99 \text{ kg/s}$$

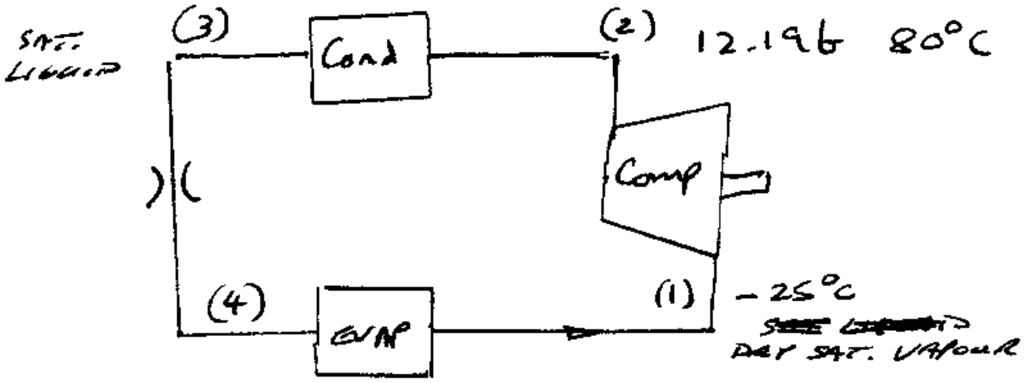
$$\text{THRUST} = m \, \Delta\text{velocity} + A \, \Delta p$$

$$= 99 \times (633.8 - 0) + 0.2(2.742 - 1) \times 10^5$$

$$= 62746 \text{ N} + 34840 \text{ N}$$

$$= \underline{\underline{97.6 \text{ kN}}}$$

Q4 2002



$P_1 = P_s$ @ $-25°C = 1.23765$
$h_1 = h_g$ @ $-25°C = 176.48$ kJ/kg
h_2 @ 12.196 80°C $t_s = 50°C$ so 30K SUPERHEAT
12.196 30K s.h. $h_2 = 230.33$ kJ/kg
$h_3 = h_f$ @ 12.196 $= 84.94$ kg/kg
$h_4 = h_3$ (THROTTLE)

Φ_{out} (CONDENSER) $= \Delta h = 230.33 - 84.94$
$\Phi_{out} = 145.39$ kJ/kg

Φ_{in} (EVAPORATOR) $= \Delta h = 176.48 - 84.94$
$\Phi_{out} = 91.54$ kJ/kg

COMPRESSOR POWER $= \Delta h = 230.33 - 176.48$
$P(in) = 53.85$ kJ/kg

C of P (REFRIG) $= \dfrac{91.54}{53.85} = \underline{1.7}$

$S_1 = 0.7127$ kJ/kgK
IDEALLY $S_2 = S_1$ AND CHECKING THIS MAKES THE VAPOUR SUPERHEATED AT (2)

CONSIDER THE IDEAL COMPRESSOR
$S_1 = S_2 = 0.7127$

FROM TABLES AT 12.19 bar 0.7127 PUTS IT BETWEEN S_g AND 15 K

S_g	θ	15 K SUPERHEAT
0.6797	0.7127	0.7166

LINEAR INTERPOLATION

$0.7127 - .6797 = 0.033$

$0.7166 - 0.6797 = 0.0369$

$$\theta = \frac{0.033}{0.0369} \times 15 = 13.4\text{ K}$$ FOR IDEAL COMPRESSION

SIMILARLY TO FIND IDEAL ENTHALPY

h_g	13.4 K	15 K
206.45	h	218.64

$218.64 - 206.45 = 12.19$

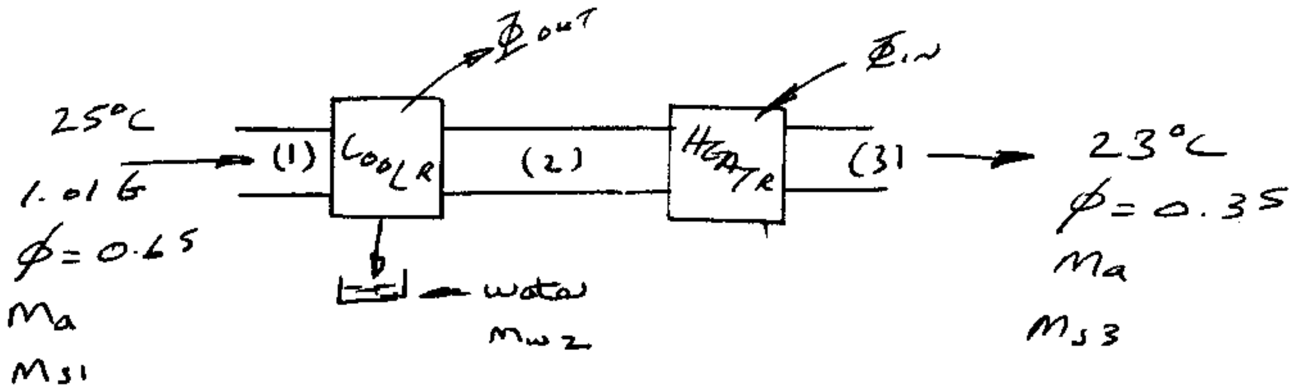
$$h - 206.45 = \frac{13.4}{15} \times 12.19 = 10.9 \qquad h = 217.3\text{ kJ/kg}$$

IDEAL $h_2 = 217.3$ kJ/kg

ACTUAL $h_2 = 230.33$

$$\eta_{is} = \frac{217.3 - 176.48}{230.33 - 176.48} = \underline{\underline{0.76}}$$

Q8 2002



m_a = MASS OF DRY AIR — CONSTANT THROUGHOUT

m_s = MASS OF VAPOUR $\quad$ m_w = MASS OF WATER

$p_g = 0.03166$ bar @ 25°C $\quad p_s = \phi p_g$

$p_{s1} = 0.65 \times 0.03166 = 0.020579$ bar

$p_a = 1.01 - p_{s1} = 0.989421$ bar

$\omega_1 = 0.622 \dfrac{p_{s1}}{p_a} = 0.012937$

$$m_a = pV/RT = \frac{0.989421 \times 10^5 \times V_A}{287 \times 298}$$

For 1 kg of DRY AIR $\quad V_A = 0.864 \text{ m}^3$

$m_{s1} = \dfrac{p_{s1} V}{RT}$ $\quad$ FOR VAPOUR VOLUME OF VAPOUR IS SAME AS VOL OF AIR

R = 462 FOR WATER VAPOUR

$$m_{s1} = \frac{0.020579 \times 10^5 \times 0.864}{462 \times 298} = 0.0129266 \text{ kg}$$

$\omega = m_s/m_a = 0.622\, p_s/p_a$

$\phi = p_s/p_g = 1.608\, p_a/p_g$

$p = 1.01$ b $\quad \theta = 25°C \quad \phi = 0.65$

INLET

@ 25°C $P_g = 0.03166$ bar

$\phi = P_s/P_g = 0.65$ $\quad P_s = 0.65 \times 0.03166$

$= 0.020579$ bar

$P_a = 1.01 - 0.020579 = 0.989421$ bar

$\omega = .622 \times \dfrac{0.020579}{0.989421} = 0.012937$

DEW POINT $\approx$ 17.8°C

EXIT $\quad \phi_3 = 0.35 = P_{s3}/P_{g3}$

$P_{g3} = P_s$ @ 23°C $= 0.02808$ bar

$P_{s3} = 0.02808 \times 0.35 = 0.009828$ bar

$P_{a3} = 1.01 - 0.009828 = 1.000172$ bar

$\omega_3 = 0.622\, P_s/P_a = 0.622 \times \dfrac{0.009828}{1.000172}$

$\omega_3 = 0.006111964$

$M_{s3} = 0.006112\, M_a$

$M_{s1} = 0.0129206\, M_a$

CONDENSATE FORMED $= M_{s1} - M_{s3} = 0.00681\, M_a$

ENERGY BALANCE ON COOLER

$P_{s2} = P_{s3} = 0.009828$ bar

$M_a C_a (T_{a1} - T_{a2}) - M_w C_w T_w + M_{s1} h_{s1}$

$- M_{s2} h_{s2} = Q_{out}$

h_{s1} @ 25°C 0.0206 bar $= 2550$ kJ/kg (Chart)

h_{s2} @ 17.8°C 0.00983 bar $= h_g = 2533$ kJ/kg

$M_{a1} = 1$ kg $\qquad C_a = 1.004$ kJ/kgK

$M_{s2} = M_{s3} \qquad C_w = 4.186$ kJ/kgK

$1 \times 1.004 (25 - 17.8) - 0.00681 \times 4.186 \times 17.8$

$+ 0.0129206 \times 2550 - 0.006112 \times 2533 = Q_{out}$

$Q_{out} = 24.187$ kJ FOR 1 kg OF DRY AIR

ENERGY BALANCE ON HEATER

$h_{s3} = 2545 \text{ kJ/kg} \quad (23°C \ 0.0098286)$

$$M_a C_a \Theta_3 + M_{s3} h_{s3} = M_a C_a \Theta_2 + M_{s2} h_{s2} + \Phi(in)$$

$$1 \times 1.004 \times 23 + 0.006112 \times 2545$$
$$= 1 \times 1.004 \times 17.8 + 0.006112 \times 2533 + \Phi(in)$$

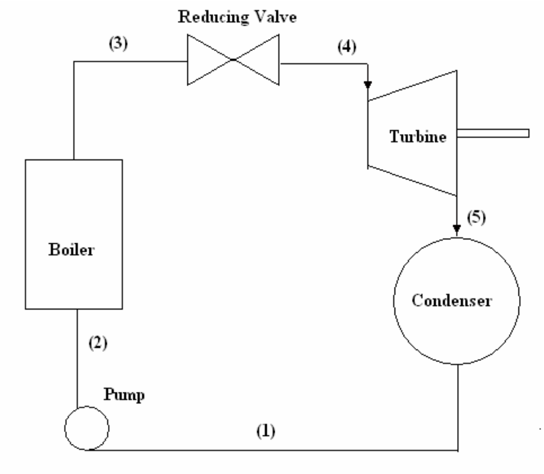
$\Phi(in) = 5.3$ kJ PER kg OF DRY AIR

$M_a = 1\text{kg}$ AT EXIT $\quad M = M_a + M_{s3}$

$$M = 1.006112 \text{ kg}$$

$$\left.\begin{aligned} \Phi(out) &= 24.04 \text{ kJ/kg} \\ \Phi(in) &= 5.27 \text{ kJ/kg} \end{aligned}\right\} \text{PER kg OF CONDITIONED AIR}$$

Q1 A schematic of a Rankine-cycle steam power plant is shown. This plant uses a boiling-water nuclear reactor as the heat source and a pressure reducing valve is located between the reactor and the turbine.



The water in the reactor is at a pressure of 7 MPa and leaves the reactor as superheated vapour at a temperature of 400°C. The pressure reducing valve lowers the steam pressure adiabatically by 2 MPa before it enters the steam turbine which has an isentropic efficiency of 80%. The steam expands through the turbine exiting at a pressure of 0.005 MPa and then is condensed at constant pressure before entering the feed-water pump. The condensate enters the feed-water pump at a pressure of 0.005 MPa and a temperature of 25°C. The pump has an isentropic efficiency of 90%. The water conditions at entry to the reactor are exactly the same as at exit from the pump and there are no pressure losses in the reactor. The net power output from the plant is 500 MW.
It may be assumed that there is no change in enthalpy across the pressure reducing valve, that is, $h_4 = h_3$.
(a) Sketch the temperature-entropy (T-s) diagram for the cycle.
(b) Determine the cycle efficiency, the mass flow rate of steam and the heat input to the boiling-water reactor.
Note. 1 bar = 10^5 N/m^2 = 10^5 Pa, and the specific heat capacity of water is 4.187 kJ/kgK.

SOLUTION

h_3 = 3158 kJ/kg (70 bar and 400°C)
h_4 = 3158 kJ/kg (50 bar)

Either by interpolation or by use of the h –s chart the temperature at point (4) is 387°C and the specific entropy is 6.592 kJ/kg K

Ideal conditions at point (5) $s_4 = s_5 = s_f + x\, s_{fg}$ at 0.05 bar
6.592 = 0.476 + 7.918x hence x = 0.772

$h_{5'} = h_f + x\, h_{fg}$ at 0.05 bar = 138 + 2423 x 0.772 = 2010 kJ/kg

Isentropic Efficiency $0.8 = \dfrac{3158 - h_5}{3158 - 2010}$ hence h_5 = 2239.6 kJ/kg

Power = 500 000 kW = m(3158-2239.6) hence m = 544.4 kg/s

Pump Ideal Power = V Δp

The volume of water is approximately 0.001 x 544.4 = 0.544 m^3/s

Pressure rise = 7 – 0.005 = 6.995 MPa Ideal Power = 6.995 x 10^6 x 0.544 = 3.8 MW

Actual Power = 3.8/0.9 = 4.228 MW Net Power = 500 – 4.228 = 495.772 MW

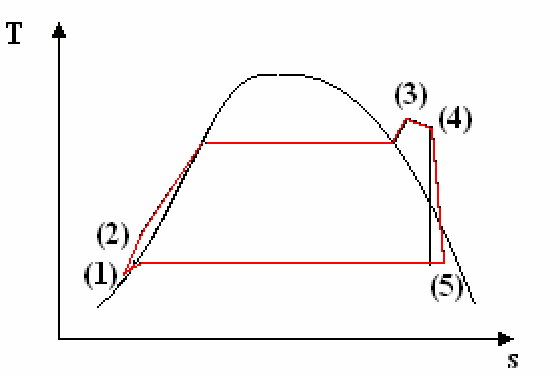
Energy added to water = 4.228/544.4 = 0.00777 MJ/kg or 7.77 kJ/kg

$h_1 = pv + mc\theta = 0.005 \times 10^6 \times 0.001 + 1 \times 4187 \times 25 = 5 + 104675 = 104680$ J/kg

$h_2 = 104.68 + 7.77 = 112.45$ kJ/k

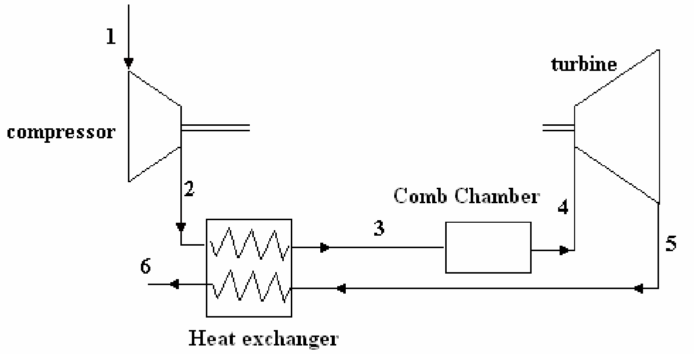
Φ(in) to boiler = $m(h_3-h_2) = 544.4(3158 - 112.45) = 1658000$ kW or 1658 MW

Cycle Efficiency = 495.772/1658 = 0.299 or 29.9%



On the T –s diagram the water is under-cooled at (1)

Q2 A schematic of a regenerative gas turbine is shown. Air ($\gamma = 1.4$) enters the compressor at a pressure of 1 bar and a temperature of 20°C. The compressor has an isentropic efficiency of 85% and a pressure ratio of 10:1. The expansion process in the turbine is polytropic, that is pv^n = constant, with n = 1.35. The plant exhaust gas temperature, that is point 6, is 20°C higher than that at the compressor outlet.



Assume that $p_6 = p_5 = p_1$ = 1 bar, T_4 = 1000°C and the specific heat capacity is constant throughout the cycle with C_P = 1.005 kJ/kgK.
(a) Sketch the T-s diagram for the cycle illustrating the regenerative heat exchange process.
(b) Calculate,
(i) the heat transfer in the heat exchanger
(ii) the heat supplied in the combustion chamber
(iii) the cycle efficiency.

SOLUTION

$$T_{2'} = T_1\left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}} = 293(10)^{\frac{1.4-1}{1.4}} = 565.7\text{ K}$$

$$\eta_{IS} = 0.85 = \frac{T_{2'} - T_1}{T_2 - T_1} = \frac{565.7 - 293}{T_2 - 293} \quad T_2 = 613.8\text{ K}$$

$T_5 = T_4/r_p^{(1-1/n)} = 1273/(10)^{0.259} = 730$ K

Heat Exchanger with same specific heat and mass flow at all points

$T_6 = T_2 + 20 = 633.8$ K

$(T_3 - T_2) = (T_5 - T_6)$ $\quad T_3 = T_5 - T_6 + T_2 = 730 - 633.8 + 613.8 = 710$ K

It willbe assumed that m = 1 kg throughout

HEAT EXCHANGER

Heat Transfer = m c_p ($T_3 - T_2$) = 1 x 1.005 x (710 - 613.6) = 99.75 kJ/kg

COMBUSTION CHAMBER

Q(in) = m c_p (T_4-T_3) = 1 x 1.005 (1273-710) = 565.8 kJ/kg

The main problem here is the turbine has a heat loss since the expansion is polytropic and we either need to find the heat loss or the power output in order to find the cycle efficiency.

For a steady flow process the work done is :

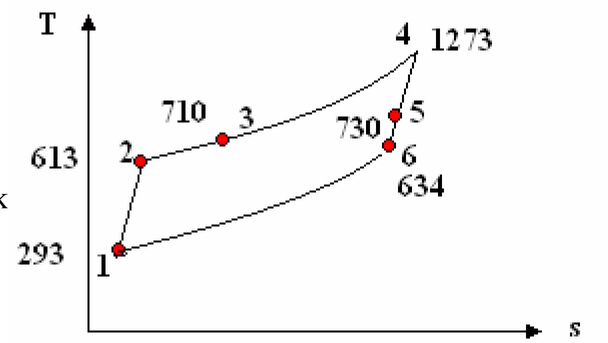
$$W(out) = \frac{mR}{n-1}(\Delta T) = \frac{1 \times 0.287}{1.35-1}(1273 - 730) = 445.36\text{ kJ/k}$$

(Turbine)

W(in)= mc_p(T_2-T_1) = 1 x 1.005 (613.8 - 293) = 322.4 kJ/kg (Compressor)

W(nett) = W(out) - W(in) = 123 kJ/kg

η_{th} = W(nett)/Q(in) = 123/565.8 = **0.22 or 22%**



Q3 A gaseous fuel has the following percentage composition by volume:
CO 13%, H_2 42%, CH_4 25%, O_2 2%, CO_2 3%, N_2 15%

Determine the wet and dry volumetric and gravimetric analyses of the products of combustion if 15% excess air is used. State all assumptions made and take air as 21% O_2 and 79% N_2 by volume. The relative atomic masses are hydrogen l, carbon 12, nitrogen 14 and oxygen 16.

VOLUMETRIC

CARBON MONOXIDE
$2CO + O_2 \rightarrow 2CO_2$
$2\ m^3 + 1\ m^3 \rightarrow 2\ m^3$
$0.13\ m^3 + 0.065\ m^3 \rightarrow 0.13\ m^3$
HYDROGEN
$2H_2 + O_2 \rightarrow 2H_2O$
$2\ m^3 + 1\ m^3 \rightarrow 2\ m^3$
$0.42\ m^3 + 0.21\ m^3 \rightarrow 0.42\ m^3$
METHANE
$CH_4 + 2O_2 \rightarrow 2H_2O + CO_2$
$1\ m^3 + 2\ m^3 \rightarrow 2\ m^3 + 1\ m^3$
$0.25\ m^3 + 0.5\ m^3 \rightarrow 0.5\ m^3 + 0.25\ m^3$
Total oxygen required is 0.065 + 0.21 + 0.5 – 0.02 = 0.755 m^3
Air required = 0.755/0.21 = 3.595 m^3
Air supplied = 3.595 x 1.15 = 4.135

PRODUCTS			WET	DRY
H_2O	0.42 + 0.5 =	0.920 m^3	18.9%	0
O_2	0.21 x 4.135 – 0.755 =	0.113 m^3	2.3%	2.9%
N_2	0.79 x 4.135 + 0.15 =	3.417 m^3	70.3%	86.7%
CO_2	0.13 + 0.25 + 0.03 =	0.410 m^3	8.4%	10.4
Total		4.86/3.94	100%	100

GRAVIMETRIC

We convert volumes to masses using the formula $\dfrac{m_i}{m} = \dfrac{(V_i / V)\tilde{m}_i}{\sum\{(V_i / V)\tilde{m}\}_i}$

				WET
i	V_i/V	$\tilde{m}_i$	$(V_i/V)\,\tilde{m}_i$	$\tilde{m}_i / m$
H_2O	0.189	18	3.40	12.3%
O_2	0.023	32	0.74	2.7%
N_2	0.703	28	19.7	71.5%
CO_2	0.084	44	3.7	13.4%
Total	1.0		27.54	100

				DRY
i	V_i/V	$\tilde{m}_i$	$(V_i/V)\,\tilde{m}_i$	$\tilde{m}_i / m$
O_2	0.029	32	0.928	3.1%
N_2	0.867	28	24.276	81.5%
CO_2	0.104	44	4.576	15.4%
Total	1.0		29.78	100

Q4 Sketch a pressure-volume diagram for the air-standard dual combustion cycle and describe the processes which occur in each part of the cycle.

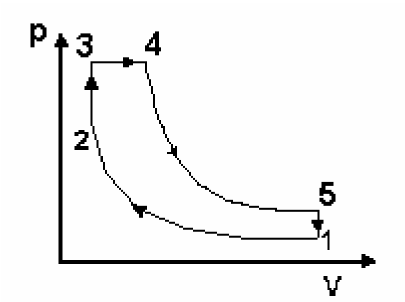
In an air-standard dual combustion cycle, the temperature and pressure at the start of compression are 300 K and 1 bar respectively. The energy added in the cycle is 1600 kJ/kg, of which three-quarters is added at the constant volume and the remainder at the constant pressure parts of the cycle. The compression ratio is 20:1 and the compression and expansion strokes are polytropic with polytropic indices of n_c = 1.45 and n_e = 1.35 respectively.

Determine:
(a) the maximum pressure in the cycle
(b) the maximum temperature in the cycle
(c) the cycle efficiency
(d) the mean effective pressure.

Assume that c_v = 0.718 kJ/kgK, c_p = 1.005 kJ/kgK and R = 0.287 kJ/kgK and all remain constant throughout the cycle.

Comment – If the compression and expansion are not adiabatic, the cycle can not be an air standard cycle. The air standard efficiency formula cannot be used in this case.

The processes are as follows.



1 - 2 reversible (polytropic??) compression.
2 - 3 constant volume heating.
3 - 4 constant pressure heating.
4 - 1 reversible (polytropic??) expansion.
5 - 1 constant volume cooling.

T_1 = 300 K p_1 = 1 bar
$V_1/V_2 = 20$
$T_2 = 300 \times 20^{n-1} = 300 \times 20^{1.45-1} = 1155$ K
$p_2 = p_1 r^n = 1 \times 20^{1.45} = 77$ bar

Heat Input at constant Volume is 0.75 x 1600 = 1200 kJ/kg
$1200 = mc_v(T_3-T_2) = 1 \times 0.718 \times (T_3 - 1155)$ $T_3 = 2826.3$ K

Heat Input at constant Pressure is 0.25 x 1600 = 400 kJ/kg

$400 = mc_p(T_4-T_3) = 1 \times 1.005 \times (T_4 - 2826.3)$ $T_4 = 3224.3$ K
This is the maximum temperature in the cycle.

$$p_3 = \frac{p_1 V_1 T_3}{V_3 T_1} = \frac{1 \times 20 \times 2826.4}{1 \times 300} = 188.42 \text{ bar}$$

p_4 = 188.42 bar This is the highest pressure in the cycle.

$$\frac{V_1}{V_4} = \frac{p_4 T_1}{p_1 T_4} = \frac{188.42 \times 300}{1 \times 3224.3} = 17.53/1 = \frac{V_5}{V_4}$$

$$p_4 V_4^n = p_5 V_5^n \qquad p_5 = p_4\left(\frac{V_4}{V_5}\right)^n = 188.42\left(\frac{1}{17.53}\right)^{1.35} = 3.95 \text{ bar}$$

$$\frac{p_5}{T_5} = \frac{p_1}{T_1} \qquad T_5 = \frac{p_5 T_4}{p_1} = \frac{3.95 \times 300}{1} = 1185 \text{ K}$$

The problem now is that because the work processes are polytropic, there is a heat transfer in these processes that makes it difficult to determine the heat rejected so we need to find the net work done. This involves a lot more work and I wonder if this is what the examiner intended?

Finding the true net work would require the work laws to be applied

COMPRESSION

$$W = \frac{p_2V_2 - p_1V_1}{n-1} = \frac{mR(T_1 - T_2)}{n-1} = \frac{1x287(300 - 1155)}{0.45} = -545.3 \times 10^3 \text{ J/kg}$$

EXPANSION

$$W = \frac{p_4V_4 - p_5V_5}{n-1} = \frac{mR(T_4 - T_5)}{n-1} = \frac{1 \times 287(3224.3 - 1185)}{0.35} = 1772.2 \times 10^3 \text{ J/kg}$$

There is also work in the constant pressure process

$$W = p_3(V_4 - V_3) = mR(T_4 - T_3) = 1 \times 287(3224.3 - 2826.3) = 114.2 \times 10^3 \text{ J/kg}$$

Net Work = 114.2 + 1772.2 – 545.3 = 1341.1 kJ/kg

η = 1341.1/1600 = 83.8%

$V_1 = mRT_1/p_1 = 1 \times 287 \times 300/(1 \times 10^5) = 0.861 \text{ m}^3$ (based on 1 kg)
$V_2 = V_1/20 = 0.04305 \text{ m}^3$ (based on 1 kg)

MEP = W(net)/Swept Volume = W(net)/($V_1 - V_2$) = 1341.1 x 10^3/(0.861 – 0.04305) = 1.64 x 10^6 Pa

This seems extremely high if anyone finds any errors in this work please contact admin@freestudy.co.uk

THERMODYNAMICS 201 2003

5 A vapour compression refrigerator uses refrigerant 12 as the working fluid and operates between temperature limits of -10°C and 60°C.

(a) Sketch the flow diagram, indicating the components of the refrigeration cycle.

(b) If the refrigerant entering the compressor is dry saturated sketch the temperature-entropy (T-s) and the pressure-enthalpy (p-h) diagrams for the two following cases;

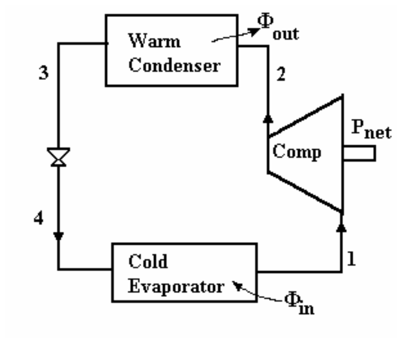
(i) the refrigerant leaves the condenser saturated

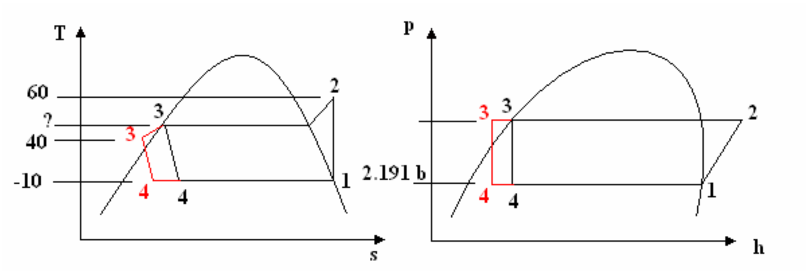
(ii) the refrigerant is sub-cooled to 40°C before entry to the throttle valve.

(c) For the case in which the refrigerant leaves the condenser and enters the throttle valve as saturated liquid and assuming isentropic processes for the compressor determine:

(i) the refrigeration effect

(ii) the coefficient of performance.





The red lines show the difference when under cooled.

The major trap to fall into here is the maximum operating temperature is not the same as the condenser temperature. Without a p - h chart this seems very difficult. If anyone knows how to complete this correctly please contact admin@freestudy.co.uk

h_1 = 183.19 kJ/kg s_1 = 0.7020 kJ/kg K

$p_1 = p_s$ at -10°C = 2.191 bar v_1 = 0.0766 m^3/kg T_1 = 263 K T_2 = 333 K

Assuming the compression is reversible and adiabatic $s_1 = s_2$. but this does not help. Clearly the refrigerant is superheated at exit from the compressor.

On the row for 60°C in the tables, s_2 = 0.7020 kJ/kg K occurs between 0 and 15 K of superheat so interpolation is needed. Using the data on 60°C row of the tables we find:

	Sat.	θ	15K
s	0.6765	0.7020	0.7146
h	209.26	h_2	222.23

$$\frac{0.7020-0.6765}{0.7146-0.6765} = 0.66929 = \frac{\theta - 0}{15-0}$$

θ = 10K so the actual saturation temperature is around 50°C

Now find the values using the 50°C row at 10 K superheat

	Sat.	10K	15K
s	0.6797	s_2	0.7166
h	206.45	h_2	218.64

$$\frac{s_2 - 0.6797}{0.7166-0.6797} = \frac{10}{15}$$

s_2 = 0.7043 kJ/kg K this is close so we will use this temperature.

$$\frac{h_2 - 206.45}{218.64 - 206.45} = \frac{10}{15} \qquad h_2 = 214.6 \text{ kJ/kg}$$

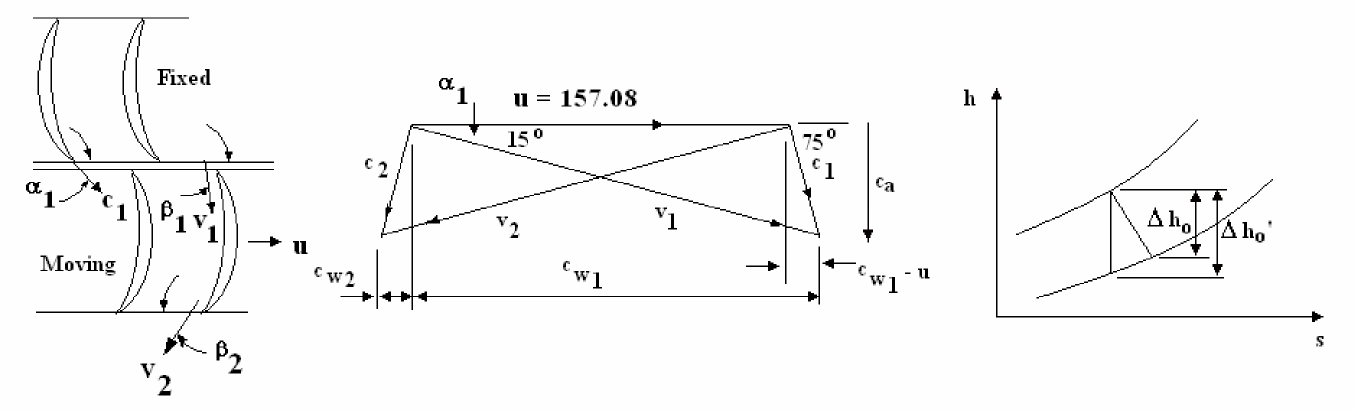
$h_3 = h_f$ at 60^oC= 95.74 kJ/kg $\qquad h_4 = h_3$

$\Phi(in) = h_1 - h_4$ = 87.45 kJ/kg = Refrigeration Effect
$P(in) = h_2 - h_1$ = 31.39 kJ/kg
C of P (refrigerator) = 87.45/31.39 = 2.8
$\Phi(out) = h_2 - h_3$ =118.56 kJ/kg
C of P (Heat Pump) = 118.56/31.39 = 3.8

Q.7 Fifteen successive stages of an axial-flow reaction steam turbine have blades with constant inlet and outlet angles of 15° and 75° respectively. The mean diameter of the blade rows is 1.0 m and the speed of rotation is 50 rev/s. The axial velocity is constant throughout the stages. The steam inlet conditions to the turbine are 15 bar and 300°C and the outlet pressure is 0.24 bar.

Determine:
(a) all relevant blade and steam velocities and sketch the velocity diagram
(b) the specific enthalpy drop per stage
(c) the overall efficiency of the turbine.

If there is a reheat factor between each turbine stage of 1.03 determine the stage efficiency.
Note. As there is constant axial velocity and all blades are of the same geometry kinetic energy can be ignored.



$u = \pi ND = \pi \times 50 \times 1 = 157.08$ m/s
$\tan\alpha_1 = c_a/c_{w1}$
$c_{w1}\tan 15 = (c_{w1} - u) \tan 75 \quad 0.269\, c_{w1} = 3.732(c_{w1} - 157.08)$
$0.269\, c_{w1} = 3.732c_{w1} - 586.23$
$586.23 = 3.463\, c_{w1}$
$c_{w1} = 169.28$ m/s
$c_{w2} = c_{w1} - u = 169.28 - 157.08 = 12.2$ m/s
$c_a = c_{w2} \tan\beta_2 = 12.2 \tan 75 = 45.55$ m/s
$\Delta c_w = 169.28 + 12.2 = 181.5$ m/s

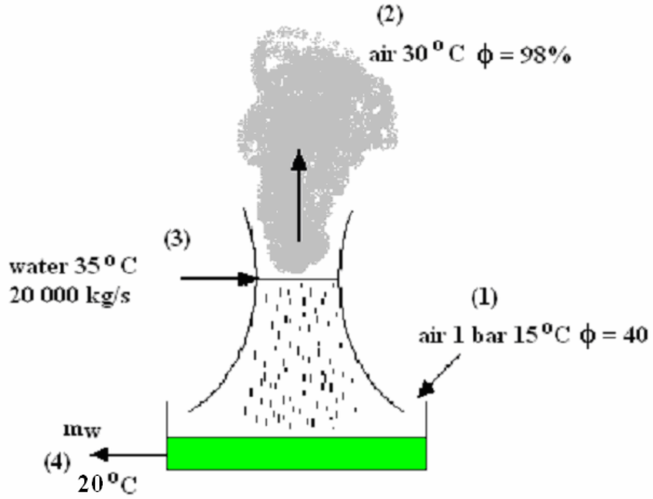
Stage enthalpy change $\Delta h_s = u\, \Delta c_w = 157.08 \times 181.5 = 28507$ J/kg
For 15 stages $\Delta h_o = 15 \times 28.507 = 427.6$ kJ/k

$h_1 = 3039$ kJ/kg $\qquad s_1 = 6.919$ kJ/kg K
$s_1 = s_2 = s_f + x_{sfg}$ at 0.24 bar
$6.919 = 0.882 + 6.962\, x \qquad x = 0.867$
$h_2 = h_f + h_{sfg}$ at 0.24 bar
$h_2 = 268 + (2348)(0.867) = 2304$ kJ/kg
Ideal enthalpy drop = 3039 – 2304 = 735 kJ/kg
Overall Efficiency $\eta_o = 427.6/735 = 58.2\%$
$\eta_o = \eta_s \times$ Reheat Factor
$0.582 = \eta_s \times 1.03$
$\eta_s = 0.565$ or 56.5 %

THERMODYNAMICS 201 2003

Q8 The water-flow rate from the condenser of a 500 MW power plant is 20 x 10^3 kg/s. The water is cooled in an array of cooling towers from a temperature of 35°C to 20°C. Atmospheric air at a pressure of 1 bar enters the towers at 15°C with a relative humidity of 40% and exits with a temperature of 30°C at 98% relative humidity.

Determine the make-up water required and the air-flow rate.
Assume that the specific heat capacity at constant pressure for air and steam are 1.005 kJlkgK and 1.86 kJ/kgK respectively and the specific heat capacity for water is 4.187 kJ/kgK.



INLET AIR

p_{g1} = 0.01704 bar at 15°C

$\phi 1 = 0.4 = ps1 / pg$

$ps1 = 0.4 \times 0.01704 = 0.006816$ bar

hence $p_{a1} = 1.0 - 0.006816 = 0.993184$ bar

$$\omega_1 = 0.622 \frac{0.006816}{0.993184} = 0.004268647$$

$m_{s1} = 0.004268647\ m_a$

OUTLET AIR

$\phi 2 = 0.98$

$ps2 = 0.98pg2 = 0.98 \times\ 0.0424242 = 0.041575716$

bar hence $p_{a2} = 0.95842428$ bar

$$\omega_2 = 0.622 \frac{0.00415757}{0.9584242} = 0.021698$$

$m_{s2} = 0.021698\ m_a$

MASS BALANCE

$m_{w4} = m_{w3} - (m_{s2} - m_{s1}) = 20000 - (0.021698\ m_a - 0.0042686\ m_a) = 20000 - 0.017429\ m_a$

ENERGY BALANCE

$h_{s2} = h_g = 2555.7$ kJ/kg

$h_{s1} = 2530$ kJ/kg (from h-s chart)

Balancing energy we get

$(20000 \times 4.86 \times 35) + (m_a \times 1.005 \times 15) + (0.0042686 \times m_a \times 2530) =$

$\{(20000 - 0.017429\ m_a) \times 4.186 \times 20\} + (0.021698 \times 2555.7\ m_a) + (m_a \times 1.005 \times 30)$

$3402000 + 15.075\ m_a + 10.8\ m_a = 1674400 - 1.459\ m_a + 70.4\ m_a + 30.15\ m_a$

$1727600 = 73.216\ m_a$

$m_a = 23596$ kg/s

$m_{s2} = 512$ kg/s

$m_{s1} = 100.72$ kg/s

Evaporation rate is 411.3 kg/s so this is the required make up water

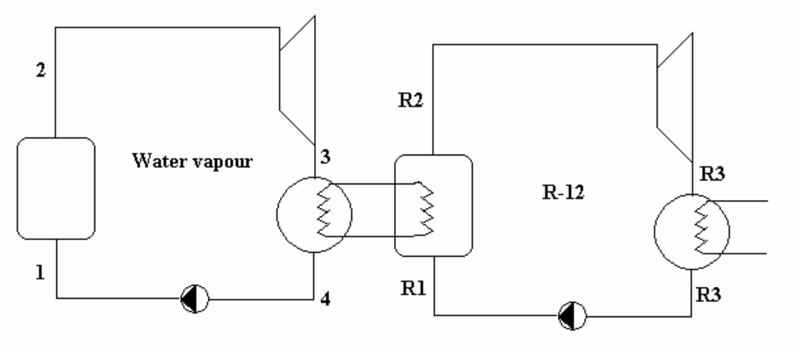
THERMODYNAMICS 201 2004

Q1 A steam power plant operates on the Rankine cycle. The high pressure steam is at 60 bar and 500°C at entry to the turbine. The turbine produces 20 MW of power. The condenser pressure is 2 bar.

During day time operation the waste heat from the condenser is used for process heating. During night time operation the waste heat is used in a R-12 power plant that also operates on the Rankine cycle. The refrigerant cycle uses vapour with no superheat at 80°C at entry to the turbine and condenses at 10°C.

Assuming no heat losses and negligible power usage at the pumps, calculate the power output from the R-12 cycle and the thermal efficiency of the plant. The isentropic efficiency of both turbines is 85%. (This question very similar to Q1 1997)

SOLUTION



WATER/VAPOUR CYCLE.

$h_4 = h_f$ @ 2 bar= 505 kJ/kg $\quad h_1 = h_4 = 505$ kJ/kg

$h_2 = h$ @ 60 bar and 500°C = 3421 kJ/kg

$s_2 = s$ @ 60 bar and 500°C = 6.879 kJ/kg K

$s_{3'} = s_2 = 6.879$ kJ/kg K = $s_f + x\, s_{fg}$ @ 2bar

6.879 = 1.530+ 5.597x $\quad$ x = 0.9557

$h_{3'} = h_f + x\, h_{fg}$ @ 2bar = 505 + 0.9557 x 2202 = 2609.4 kJ/kg

Power out = 20 000 kW = m_s x η_I (3421 - 2609.4)

20 000 = m_s x 0.85 (3421 - 2609.4)

m_s = 20 000/689.8 = 29 kg/s

We need to find h_3. $\quad \dfrac{3421 - h_3}{3421 - 2609.4} = 0.85 \quad h_3 = 2731$ kJ/kg

Check Power out = 29(3421 - 2731) = 20 000 kW

Heat lost from the condenser = $29(h_3 - h_4)$ = 29(2731 – 505) = 64554 kW

This becomes the heat input to the evaporator in the R-12 cycle.

R-12 CYCLE

$h_{R2} = h_g$ at 80°C = 212.83 kJ/kg

$h_{R1} = h_{R4} = h_f$ @ 10°C = 45.37 kJ/kg

Φ(in) = 64554 = m_R(212.83 - 45.37) $\quad m_R$ = 385.48 kg/s

$s_{R2} = s_g$ at 80°C = 0.6673 kJ/kg K = $s_{R3} = s_f + x\, s_{fg}$ @ 10°C = 0.1752 +x(0.6921 - 0.1752)

x = (0.6673 - 0.1752)/0.5169 = 0.952

$h_{R3} = h_f + x\, h_{fg}$ @ 10°C = 45.37 + 0.952(191.74 - 45.37) = 184.72 kJ/kg

Power output = m_R (h_{R2} - h_{R3}) = 385.48 (212.83 - 184.72) = 10 837 kW

Thermal efficiency P(out)/ Φ(in) = 10 837/ 64554 = 0.168 or 16.8%

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.

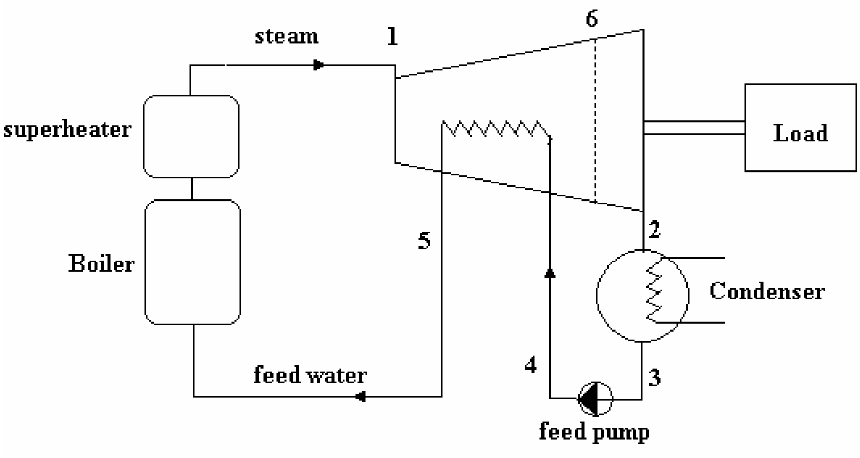
(c) Determine the temperature of the steam at (6), the dryness fraction and enthalpy.

(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.

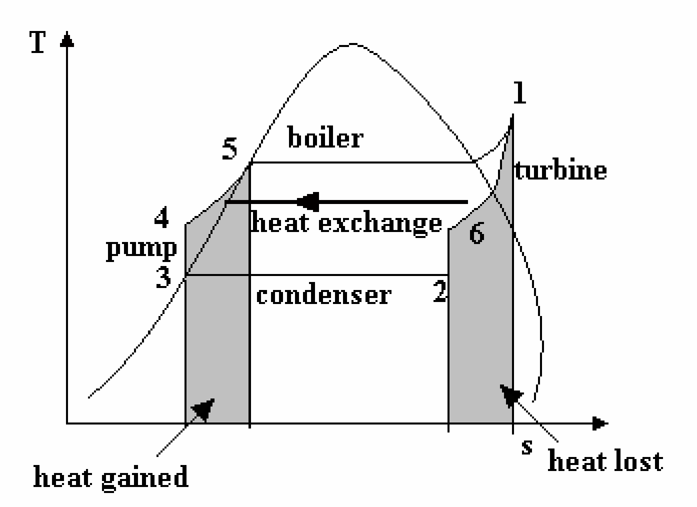
COMMENT

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

$\eta = 40\% = W(nett)/Q(in)$

NETT WORK

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

PUMP

Work input = volume x $\Delta p = 0.001 \text{ m}^3/\text{kg} \times (200 - 0.05) \times 10^5 = 19995$ J/kg or 20 kJ/kg

Pump work = 20 kJ/kg = $c\,\Delta\theta$ $\Delta\theta = 20/4.187 = 4.8$ K

$\theta_3 = t_s$ @ 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 $\eta = 1 - Q(out)/Q(in) = 1 - 1026/1710 = 40\%$

$h_2 = h_3 + Q(out) = 138 + 1026 = 1164$ 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.

$\theta_4 = 32.9 + 4.8 = 37.7\,°C$

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

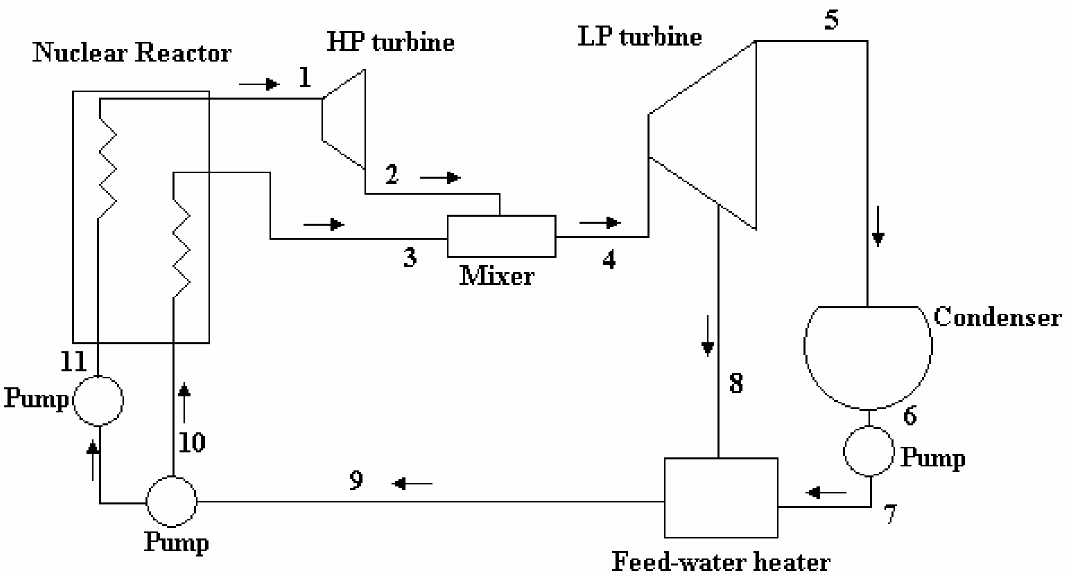
Q3 In a water-cooled nuclear reactor the coolant water to the reactor is divided into high-pressure and low-pressure circuits. The high-pressure circuit generates 200 kg/s of steam at 100 bar and 500 °C. The low-pressure circuit generates 100 kg/s of dry saturated steam at 30 bar. A line diagram of the plant is shown.

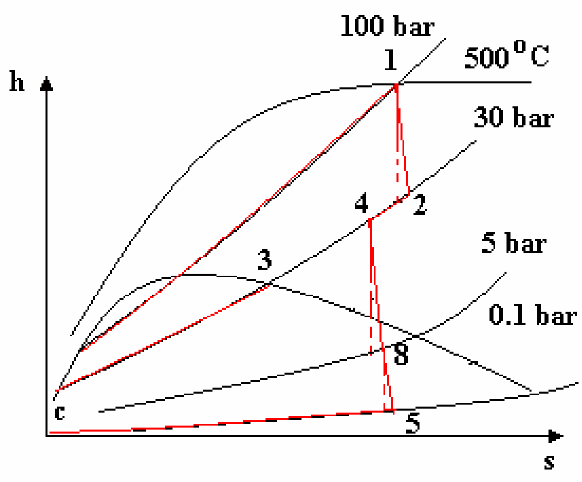
The high-pressure steam expands in a high-pressure turbine to 30 bar with an isentropic efficiency of 90%, and the exhaust is mixed adiabatically with the low-pressure steam all of which is then expanded in a low-pressure turbine to 0.10 bar with an isentropic efficiency of 92%. The optimum quantity of dry saturated steam is bled at 5 bar from the low-pressure turbine into an open-type feed-water heater positioned prior to the separation into the two coolant-water circuits.

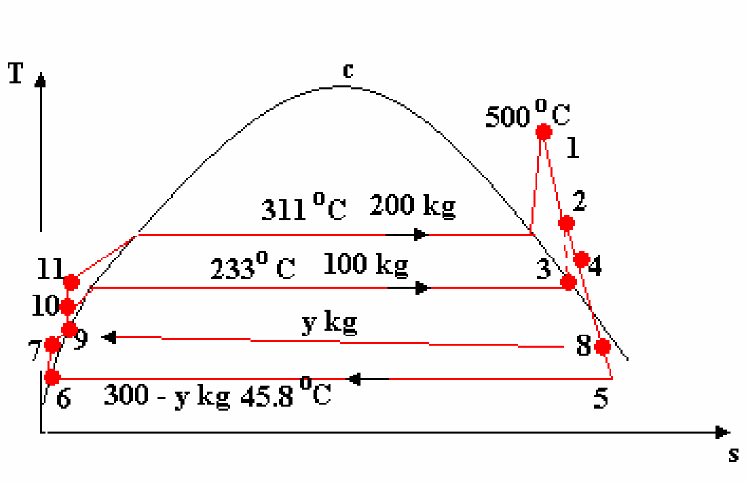
(a) Sketch the T-s and h-s diagrams for the cycle.

(b) Calculate the power developed and the cycle efficiency.

Neglect the feed-pumps work, and assume a straight line of condition for the low-pressure turbine.







Start with known points.

Point 1	100 bar	500^oC	h = 3373 kJ/kg	s = 6.596 kJ/kg K
Point 2	30 bar			
Point 3	30 bar	dss	h = 2803 kJ/kg	s = 6.186 kJ/kg K
Point 4	30 bar			
Point 5	0.1 bar			
Point 6	0.1 bar	sw	h = 192 kJ/kg (assumed to be saturated water in absence of information)	
Point 8	5 bar			
Point 9	5 bar	sw	h = 640 kJ/kg (assumed to be saturated water in absence of information)	

HP Turbine m = 200 kg/s

Ideal expansion $s_2 = s_1 = 6.596$ From h – s chart the steam is superheated at 30 bar and 310^oC

$h_{2'} = 3020$ kJ/kg

$\eta = 0.9 = \dfrac{3373 - h_2}{3373 - 3020}$ $h_2 = 3055.3$ kJ/kg – the actual enthalpy

Power output = $200(h_1 - h_2) = 63\,540$ kW

MIXING $200\, h_2 + 100\, h_3 = 300\, h_4$ 200(3055.3) + 100(2803) = 891360 = $300\, h_4$

$h_4 = 2971.2$ kJ/kg

LP TURBINE

First expansion to 5 bar

Point 4 30 bar $h_4 = 2971.2$ kJ/kg Locate on h – s chart and find h_8'=2620 kJ/kg

$\eta = 0.92 = \dfrac{2971.2 - h_8}{2971.2 - 2620}$ $h_8 = 2648.1$ kJ/kg

Power out = 300 (2971.2 - 2648.1) = 96931.2 kW

Expansion to 0.1 bar

Locate point 8 and then point '5 h_5' = 2090 kJ/kg

$\eta = 0.92 = \dfrac{2648.1 - h_5}{2648.1 - 2090}$ $h_5 = 2134.6$ kJ/kg

Power out = m(2648.1 - 2134.6) = 513.45 m kW m = mass flowing to condenser.

FEED HEATER

$y\, h_8 + (300 - y)\, h_7 = 300\, h_9$ y = mass bled at 5 bar

$h_6 = h_7 = 192$ kJ/kg

y 2648.1 + (300 –y) 192 = 300 x 640

2648.1y + 57600 – 192 y = 192000

2456.1 y = 134400 y = 54.72 kg/s

m = 300 – 54.72 = 245.28 Power out of second part of expansion 513.45 m = 125938.6 kW

Total power from LP turbine = 96931.2 + 125938.6 = 222869.7 kW

Total power out from both turbines = 222869.7 + 63 540 = 286409.7 kW say 286.41 MW

BOILER

$\Phi(in) = 200(h_1 - h_{11}) + 100(h_3 - h_{10})$ $h_{11} = h_{10} = h_9 = 640$ kJ/kg

Φ(in)= 200(3373 - 640) + 100 (2803 – 640) = 762900 kW say 762.9 MW

CONDENSER

$\Phi(out) = (300 - 54.72)(h_5 - h_6) = (300 - 54.72)(2134.6 - 192) = 476481$ kW

Check P = Φ(in)- Φ(out) = 762.9 – 476.48 = 286.4 MW

$\eta = P/\Phi = 286.41/1036.2 = 27.6\%$

4 (a) Show for helium that $\gamma = 5/3$ where γ is the adiabatic constant.

A closed-cycle single-shaft gas turbine plant using helium as the working fluid incorporates the following components in the given order: (a) a compressor, (b) a heater, (c) a two-stage turbine with reheater and (d) a cooler.

The maximum and minimum pressures and temperatures in the cycle are 40 bar and 700 °C, and 10 bar and 25 °C respectively, with reheat to 700 °C. The pressure in the reheater is optimum for maximum specific power (power per kg/s of gas flow).

The molar mass of helium is 4 kg/kmol and the molar heat capacity at constant volume for helium is $3/2\,\tilde{R}$ where $\tilde{R} = 8.3145$ kJ/kmol K is the universal molar gas constant.

(b) Sketch the T-s diagram for the plant and indicate pressures and temperatures between the components if

(i) the reheater is used,

(ii) the reheater is by-passed.

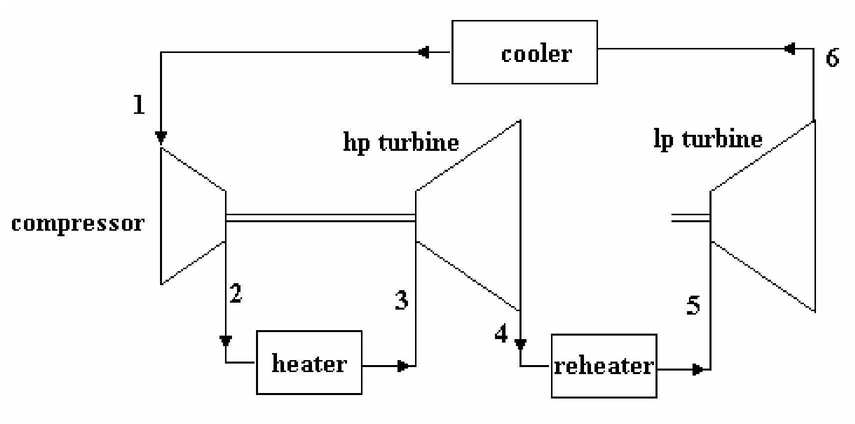
(c) Calculate the ideal cycle efficiency and specific power for each case. Assume that there are no losses in the cycle.

(a) For Helium $\tilde{m} = 4$ (mol mass) $R = \tilde{R}/\tilde{m} = 8.3145/4 = 2.0786$ kJ/kg K

$$\tilde{c}_v = \frac{3\tilde{R}}{2}$$

$$\tilde{c}_p = \tilde{R} + \tilde{c}_v = \tilde{R} + \frac{3\tilde{R}}{2} = \frac{5\tilde{R}}{2}$$

$$c_v = \frac{3R}{2} = 3.1179 \text{ kJ/kg K}$$



$c_p = R + c_v = 5.1966$ kJ/kg K

$\gamma = c_p / c_v = 1.667$

$p_1 = 10$ bar $\theta_1 = 25^oC$ $T_1 = 298$ K

$p_2 = 40$ bar

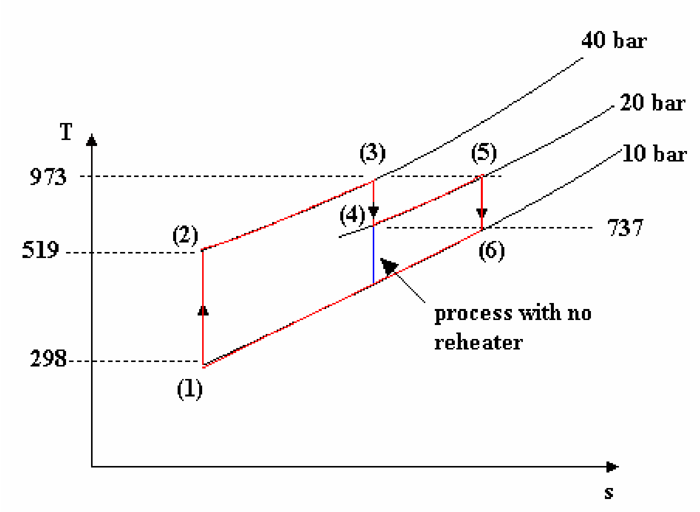
$p_3 = 40$ bar $\theta_3 = 700^oC$ $T_3 = 973$ K

For optimal turbine work $p_{4/5} = \sqrt{(40)(10)} = \sqrt{400} = 20$ bar $\theta_5 = 700^oC$ $T_5 = 973$ K

$$T_2 = T_1\left(\frac{p_2}{p_1}\right)^{1-\frac{1}{\gamma}} = 298\left(\frac{40}{10}\right)^{1-\frac{1}{1.667}} = 518.9\text{ K}$$

$$T_4 = T_3\left(\frac{p_4}{p_3}\right)^{1-\frac{1}{\gamma}} = 973\left(\frac{20}{40}\right)^{1-\frac{1}{1.667}} = 737.4\text{ K}$$

$$T_6 = T_5\left(\frac{p_6}{p_5}\right)^{1-\frac{1}{\gamma}} = 973\left(\frac{10}{20}\right)^{1-\frac{1}{1.667}} = 737.4\text{ K}$$



HEAT INPUT

$\Phi(in) = c_p(T_3 - T_2) + c_p(T_5 - T_4) = 5.1966(973 - 518.9) + 5.1966(973 - 734.7) = 3598.1$ kW

HEAT OUTPUT

$\Phi(out) = c_p(T_6 - T_1) = 5.1966(734.7 - 298) = 2269.4$ kW

Nett Power Out = 3598.1- 2269.4 = 1328.7 kW per kg/s of gas flow

Cycle efficiency $\eta = P/\Phi(in) = 1328.7/3598.1 = 0.369$ or 36.9 % with reheater

With the reheater bypassed we have a standard Joule cycle.

$$\eta = 1 - r_p^{\frac{1}{\gamma} - 1} = 1 - \left(\frac{40}{10}\right)^{\frac{1}{1.667} - 1} = 0.426$$

HEAT INPUT

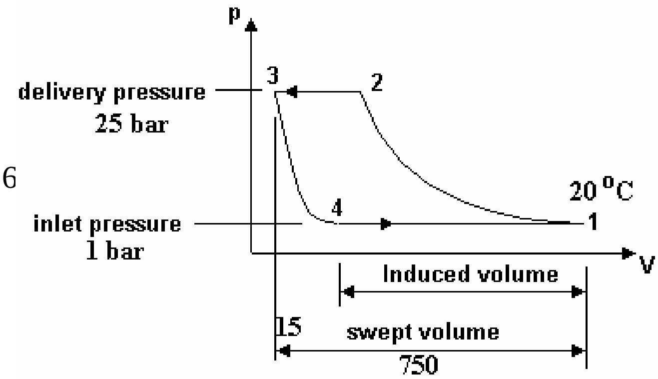
$\Phi(in) = c_p(T_3 - T_2) = 5.1966(973 - 518.9) = 2360$ kW

Nett Power Out = η x 2360 = 1005 kW per kg/s of gas flow

5. A single-stage air compressor has a clearance volume of 15 x 10^{-6} m^3 and a swept volume of 750 x 10^{-6} m^3. Air enters the compressor at a temperature of 20°C and a pressure of 1 bar. The delivery pressure is 25 bar and the compressor speed is 600 rev/min. Assume for the compression and expansion strokes that the polytropic indices are identical and equal to 1.45 respectively, and the gas constant for air is 0.287 kJ/kgK.
(a) Sketch the ideal indicator diagram.
(b) Determine
(i) The delivery temperature.
(ii) The mass flow rate.
(iii) The indicated power.

(c) Show how an actual indicator diagram would differ from the ideal diagram and explain why.

The ideal cycle is as shown.



DELIVERY TEMPERATURE

$$T_2 = T_1 r_p^{\frac{n-1}{n}} = 293 \text{ x } 25^{\frac{1.45-1}{1.45}} = 293 \text{ x } 25^{0.310} = 795.6$$

VOLUMETRIC EFFICIENCY
Clearance ratio c = 15/750

$$\eta_{vol} = 1 - c\left(r_p^{\frac{1}{n}} - 1\right) = 1 - \frac{15}{750}\left(25^{0.6896} - 1\right) = 1 - \frac{15}{750}(8.2065) = 0.8359$$

Induced volume = 0.8359 x 750 = 626.9 cm^3

Induced flow rate = 626.9 x 10^{-6} x 600 rev/min = 0.376 m^3/min
Mass flow rate

$$m = \frac{pV}{RT} = \frac{1\text{x}10^5 \text{ x } 0.376}{287 \text{ x } 293} = 0.447 \text{ kg/min} = 0.007455 \text{ kg/s}$$

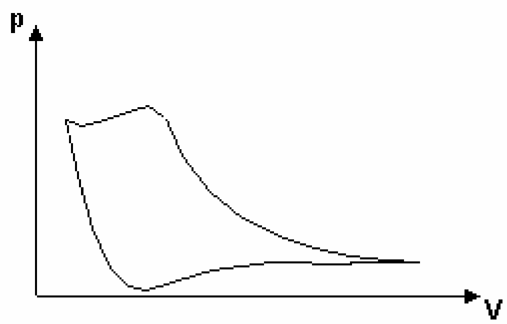
INDICATED POWER
There are various ways to find this. A derived formula for the standard cycle is as follows.

$$P = mRT_1\left(\frac{n}{n-1}\right)\left\{r_p^{\frac{n-1}{n}} - 1\right\} = 0.007455 \text{ x } 287 \text{ x } 293\left(\frac{1.45}{0.45}\right)\left\{25^{0.310} - 1\right\} = 3465 \text{ W}$$

or

$$P = \frac{nmR}{n-1}(T_2 - T_1) = \frac{1.45 \text{ x } 0.007455 \text{ x } 287}{0.45}(795.6 - 293) = 3465 \text{ W}$$

In practice there is restriction when the air is being sucked in and pushed out and the valves move on their springs so actual cycle is more like this.



6 A single-shaft gas-turbine jet engine is used as the propulsion unit on a small aircraft. The aircraft is flying at a velocity of 200 m/s at sea level where atmospheric pressure p is 1 bar and temperature T is 293 K. The pressure ratio over the compressor is 30. The compressor is adiabatic with an isentropic efficiency of 85%. After combustion, the hot gases enter the turbine with a temperature of 1200 K and expand adiabatically through the turbine. The turbine has an isentropic efficiency of 90% and it generates just sufficient power to drive the compressor. Finally the gases expand reversibly and adiabatically through a convergent propulsion nozzle, the outlet of which is choked.

(a) Determine the pressures at turbine and nozzle exits, the mass flow rate and the thrust developed if the nozzle has an exit area of 0.15 m^2.

(b) Also determine the power being generated to propel the aircraft.

Assume that the engine intake is isentropic, the working fluid throughout the engine is air with a gas constant R of 0.287 kJ/kgK, a specific heat capacity at constant pressure C_P of 1.0 kJ/kgK and an adiabatic constant γ of 1.4. Further assume that air is a perfect gas, and neglect all mechanical losses.

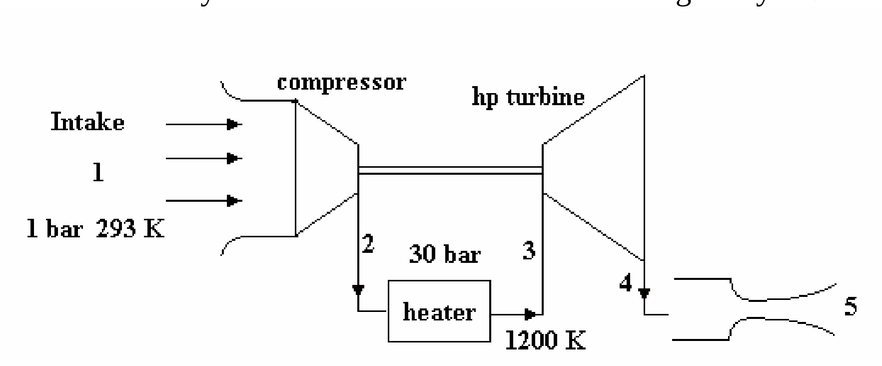
The critical temperature ratio in an isentropic nozzle is $\frac{2}{\gamma+1}$ and the velocity of sound is $\frac{\gamma p}{\rho}$

Where ρ is density.

The stagnation and static pressures p_o and p respectively are linked to the Mach number *M* by

$$\frac{p}{p_o} = \left[1+\left(\frac{\gamma-1}{2}\right)M^2\right]^{\frac{2}{\gamma-1}}$$

(c) Show that an aircraft velocity of 200 m/s has an effect on the engine cycle.



COMPRESSOR

$$T_o = T_1 + \frac{u_1^2}{2c_p} = 293 + \frac{200^2}{2000} = 313\,K$$

$$T_2' = T_o\left(r_p\right)^{\frac{\gamma-1}{\gamma}} = 313 \text{ x } 30^{0.2857} = 827\,K$$

$$\eta_i = 0.85 = \frac{827-313}{T_2 - 313} \quad T_2 = 917.7\text{ K}$$

Specific Power Input = c_p ΔT = 1 x (917.7- 313) = 604.7 kW

TURBINE

Power Out = Power In = 604.7 = c_p ΔT = 1 x (1200 – T_4) T_4 = 595.3 K

This is the actual temperature. Find the ideal temperature.

$$\eta_i = 0.9 = \frac{1200 - 595.3}{1200 - T_4'} \quad T_4' = 528.1 \text{ K}$$

$$\frac{T_4'}{T_3} = \left(\frac{p_4}{p_3}\right)^{\frac{\gamma-1}{\gamma}} \quad \frac{528.1}{1200} = \left(\frac{p_4}{30}\right)^{0.2857} \quad p_4 = 1.696 \text{ bar}$$

NOZZLE

$$T_5 = T_4\left(\frac{2}{\gamma+1}\right) = 595.3 \text{ x } 0.833 = 496.1 \text{ K}$$

$$\frac{T_4}{T_5} = \frac{595.3}{496.1} = \left(\frac{p_4}{p_5}\right)^{0.2857} \quad 1.2 = \left(\frac{1.696}{p_5}\right)^{0.2857} \quad p_5 = 0.896 \text{ bar}$$

or $$p_5 = p_4\left(\frac{2}{\gamma+1}\right)^{\frac{\gamma}{\gamma-1}} = 1.696\left(\frac{2}{2.4}\right)^{3.5} = 0.896 \text{ bar}$$

This pressure is less than atmospheric so there must be shock waves????

Apply conservation of energy.
$c_p T_4 = c_p T_5 + u^2/2$

$1000 \text{ x } 595.3 = 1000 \text{ x } 496.1 + u^2/2$ $\quad u = 951.5$ m/s

$V = A_2 u = 0.15 \text{ x } 951.5 = 142.725 \text{ m}^3/\text{s}$

$m = pV/RT = (0.896 \text{ x } 10^5 \text{ x } 142.725)/(287 \text{ x } 496.1) =$ kg/s

THRUST

$F_T = m(v - u) + A_2(p_2 - p_a) = 89.82 (951.5 - 200) + 0.015 (0.896 - 1.013) \text{ x } 10^5 = 67497 - 175.5$
$F_T = 67.32$ kN

NB I am not sure about the low pressure p_5. There must be some affect due to the pressure rise to atmospheric.

(b) POWER DEVELOPED

$P = F_T \; v = 67.32 \text{ x } 200 = 13464$ kW or 13.46 MW

(c) The entrance to the compressor must be a duct and a ram jet affect is achieved which affects the pressure rise and temperature rise over the compressor. I thought this was taken into account with the use of stagnation temperature and pressure so I don't see the relevance of this part of the question. Anyone knowing the answer, please let me know.

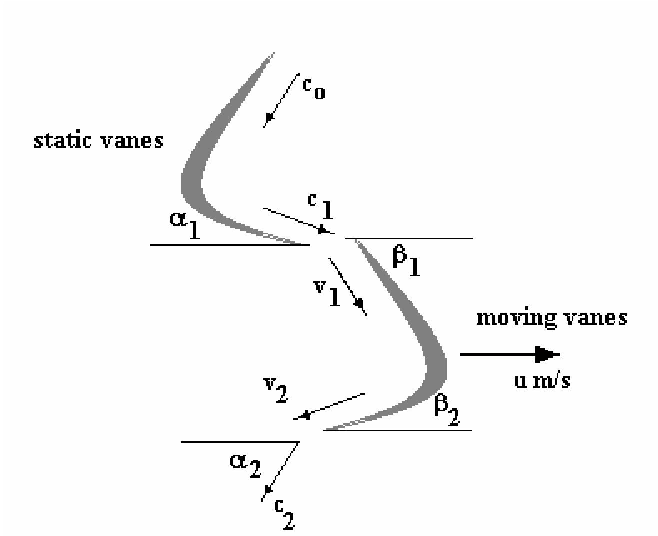
7 (a) Sketch the velocity diagram for the mean-diameter stator and rotor sections of a stage of an axial-flow reaction turbine. Assume equal inlet and outlet velocities to the stage and constant axial flow velocity. Indicate on the diagram all the angles which the absolute and relative velocity vectors make with the tangential, which is the whirl, direction.

(b) The degree of reaction **DR** is the ratio of the rotor enthalpy drop to the stage enthalpy drop. Prove that

$$DR = \frac{V_a}{2U}\left(\cot\beta_2 - \cot\beta_1\right)$$

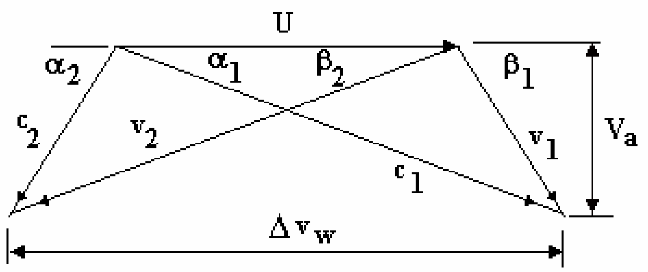
where $\frac{V_a}{U}$ is the ratio of the axial flow velocity to the rotor blade velocity, and β_1, and β_2, are the rotor blade inlet and outlet angles respectively.

(c) The mean-diameter section of a stage with **DR** = 0.5, has a blade velocity of 150 m/s and an axial gas velocity of 120 m/s. If the temperature drop across the stage is 25 °C and the specific heat capacity at constant pressure C_P is 1.0 kJ/kgK, calculate all stator and rotor angles.



The stationary vane makes an angle α_1 with the direction of rotation. The moving vane has an angle β_1 at inlet and β_2 at outlet. c is the absolute velocity of the steam and v is the relative velocity. The velocity diagram is as shown if the absolute velocity entering the stationary vanes is the same as the absolute velocity c_2 at exit from the moving rotor. In this event it follows that $\beta_1 = \alpha_2$ and $\beta_2 = \alpha_1$.

U = blade velocity. V_a = Axial velocity.
Δv_w = change in velocity in whirl direction.
Enthalpy at entry to stage = h_o
Enthalpy at exit from stage = h_2
Change in enthalpy = work given to the rotor
$h_o - h_2 = U\,\Delta v_w$ $\quad\quad$ $\Delta v_w = V_a(\cot\beta_1 + \cot\beta_2)$
h_1 = enthalpy at entry to the rotor. Change in enthalpy over the rotor = change in KE over the rotor



$$h_1 - h_2 = \frac{v_2^2 - v_1^2}{2}$$

$v_2 = V_a \operatorname{cosec}\beta_2$ $\quad\quad$ $v_1 = V_a \operatorname{cosec}\beta_1$

$$h_1 - h_2 = V_a^2\left\{\frac{\left(\operatorname{cosec}^2\beta_2 - \operatorname{cosec}^2\beta_1\right)}{2}\right\}$$ but since

$(\operatorname{cosec}\beta)^2 = (\cot\beta)^2 + 1$

$$h_1 - h_2 = V_a^2\left\{\frac{\left(\cot^2\beta_2 - \cot^2\beta_1\right)}{2}\right\}$$

$$h_1 - h_2 = \frac{V_a^2}{2}\left(\cot\beta_2 + \cot\beta_1\right)\left(\cot\beta_2 - \cot\beta_1\right)$$

$$DR = \frac{h_1 - h_2}{h_o - h_2} = \frac{V_a^2}{2}\left\{\frac{\left(\cot\beta_2 + \cot\beta_1\right)\left(\cot\beta_2 - \cot\beta_1\right)}{UV_a\left(\cot\beta_2 + \cot\beta_1\right)}\right\} = \frac{V_a}{2U}\left(\cot\beta_2 - \cot\beta_1\right)$$

DR = 0.5 $\quad$ U = 150 m/s $\quad\quad$ V_a = 120 m/s $\quad$ ΔT = 25 K $\quad\quad$ C_P is 1.0 kJ/kgK

$C_p\ \Delta T$ = change in enthalpy over the stage = $U\ \Delta v_w$
$C_p\ \Delta T$ = change in enthalpy over the stage = Δv_w
$\Delta v_w = C_p\ \Delta T/U = 45\ 000/150 = 300$ m/s
$\Delta v_w = V_a(\cot \beta_1 + \cot \beta_2)$ $\quad 300 = 120(\cot \beta_1 + \cot \beta_2)$ $\quad \cot \beta_1 = 2.5 - \cot \beta_2$

$$DR = 0.5 = \frac{V_a}{2U}\left(\cot \beta_2 - \cot \beta_1\right)$$

$$0.5 = \frac{120}{2x\,150}\left(\cot \beta_2 - \cot \beta_1\right)$$

$$1.25 = \left(\cot \beta_2 - \cot \beta_1\right)$$

$$\cot \beta_1 = \cot \beta_2 - 1.25$$

$\cot \beta_1 = 2.5 - \cot \beta_2 = \cot \beta_2 - 1.25$
$2\cot \beta_2 = 3.75$ $\quad \cot \beta_2 = 1.875$ $\quad \tan \beta_2 = 0.5333$ $\quad \beta_2 = 28^o = \alpha_1$
$\cot \beta_1 = 2.5 - \cot \beta_2 = 0.625$ $\quad \tan \beta_1 = 1.6$ $\quad \beta_2 = 58^o = \alpha_2$

8 The analysis by mass of a solid fuel is as follows:

Carbon 70%, Hydrogen 15%, Oxygen 5%, Ash 10 %.

The fuel is burnt with 20% excess air. Assuming complete combustion, calculate

(a) the composition by mass of the products of combustion,
(b) the dewpoint,
(c) for each kg of fuel burnt, the mass of water which will condense when the products of combustion are cooled at a constant pressure to 20 °C.

Assume that the barometric pressure is 1 atm.

C +	$O_2 \leftrightarrow$	CO_2	$2H_2$ +	$O_2 \leftrightarrow$	$2H_2O$
12	32	44	4	32	36
0.7	1.867	2.57	0.15	1.2	1.35

There are .7/12 = 0.05833 kmol of C and 0.15/2 = 0.075 kmol of H_2

Total O_2 needed = 1.867 + 1.2 – 0.05 = 3.0167 kg
Air needed = 3.0167/0.233 = 12.947 kg
Actual air 12.947 x 1.2 = 15.537 kg
Nitrogen in this air = 0.77 x 15.537= 11.963 kg
oxygen in this air = 3.620 Oxygen used = 3.0167 Oxygen left over = 0.603 kg

PRODUCTS

	kmol	mass	%
N_2	0.427	11.963	72.6
CO_2	0.0584	2.57	15.6
H_2O	0.075	1.35	8.2
O_2	0.01884	0.603	3.6
Total	0.5792	16.486	100

If everything ends up as gas then the partial pressure of H_2O is
p_{H2O} = (0.075/0.5792) x 1 atm = 0.1295 atm = 0.131 bar

The corresponding saturation temperature is 51.2°C (The dew Point)

If cooled to 20°C some condensation must occur and the vapour left will be dry saturated vapour.
ps at 20°C is 0.02337 bar
Let the kmol of H_2O vapour be x. The total kmol is the same = .5792 – 0.075 + x = 0.5042 + x

$$p_{H_2O} = \frac{x}{0.5042 + x} \times 1.013 = 0.02337 \text{ bar}$$

$$0.01163 + 0.02307x = x$$

$$0.01163 = 0.9769x \qquad x = 0.0119 \text{ kmol}$$

The mass of vapour is m = 0.0119 x 18 = 0.2142 kg
Condensate formed is 1.35 - 0.2142 = 1.1358 kg

9 As hydrocarbon fuels become scarcer, and the cost of extraction from the earth increases, it is essential that all of us become efficient energy managers. In most factories, offices, apartment blocks and homes, energy is wasted, usually in the form of hot fluids. Heat recovery is not a new technology, but it is a technology which needs wider application with particular emphasis on smaller units.

There are various types of small scale recuperators in which the fluids exchanging heat are separated by a dividing wall. Some examples are parallel flow, counter flow, cross flow, multipass, mixed flow and extended surface.

Explain the basic operating principles of recuperators and indicate which is most advantageous for small scale application.

A recupurator is a heat exchanger that removes heat from a waste fluid and adds it to another fluid where it will be useful.

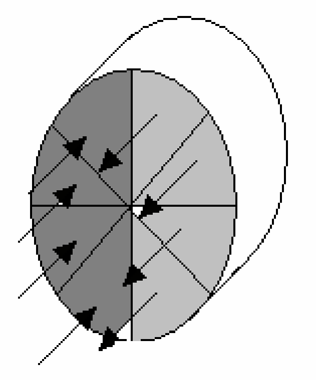
On large boiler plant they are used to remove heat from flue gas and add it to the air supplied for combustion. This could be applied to central heating boilers or boilers supplying process heat. The capital cost is high and hard to recover through the economy made.

Factories with a large amount of waste heat may find it economical to recover heat. Waste steam is relative easy to recover by condensing it and recycling it using it for space heating

Hot waste air and other gasses are more difficult to recover and recuperators are often better than other forms of heat exchangers for this purpose.

In domestic and office situations they are more likely to be used to remove heat from stale air being removed from the building (e.g. from kitchens venting the fumes from cooking) and added to the fresh air being drawn into the building hence saving on cost of heating the building.

The regenerative type is a rotating drum with half in the path of one fluid and half in the path of another. The hot fluid passes through a heat absorbent material in a drum. The drum rotates and the heated material rotates into the path of the cool fluid and warms it up.



Others work by conduction of heat from the warm fluid to the cool through metal plates with the maximum exposed surface area possible.

Heat pipes contain a fluid that transports heat from one fluid to the other and makes use of the latent heat of the fluid to transport large quantities of heat. These are very effective.

THE FOLLOWING IS TYPICAL OF INFORMATION THAT CAN BE FOUND ON THE INTERNET BY SIMPLY SEARCHING FOR RECUPERATORS.

Heat Recuperators

It is also possible to use the recuperated heat to heat water for cleaning purposes or air for heating rooms. In the following only preheating of the drying air is discussed.

In principle, there are two different recuperating systems:

- Air-to-Air
- Air-Liquid-Air

Air-to-Air Heat Recuperator

In the heat recuperator type air-to-air, see Fig. 98, the drying air is preheated by means of the outgoing air passing counter-currently over the heat surface of the recuperator. This surface is formed as a number of tubes, inside of which the outgoing warm air is passing while the cold air is passing on the outside.

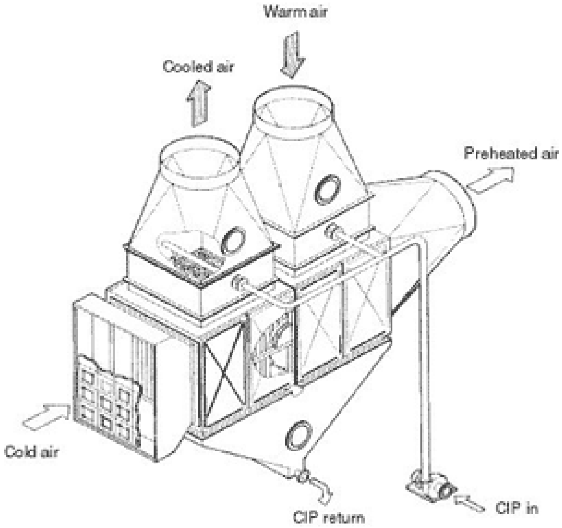


Fig. 98 Heat recuperator type air-to-air

The incorporation of this equipment in an existing plant may prove difficult and ex-pensive, as it may require large and long air ducts from which part of the recuperated energy is lost due to radiation, if the ducts are not insulated. In new installations it is easier to incorporate this type of heat recuperator, as the arrangement can be optimized with short air ducts. See Fig. 99.

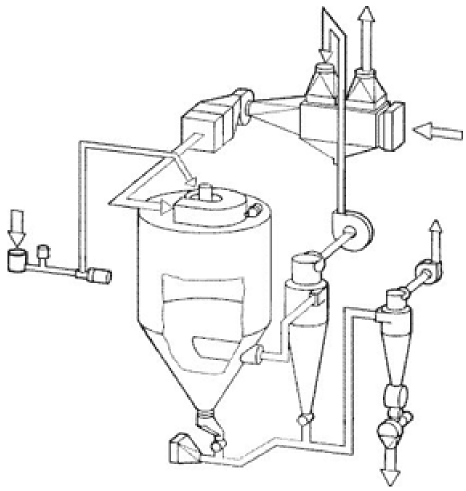


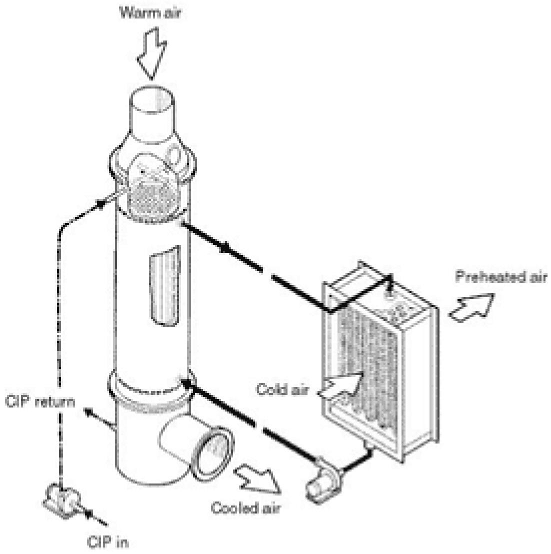
Fig. 99 One-stage spray dryer with hear recuperator type air-to-air

The temperature to which the air can be preheated depends upon the temperature of the outgoing air. Therefore, this type of heat recuperator is most beneficial in combination with a one-stage spray dryer where the temperature of the outgoing air is high. The figures mentioned below are based upon a one-stage plant as mentioned in the table on page 139.

Ambient air preheated from 10ºC to 52ºC
Outgoing air cooled from 93ºC to 51ºC:

Air-Liquid-Air Heat Recuperator

Another system, more flexible regarding the installation, is the air-liquid-air heat re-cuperator, see Fig. 100. This system is divided in two heat exchangers, in between which a heat transfer liquid is circulated, for example water. See Fig. 100a. If, due to low air temperatures during winter, it may be expected that the temperature of the water gets below zero, an anti-freeze agent is added to the water. As the heat transfer co-efficient is higher for air-liquid than for air-air, this system is more efficient than the air-to-air heat recuperator despite the fact that two heat surfaces are needed.



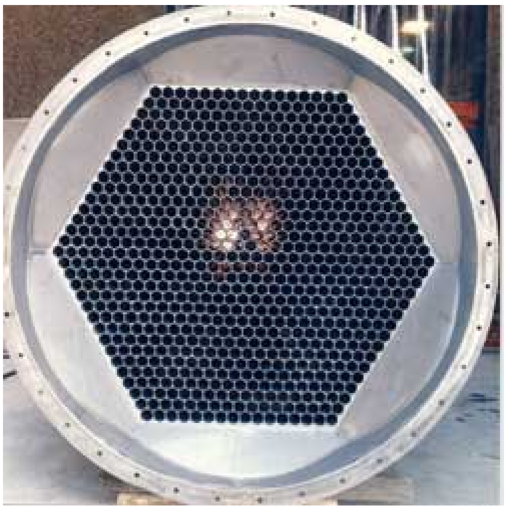


Fig. 100 Heat recuperator type air-liquid-air

The heat transfer surface placed in the outgoing air is formed as a bundle of tubes inside which the dust-loaded air is passed. On the outside of the tubes the water streams counter-currently. The heat transfer surface placed in the inlet air is a normal finned tube heat exchanger. Water is recycled by means of a centrifugal pump.

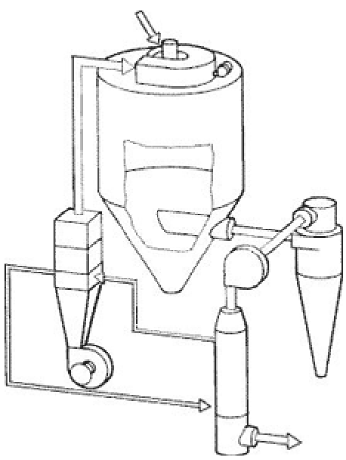


Fig. 100a One-stage spray dryer with heat recuperator type air-liquid-air

If indirect oil- or gas-fired air heaters are used, the heat transfer liquid can - after the passage through the exhaust air heat exchanger - be passed through a heat exchanger placed in the combustion air duct, whereby even further savings can be achieved.

Tubular Heat Recuperators

Exothermics Tubular Heat Recuperators (THR) are air-to-air heat recovery units that effectively reclaim heat from catalytic incinerators, furnaces, thermal oxidizers and many other high temperature process and environmental applications.

But that's just the beginning. They also help you lower energy costs, easily and effectively control process air temperatures and reclaim a fast return on investment.

No other company manufactures a more effective Tubular Heat Recuperator than Exothermics. Our units are installed in hundreds of sites around the world, and we are quickly becoming the preferred choice for high temperature heat recovery equipment. Here's why:

Our Tubular Heat Recuperators are accepted and endorsed worldwide because they simply perform better. Features include:

Boundary Layer Breakdown

Exothermics Tubular Heat Recuperators have a proprietary tubular core design in which the placement of the heat recovery tubes assures a breakdown of air boundary layers in and around the tubes. The design creates a turbulent movement of the hot gas and process airstreams, resulting in more efficient heat transfer and optimum heat recovery.

Multi-Pass Designs

Crossflow and multiple pass designs are available. Multiple pass designs are used when the application requires greater effectiveness. Units can be manufactured so that the multiple passes are on the shell side, where the gas stream passes over the tubes several times before exiting the recuperator. Other applications may require a multiple tube pass design.

Insulation

Various options are available. Our Tubular Heat Recuperators can be ordered without insulation or with external insulation when a hot flange connection is required. Where cold flange connections are involved, the unit is designed with internal ceramic fiber insulation.

Rugged Construction

Exothermics Tubular Heat Recuperators are all welded assemblies constructed from stainless steel or other high temperature alloys. Each unit is custom engineered, then carefully fabricated and quality tested by certified welders and experienced craftsmen. Where required, a mechanism for accommodating thermal expansion is provided. And because our tubular heat recuperators are of all welded construction, internal cross contamination is virtually eliminated.