Question 1:

We need to calculate $m = \sqrt{\frac{hP}{kA}} = \sqrt{20\frac{\pi(0.01)}{52\left(\pi(\frac{0.01}{2})^2\right)}} = \sqrt{153.8} = 12.4\, m^{-1}$ (2 marks)

We should rearrange the long fin equation to have x as the argument.

$$x = -\frac{1}{m}\ln\left(\frac{\theta}{\theta_0}\right)$$

We know that $\theta_0 = (600 - 28) = 572C$ and $\theta = (40 - 28) = 12C$ (2 marks)

So the distance for safe handling from the base is:

$$x = -\frac{1}{12.4}\ln\left(\frac{12}{572}\right) = 0.31\, m \qquad (1\, mark)$$

So the safe distance from the surface of the fire is: 310 mm. The poker is 100 mm in the fire and 600 mm long in total, so it can be held safely 410 mm from the base, which is 190 mm from the top. (1 marks)

In other words you can safely touch the top 19 cm of the poker without getting burnt. [6 marks total]

Question2:

The fin effectiveness is given by the ratio of the actual heat loss through the fin/the heat loss without the fin (1 mark for exact solution) it can have values from 0 to infinity. (1 mark)

The fin efficiency is the ratio of the actual heat loss through the fin / the heat loss from an ideal fin (1 mark). It can have values from 0 to1 (1 mark).

An ideal fin is one where the surface temperature of the extruded surface is **the same as** the **base of the fin**. (2 marks explaining highlighted concepts)

Question 3:

The Shape factor S is given as: $S = \frac{2\pi L}{\ln\left(\frac{4L}{D}\right)} = \frac{2\pi(1.5)}{\ln\left(\frac{4(1.5)}{0.5}\right)} = 3.79\, m$ (2 mark)

The heat flow is given as $\dot{q} = kS(T_1 - T_2) = 1.2(3.79)(60 - 10) = 227W$ (2 mark)

This is the heat loss from the drum which needs to be replaced, so a value of 227W will be correct. [4 marks total]

Question 4:

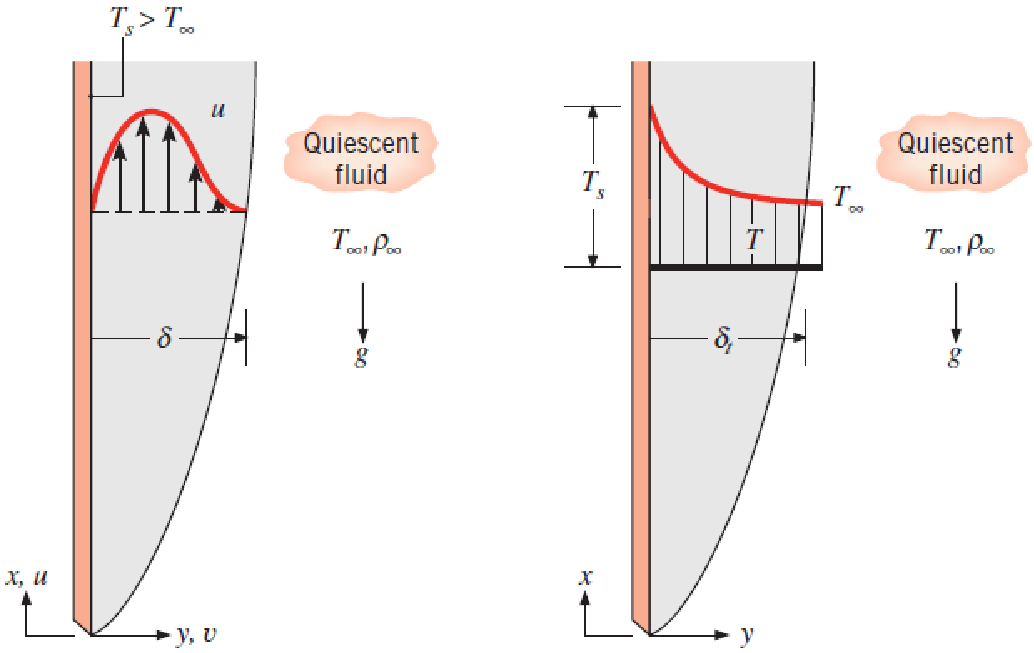
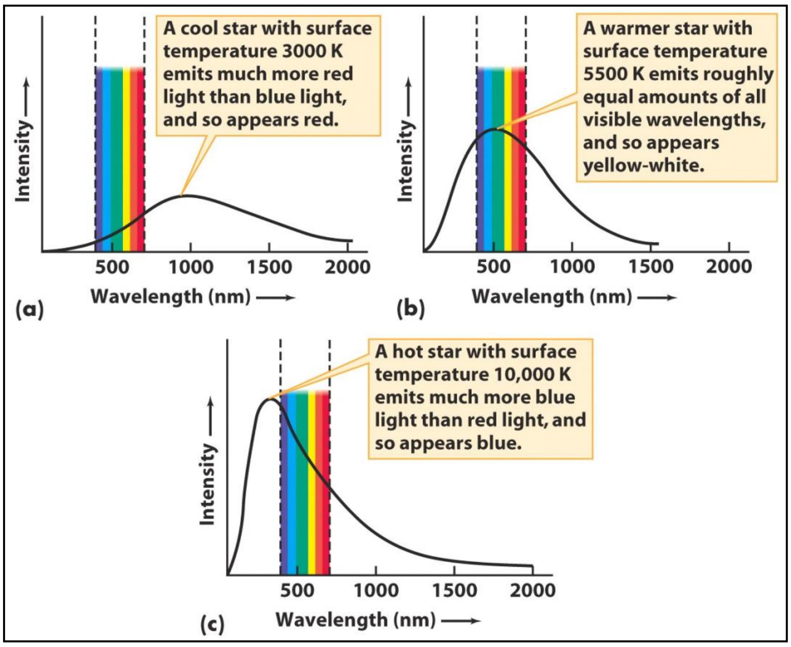


Figure 1: (1 marks for each diagram being present. 1 marks for clarity of presetnation.)

Main points to mention in text:
Velocity: Not monotonic. Zero near wall and outside velocity boundary layer with maximum in between. (2 marks)
Temperature: High close to the wall and falls monotonically to the ambient temperature at the edge of the boundary layer. (2 marks)
Boundary layer thickness: Prandtl number is 1 so they are the same thickness. (1 marks) [7 marks]

Question 5:



(1 mark) for black body emission diagrams. (1 mark) for indication of visible spectra. (2 marks) for explanation of apparent colour.

Question 6.

The reciprocity relationship states that the two view factors are linked through the areas of the surfaces by:

$$A_1F_{1-2} = A_2F_{2-1} \rightarrow F_{2-1} = \frac{A_1}{A_2}F_{1-2} = \frac{0.3}{0.15}\,0.15 = 0.3$$

So the view factor $F_{2-1} = 0.3$ (1 mark)

This **does not affect the view factor** for diffuse radiation. All the radiation **passes through the virtual surface** which is the same as the original surface. (2 marks)

Question 7.

Quite simply replace the Nusselt number with the Sherwood number and the Prandtl number with the Schmidt number. [2 marks]

$$\overline{Nu_L} = 0.644Pr^{\frac{1}{3}}Re_L^{\frac{1}{2}} \quad for\ Pr \geq\ 0.5$$

Becomes

$$\overline{Sh_L} = 0.644Sc^{\frac{1}{3}}Re_L^{\frac{1}{2}} \quad for\ Sc \geq\ 0.5$$

Question 8.

Molar Gibbs Function

$$\tilde{g} = \tilde{h} - T\tilde{s} \qquad [1]$$

$$(\tilde{g}_{NO})_{1200,5\,bar} = (\tilde{h}_{fo} + \Delta\tilde{h}) - T\left(\tilde{s}^{o} - R\ln\frac{p}{p^{o}}\right) \qquad [2]$$

$$= (90\,290 + 29121) - 1200(257.71 - 8.314\ln 5)$$

$$= 119411 - 293195$$

$$(\tilde{g}_{NO}) = -173783 \text{ kJ/kmol at 1200K, 5 bar} \qquad [2]$$

Question 9

In an open feed heater there is direct contact (mixing) between the steam and water being heated and the steam and water leave in the same flow and must at the same pressure [1]

In a closed feed heater the steam is separated from the water being heated and does not need to be at the same pressure. [1]

Question 10.

11. The internal energy of the steam in the filled vessel is equal to the enthalpy of the steam entering the vessel (from NSFEE)

$h_{feed} = u_{vessel}$ Final pressure in vessel = 15 bar [2]

Dry saturated steam at 15 bar $h = 2792$ kJ/kg

At 15 bar ($u = 2595$ kJ/kg at saturation (198.3°C))

" $u = 2784$ kJ/kg at 300°C

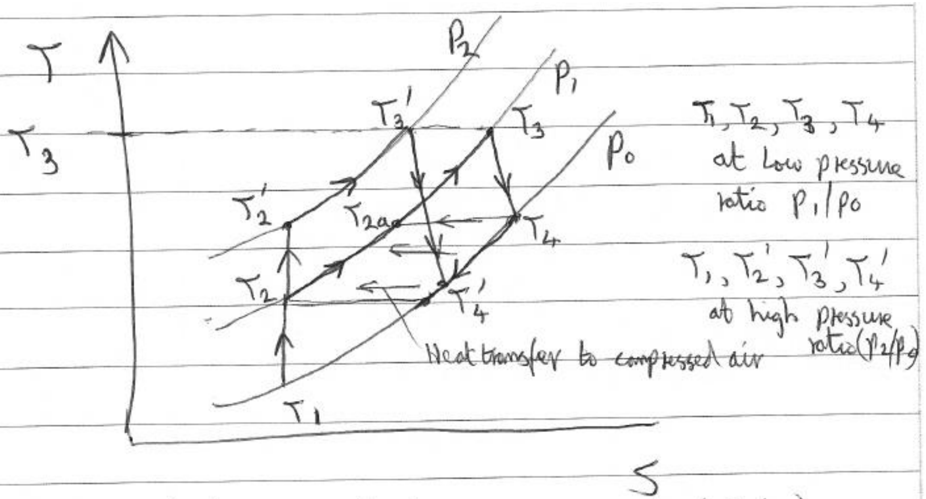
2868 kJ/kg at 350°C

By linear interpolation

$T = 305°C$ at $u = 2792$ kJ/kg [2]

Question 11

Exhaust recuperation reduces the heat input required in the combustion chamber of a gas turbine by recovering heat from the exhaust gases to preheat the air leaving the compressor before entering the combustion chamber. [3]

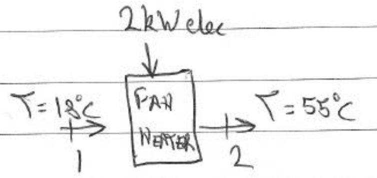


For gas turbine with Low pressure ratio (P_1/P_0) Exhaust gases leave at a temperature well above compressor outlet temperature providing opportunity for heat recovery. [2]

But for a high pressure ratio (p_2/p_0) the compressor outlet temperature may be higher than the exhaust gas outlet temperature, thereby not allowing recuperation.

[2]

Question 12



$T_0 = 7°C$ (280 K)

To find air mass flow rate

$$Q_{in} = 2\text{ kW} = \dot{m} c_p (\Delta T)$$

$$= \dot{m} \times 1.0 (55 - 18) \qquad [1]$$

$$\dot{m} = 0.054 \text{ kg/s}$$

$$\text{Rational Efficiency} = \frac{\text{Exergy Output}}{\text{Exergy input}} \qquad [1]$$

Exergy output is gain in exergy of air

$$E = \dot{m}[(h - h_0) - T_0(s - s_0)]$$

$$\Delta E = \dot{m}[(h_2 - h_1) - T_0(s_2 - s_1)] \qquad [1]$$

$$= \dot{m}\left[c_p \Delta T - T_0\left(c_p \ln \frac{T_2}{T_1} - R \ln \frac{p_2}{p_1}\right)\right]$$

$$= 0$$

$$= 0.054\left[1 \times (55 - 18) - 280\left(1 \times \ln \frac{(273+55)}{(273+18)}\right)\right]$$

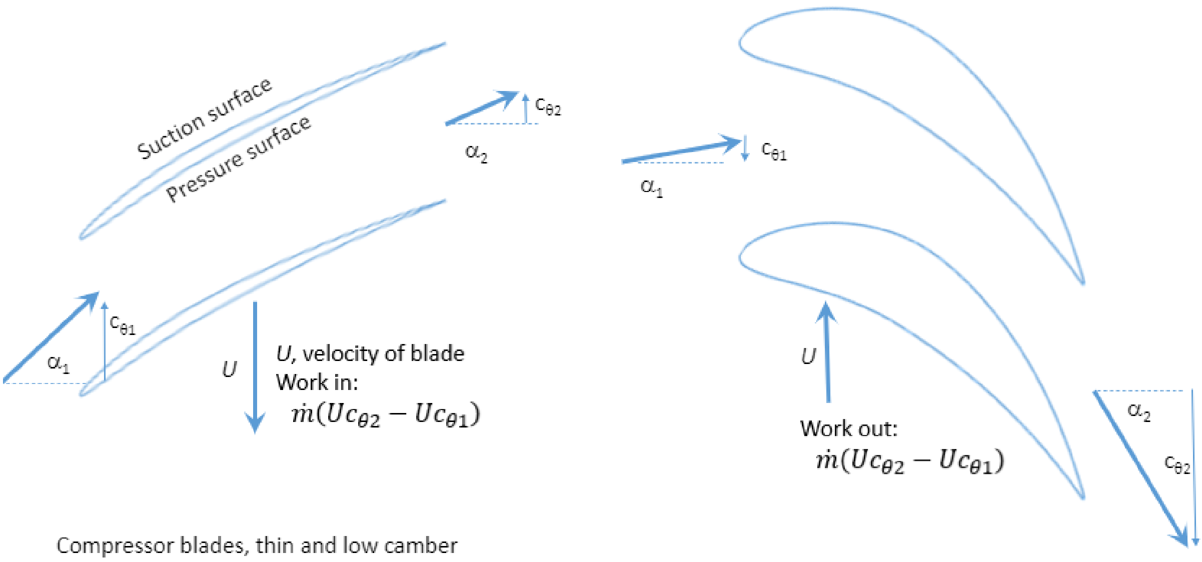
$$\Delta E_{output} = 0.188 \text{ kW} \qquad [1]$$

Exergy input is 2kW [1]

$$\therefore \text{Rational Efficiency} = \frac{0.188}{2} = 9.4\% \qquad [1]$$

Electricity could be used more efficiently by using a heat pump to provide heating. [1]

13. For compressor blades, choose a thin blade and a small camber, the turning angle is limited, and due to the pressure increase across the stage, the boundary layer thickens quickly [2].
For turbine blades the process is of expansion, and the boundary layer decreases in height because of this, and it is therefore possible to achieve much higher stage loading, which is characterised by large turning angles of the blades [2].

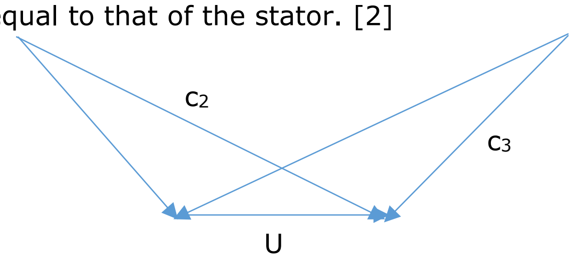


Compressor blades, thin and low camber

Turbine blades, thick and high camber

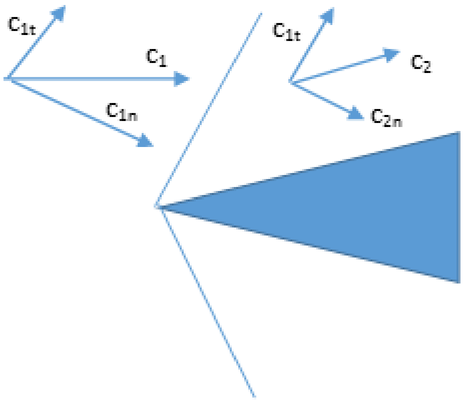
[2]

14. 50% reaction means that the velocity triangles are symmetrical. Reaction indicates the amount of relative turning of the flow in the blade passages, and 50% means that the enthalpy change in the rotor is 50% of the enthalpy change in the stage, which for an axial flow turbine is related to the static pressure change directly, so that the rotor pressure drop is equal to that of the stator. [2]



[1]

15. Oblique shock will preserve the velocity tangential to the shock wave and the velocity normal to the shock will behave as a normal shock for that component. The normal and total velocity are modified by the shock. [2]



Prandtl-Mayer flow occurs where there is an expansion. In this case the flow turns on a convex corner. The shock wave occurs in a succession of waves which are each weak and lead to a Prandtl-Mayer fan of weak Mach waves.

[2]

Question B1:
a)

i) Stage loading is from the formula sheet:

$$\psi = \frac{h_0}{U^2} = \frac{w_{\theta 3} - w_{\theta 2}}{U} = \emptyset(tan\beta_3 + tan\beta_2)$$

Reaction is from the formula sheet:

$$R = \phi\,(tan\beta_3 - tan\beta_2)/2$$

Combining these equations yields:

$$tan\beta_3 = \frac{\psi/_2 + R}{\phi}$$

and

$$tan\beta_2 = \frac{\psi/_2 - R}{\phi}$$

[2]

Since the values are given, the angles are:

$$R = 0.5, \psi = 1.8, \phi = 0.9$$

So:

$$tan\beta_3 = \frac{1.8/_2 + 0.5}{0.9} = 1.55, \beta_3 = 57.3\,^{o}$$

$$tan\beta_2 = \frac{1.8/_2 - 0.5}{0.9} =, \beta_3 = 24.0\,^{o}$$

[2]

for a 50% reaction, the absolute angles will be:

$$\alpha_2 = \beta_3 \; and \; \alpha_3 = \beta_2$$

[1]

ii) work out the flow through the blade passage annulus:

Blade speed at mean radius is:

$$\frac{18000}{60} 2\pi \times 0.25 = 471\, m/s$$

The axial velocity:

$$c_x = \phi U_m = 0.9 \times 471 = 424 \frac{m}{s}$$

[2]

and the velocity of the gas at nozzle exit is:

$$c_2 = \frac{c_x}{cos\alpha_2} = \frac{424}{cos24} = 464\, m/s$$

[2]

iii) Mass flow rate is based on annulus area:

$$\dot{m} = \rho_2 A_2 c_{x2} = \frac{p}{RT} A_2 c_{x2} = \frac{24.4 \times 10^5}{287 \times 1807} 0.0157 \times 424 = 31.2\, kg/s$$

[2]

Solution:
b)

i. First calculate film temperature: =(1900+1700)/2=1800 K (1 mark)
At this temperature the kinematic viscosity of the air is: $\nu = 29.72 \times 10^{-5}\, m^2/s$
So: (1 mark)

$$Re = \frac{175(0.1)}{29.72 \times 10^{-5}} = 60345$$

ii. This Reynolds number can be used to calculate the heat transfer coefficient. For this we also need the Prandtl number of the fluid and the thermal conductivity.
At the film temperature these are: $Pr = 0.728$, and $k = 9.899 \times 10^{-2} W/mK$ (2 marks)
The fluid is being cooled, so n=0.3. (1 mark+ 1mark)

$$Nu_L = 0.023(60345)^{0.8}(0.728)^{0.3} = 139.5$$

The heat transfer coefficient is therefore: (1 mark)

$$h = \frac{Nu_L k}{L} = \frac{139.5(9.899 \times 10^{-2})}{0.1} = 138W/m^2K$$

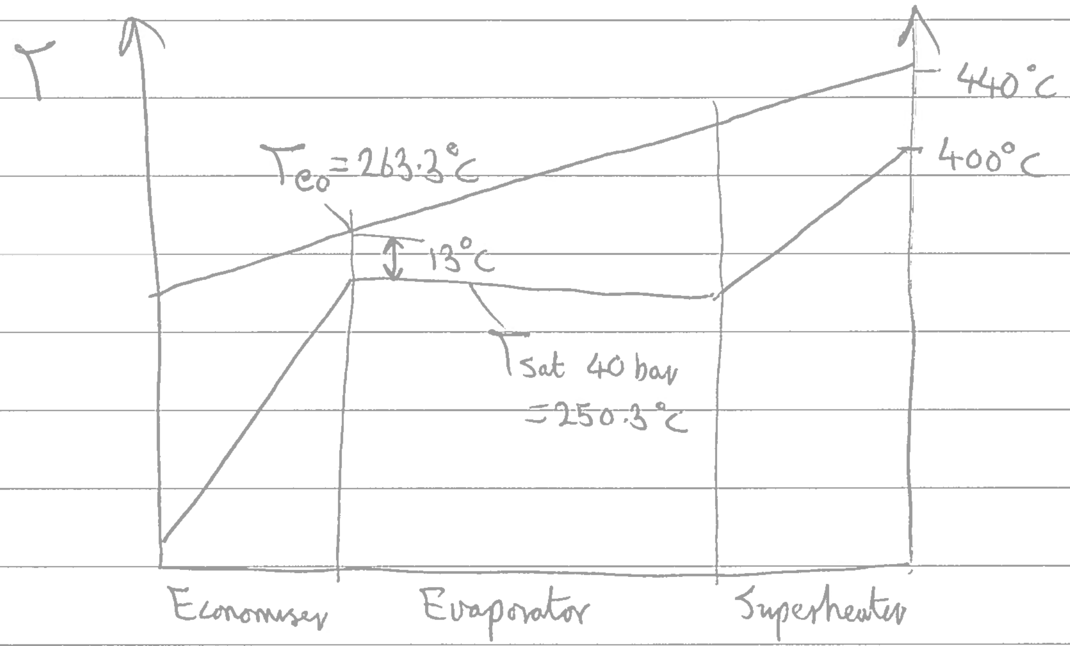
So for one side of the blade the heat transfer coefficient is $h = 138\ W/m^2K$

iii. heat loss for both sides of the airfoil is therefore:

$$\dot{Q}'' = h(2L)(1)(1900 - 1700) = 138(0.2)(200) = 5.2kW$$

So we need to remove 5.3 kW per turbine blade. (2 mark)

15. In heat recovery boiler



[4]

In evaporator and superheater

Enthalpy of feedwater in = 1087 kJ/kg

Enthalpy of superheated steam = 3214 kJ/kg

Enthalpy balance on evap & superheater

$$\dot{m}_{st}(3214 - 1087) = 14.5 \times 1.05 \times (440 - 263.3)$$

$$\dot{m}_{st} = \frac{2690.3}{2127} = 1.26 \text{ kg/s}$$ [3]

In economiser heat gain in water = $\dot{m}$ (h_f at 40 bar − enthalpy of feedwater)

Feedwater enthalpy is that of saturated water at condenser pressure 0.1 bar

= 192 kJ/kg

Enthalpy balance on economiser

$$\dot{m}_{st}(\Delta h) = \dot{m}_{gas}\, c_p\, \Delta T_{gas}$$

$$\Delta T_{gas} = 1.26\,(1087 - 192) / 14.5 \times 1.05$$
$$= 74.1°C$$

So exhaust gas temperature at outlet of boiler is

$263.3 - 74.1 = \underline{189.2\ °C}$ [1]

Enthalpy and exergy of steam in cycle.

Use steam chart for enthalpy at turbine outlet

Exergy $= \dot{m}\left((h - h_0) - T_0(s - s_0)\right)$ $T_0 = 280\,K$

	Temp (°C)	Pressure (bar)	h kJ/kg	s kJ/kgK	Specific Exergy kJ/kg	Flow Exergy kW
1	45.8°C	0.1	192	0.649	10.6	13.4
2	400	40	3214	6.769	1319.0	1662
3′		0.1	2145	6.769	–	
3		0.1	2316	7.30	272.3	342.7
4	45.8	0.1	192	0.649	10.6	13.4
Env Point	7°C (280 K)	1	29.4	0.106		

[2]

At turbine outlet h_3' (isentropic condition) $= 2145$ kJ/kg

$\therefore\ \Delta h' = 3214 - 2145 = 1065$

Actual $\Delta h = \eta_s \Delta h' = 0.84 \times 1065 = 898$ kJ/kg

$\therefore\ h_3 = h_2 - \Delta h = 3214 - 898 = 2316$ kJ/kg [2]

Work output from steam turbine $= \dot{m}(h_2 - h_3)$

$= 1.26(3214 - 2316) = 1131.5$ kW

Electricity output $= \eta_{generator} \times 1131.5$

$= 0.93 \times 1131.5 = 1052$ kW [2]

Heat input to steam cycle is $\dot{m}_{st}(h_2 - h_1)$

$= 1.26(3214 - 192)$

3808 kW [4]

Thermal efficiency of steam cycle is 1052/3808

(or 1131.5/3808 = 29.7%) = 27.6 %

Exergy in the exhaust gases $E = \dot{m}[(h - h_0) - T_0(s - s_0)]$

For perfect gas $s_1 - s_0 = c_p \log_e(T_1/T_0) - R\log_e(p_1/p_0)$

As gas is at constant pressure in boiler

$$s_1 - s_0 = c_p \log_e\left(\frac{T_1}{T_0}\right)$$

Exergy Loss in exhaust gas is

$$\Delta E = \dot{m}\left[c_p(T_A - T_B) - T_0\left(c_p \log_e\left(\frac{T_A}{T_B}\right)\right)\right]$$

$$= 14.5\left[1.05(440 - 189.2) - 280\left(1.05 \log_e\frac{(440+273)}{(189.2+273)}\right)\right]$$

$= 1971$ kW. [2]

Irreversibility in boiler = Loss in exergy in boiler − gain in exergy in steam

$= 1971 - (1662 - 13.4)$

$= 322.5$ kW [2]

Irreversibility in steam turbine generator = Loss in exergy steam − electricity output

$= (1662 - 342.7) - 1052 = 267.3$ kW [2]

Irreversibility in condenser = Loss in exergy in steam, There is no exergy output.

$= (342.7 - 13.4) = 329.3$ kW [1]

c) Power output could be increased by reducing irreversibilities by:

- Reducing pinch temperature difference in boiler or adopting a double pressure steam cycle
- Increasing isentropic efficiency of steam turbine and increasing efficiency of generator
- Reducing condenser pressure and temperature
- Feedheating can reduce irreversibility in the boiler and increase steam cycle efficiency but the exhaust outlet temperature increases, reducing heat input. [5]