1. What do you understand by the Air standard efficiency?

   The cycles encountered in actual devices are difficult to analyse because of the presence of friction, and the absence of sufficient time for establishment of equilibrium conditions during the cycle. In order to make an analytical study of a cycle feasible, we have to make some idealizations by neglecting internal Irreversibility's and complexities. Such cycles resemble the actual cycles closely but are made up of internal reversible processes.These cycles are called ideal cycles.
   State the assumptions made in analysis of air standard cycles.Air standard cycles are thermodynamic cycles with air assumed as the working substance.In these cycles the combustion process is replaced by heat addition and exhaust process by heat rejection process. The efficiency of such cycles are called as air standard efficiency given by

$$air\ standard\ Efficiency = \frac{Q_s - Q_r}{Q_s}$$

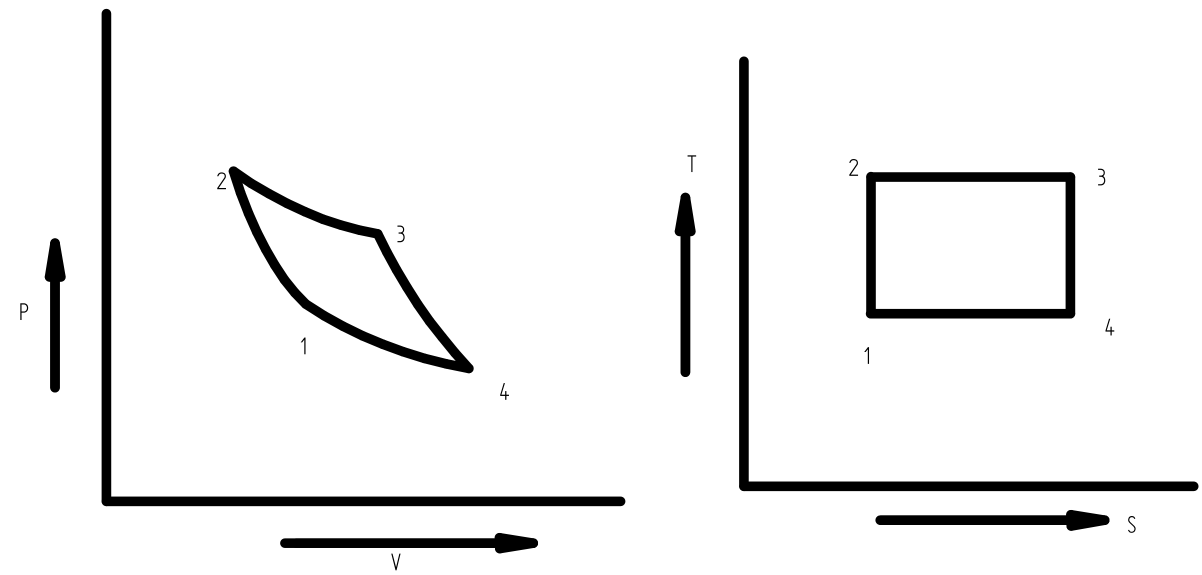
Actual efficiency is always less than air standard efficiency given by term relative efficiency

$$Relative\ Efficiency = \frac{Actual\ thermal\ efficiency}{Air\ standard\ efficiency}$$

2. State the assumptions made in air standard cycle analysis

- The gas in the cycle is perfect gas and obeys gas laws, has constant specific heats.
- The physical constants of gas are same as that of air.
- The expansion and compression are adiabatic process.
- No chemical reaction takes place in cylinder. Heat is supplied or rejected by source or sink.
- The cycle is considered as closed.

3. Derive an expression for the air standard efficiency of carnot cycle. Show the cycle on PV and T-S diagrams.



The Carnot cycle consists of an alternate series of two reversible isothermal and two reversible adiabatic processes. Since the processes in the cycle are all reversible the Carnot cycle as a whole is reversible

Process 2–3: Gas expands isothermally absorbing heat Q_s from the source at Temperature T_H. Work done during this process is given by the area under 2 – 3 (W_{2-3})
Process 3–4: During this process cylinder is thermally isolated from the heat reservoir and the head is insulated by the piece of perfect insulator. Gas expands reversibly and adiabatically to temperature T_L (point 4). Work done is W_{34}.
Process 4–1: Cylinder is in contact with the heat reservoir at T_L. Gas is isothermally and reversibly compressed to point 1 rejecting an amount of heat Q_R to the sink. The work done on the W_{41}.
Process 1-2: Cylinder is again isolated thermally from the thermal reservoir; gas is recompressed adiabatically and reversibly to point 2. The cycle is now complete. Work done is W_{12}

The efficiency of the Carnot engine is given by,

$$\eta_{Cornot} = \frac{Q_s - Q_r}{Q_s}$$

1-2 Process Adiabatic

$$Q_{12} = 0$$

2-3 Process: Isothermal Heat addition

$$Q_{23} = mRT_H \ln\frac{V_3}{V_2}$$

3-4 Process Adiabatic

$Q_{34} = 0$

4-1 Process: Isothermal Heat Rejection

$$Q_{4-1} = mRT_L \ln \frac{V_4}{V_1}$$

$$Net\ Heat\ supplied\ in\ the\ cycle\ \ Q_s = mRT_H \ln \frac{V_3}{V_2}$$

$$Net\ Heat\ rejected\ \ in\ the\ cycle\ Q_R = mRT_L \ln \frac{V_4}{V_1}$$

Substituting $Q_s and\ Q_R$ in efficiency formula

$$\eta_{Cornot} = \frac{mRT_H \ln\frac{V_3}{V_2} - mRT_L \ln\frac{V_4}{V_1}}{mRT_H \ln\frac{V_3}{V_2}} \text{ --------------A}$$

1-2 process adiabatic

$$\frac{T_2}{T_1} = \frac{T_H}{T_L} = \left(\frac{V_1}{V_2}\right)^{\gamma-1}$$

$$\frac{T_H}{T_L} = \left(\frac{V_1}{V_2}\right)^{\gamma-1} \text{------i)}$$

3-4 process adiabatic

$$\frac{T_3}{T_4} = \frac{T_H}{T_L} = \left(\frac{V_4}{V_3}\right)^{\gamma-1}$$

$$\frac{T_H}{T_L} = \left(\frac{V_4}{V_3}\right)^{\gamma-1} \text{----------ii)}$$

Comparing i) and ii)

$$\left(\frac{V_1}{V_2}\right)^{\gamma-1} = \left(\frac{V_4}{V_3}\right)^{\gamma-1}; \quad \frac{V_1}{V_2} = \frac{V_4}{V_3}\ ; \quad \frac{V_3}{V_2} = \frac{V_4}{V_1}$$

Substituting $\frac{V_3}{V_2} = \frac{V_4}{V_1}$ in Equation A

$$\eta_{Cornot} = \frac{T_H - T_L}{T_H}$$

4. Define work ratio and Mean effective pressure

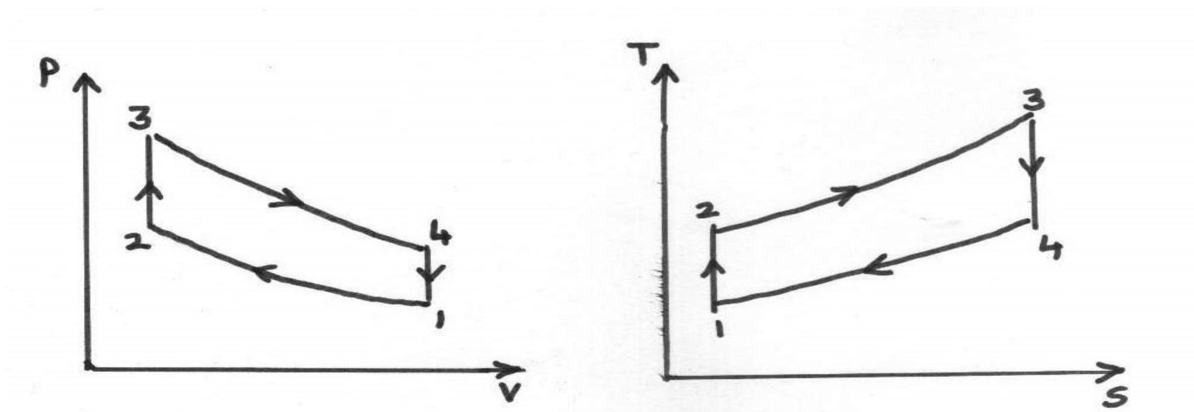
Work ratio is define as ratio of net work done to positive work

$$Work\ ratio = \frac{Wnet}{Posotive\ work\ done}$$

Mean effective pressure may be defined as the theoretical pressure which, if it is maintained constant throughout the volumechange of the cycle, would give the same work output as that obtained from the cycle.

$$MEP = \frac{Wnet}{Stroke\ Volume}$$

5. Derive an expression for the air standard efficiency of Otto combustion cycle in terms of compression ratio. Show the cycle on PV and T-S diagrams.



The cycle consists of two adiabatic processes and two constant volume processes as shown in P-V and T-S diagrams.

Process 1–2: Gas is compressed isentropically. Q=0

Process 2–3: During this process heat is supplied from the heat reservoir at constant volume. $Q_s = C_v(T_3 - T_2)$

Process 3–4: Gas expands isentropically to point 4. Q=0

Process 4–1: Heat is rejected to sink at constant volume. $Q_r = C_v(T_4 - T_1)$

$$\eta_{air\ standard} = \frac{Q_s - Q_r}{Q_s}$$

$$\eta_{air\ standard} = \frac{C_v(T_3 - T_2) - C_v(T_4 - T_1)}{C_v(T_3 - T_2)}$$

$$\eta_{air\ standard} = 1 - \frac{C_v(T_4 - T_1)}{C_v(T_3 - T_2)}$$

$$\eta_{air\ standard} = 1 - \frac{T_1(\frac{T_4}{T_1} - 1)}{T_2(\frac{T_3}{T_2} - 1)} \text{ ----------------A}$$

1-2 Process is adiabatic

$$\frac{T_2}{T_1} = \left(\frac{V_1}{V_2}\right)^{\gamma-1} = R_c^{\gamma-1}$$

Also,

3-4 adiabatic process

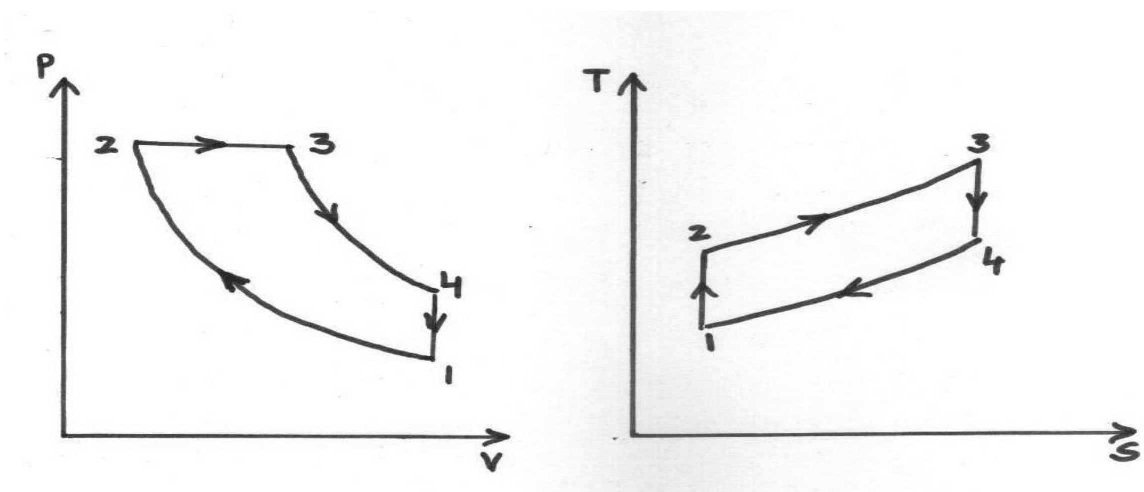
$$\frac{T_3}{T_4} = \left(\frac{V_4}{V_3}\right)^{\gamma-1} = \left(\frac{V_1}{V_2}\right)^{\gamma-1} = R_c^{\gamma-1}$$

$\frac{T_2}{T_1} = \frac{T_3}{T_4}$ or $\frac{T_4}{T_1} = \frac{T_3}{T_2}$

Substituting $\frac{T_4}{T_1} = \frac{T_3}{T_2}$ in equation A

$$\eta_{air\ standard} = 1 - \frac{1}{R_c^{\gamma-1}}$$

6. Derive an expression for the air standard efficiency of Diesel combustion cycle in terms of compression ratio and cut off ratio. Show the cycle on PV and T-S diagrams



This cycle is also known as constant pressure cycle because heat is added at constant pressure.

Process 1–2: Gas is compressed isentropically. Q=0

Process 2–3: During this process heat is supplied from the heat reservoir at constant pressure. $Q_s = C_p(T_3 - T_2)$
Process 3–4: Gas expands isentropically to point 4. Q=0
Process 4–1: Heat is rejected to sink at constant volume. $Q_r = C_v(T_4 - T_1)$

$$\eta_{air\ standard} = \frac{Q_s - Q_r}{Q_s}$$

$$\eta_{air\ standard} = \frac{C_p(T_3 - T_2) - C_v(T_4 - T_1)}{C_p(T_3 - T_2)}$$

$$\eta_{air\ standard} = 1 - \frac{C_v(T_4 - T_1)}{C_p(T_3 - T_2)}$$

$\eta_{air\ standard} = 1 - \frac{1}{\gamma}\frac{(T_4 - T_1)}{(T_3 - T_2)}$ --------------A

Let Compression ratio $R_c = \frac{V_1}{V_2}$, Cut off ratio $\rho = \frac{T_3}{T_2} = \frac{V_3}{V_2}$, Expansion Ratio $R_e = \frac{V_4}{V_3}$

$$R_e = \frac{V_4}{V_3} = \frac{V_1}{V_3} = \frac{V_1}{V_2} x \frac{V_2}{V_3} = \frac{R_c}{\rho}$$

1-2 adiabatic process

$\frac{T_2}{T_1} = \left(\frac{V_1}{V_2}\right)^{\gamma-1} = R_c^{\gamma-1}$ or $T_2 = T_1 R_c^{\gamma-1}$------------i)

2-3 Process is constant Pressure process $P_2 = P_3$

$\frac{P_2 V_2}{T_2} = \frac{P_3 V_3}{T_3}$ ie $\frac{V_2}{T_2} = \frac{V_3}{T_3}$ ie $\frac{V_3}{V_2} = \frac{T_3}{T_2}$

$\rho = \frac{T_3}{T_2}$; $T_3 = T_2 \rho$; $T_3 = T_1 R_c^{\gamma-1} \rho$--------------ii)

3-4 Adiabatic process

$\frac{T_3}{T_4} = \left(\frac{V_4}{V_3}\right)^{\gamma-1} = R_e{}^{\gamma-1}$ or $T_4 = \frac{T_3}{R_e{}^{\gamma-1}}$ or $T_4 = \frac{T_1 R_c^{\gamma-1} \rho \rho^{\gamma-1}}{R_c^{\gamma-1}} = T_1 \rho^{\gamma}$

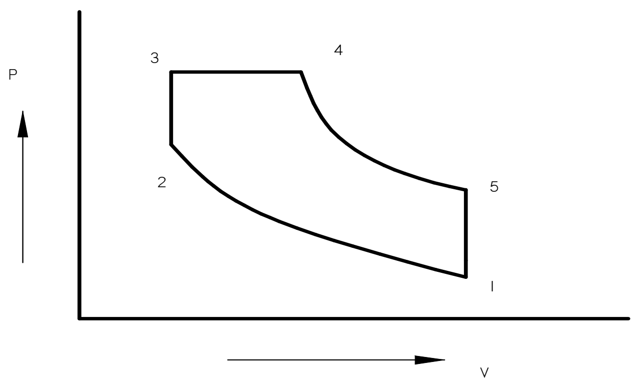
$T_4 = T_1 \rho^{\gamma}$--------------iii)

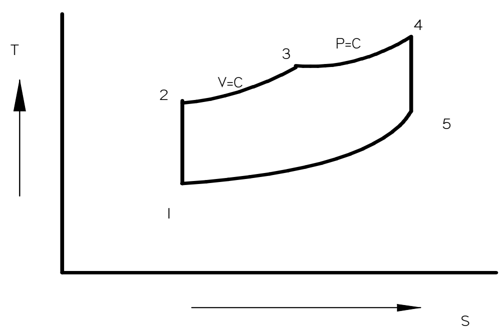
Substituting i) ii) and iii) in A

$$\eta_{air\ standard} = 1 - \frac{1}{\gamma}\frac{(T_1 \rho^{\gamma} - T_1)}{(T_1 R_c^{\gamma-1} \rho - T_1 R^{\gamma-1})}$$

$$\eta_{air\ standard} = 1 - \frac{1}{\gamma}\left[\frac{(\rho^{\gamma} - 1)}{R_c^{\gamma-1}(\rho - 1)}\right]$$

7. Derive an expression for the air standard efficiency of dual combustion cycle in terms of compression ratio , cut-off ratio and explosion ratio. Show the cycle on PV and T-S diagrams (March2001)





Process 1–2: Gas is compressed isentropically. Q=0

Process 2–3: During this process heat is supplied from the heat reservoir at constant volume.

$$Q_{sv} = C_v(T_3 - T_2)$$

Process 3–4: During this process heat is supplied from the heat reservoir at constant pressure.

$$Q_{sp} = C_p(T_4 - T_3)$$

Process 4–5: Gas expands isentropically to point 4. Q=0

Process 5–1: Heat is rejected to sink at constant volume. $Q_r = C_v(T_5 - T_1)$

$Q_s = Q_{sv} + Q_{sp} = C_v(T_3 - T_2) + C_p(T_4 - T_3)$

$$Q_r = C_v(T_5 - T_1)$$

$$\eta_{air\ standard} = \frac{Q_s - Q_r}{Q_s}$$

$$\eta_{air\ standard} = \frac{\left[C_v(T_3 - T_2) + C_p(T_4 - T_3)\ \right] - C_v(T_5 - T_1)}{C_v(T_3 - T_2) + C_p(T_4 - T_3)}$$

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

$$\eta_{air\ standard} = 1 - \frac{(T_5 - T_1)}{(T_3 - T_2) + \gamma(T_4 - T_3)}$$

Let Compression ratio $R_c = \frac{V_1}{V_2}$, Cut off ratio $\rho = \frac{T_4}{T_3} = \frac{V_4}{V_3}$, Expansion Ratio $R_e = \frac{V_5}{V_4}$

Explosion ratio $\alpha = \frac{p_3}{p_2}$

$R_c = \frac{V_1}{V_2} = \frac{V_5}{V_3} = \frac{V_5}{V_4} x \frac{V_4}{V_3} = R_e\rho$ as $V_2=V_3$, $V_5=V_4$

$$R_e = \frac{R_c}{\rho}$$

1-2 process adiabatic

$\frac{T_2}{T_1} = \left(\frac{V_1}{V_2}\right)^{\gamma-1} = R^{\gamma-1}$ or $T_2 = T_1 R_c^{\gamma-1}$

2-3 process constant volume $V_2 = V_3$

$$\frac{P_2 V_2}{T_2} = \frac{P_3 V_3}{T_3}$$

$\frac{p_2}{T_2} = \frac{p_3}{T_3}$ ie $T_3 = \frac{p_3}{p_2} x T_2 = \alpha T_1 R_c^{\gamma-1}$

3-4 constant pressure process $P_3 = P_4$

$\frac{P_3 V_3}{T_3} = \frac{P_4 V_4}{T_4}$; $\frac{V_3}{T_3} = \frac{V_4}{T_4}$ ie $T_4 = \frac{V_4}{V_3} x T_3 = \rho\ \alpha T_1 R_c^{\gamma-1}$

4-5 process adiabatic

$\frac{T_4}{T_5} = \left(\frac{V_5}{V_4}\right)^{\gamma-1} = R_e{}^{\gamma-1}$ or $T_4 = \frac{T_3}{R_e{}^{\gamma-1}}$ or $T_4 = \frac{\rho\alpha T_1 R_c^{\gamma-1} \rho^{\gamma-1}}{R_c^{\gamma-1}} = \rho^\gamma \alpha T_1$

$$\eta_{air\ standard} = 1 - \frac{(\rho^\gamma \alpha T_1 - T_1)}{\gamma\left(\rho\alpha T_1 R_c^{\gamma-1} - \alpha T_1 R_c^{\gamma-1}\right) + \left(\alpha T_1 R_c^{\gamma-1} - T_1 R_c^{\gamma-1}\right)}$$

$$\eta_{air\ standard} = 1 - \frac{(\rho^\gamma \alpha - 1)}{R_c^{\gamma-1}[\alpha\gamma(\rho - 1) + (\alpha - 1)]}$$

8. Derive an expression for mean effective pressure for otto cycle in terms of compression ratio and explosion ratio.

R= gas constant, r=compression ratio

$$MEP = \frac{Wnet}{Stroke\ Volume}$$

$$Wnet = heat\ supplied - heat\ rejected = Q_s - Q_r$$

$$MEP = \frac{Q_s - Q_r}{Stroke\ Volume}$$

$$Stroke\ volume = V_1 - V_2 = V_1\left(1 - \frac{V_2}{V_1}\right) = \frac{RT_1}{P_1}\left(1 - \frac{1}{R_c}\right) = \frac{RT_1}{P_1 R_c}(R_c - 1)$$

$$MEP = \frac{Q_s - Q_r}{\frac{RT_1}{P_1 R_c}(R_c - 1)}$$

$$MEP = \frac{Q_s - Q_r}{\frac{(C_p - C_v)T_1}{P_1 R_c}(R_c - 1)}$$

$$MEP = P_1 R_c x \frac{Q_s - Q_r}{C_v \frac{(C_p - C_v)T_1}{C_v}(R_c - 1)}$$

$$MEP = P_1 R_c x \frac{Q_s - Q_r}{C_v(\gamma - 1)(R_c - 1)T_1}$$

$MEP = \frac{P_1 R_c}{T_1} x \frac{C_v(T_3 - T_2) - C_v(T_4 - T_1)}{C_v(\gamma - 1)(R_c - 1)}$ ---------------------1

1-2 process adiabatic

$\frac{T_2}{T_1} = \left(\frac{V_1}{V_2}\right)^{\gamma - 1} = R_c^{\gamma - 1}$or $T_2 = T_1 R_c^{\gamma - 1}$

2-3 process constant volume $V_2 = V_3$

$$\frac{P_2 V_2}{T_2} = \frac{P_3 V_3}{T_3}$$

$\frac{p_2}{T_2} = \frac{p_3}{T_3}$ ie $T_3 = \frac{p_3}{p_2} x T_2 = \alpha T_1 R_c^{\gamma - 1}$

3-4 process adiabatic

$$\frac{T_3}{T_4} = \left(\frac{V_4}{V_3}\right)^{\gamma-1} = \left(\frac{V_1}{V_2}\right)^{\gamma-1} = R_c^{\gamma-1} \text{ as } V_4 = V_1 \; and \; V_3 = V_2$$

$$T_4 = \frac{T_3}{R_c^{\gamma-1}} = \frac{\alpha T_1 R_c^{\gamma-1}}{R_c^{\gamma-1}} = \alpha T_1$$

$$MEP = \frac{P_1 R_c}{T_1} x \frac{C_v(T_3 - T_2) - C_v(T_4 - T_1)}{C_v(\gamma - 1)(R_c - 1)}$$

$$MEP = \frac{P_1 R_c}{T_1} x \frac{C_v(\alpha T_1 R_c^{\gamma-1} - T_1 R_c^{\gamma-1}) - C_v(\alpha T_1 - T_1)}{C_v(\gamma - 1)(R_c - 1)}$$

$$MEP = \frac{P_1 R_c}{T_1} \frac{T_1\left[R_c^{\gamma-1}(\alpha - 1) - (\alpha - 1)\right]}{(\gamma - 1)(R_c - 1)}$$

$$MEP = \frac{P_1 R_c\left[(\alpha - 1)(R_c^{\gamma-1} - 1)\right]}{(\gamma - 1)(R_c - 1)}$$

9. Derive an expression for mean effective pressure for Diesel cycle in terms of compression ratio and cut-off ratio

R= gas constant, r=compression ratio

$$MEP = \frac{Wnet}{Stroke\ Volume}$$

$$Wnet = heat\ supplied - heat\ rejected = Q_s - Q_r$$

$$MEP = \frac{Q_s - Q_r}{Stroke\ Volume}$$

$$Stroke\ volume = V_1 - V_2 = V_1\left(1 - \frac{V_2}{V_1}\right) = \frac{RT_1}{P_1}\left(1 - \frac{1}{R_c}\right) = \frac{RT_1}{P_1 R_c}(R_c - 1)$$

$$MEP = \frac{Q_s - Q_r}{\frac{RT_1}{P_1 R_c}(R_c - 1)}$$

$$MEP = \frac{Q_s - Q_r}{\frac{(C_p - C_v)T_1}{P_1 R_c}(R_c - 1)}$$

$$MEP = P_1 R_c x \frac{Q_s - Q_r}{C_v \frac{(C_p - C_v)T_1}{C_v}(R_c - 1)}$$

$$MEP = P_1 R_c x \frac{Q_s - Q_r}{C_v(\gamma - 1)(R_c - 1)T_1}$$

$$R_e = \frac{V_4}{V_3} = \frac{V_1}{V_3} = \frac{V_1}{V_2} x \frac{V_2}{V_3} = \frac{R_c}{\rho}$$

1-2 adiabatic process

$\frac{T_2}{T_1} = \left(\frac{V_1}{V_2}\right)^{\gamma-1} = R_c^{\gamma-1}$or $T_2 = T_1 R_c^{\gamma-1}$

2-3 process (constant volume process. $\frac{V}{T} = C$)

$\frac{V_2}{T_2} = \frac{V_3}{T_3}$ ie $T_3 = \frac{V_3}{V_2} x T_2$

$T_3 = \rho T_2 = \rho T_1 R_c^{\gamma-1}$

3-4 process (adiabatic process)

$\frac{T_3}{T_4} = \left(\frac{V_4}{V_3}\right)^{\gamma-1} = {R_e}^{\gamma-1}orT_4 = \frac{T_3}{{R_e}^{\gamma-1}} = \frac{T_3}{R_c^{\gamma-1}} x \rho^{\gamma-1}$

$$T_4 = \frac{\rho T_1 R_c^{\gamma-1} \rho^{\gamma-1}}{R_c^{\gamma-1}} = \rho^{\gamma} T_1$$

$$MEP = P_1 R_c x \frac{Q_s - Q_r}{C_v(\gamma - 1)(R_c - 1)T_1}$$

$$MEP = P_1 R_c \left\{ \frac{C_p(\rho T_1 R_c^{\gamma-1} - T_1 R_c^{\gamma-1}) - C_v(\rho^{\gamma} T_1 - T_1)}{C_v(\gamma - 1)(R_c - 1)T_1} \right\}$$

$$MEP = P_1 R_c C_v T_1 \left\{ \frac{\gamma(\rho R_c^{\gamma-1} - R_c^{\gamma-1}) - (\rho^{\gamma} T_1 - T_1)}{C_v(\gamma - 1)(R_c - 1)T_1} \right\}$$

$$MEP = \frac{P_1 R_c \; [\gamma R_c^{\gamma-1}(\rho - 1) - (\rho^{\gamma} - 1)}{(\gamma - 1)(r - 1)}$$

Expression for cut-off ratio in terms of k:Let ‘k’ be the cut-off in percentage of stroke (from V1 to V2)

$$\rho = \frac{V_3}{V_2} = \frac{V_c + kV_s}{V_c} = 1 + \frac{kV_s}{V_c} = 1 + \frac{k(V_1 - V_c)}{V_c} = 1 + k(R - 1)$$

10. Derive an expression for mean effective pressure for Duel cycle in terms of compression ratio, cut-off ratio and explosion ratio

R= gas constant, r=compression ratio

$$MEP = \frac{Wnet}{Stroke\ Volume}$$

$$Wnet = heat\ supplied - heat\ rejected = Q_s - Q_r$$

$$MEP = \frac{Q_s - Q_r}{Stroke\ Volume}$$

$$Stroke\ volume = V_1 - V_2 = V_1\left(1 - \frac{V_2}{V_1}\right) = \frac{RT_1}{P_1}\left(1 - \frac{1}{R_c}\right) = \frac{RT_1}{P_1R_c}(R_c - 1)$$

$$MEP = \frac{Q_s - Q_r}{\frac{RT_1}{P_1R_c}(R_c - 1)}$$

$$MEP = \frac{Q_s - Q_r}{\frac{(C_p - C_v)T_1}{P_1R_c}(R_c - 1)}$$

$$MEP = P_1R_c x \frac{Q_s - Q_r}{C_v\frac{(C_p - C_v)T_1}{C_v}(R_c - 1)}$$

$$MEP = P_1R_c x \frac{Q_s - Q_r}{C_v(\gamma - 1)(R_c - 1)T_1}$$

$$Q_s = Q_{sv} + Q_{sp}$$

$$Q_{sv} = C_v(T_3 - T_2);\ Q_{sp} = C_p(T_4 - T_3)$$

$$Q_s = C_v(T_3 - T_2) + C_p(T_4 - T_3)$$

$$Q_R = C_v(T_5 - T_1)$$

$$MEP = P_1 R_c x \frac{Q_s - Q_r}{C_v(\gamma - 1)(R_c - 1)T_1}$$

$$MEP = P_1 R_c x \frac{\left(C_v(T_3 - T_2) + C_p(T_4 - T_3)\right) - C_v(T_5 - T_1)}{C_v(\gamma - 1)(R_c - 1)T_1}$$

$$MEP = P_1 R_c x \frac{C_v\left((T_3 - T_2) + \gamma(T_4 - T_3) - (T_5 - T_1)\right)}{C_v(\gamma - 1)(R_c - 1)T_1}$$

1-2 process adiabatic

$\frac{T_2}{T_1} = \left(\frac{V_1}{V_2}\right)^{\gamma-1} = R^{\gamma-1}$or $T_2 = T_1 R_c^{\gamma-1}$

2-3 process constant volume

$\frac{p_2}{T_2} = \frac{p_3}{T_3}$ ie $T_3 = \frac{p_3}{p_2} x T_2 = \alpha T_1 R_c^{\gamma-1}$

3-4 process constant pressure process

$\frac{V_4}{T_4} = \frac{V_3}{T_3}$ ie $T_4 = \frac{V_4}{V_3}\ x\ T_3$

$T_4 = \rho\ \alpha T_1 R_c^{\gamma-1}$

4-5 process adiabatic

$\frac{T_4}{T_5} = \left(\frac{V_5}{V_4}\right)^{\gamma-1} = R_e^{\gamma-1}$

$\frac{V_1}{V_2} = \frac{V_5}{V_3} = \frac{V_5}{V_4}\, x \frac{V_4}{V_3}$

$R_c = R_e \rho$

$$T_5 = \frac{T_4}{R_e^{\gamma-1}} = \frac{\rho\alpha T_1 R_c^{\gamma-1}}{\left(\frac{R_c}{\rho}\right)^{\gamma-1}} = \alpha\rho^{\gamma} T_1$$

$$MEP = P_1 R_c x \frac{C_v\left((T_3 - T_2) + \gamma(T_4 - T_3) - (T_5 - T_1)\right)}{C_v(\gamma - 1)(R_c - 1)T_1}$$

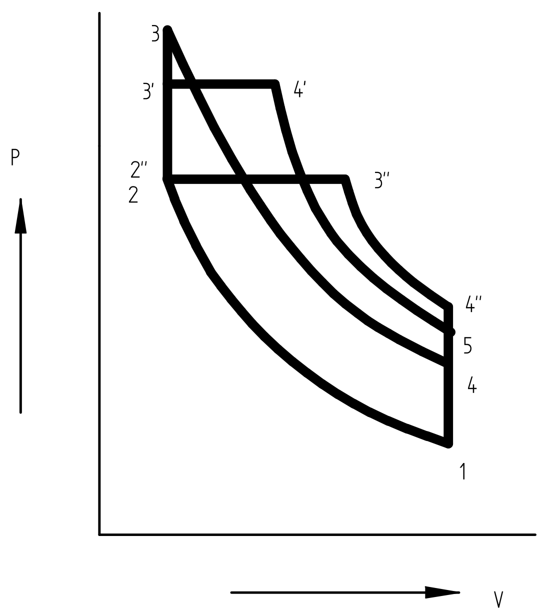
$$MEP = P_1 R_c x \frac{C_v \left(\left(\alpha T_1 R_c^{\gamma-1} - T_1 R_c^{\gamma-1}\right) + \gamma\left(\rho\ \alpha T_1 R_c^{\gamma-1} - \alpha T_1 R_c^{\gamma-1}\right) - (\alpha\rho^{\gamma} T_1 - T_1)\right)}{C_v(\gamma - 1)(R_c - 1)T_1}$$

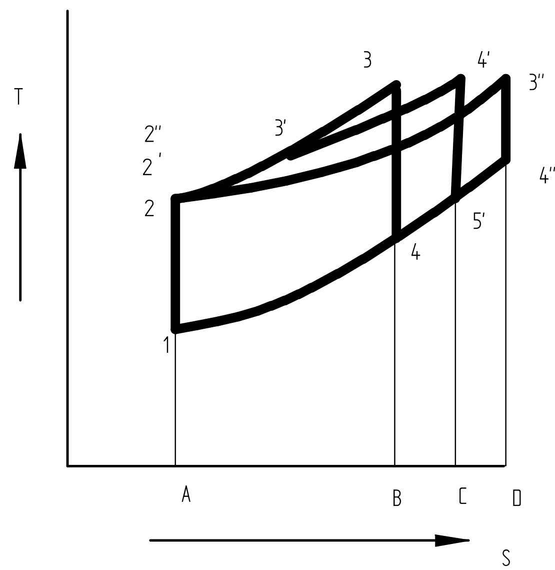
$$MEP = P_1 R_c x \frac{C_v T_1 \left(\left(\alpha R_c^{\gamma-1} - R_c^{\gamma-1} \right) + \gamma \left(\rho \, \alpha R_c^{\gamma-1} - \alpha R_c^{\gamma-1} \right) - \left(\alpha \rho^{\gamma} - 1 \right) \right)}{C_v (\gamma - 1)(R_c - 1) T_1}$$

$$MEP = \frac{P_1 R_c \left[R_c^{\gamma-1} (\alpha - 1) + \gamma \alpha R_c^{\gamma-1} (\rho - 1) \right] - (\rho^{\gamma} \alpha - 1))}{(\gamma - 1)(r - 1)}$$

$$MEP = \frac{P_1 R_c \left[R_c^{\gamma-1} \left((\alpha - 1) + \gamma \alpha (\rho - 1) \right) \right] - (\rho^{\gamma} \alpha - 1))}{(\gamma - 1)(R_c - 1)}$$

11. Prove that for same compression ratio and heat input, Otto cycle efficiency is more than Diesel cycle





The comparison of these cycles for the same compression ratio and same heat supply are shown in on both p – V and T – S diagrams. In these diagrams, cycle 1-2-3-4-1 represents Otto Cycle, cycle 1-2’-3’-4’-5’-1 represents Dual cycle and cycle 1-2”-3”-4”- 1 represents the diesel combustion cycle for the same compression ratio and heat supply.

From the data heat supplied for all the cycle is same

Hence area under heat supply curve in T-S diagram for all the cycle is same

Hence area A23B = area A2’3’4’C = area A2”3”D as All the cycles start from the same initial point 1 and the air is compressed from state 1 to state 2 as the compression ratio is same.

$$\eta = 1 - \frac{Q_R}{Q_s}$$

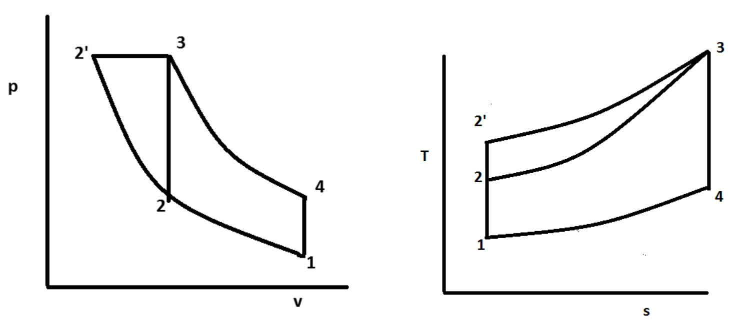
From the above equation it can be seen that since Q_s is same for all the three cycles efficiency of the cycle depends on Q_R and efficiency is inversely proportional to heat of rejection

It is seen from the T-s diagram, area A14B $(Q_R\ for\ otto\ cycle)$ < Area A1"5"C $(Q_R\ for\ dualcycle)$ < Area A1'4'D $(Q_R\ for\ diesel\ cycle)$

Consequently Otto cycle has the highest work output and efficiency. Diesel cycle has the least efficiency and dual cycle has the efficiency between the two.Therefore for the same compression ratio and same heat rejection, Otto cycle is the most efficient while the Diesel cycle is the least efficient.

$$\eta_{Otto} < \eta_{Diesel} < \eta_{Dual}$$

12. Prove that for same Maximum pressure and maximum temperature Diesel cycle efficiency is more than Otto cycle or Prove that for same Maximum pressure and same output Diesel cycle efficiency is more than Otto cycle



12341 represents otto and 12'3'4'1 diesel cycle respectively in T-S diagram and P-V diagram

Since Maximum pressure and maximum temperature is same for both cycles 3 is the common point for both cycles in PV and TS diagram

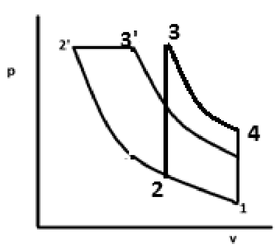
$$\eta = 1 - \frac{Q_R}{Q_s}$$

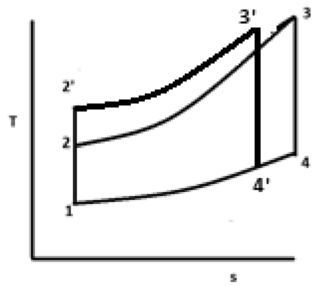
From the above equation it can be seen that since Q_R is same for diesel and otto cycles efficiency of the cycle depends on Q_S and efficiency is proportional to heat of supply

It is seen from the T-s diagram, area A23B $(Q_s\ for\ otto\ cycle)$ < Area A2'3'B$(Q_R\ for\ diesel\ cycle)$

Hence, $\eta_{Otto} < \eta_{Diesel}$

13. Prove that for same Maximum pressure and same heat input Diesel cycle efficiency is more than Otto cycle





12341 represents otto and 12’3’4’1 diesel cycle respectively in T-S diagram and P-V diagram

From the data heat supplied for all the cycle is same

Hence area under heat supply curve in T-S diagram for both the cycle is same

Hence area A23B = area A2’3’C as All the cycles start from the same initial point 1

$$\eta = 1 - \frac{Q_R}{Q_S}$$

From the above equation it can be seen that since Q_S is same for both the cycles efficiency of the cycle depends on Q_R and efficiency is inversely proportional to heat of rejection

It is seen from the T-s diagram, area A14B $(Q_R\ for\ otto\ cycle) >$ Area A14’C$(Q_R\ for\ diesel\ cycle)$

From figure heat rejection in diesel cycle is less than Otto cycle

$\eta_{Diesel} > \eta_{Otto}$

14. In an otto cycle the upper and lower limits for the absolute temperature respectively are T_3 and T_1. Show that for the maximum work the ratio of compression should have the value

$$R_c = \left(\frac{T_3}{T_1}\right)^{1.25}$$

$$Wnet = heat\ supplied - heat\ rejected = C_V(T_3 - T_2) - C_v(T_4 - T_1)$$

$$\frac{T_2}{T_1} = \left(\frac{V_1}{V_2}\right)^{\gamma-1} = R_c^{\gamma-1} \quad \text{or} \quad T_2 = T_1 R_c^{\gamma-1}$$

$\frac{T_3}{T_4} = \left(\frac{V_4}{V_3}\right)^{\gamma-1} = R_c^{\gamma-1}$ or $\quad T_4 = \frac{T_3}{R_c^{\gamma-1}}$

$$Wnet = C_V(T_3 - T_1R_c^{\gamma-1}) - C_v\left(\frac{T_3}{R_c^{\gamma-1}} - T_1\right)$$

$$\frac{dw}{dR_c} = 0\, for\ maximum\ work$$

T_1, T_3 are constants

$$0 - C_VT_1(\gamma-1)R_c^{\gamma-1-1} - C_VT_3(-1)(\gamma-1)R_c^{-(\gamma-1)-1} - 0 = 0$$

$$T_1R_c^{\gamma-1-1} = T_3\ R_c^{-(\gamma-1)-1}$$

$$T_1R_c^{\gamma-2} = \frac{T_3}{R_c^{\gamma}}$$

$$T_1R_c^{\gamma-2}R_c^{\gamma} = T_3$$

$$T_1R_c^{2\gamma-2} = T_3$$

$$T_1R_c^{2(\gamma-1)} = T_3$$

$$R_c = \left(\frac{T_3}{T_1}\right)^{\frac{1}{2(\gamma-1)}} = \left(\frac{T_3}{T_1}\right)^{1.25}$$

15. An engine working on otto cycle in which the salient points are 1,2,3 and 4 has upper and lower temperature limits T_3 and T_1. If the maximum work per kg of air is to be done, show that the intermediate temperatures are given by $T_2=T_4=\sqrt[2]{T_1T_3}$

$$T_4 = \frac{T_3}{R_c^{\gamma-1}}$$

But $R_c = \left(\frac{T_3}{T_1}\right)^{\frac{1}{2(\gamma-1)}}$ from problem 14.

$$T_4 = \frac{T_3}{\left(\left(\frac{T_3}{T_1}\right)^{\frac{1}{2(\gamma-1)}}\right)^{\gamma-1}}$$

$$T_4 = \frac{T_3}{\left(\frac{T_3}{T_1}\right)^{\frac{1}{2}}}$$

$T_4 = \sqrt[2]{T_1 T_3}$

$$T_2 = T_1 R_c^{\gamma-1}$$

But $R_c = \left(\frac{T_3}{T_1}\right)^{\frac{1}{2(\gamma-1)}}$ from problem 14.

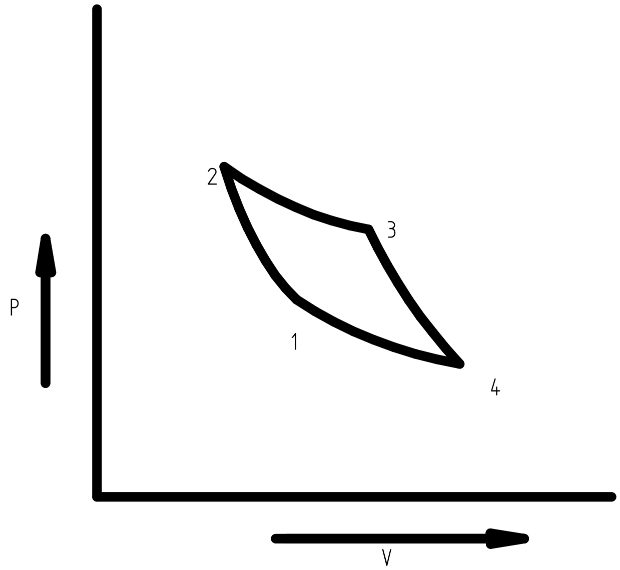
$$T_2 = T_1 \left[\left(\frac{T_3}{T_1}\right)^{\frac{1}{2(\gamma-1)}}\right]^{\gamma-1}$$

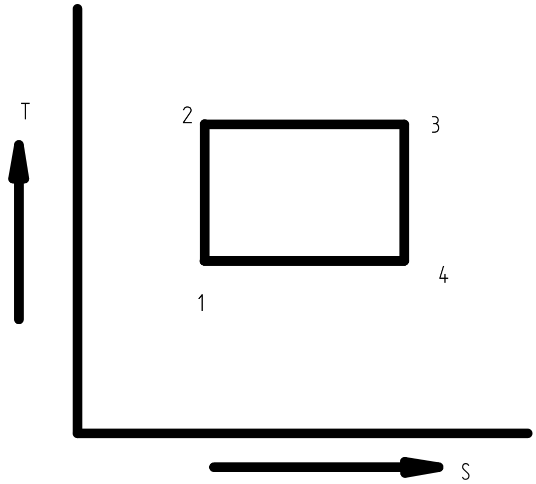
$T_2 = \sqrt[2]{T_1 T_3}$

$T_2 = T_4 = \sqrt[2]{T_1 T_3}$

NUMERICALS :

1. The minimum pressure and temperature of the air standard Carnot cycle are 1 bar and 15^{o}C respectively. The pressure after isothermal compression is 3.5 bar and the pressure after isentropic compression is 10.5 bar. Determine i) Efficiency ii) Mean effective pressure and the Power developed if the Carnot engine makes 2 cycles/s. Take for air R=.287kJ/kgK and γ=1.4 (May June 2010)





Given:

The minimum pressure and temperature of the air standard Carnot cycle are 1 bar and 15°C respectively. P_4=1bar; T_1=288K

The pressure after isothermal compression is 3.5 bar, P_1=3.5bar

the pressure after isentropic compression is 10.5 bar. P_2=10.5bar

$T_1 = T_4 = T_L$=288K

$T_2 = T_3 = T_H$

1-2 Process adiabatic

$\frac{T_2}{T_1} = \left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}}; \frac{T_2}{288} = \left(\frac{10.5}{3.5}\right)^{\frac{1.4-1}{1.4}}$ $ie\ T_2 = 394.2K$

Ie $T_2 = T_3 = T_H = 394.2K$

$$\eta = \frac{T_h - T_l}{T_h} = \frac{394.2 - 288}{394.2} = 0.269$$

Work Net /kg= $Q_s - Q_R$

Q_s *is during isothermal process* 2-3 process

$Q_s = P_2V_2 ln\frac{V_3}{V_2} = mRT_2\ ln\frac{V_3}{V_2} = mRT_H\ ln\frac{V_3}{V_2}$

Q_R *is during isothermal process* 3-4 process

$Q_s = P_3V_3 ln\frac{V_3}{V_2} = mRT_3\ ln\frac{V_4}{V_1}$

$\frac{T_2}{T_1} = \frac{T_H}{T_L} = \left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}}$ ---------1

$\frac{T_3}{T_4} = \frac{T_H}{T_L} = \left(\frac{P_3}{P_4}\right)^{\frac{\gamma-1}{\gamma}}$ ---------------2

From 1 and 2

$$\frac{T_H}{T_L} = \left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}} = \left(\frac{P_3}{P_4}\right)^{\frac{\gamma-1}{\gamma}}$$

$\frac{P_2}{P_1} = \frac{P_3}{P_4}$

$\frac{10.5}{3.5} = \frac{P_3}{1}$ ie $P_3 = 3\ bar$

2-3 Process is Isothermal

$P_2V_2 = P_3V_3$; $\frac{P_2}{P_3} = \frac{V_3}{V_2}$; $\frac{10.5}{3} = \frac{V_3}{V_2}$; $\frac{V_3}{V_2} = 3.5$

4-1 Process is Isothermal

$P_4V_4 = P_1V_1$; $\frac{P_1}{P_4} = \frac{V_4}{V_1}$; $\frac{3.5}{1} = \frac{V_4}{V_1}$; $\frac{V_4}{V_1} = 3.5$

$Q_s = mRT_H\ ln\frac{V_3}{V_2}$=1x0.287x394.2 x ln3.5=141.73 kJ/kg

$Q_R = mRT_L\ ln\frac{V_1}{V_4}$=1x0.287x288 x ln3.5 = 103.54 kJ/kg

W_{net}= $Q_s -\ Q_R$ =141. 73 − 103.54 = 38.19kJ/kg

$p_m = \frac{W_{net}}{Swept\ Volume}$; $p_m = \frac{W_{net}}{V_4 - V_2}$

$P_4V_4 = mRT_4$

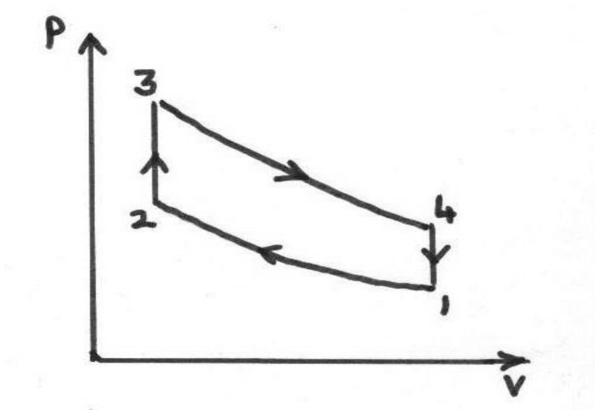
$V_4 = \frac{mRT_4}{P_4} = \frac{mRT_L}{P_4} = \frac{1x0.287x288}{1x10^2}$ =0.82656 m^3/kg

$P_2V_2 = mRT_2$

$V_2 = \frac{mRT_2}{P_2} = \frac{mRT_H}{P_2} = \frac{1x0.287x394.2}{10.5x10^2}$ =0.10778 m^3/kg

$p_m = \frac{W_{net}}{V_4 - V_2} = \frac{33.48}{0.82656 - 0.10778}$ = 46.57kPa

2. A petrol engine works on Otto cycle under ideal conditions. The initial pressure before the beginning of compression is 101kPa at 340K. The pressure at the end of heat addition process is 3.5Mpa. As per the details furnished by the manufacturer engine has stroke length twice the bore. Engine bore is 300mm and clearance volume $4x10^{-3}m^3$. Determine: i) compression ratio ii) The air standard efficiency iii) the mean effective pressure (Dec 09- Jan10).



Given:

The initial pressure before the beginning of compression is 101kPa at 340K.

P_1=101kPa,T_1=340K,

The pressure at the end of heat addition process is 3.5Mpa.

P_3=3.5Mpa =35bar,

As per the details furnished by the manufacturer engine has stroke length twice the bore L=2d,

Engine bore is 300mm and clearance volume $4x10^{-3}m^3$.

D=0.3m,$V_c=4x10^{-3}m^3$.

$R_c = \frac{V_1}{V_2} = \frac{V_C + V_S}{V_C}$

$V_s = \frac{\pi D^2}{4} x l = \frac{\pi 0.3^2}{4} x0.6 = 0.0424m^3$

$R_c = \frac{0.0424+4x10^{-3}}{4x10^{-3}}$=11.6

ŋ air standard efficiency $= 1 - \frac{1}{R_c^{\gamma-1}}$

ŋ air standard efficiency $= 1 - \frac{1}{11.6^{1.4-1}} = 0.625$

$$MEP = \frac{R_c P_1(\alpha - 1)(R_c^{\gamma-1} - 1)}{(\gamma - 1)(R - 1)}$$

$P_1 V_1{}^{\gamma} = P_2 V_2{}^{\gamma}$; $\frac{p_2}{p_1} = \left(\frac{V_1}{V_2}\right)^{\gamma}$; $P_2 = P_1 R_c^{\gamma}$

$P_2 = 101x11.6^{1.4} = 3122.9kPa$

$\alpha = \frac{P_3}{P_2} = \frac{3.5}{3.1229} = 1.12$

$$MEP = \frac{11.6x101(1.12 - 1)(11.6^{1.4-1} - 1)}{(1.4 - 1)(11.6 - 1)} = 55.22kPa$$

Alternative Method

$P_2 = P_1 R_c^{\gamma}$; $P_3 = \alpha P_2$; $P_4 = \frac{P_3}{R_c^{\gamma}}$
$T_2 = T_1 R_c^{\gamma-1}$; $T_3 = \alpha T_2 = \alpha T_1 R_c^{\gamma-1}$; $T_4 = \alpha T_1$
$P_2 = P_1 R_c^{\gamma}$ =1.01x11.6$^{1.4}$=31.22 bar
$T_2 = T_1 R_c^{\gamma-1}$=340x11.6$^{1.4-1}$=906.27K
P_3 =35bar(given)
$P_3 = \alpha P_2$; 35= $\alpha x 31.22$
Explosion ratio, α =1.12
$T_3 = \alpha T_2$; $T_3 = 1.12x906.27$=1015.02K
$T_4 = \alpha T_1$; $T_4 = 1.12x340$; $T_4 = 380.8K$

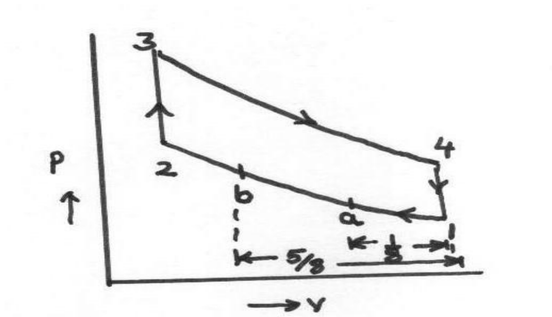
$P_4 = \frac{P_3}{R_c^{\gamma}} = \frac{35}{11.6^{1.4}}$=1.13 bar
$Q_s = C_v(T_3 - T_2) = 0.718(1015.02 - 906.27) = 78.08kJ/kg$
$Q_R = C_v(T_4 - T_1) = 0.718(380.8 - 340) = 29.29kJ/kg$
$\eta = 1 - \frac{\boldsymbol{Q_R}}{\boldsymbol{Q_s}}$ =1 $- \frac{29.29}{78.083}$ =0.6248 ie $\eta = 46.9\%$
$W = \boldsymbol{Q_s} - \boldsymbol{Q_R} = 78.08 - 29.29 = 48.79\ kJ/kg$

$$p_m = \frac{W_{net}}{Swept\ Volume} = \frac{W_{net}}{V_1 - V_2} = \frac{W_{net}}{V_1\left(1 - \frac{V_2}{V_1}\right)} = \frac{W_{net}}{\frac{RT_1}{P_1}\left(1 - \frac{1}{R_c}\right)} = \frac{P_1 R_c W_{net}}{RT_1(R_c - 1)}$$

$$p_m = \frac{1x11.6x48.79}{0.287x340(11.6-1)} = 0.547 \text{ bar} = 54.7\text{kPa}$$

3. From PV diagram of an engine working on otto cycle, it is found that the pressure in the cylinder after 1/8th of the compression stroke is executed is 1.4 bar. After 5/8th of the compression stroke, the pressure is 3.5 bar. Compute the compression ratio and the air standard efficiency. Also if the maximum cycle temperature is limited to 1000°C, find the network output (Jan/Feb2005)

Solution: the maximum cycle temperature is limited to 1000°C ie T_3=1000°C



the pressure in the cylinder after 1/8th of the compression stroke is executed is 1.4 bar

Compression process 12 starts from 1 Let 'a ' the point on the curve 12 after 1/8th of the compression stroke and p_a=1.4 bar

Ie $V_a = V_1 - \frac{1}{8}V_s$ refering from origin of pv diagram where $V_s = V_1 - V_2$

$V_a = V_1 - \frac{1}{8}(V_1 - V_2)$ where as $V_2 = V_c$

After 5/8th of the compression stroke, the pressure is 3.5 bar

Let 'b ' the point on the curve 12 after 5/8th of the compression stroke and p_b=3.5bar

$V_b = V_1 - \frac{5}{8}V_s$; $\quad V_b = V_1 - \frac{5}{8}(V_1 - V_2)$

a to b is adiabatic process

$$P_a V_a{}^{\gamma} = P_b V_b{}^{\gamma}$$

$$\frac{V_a}{V_b} = \left(\frac{P_b}{P_a}\right)^{\frac{1}{\gamma}} = \left(\frac{3.5}{1.4}\right)^{\frac{1}{1.4}} = 1.924$$

$$\frac{V_1 - \frac{1}{8}(V_1 - V_2)}{V_1 - \frac{5}{8}(V_1 - V_2)} = 1.924$$

$$\frac{\frac{V_1}{V_2} - \frac{1}{8}\left(\frac{V_1}{V_2} - 1\right)}{\frac{V_1}{V_2} - \frac{5}{8}\left(\frac{V_1}{V_2} - 1\right)} = 1.924$$

$$\frac{R_c - \frac{1}{8}(R_c - 1)}{R_c - \frac{5}{8}(R_c - 1)} = 1.924$$

ie $\frac{\frac{7}{8}R_c + \frac{1}{8}}{\frac{3}{8}R_c + \frac{5}{8}}$ =1.924 ; $7R_c + 1 = 1.924(3R_c + 5)$

$$7R_c = 1.924x3R_c + 1.924x5 - 1$$

$$R_c(7 - 1.924x3) = 8.62$$

$$R_c = 7.01$$

ŋ air standard efficiency $= 1 - \frac{1}{R_c^{\gamma-1}}$; ŋ air standard efficiency $= 1 - \frac{1}{7.01^{1.4-1}} = 0.54$

$$T_2 = T_1 R_c^{\gamma-1} = 300x7.01^{1.4-1} = 653.37K$$

$$T_4 = \frac{T_3}{R_c^{\gamma-1}} = \frac{1273}{7.01^{1.4-1}} = 584.5K$$

$$Work\ output = C_v(T_3 - T_2) - C_v(T_4 - T_1)$$

$$Work\ output = 0.718(1273 - 653.37) - 0.718(584.5 - 300) = 240.6kJ/kg$$

Alternative method

$P_2 = P_1 R_c^{\gamma}$; $P_3 = \alpha P_2$; $P_4 = \frac{P_3}{R_c^{\gamma}}$

$T_2 = T_1 R_c^{\gamma-1}$; $T_3 = \alpha T_2 = \alpha T_1 R_c^{\gamma-1}$; $T_4 = \alpha T_1$

From fig, $V_a = V_2 + (V_s - \frac{1}{8}V_s)$ refering from origin of pv diagram where $V_s = V_1 - V_2$

$$P_a {V_a}^{\gamma} = P_b {V_b}^{\gamma}$$

$$\frac{V_a}{V_b} = \left(\frac{P_b}{P_a}\right)^{\frac{1}{\gamma}} = \left(\frac{3.5}{1.4}\right)^{\frac{1}{1.4}} = 1.924$$

$$\frac{V_1 - \frac{1}{8}(V_1 - V_2)}{V_1 - \frac{5}{8}(V_1 - V_2)} = 1.924$$

$$\frac{\frac{V_1}{V_2} - \frac{1}{8}\left(\frac{V_1}{V_2} - 1\right)}{\frac{V_1}{V_2} - \frac{5}{8}\left(\frac{V_1}{V_2} - 1\right)} = 1.924$$

$$\frac{R_c - \frac{1}{8}(R_c - 1)}{R_c - \frac{5}{8}(R_c - 1)} = 1.924$$

ie $\frac{\frac{7}{8}R_c + \frac{1}{8}}{\frac{3}{8}R_c + \frac{5}{8}}$ =1.924 ; $7R_c + 1 = 1.924(3R_c + 5)$

$$7R_c = 1.924x3R_c + 1.924x5 - 1$$

$$R_c(7 - 1.924x3) = 8.62$$

$R_c = 7.01$

$P_2 = P_1 R_c^{\gamma}$ =1x7.01$^{1.4}$=15.24 bar
$T_2 = T_1 R_c^{\gamma-1}$=300x7.01$^{1.4-1}$=653.37K
Explosion ratio, α =1.12
$T_3 = \alpha T_2$; $1273 = \alpha x 653.37$; Explosion ratio, $\alpha = 1.948$
$P_3 = \alpha P_2$; P_3= $1.948x15.24$; $P_3 = 29.69 bar$

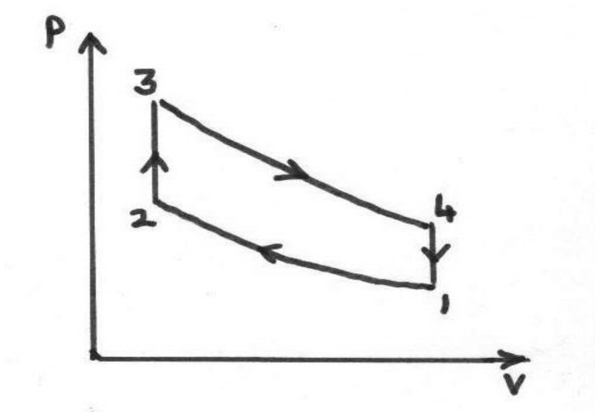
$T_4 = \alpha T_1$; $T_4 = 1.948x300$; $T_4 = 584.4K$

$P_4 = \frac{P_3}{R_c^{\gamma}} = \frac{29.69}{7^{1.4}}$=1.95 bar
$Q_s = C_v(T_3 - T_2) = 0.718(1273 - 653.3) = 444.89 kJ/kg$
$Q_R = C_v(T_4 - T_1) = 0.718(584.4 - 300) = 204.2 kJ/kg$
$\eta = 1 - \frac{\boldsymbol{Q_R}}{\boldsymbol{Q_s}}$ =1 $- \frac{204.2}{444.89}$ =0.541 ie $\eta = 54.1\%$
$W = \boldsymbol{Q_s} - \boldsymbol{Q_R} = 444.89 - 204.2 = 240.69\ kJ/kg$

$$p_m = \frac{W_{net}}{Swept\ Volume} = \frac{W_{net}}{V_1 - V_2} = \frac{W_{net}}{V_1\left(1 - \frac{V_2}{V_1}\right)} = \frac{W_{net}}{\frac{RT_1}{P_1}\left(1 - \frac{1}{R_c}\right)} = \frac{P_1 R_c\, W_{net}}{RT_1(R_c - 1)}$$

$p_m = \frac{1x7x240.69}{0.287x300(7-1)}$ =3.26 bar

4. An engine working on otto cycle rejects 535kJ/kg and has an air standard efficiency of 52% the pressure and temperature of the air at the compression are 1 bar and 50°C. Compute (i) The compression ratio of an engine (ii) The work done per kg of air (iii) The pressure and temperature at the end of compression and (iv) The minimum pressure in the cycle



Given:,

An engine working on otto cycle rejects 535kJ/kg ie Q_R=535kJ/kg

has an air standard efficiency of 52% efficiency=0.52,

The pressure and temperature of the air at the compression are 1 bar and 50°C.

P_1=100kPa,T_1=323K

ŋ air standard efficiency $= 1 - \frac{1}{R_c^{\gamma-1}}$; $0.52 = 1 - \frac{1}{R_c^{1,4-1}}$; R_c=6.27

$T_2 = T_1 R_c^{\gamma-1} = 323x6.27^{1.4-1} = 673.15K$

$Q_R = C_v(T_4 - T_1)$; $535 = 0.718(T_4 - 323)$; T_4=1068.12K

$T_4 = \frac{T_3}{R_c^{\gamma-1}}$; $1068.12 = \frac{T_3}{6.27^{1.4-1}}$; T_3=2226K

$$Q_s = C_v(T_3 - T_2) = 0.718(2226 - 673.15) = 1114.9kJ/kg$$

$Work\ output = Q_s - Q_R$

$Work\ output = 1114.9 - 535$ =579.95kJ/kg

$$p_m = \frac{W_{net}}{Swept\ Volume} = \frac{W_{net}}{V_1 - V_2} = \frac{W_{net}}{V_1\left(1 - \frac{V_2}{V_1}\right)} = \frac{W_{net}}{\frac{RT_1}{P_1}\left(1 - \frac{1}{R_c}\right)} = \frac{P_1 R_c W_{net}}{RT_1(R_c - 1)}$$

$p_m = \frac{1x6.27x579.95}{0.287x323(6.27-1)}$ =7.44bar

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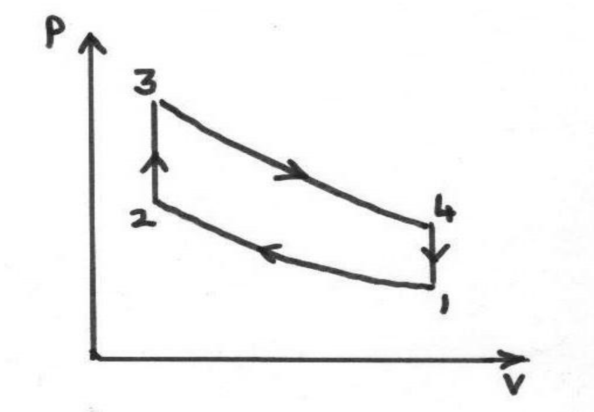
5. If an engine works on Otto cycle between temperature limits 1450K and 310K. Find the maximum power developed by the engine assuming the circulation of air per minute as 0.38kg/min.



If an engine works on Otto cycle between temperature limits 1450K and 310K

Ie T_3 =1450K and T_1=310K

For maximum work, $R_C = \left(\frac{T_3}{T_1}\right)^{\frac{1}{2(\gamma-1)}}$

$$R_C = \left(\frac{1450}{310}\right)^{\frac{1}{2(1.4-1)}} = 6.88$$

$$T_2 = T_1 R_c^{\gamma-1} = 310x6.88^{1.4-1} = 670.5K$$

$T_3 = \alpha T_2$;   1450= $\alpha x 670.5$; α =2.1625

$T_4 = \alpha T_1$; $T_4 = 2.1625x310$ =670.4

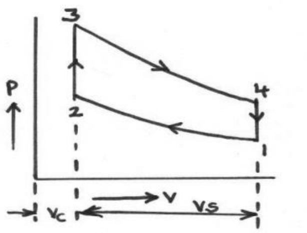
Alternatively

$$T_4 = \frac{T_3}{R_c^{\gamma-1}} = \frac{1450}{6.88^{1.4-1}} = 670.4K$$

$$Work\ output = mC_v(T_3 - T_2) - mC_v(T_4 - T_1)$$

$$Work\ output = \frac{1}{60}[0.718(1450 - 670.5) - 0.718(670.4 - 310)] = 5.02kW$$

6. A four stroke four cylinder petrol engine of 250mm bore and 375 mm stroke works on the Otto cycle. The clearance volume is 0.01052m^3. The initial pressure and temperature are 1 bar and 47^{0}C. If the maximum pressure is limited to 25 bar, find the following: air standard efficiency and mean effective pressure.(Dec2011).



A four stroke four cylinder petrol engine of 250mm bore and 375 mm stroke ie D=0.25m ,L =0.375m

The clearance volume is 0.01052m^3 V_c=0.01052m^3

$$R_C = \frac{V_C + V_S}{V_C}$$

$V_s = \frac{\pi d^2}{4} xl = \frac{\pi 0.25^2}{4} x0.375$ =0.0184m^3

$R_C = \frac{0.0184+0.01052}{0.01052}$=2.75

ŋ air standard efficiency $= 1 - \frac{1}{R_c^{\gamma-1}}$; ŋ air standard efficiency $= 1 - \frac{1}{2.75^{1.4-1}} = 0.333$

$$P_2 = P_1 R_c^{\gamma}; P_2 = 100x2.75^{1.4} = 412.16kPa$$

$$\alpha = \frac{P_3}{P_2} = \frac{2500}{412.16} = 6.07$$

$$MEP = \frac{R_C P_1(\alpha - 1)(R_c^{\gamma-1} - 1)}{(\gamma - 1)(R_C - 1)}$$

$$MEP = \frac{2.75x100(6.07 - 1)(2.75^{1.4-1} - 1)}{(1.4 - 1)(2.75 - 1)} = 993.4kPa$$

Alternative method

$$P_2 = P_1 R_c^{\gamma} \;;\; P_3 = \alpha P_2 \;; P_4 = \frac{P_3}{R_c^{\gamma}}$$

$$T_2 = T_1 R_c^{\gamma-1} \;;\; T_3 = \alpha T_2 = \alpha T_1 R_c^{\gamma-1} \;; T_4 = \alpha T_1$$

$$R_C = \frac{V_C + V_S}{V_C}$$

$V_s = \frac{\pi d^2}{4} xl = \frac{\pi 0.25^2}{4} x0.375$ =0.0184m^3

$R_C = \frac{0.0184+0.01052}{0.01052}$=2.75

$P_2 = P_1 R_c^{\gamma}$ =1x2.75$^{1.4}$=4.12 bar
$T_2 = T_1 R_c^{\gamma-1}$=320x2.75$^{1.4-1}$=479.60K
$P_3 = 25\ bar\ (given)$
$P_3 = \alpha P_2;\ 25 = \alpha x4.12;$
Explosion ratio, α =6.07
$T_3 = \alpha T_2;\ \ 1273 = 6.07x479.60\ ;\ T_3 = 2911.17K$
$P_3 = \alpha P_2;\ P_3 = 1.948x15.24;\ \ P_3 = 29.69bar$
$T_4 = \alpha T_1;\ T_4 = 6.07x320;\ T_4 = 1942.4K$

$P_4 = \frac{P_3}{R_c^{\gamma}} = \frac{25}{2.75^{1.4}}$=6.06 bar

$$Q_s = C_v(T_3 - T_2) = 0.718(2911{,}17 - 479.6) = 1745.86kJ/kg$$

$$Q_R = C_v(T_4 - T_1) = 0.718(1942 - 320) = 1164.59kJ/kg$$

$\eta = 1 - \frac{\boldsymbol{Q_R}}{\boldsymbol{Q_s}} = 1 - \frac{1164.59}{1745.86}$ =0.3329 ie $\eta = 33.29\%$
$W = \boldsymbol{Q_s} - \boldsymbol{Q_R} = 1745.86 - 1164.59 = 581.27\ kJ/kg$

$$p_m = \frac{W_{net}}{Swept\ Volume} = \frac{W_{net}}{V_1 - V_2} = \frac{W_{net}}{V_1\left(1 - \frac{V_2}{V_1}\right)} = \frac{W_{net}}{\frac{RT_1}{P_1}\left(1 - \frac{1}{R_c}\right)} = \frac{P_1 R_c\, W_{net}}{RT_1(R_c - 1)}$$

$p_m = \frac{1x2.75x581.27}{0.287x300(2.75-1)}$ =10.60 bar

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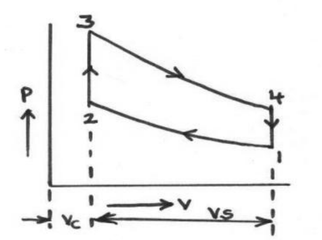
7. A four stroke four cylinder petrol engine of 250mm bore and 375 mm stroke works on the Otto cycle. The clearance volume is 0.00263m^3. The initial pressure and temperature are 1 bar and 50^{0}C. If the maximum pressure is limited to 25 bar, find the following: air standard efficiency and mean effective pressure.(July2011).



A four stroke four cylinder petrol engine of 250mm bore and 375 mm stroke ie D=0.250m L=0.375m

$$R_C = \frac{V_C + V_S}{V_C}$$

$V_s = \frac{\pi d^2}{4} xl = \frac{\pi 0.25^2}{4} x0.375$ =0.0184m^3

$R_C = \frac{0.0184+0.00263}{0.00263}$=8

$$\eta \text{ air standard efficiency} = 1 - \frac{1}{R_c^{\gamma-1}}$$

$$\eta \text{ air standard efficiency} = 1 - \frac{1}{8^{1.4-1}} = 0.565$$

$$P_2 = P_1 R_c^{\gamma}$$

$$P_2 = 100x8^{1.4} = 1837.9kPa$$

$$\alpha = \frac{P_3}{P_2} = \frac{2500}{1837.9} = 1.36$$

$$MEP = \frac{R_C P_1(\alpha - 1)(R_c^{\gamma-1} - 1)}{(\gamma - 1)(R_C - 1)}$$

$$MEP = \frac{8x100(1.36 - 1)(8^{1.4-1} - 1)}{(1.4 - 1)(8 - 1)} = 133.45kPa$$

Alternative method

$P_2 = P_1 R_c^{\gamma}$; $P_3 = \alpha P_2$; $P_4 = \frac{P_3}{R_c^{\gamma}}$

$T_2 = T_1 R_c^{\gamma-1}$; $T_3 = \alpha T_2 = \alpha T_1 R_c^{\gamma-1}$; $T_4 = \alpha T_1$

$$R_C = \frac{V_C + V_S}{V_C}$$

$V_s = \frac{\pi d^2}{4} xl = \frac{\pi 0.25^2}{4} x0.375 = 0.0184 m^3$

$R_C = \frac{0.0184+0.00263}{0.00263} = 8$

$P_2 = P_1 R_c^{\gamma} = 1x8^{1.4} = 18.38$ bar

$T_2 = T_1 R_c^{\gamma-1} = 323x8^{1.4-1} = 742.05K$

Explosion ratio, $\alpha = 1.12$

$P_3 = \alpha P_2$; $25 = \alpha x18.38$; Explosion ratio, $\alpha = 1.36$

$T_3 = \alpha T_2$; $T_3 = 1.36x742.05$; $T_3 = 1009.19K$

$T_4 = \alpha T_1$; $T_4 = 1.36x323$; $T_4 = 439.28K$

$P_4 = \frac{P_3}{R_c^{\gamma}} = \frac{25}{8^{1.4}} = 1.36\ bar$

$Q_s = C_v(T_3 - T_2) = 0.718(1009.19 - 742.05) = 191.80kJ/kg$

$Q_R = C_v(T_4 - T_1) = 0.718(439.28 - 323) = 83.49kJ/kg$

$\eta = 1 - \frac{\boldsymbol{Q_R}}{\boldsymbol{Q_s}} = 1 - \frac{83.49}{191.8} = 0.564$ ie $\eta = 56.4\%$

$W = \boldsymbol{Q_s} - \boldsymbol{Q_R} = 191.80 - 83.49 = 108.31kJ/kg$

$$p_m = \frac{W_{net}}{Swept\ Volume} = \frac{W_{net}}{V_1 - V_2} = \frac{W_{net}}{V_1\left(1 - \frac{V_2}{V_1}\right)} = \frac{W_{net}}{\frac{RT_1}{P_1}\left(1 - \frac{1}{R_c}\right)} = \frac{P_1 R_c W_{net}}{RT_1(R_c - 1)}$$

$p_m = \frac{1x8x108.31}{0.287x323(8-1)}$ =1.34 bar

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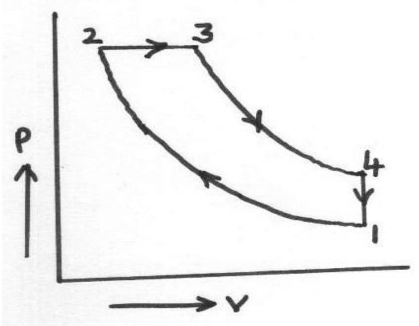
8. In an air standard diesel cycle, the compression ratio is 16. At the beginning of isentropic compression, the temperature is 15^o C and pressure is 0.1 Mpa. Heat is added until the temperature at the end of the constant pressure process is 1480^{o}C . Calculate i) cut off ratio ii) heat supplied per kg of air iii) cycle efficiency and iv) mean effective pressure (Dec08/09)



the compression ratio is 16. Ie R_c=16

At the beginning of isentropic compression, the temperature is 15^o C and pressure is 0.1 Mpa ie T_1=15^{o}C and p_1=1 bar=100kPa

Heat is added until the temperature at the end of the constant pressure process is 1480^{o}C ie T_3=1480^{o}C

$$P_2 = P_1 R_c^{\gamma}$$

$$P_2 = 100x16^{1.4} = 4850.29kPa$$

$$T_2 = T_1 R_c^{\gamma-1} = 288x16^{1.4-1} = 873.05K$$

$$\rho = \frac{V_3}{V_2} = \frac{T_3}{T_2} = \frac{1480}{873.05} = 1.7$$

$$Q_s = C_p(T_3 - T_2)$$

$$Q_s = 1.005(1450 - 873.05) = 579.83kJ/kg$$

$$\text{ŋ air standard efficiency} = 1 - \frac{1}{R_c^{\gamma-1}}\left[\frac{(\rho^{\gamma} - 1)}{\gamma(\rho - 1)}\right]$$

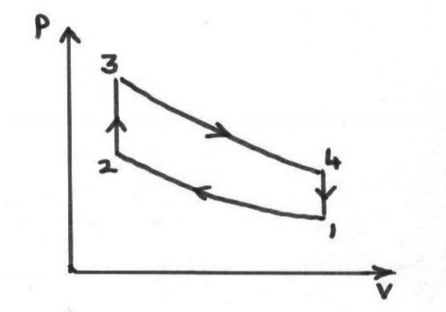
$$\text{ŋ air standard efficiency} = 1 - \frac{1}{16^{1.4-1}}\left[\frac{(1.7^{1.4} - 1)}{1.4(1.7 - 1)}\right] = 0.629$$

$$Mean\ efficitve\ pressure = \frac{P_1 R_C\left[\gamma R_c^{\gamma-1}(\rho - 1) - (\rho^{\gamma} - 1)\right]}{(\gamma - 1)(R - 1)}$$

$$Mean\ effecitve\ pressure = \frac{100x16[1.4x16^{1.4-1}(1.7-1)-(1.7^{1.4}-1)]}{(1.4-1)(16-1)}$$

Mean effective Pressure=498.35kPa

9. Two engines are to operate on otto and diesel cycles with the following data. Maximum temperature: 1400K, Exhaust temperature: 700K. State of air at the beginning of compression 0.1Mpa, 300K. Estimate the compression ratio, the maximum pressures, efficiencies and rate of work output (for I kg/s of air) of the respective cycles (June/July08)



Otto cycle

Maximum temperature: 1400K, Exhaust temperature: 700K. Ie T_3=1400K T_4=700K

$$R_e = R_C = \frac{V_4}{V_3} = \left(\frac{T_3}{T_4}\right)^{\frac{1}{\gamma-1}} = \left(\frac{1400}{700}\right)^{\frac{1}{1.4-1}} = 5.66$$

Compression ratio =5.66

$$\eta \text{ air standard efficiency} = 1 - \frac{1}{R_c^{\gamma-1}}$$

$$\eta \text{ air standard efficiency} = 1 - \frac{1}{5.66^{1.4-1}} = 0.499 = 49.9\%$$

$$P_2 = P_1 R_c^{\gamma}$$

$$P_2 = 100x5.66^{1.4} = 1132.25kPa$$

$$T_2 = T_1 R_c^{\gamma-1} = 300x5.66^{1.4-1} = 600.13K$$

$$\frac{P_3}{P_2} = \frac{T_3}{T_2}$$

$Max\ pressure\ P_3 = 1132.25\frac{1400}{600.13}$=2641.33kPa

$$Work\ output = C_v(T_3 - T_2) - C_v(T_4 - T_1)$$

$$Work\ output = [0.718(1400 - 600.13) - 0.718(700 - 300)] = 287.11kJ/kg$$

$P_2 = P_1 R_c^\gamma$; $P_3 = \alpha P_2$; $P_4 = \frac{P_3}{R_c^\gamma}$

$T_2 = T_1 R_c^{\gamma-1}$; $T_3 = \alpha T_2 = \alpha T_1 R_c^{\gamma-1}$; $T_4 = \alpha T_1$

$T_3 = 1400K;\ T_4 = 700K$ (Given)

$R_e = R_C = \frac{V_4}{V_3} = \left(\frac{T_3}{T_4}\right)^{\frac{1}{\gamma-1}} = \left(\frac{1400}{700}\right)^{\frac{1}{1.4-1}} = 5.66$

$P_2 = P_1 R_c^\gamma;\quad P_2 = 1x5.66^{1.4} = 11.32bar$

$T_2 = T_1 R_c^{\gamma-1};\ T_2 = 300x5.66^{1.4-1} = 600.13K$

$T_4 = \alpha T_1;$ 700= $\alpha x300$; explosion ratio, $\alpha = 2.33$

$T_3 = \alpha T_2;\ T_3$ = 2.33x600.13 = 1398.3K

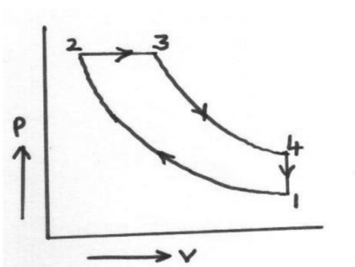
$Q_s = C_v(T_3 - T_2) = 0.718(1400 - 600.13) = 574.31kJ/kg$

$Q_R = C_v(T_4 - T_1) = 0.718(700 - 300) = 287.2kJ/kg$

$\eta = 1 - \frac{\boldsymbol{Q_R}}{\boldsymbol{Q_s}}$ =$1 - \frac{287.2}{574.31}$ =0.50 ie $\eta = 50\%$

$W = \boldsymbol{Q_s} - \boldsymbol{Q_R} = 574.31 - 287.2 = 287.11kJ/kg$

Diesel Cycle



Maximum temperature: 1400K, Exhaust temperature: 700K. Ie T_3=1400K T_4=700K

$$T_4 = T_1 \rho^\gamma$$

700=300 ρ^γ

Cut off ratio, $\rho = \left(\frac{T_4}{T_1}\right)^{\frac{1}{\gamma}} = \left(\frac{700}{300}\right)^{\frac{1}{1.4}} = 1.83$

$$\frac{T_3}{T_4} = \left(\frac{V_4}{V_3}\right)^{\gamma-1}$$

$R_e = \frac{V_4}{V_3} = \left(\frac{T_3}{T_4}\right)^{\frac{1}{\gamma-1}} = \left(\frac{1400}{700}\right)^{\frac{1}{1.4-1}} = 5.66$

$R_e = \frac{R_C}{\rho}$

$R_C = 5.66x1.83 = 10.36$

$P_2 = P_1 R_c^{\gamma}$

$P_2 = 100x10.36^{1.4} = 2639.39kPa$

$T_2 = T_1 R_c^{\gamma-1} = 300x10.36^{1.4-1} = 764.3K$

$\rho = \frac{V_3}{V_2}$

Max pressure $P_3 = P_2 = 2639.39kPa$

ŋ air standard efficiency $= 1 - \frac{1}{R_c^{\gamma-1}}\left[\frac{(\rho^{\gamma}-1)}{\gamma(\rho-1)}\right]$

ŋ air standard efficiency $= 1 - \frac{1}{10.36^{1.4-1}}\left[\frac{(1.83^{1.4}-1)}{1.4(1.83-1)}\right] = 0.55$

$Work\ output = C_p(T_3 - T_2) - C_v(T_4 - T_1)$

$Work\ output = [1.005(1400 - 764.3) - 0.718(700 - 300)] = 351.68kJ/kg$

Note you can apply direct formula to find out temperatures and pressures

$P_2 = P_1 R_c^{\gamma}$; $P_3 = P_2$; $P_4 = \frac{P_3}{R_e^{\gamma}}$

$T_2 = T_1 R_c^{\gamma-1}$; $T_3 = \rho T_1 R_c^{\gamma-1}$; $T_4 = \rho^{\gamma} T_1$

$T_3 = 1400K$; $T_4 = 700K$ (Given)

Expansion ratio, $R_e = \frac{V_4}{V_3} = \left(\frac{T_3}{T_4}\right)^{\frac{1}{\gamma-1}} = \left(\frac{1400}{700}\right)^{\frac{1}{1.4-1}} = 5.66$

$T_4 = \rho^{\gamma} T_1$ in an Diesel cycle

$T_4 = 700K;\ T_1 = 300K\ (given)$

$700 = \rho^{1.4} x300$

$$cut\ off\ ratio\ \rho = \left(\frac{700}{300}\right)^{\frac{1}{1.4}} = 1.83$$

$R_C = R_e x\rho$

$Compression\ ratio\ R_C = 5.66x1.83 = 10.36$

$P_2 = P_1 R_c^{\gamma} = 1x10.36^{1.4}$

$P_2 = 1x10.36^{1.4} = 26.39bar$

$T_2 = T_1 R_c^{\gamma-1};\ T_2 = 300x10.36^{1.4-1} = 764.3K$

$T_3 = \rho T_2;\quad T_3 = \rho T_2$

$T_4 = 700K\ (given)$

$P_3 = P_2 = 26.39\ bar$

$P_4 = \frac{P_3}{R_e^{\gamma}} = \frac{26.39}{5.66^{1.4}} = 2.33bar$

$Q_s = C_p(T_3 - T_2) = 1.005(1400 - 764.3) = 638.87kJ/kg$

$Q_R = C_v(T_4 - T_1) = 0.718(700 - 300) = 287.2kJ/kg$

$\eta = 1 - \frac{\boldsymbol{Q_R}}{\boldsymbol{Q_s}} = 1 - \frac{287.2}{638.87} = 0.5504$ ie $\eta = 55.04\%$

$W = \boldsymbol{Q_s} - \boldsymbol{Q_R} = 638.87 - 287.2 = 351.67kJ/kg$

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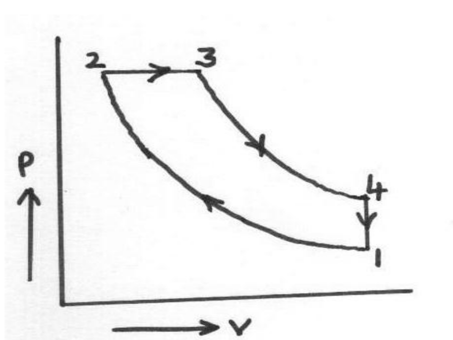
10. The minimum and maximum temperatures in an engine working on constant pressure cycle are 300K and 1500K and the heat addition during combustion is 500kJ/kg of air. Another engine working on semi diesel cycle between the same temperature limits of 300K and 1500K has a heat addition of 500kJ/kg of air which is shared equally between the two heat addition processes. Compare their – i) Efficiencies and ii) Work outputs (Dec07/Jan08)



The minimum and maximum temperatures in an engine working on constant pressure cycle are 300K and 1500K ie T_1=300K, T_3=1500K

the heat addition during combustion is 500kJ/kg of air $Q_s = 500kJ/kg$

$$Q_s = C_p(T_3 - T_2)$$

$$500 = 1.005(1500 - T_2)$$

T_2=1002.5K

$$\rho = \frac{V_3}{V_2} = \frac{T_3}{T_2} = \frac{1500}{1002.5} = 1.5$$

$$R_C = \frac{V_1}{V_2} = \left(\frac{T_2}{T_1}\right)^{\frac{1}{\gamma-1}} = \left(\frac{1002.5}{300}\right)^{\frac{1}{1.4-1}} = 20.41$$

$$\text{ŋ air standard efficiency} = 1 - \frac{1}{R_c^{\gamma-1}}\left[\frac{(\rho^\gamma - 1)}{\gamma(\rho - 1)}\right]$$

$$\text{ŋ air standard efficiency} = 1 - \frac{1}{20.41^{1.4-1}}\left[\frac{(1.5^{1.4} - 1)}{1.4(1.5 - 1)}\right] = 0.673$$

$$R_e = \frac{R_C}{\rho} = \frac{20.41}{1.5} = 13.6$$

$$T_4 = \frac{T_3}{{R_e}^{\gamma-1}} = \frac{1500}{13.6^{1.4-1}} = 528.1K$$

$$Work\ output = C_p(T_3 - T_2) - C_v(T_4 - T_1)$$

$$Work\ output = [1.005(1500 - 1002.5) - 0.718(528.1 - 300)] = 336.2kJ/kg$$

Note you can apply direct formula to find out temperatures and pressures

$P_2 = P_1 R_c^\gamma$; $P_3 = P_2$; $P_4 = \frac{P_3}{R_e^\gamma}$

$T_2 = T_1 R_c^{\gamma-1}$; $T_3 = \rho T_2 = \rho T_1 R_c^{\gamma-1}$; $T_4 = \rho^\gamma T_1$

he heat addition during combustion is 500kJ/kg of air $Q_s = 500kJ/kg$

$Q_s = C_p(T_3 - T_2)$

$500 = 1.005(1500 - T_2)$; T_2=1002.5K

$T_2 = T_1 R_c^{\gamma-1}$; 1002.5=300$x R_c^{\gamma-1}$;

$$R_C = \left(\frac{1002.5}{300}\right)^{\frac{1}{1.4-1}} = 20.41$$

$T_3 = \rho T_2;\ 1500 = \rho x 1002.5;\ cut\ off\ ratio\ \rho = 1.496$

$T_4 = \rho^{\gamma} T_1;\ T_4 = 1.496^{1.4} x 300;\ T_4 = 527.26\text{K}$

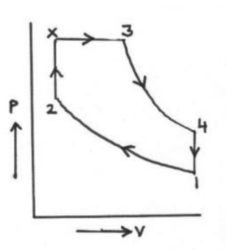
$P_2 = P_1 R_c^{\gamma};\ \ P_2 = 1x20.41^{1.4} = 68.2bar$

$P_3 = P_2 = 68.2\ bar$

$R_e = \frac{R_C}{\rho} = \frac{20.41}{1.5} = 13.6$

$P_4 = \frac{P_3}{R_e^{\gamma}};\ P_4 = \frac{68.2}{13.6^{1.4}}$=1.765bar

$Q_s = C_p(T_3 - T_2) = 1.005(1500 - 1002.5) = 499.98kJ/kg$
$Q_R = C_v(T_4 - T_1) = 0.718(527.6 - 300) = 163.42kJ/kg$
$\eta = 1 - \frac{\boldsymbol{Q_R}}{\boldsymbol{Q_s}}$ =$1 - \frac{163.42}{499.98}$ =0.6731 ie $\eta = 67.31\%$
$W = \boldsymbol{Q_s} - \boldsymbol{Q_R} = 499.98 - 163.42 = 336.56kJ/kg$



The minimum and maximum temperatures in an engine working on constant pressure cycle are 300K and 1500K ie T_1=300K, T_4=1500K

the heat addition during combustion is 500kJ/kg of air $Q_s = 500kJ/kg$

$Q_s = Q_{sV} + Q_{sP}$

a heat addition of 500kJ/kg of air which is shared equally between the two heat addition processes

ie $Q_s = Q_{sV} + Q_{sP}$=500

$and\ Q_{sV} = Q_{sP}$

Hence $Q_{sV} = Q_{sP} = \frac{500}{2} = 250kJ/kg$

$Q_{sp} = C_p(T_4 - T_3)$

$250 = 1.005(1500 - T_3)$

T_3=1251.24K

$Q_{sV} = C_v(T_3 - T_2)$

$250 = 0.718(1251.24 - T_2)$

T_2=903.1K

$$\rho = \frac{V_4}{V_3} = \frac{T_4}{T_3} = \frac{1500}{1251.24} = 1.2$$

$$\alpha = \frac{P_3}{P_2} = \frac{T_3}{T_2} = \frac{1251.24}{903.1} = 1.4$$

$$R_C = \frac{V_1}{V_2} = \left(\frac{T_2}{T_1}\right)^{\frac{1}{\gamma-1}} = \left(\frac{903.1}{300}\right)^{\frac{1}{1.4-1}} = 15.72$$

$$\text{ŋ air standard efficiency} = 1 - \frac{1}{R_c^{\gamma-1}}\left[\frac{(\alpha\rho^\gamma - 1)}{(\alpha - 1) + \gamma\alpha(\rho - 1)}\right]$$

$$\text{ŋ air standard efficiency} = 1 - \frac{1}{15.72^{1.4-1}}\left[\frac{(1.4x1.2^{1.4} - 1)}{(1.4 - 1) + 1.4x1.4(1.2 - 1)}\right] = 0.661$$

$$R_e = \frac{R_C}{\rho} = \frac{15.72}{1.2} = 13.1$$

$$T_4 = \frac{T_3}{{R_e}^{\gamma-1}} = \frac{1500}{13.1^{1.4-1}} = 537.67K$$

$$Work\ output = \left[C_p(T_4 - T_3) + C_v(T_3 - T_2)\right] - C_v(T_4 - T_1)$$

$$Work\ output = [500 - 0.718(537.67 - 300)] = 329.35kJ/kg$$

$P_2 = P_1 R_c^\gamma$; $P_3 = \alpha P_2$; $P_4 = P_3$; $P_5 = \frac{P_4}{R_e^\gamma}$

$T_2 = T_1 R_c^{\gamma-1}$; $T_3 = \alpha T_2 = \alpha T_1 R_c^{\gamma-1}$; $T_4 = \rho T_3 = \alpha\rho T_1 R_c^{\gamma-1}$; $T_5 = \alpha\rho^\gamma T_1$

the heat addition during combustion is 500kJ/kg of air $Q_s = 500kJ/kg$

$Q_s = Q_{sV} + Q_{sP}$

a heat addition of 500kJ/kg of air which is shared equally between the two heat addition processes

ie $Q_s = Q_{sV} + Q_{sP}$=500

$and\ Q_{sV} = Q_{sP}$

Hence $Q_{sV} = Q_{sP} = \frac{500}{2} = 250kJ/kg$

$Q_{sp} = C_p(T_4 - T_3)$

$250 = 1.005(1500 - T_3)$

T_3=1251.24K

$Q_{sV} = C_v(T_3 - T_2)$

$250 = 0.718(1251.24 - T_2)$

T_2=903.1K

$T_2 = T_1 R_c^{\gamma-1}$; 903.1 =$300xR_c^{1.4-1}$

$$R_C = \left(\frac{903.1}{300}\right)^{\frac{1}{1.4-1}} = 15.72$$

$T_3 = \alpha T_2$; 1251.24= $\alpha x903.14$; $\alpha = 1.386$

$T_4 = \rho T_3$;1500=$\rho x1251.24$; cut off ratio ρ =1.19

534.24

$T_5 = \alpha\rho^{\gamma}T_1; T_5 = 1.386x1.19^{1.4}x300; T_5 = 530.45K$

$Q_R = C_v(T_5 - T_1); Q_R = 0.718x(530.45 - 300); Q_R = 165.46kJ/kg$

$\eta = 1 - \frac{\boldsymbol{Q_R}}{\boldsymbol{Q_s}}$ =$1 - \frac{165.46}{500}$ =0.669 ie $\eta = 66.9\%$

$W = \boldsymbol{Q_s} - \boldsymbol{Q_R} = 500 - 165.46 = 334.54kJ/kg$

11. The stroke and cylinder diameter of a compression ignition engine working on theoretical diesel cycle are 250mm and 150mm respectively. The clearance volume is 0.0004m^3. The fuel injection at constant pressure takes place for 5% of the stroke. Calculate the efficiency of the engine.

The stroke and cylinder diameter of a compression ignition engine working on theoretical diesel cycle are 250mm and 150mm respectively. Ie D=0.25m, L=0.15m

The clearance volume is 0.0004m^3 V_c=0.004m^3

The fuel injection at constant pressure takes place for 5% of the stroke. K=5

$$R_C = \frac{V_C + V_S}{V_C}$$

$V_s = \frac{\pi d^2}{4} x l = \frac{\pi 0.15^2}{4} x 0.25 = 0.0044 m^3$

$R_C = \frac{0.0044+0.0004}{0.0004} = 12$

$$\rho = \frac{V_3}{V_2} = \frac{V_c + kV_s}{V_c} = 1 + \frac{kV_s}{V_c} = 1 + \frac{k(V_1 - V_c)}{V_c} = 1 + k(R_C - 1)$$

$$\rho = 1 + 0.05(12 - 1) = 1.55$$

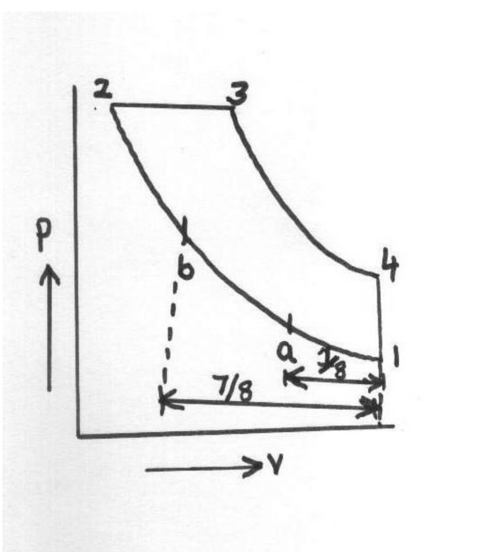
$$\text{ŋ air standard efficiency} = 1 - \frac{1}{R_C{}^{\gamma-1}}\left[\frac{(\rho^\gamma - 1)}{\gamma(\rho - 1)}\right]$$

$$\text{ŋ air standard efficiency} = 1 - \frac{1}{12^{1.4-1}}\left[\frac{(1.55^{1.4} - 1)}{1.4(1.55 - 1)}\right] = 0.593$$

12. An indicator diagram taken on diesel engine shows that the compression curve follows the law $PV^{1.4}$ = Constant. At two points lying on the compression curve 1/8th and 7/8th of the stroke the pressures are 1.6 bar and 16 bar respectively. Find the compression ration of the engine. If cut off occurs at 6% of the stroke, calculate air standard efficiency of the engine (June/July2009)

An indicator diagram taken on diesel engine shows that the compression curve follows the law $PV^{1.4}$ = Constant. Ie compression is adiabatic since n=1.4= γ

At two points lying on the compression curve 1/8th and 7/8th of the stroke the pressures are 1.6 bar and 16 bar respectively.



From fig, $V_a = V_c + (V_s - \frac{1}{8}V_s)$

$$V_a = V_c + \frac{7}{8}V_s$$

$$V_b = V_c + (V_s - \frac{7}{8}V_s)$$

$$V_b = V_c + \frac{1}{8}V_s$$

$$P_a {V_a}^{\gamma} = P_b {V_b}^{\gamma}$$

$$\frac{V_a}{V_b} = \left(\frac{P_b}{P_a}\right)^{\frac{1}{\gamma}} = \left(\frac{16}{1.6}\right)^{\frac{1}{1.4}} = 5.18$$

$$\frac{V_c + \frac{7}{8}V_s}{V_c + \frac{1}{8}V_s} = 5.18$$

$$V_c + \frac{7}{8}V_s = 5.18\left(V_c + \frac{1}{8}V_s\right)$$

$V_s\left(\frac{7}{8} - \ 5.18\ \left(\frac{1}{8}\right)\right) = V_c(5.18 - 1)$

$0.228V_s = 4.18V_c$

$$\frac{V_s}{V_c} = 18.33$$

$$R_C = 1 + \frac{V_S}{V_C} = 1 + 18.33 = 19.33$$

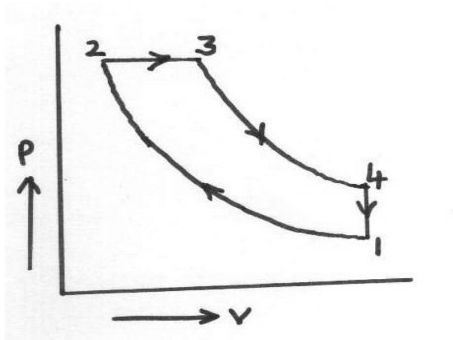
$$\rho = \frac{V_3}{V_2} = \frac{V_c + kV_s}{V_c} = 1 + \frac{kV_s}{V_c} = 1 + \frac{k(V_1 - V_c)}{V_c} = 1 + k(R_C - 1)$$

$$\rho = 1 + 0.06(19.33 - 1) = 2.1$$

$$\text{ŋ air standard efficiency} = 1 - \frac{1}{{R_C}^{\gamma-1}}\left[\frac{(\rho^{\gamma} - 1)}{\gamma(\rho - 1)}\right]$$

$$\text{ŋ air standard efficiency} = 1 - \frac{1}{19.33^{1.4-1}}\left[\frac{(2.1^{1.4} - 1)}{1.4(2.1 - 1)}\right] = 0.637$$

13. An air standard diesel cycle has a compression ratio of 16. The compression ratio of 16. The temperature before compression is 27^{o}C and the temperature after expansion is 627^{o}C. Determine (i) The network output per unit mass of air (ii) Thermal efficiency (iii) Specific air consumption in kg/kWh (July/Aug 2005)



An air standard diesel cycle has a compression ratio of 16. Ie $R_C = 16$
The temperature before compression is 27°C and the temperature after expansion is 627°C. T_1=27°C T_4=627°C

$$T_2 = T_1 {R_C}^{\gamma-1} = 300x16^{1.4-1} = 909.4K$$

$$T_4 = \rho^{\gamma} T_1$$

$$\rho = \left(\frac{T_4}{T_1}\right)^{\frac{1}{\gamma}} = \left(\frac{900}{300}\right)^{\frac{1}{1.4}} = 2.19$$

T_3=909.4x2.19=1993.22K

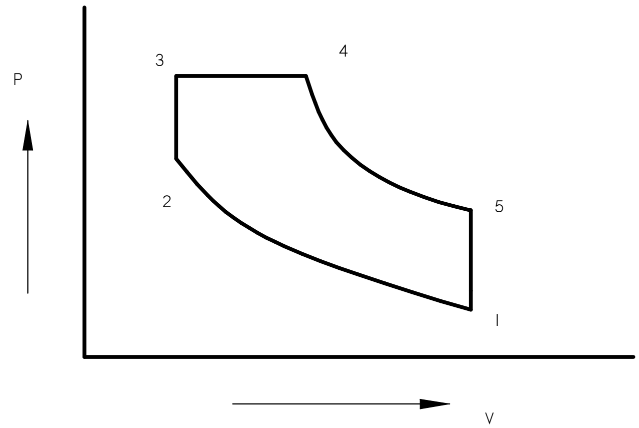
$$Work\ output = C_p(T_3 - T_2) - C_v(T_4 - T_1)$$

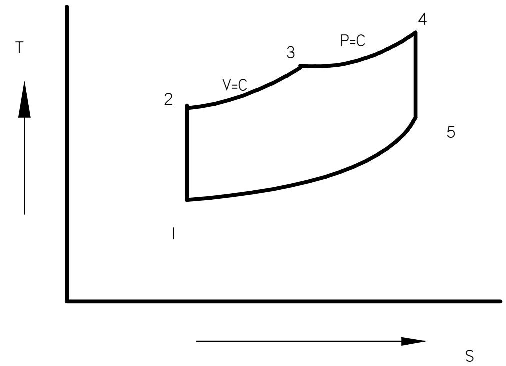
$$Work\ output = [1.005(1993.22 - 909.4) - 0.718(900 - 300)] = 658.44kJ/kg$$

$$\eta \text{ thermal} = \frac{Wnet}{Q} = \frac{658.44}{1.005(1993.22 - 909.4)} = 0.605$$

SFC=(3600/W_{net})=(3600/658.44)=5.47kg/kWh

14. An air standard limited pressure cycle has a compression has a compression ratio of 15 and compression begins at 0.1MPa,40°C. The maximum pressure is limited to 6MPa and the heat added is 1.675MJ/kg. Compute (i) the heat supplied at constant volume per kg of air ii) the heat supplied at constant pressure per kg of air (iii) the work done per kg of air (iv) the cycle efficiency (v) the cut-off ratio (vi) m.e.p of the cycle (July07)





An air standard limited pressure cycle has a compression has a compression ratio of 15 and compression begins at 0.1MPa,40°C *ie* $R_C = 15,\ P_1 = 100kPa\ and\ T_1 = 40^o$ C,

$$T_2 = T_1 R_C{}^{\gamma-1} = 313x15^{1.4-1} = 924.65K$$

$$P_2 = P_1 R_C{}^{\gamma} = 100x15^{1.4} = 4431.26KPa$$

$$\alpha = \frac{P_3}{P_2} = \frac{6000}{4431.26} = 1.35$$

Also, $\alpha = \frac{T_3}{T_2}$ $\quad 1.35 = \frac{T_3}{T_2} = \frac{T_3}{924.65}$

T_3=1248.28K

$$Q = \left[C_p(T_4 - T_3) + C_v(T_3 - T_2)\right]$$

$$1675 = [1.005(T_4 - 1248.28) + 0.718(1248.28 - 924.65)]$$

T_4=2683.74K

$$Q_{sV} = C_v(T_3 - T_2) = 0.718(1248.28 - 924.65) = 232.37kJ/kg$$

$$Q_{sp} = C_p(T_4 - T_3) = 1.005(2683.74 - 1248.28) = 1442.63kJ/kg$$

$$\rho = \frac{T_4}{T_3} = \frac{2683.74}{1248.28} = 2.15$$

$$R_e = \frac{R_C}{\rho} = \frac{15}{2.15} = 6.98$$

$$T_5 = \frac{T_4}{{R_e}^{\gamma-1}} = \frac{2683.74}{6.98^{1.4-1}} = 1233.67K$$

$$Work\ output = \left[C_p(T_4 - T_3) + C_v(T_3 - T_2)\right] - C_v(T_5 - T_1)$$

$$Work\ output = [1675 - 0.718(1233.67 - 313)] = 1013.95kJ/kg$$

$$\text{ŋ air standard efficiency} = 1 - \frac{1}{{R_C}^{\gamma-1}}\left[\frac{(\alpha\rho^{\gamma} - 1)}{(\alpha - 1) + \gamma\alpha(\rho - 1)}\right]$$

$$\text{ŋ air standard efficiency} = 1 - \frac{1}{15^{1.4-1}}\left[\frac{(1.35x2.15^{1.4} - 1)}{(1.35 - 1) + 1.4x1.35(2.15 - 1)}\right] = 0.605$$

$$Mean\ efficitve\ pressure = \frac{P_1 R_C\left[{R_C}^{\gamma-1}\big((\alpha - 1) + \alpha\gamma(\rho - 1)\big) - (\alpha\rho^{\gamma} - 1)\right]}{(\gamma - 1)(R_C - 1)}$$

$$Mean\ efficitve\ pressure = \frac{100x15\left[15^{1.4-1}\big((1.35 - 1) + (1.35x1.4)(2.15 - 1)\big) - (1.35x2.15^{1.4} - 1))\right]}{(1.4 - 1)(15 - 1)}$$

MEP=1200kPa

$P_2 = P_1 R_c^{\gamma}$; $P_3 = \alpha P_2$; $P_4 = P_3$; $P_5 = \frac{P_4}{R_e^{\gamma}}$

$T_2 = T_1 R_c^{\gamma-1}$; $T_3 = \alpha T_2 = \alpha T_1 R_c^{\gamma-1}$; $T_4 = \rho T_3 = \alpha\rho T_1 R_c^{\gamma-1}$; $T_5 = \alpha\rho^{\gamma} T_1$

$T_2 = T_1 {R_C}^{\gamma-1} = 313x15^{1.4-1} = 924.65K$

$P_2 = P_1 {R_C}^{\gamma} = 1x15^{1.4} = 44.31bar$

$\alpha = \frac{T_3}{T_2} = \frac{P_3}{P_2} = \frac{60}{44.31} = 1.35$

$T_3 = \alpha T_2$; $T_3 = 1.35x\ 924.65$; $T_3 = 1248.27K$

$Q_{sV} = C_V(T_3 - T_2)$; $Q_{sV} = 0.718(1248.27 - 924.65)$; $Q_{sV} = 232.35kJ/kg$

$Q = 1675$kJ/kg (given)

$$Q = Q_{sV} + Q_{sP}\ in\ Dual\ Engines$$

1675=232.35+Q_{sP}; $Q_{sP} = 1442.65kJ$/kg

$Q_{sP} = C_P(T_4 - T_3)$; 1442.65= $1.005(T_4 - 1248.27)$; $T_4 = 2683.74K$

T_4=ρT_3;   ρ $= \frac{T_4}{T_3}$; ρ= $\frac{2683.74}{1248.27} = 2.15$

$T_5 = \alpha\rho^{\gamma}T_1$; $T_5 = 1.35x2.15^{1.4}x313$; $T_5 = 1233.90K$

$Q_R = C_v(T_5 - T_1)$; $Q_R = 0.718(1233.9 - 313)$; $Q_R = 661.21k/kg$

$\eta = 1 - \frac{\boldsymbol{Q_R}}{\boldsymbol{Q_s}}$ =$1 - \frac{661.21}{1675}$ =0.6052ie $\eta = 60.52\%$

$W = \boldsymbol{Q_s} - \boldsymbol{Q_R} = 1675 - 661.21 = 1013.79\ kJ/kg$

$$p_m = \frac{W_{net}}{Swept\ Volume} = \frac{W_{net}}{V_1 - V_2} = \frac{W_{net}}{V_1\left(1 - \frac{V_2}{V_1}\right)} = \frac{W_{net}}{\frac{RT_1}{P_1}\left(1 - \frac{1}{R_c}\right)} = \frac{P_1 R_c\, W_{net}}{RT_1(R_c - 1)}$$

$p_m = \frac{1x15x1013.79}{0.287x313(15-1)}$ =12.09 bar

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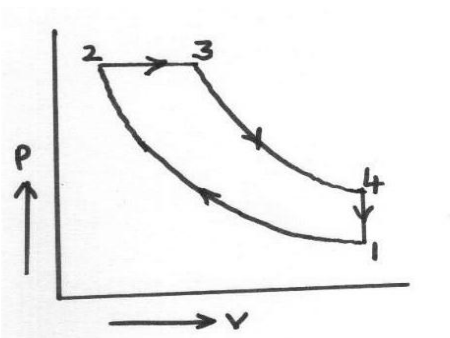
15. An ideal diesel engine has a compression ratio of 20 and uses air as the working fluid. The state of air at the beginning of the compression process is 95kPa and 20°C. If the maximum temperature in the cycle is not to exceed 2200K, determine (i) the thermal efficiency (ii) the mean effective pressure. Assume constant specific heats for air at room temperature. (Dec /2010)



An ideal diesel engine has a compression ratio of 20 *ie* $R_c = 20$

The state of air at the beginning of the compression process is 95kPa and 20°C ie $P_1 = 95kPa\ and\ T_1 = 20^{o}$ C

$$T_2 = T_1 {R_C}^{\gamma-1} = 293x20^{1.4-1} = 971.14K$$

$$P_2 = P_1 {R_C}^{\gamma} = 95x20^{1.4} = 6297.46KPa$$

$$\rho = \frac{V_3}{V_2} = \frac{T_3}{T_2} = \frac{2200}{971.14} = 2.27$$

$$T_4 = \rho^{\gamma} T_1$$

T_4=293x2.27$^{1.4}$=923.2K

$$Work\ output = C_p(T_3 - T_2) - C_v(T_4 - T_1)$$

$$Work\ output = [1.005(2200 - 971.14) - 0.718(923.2 - 293)] = 782.52kJ/kg$$

$$\eta\ \text{thermal} = \frac{Wnet}{Q} = \frac{782.52}{1.005(2200 - 971.14)} = 0.634$$

$$Mean\ efficitve\ pressure = \frac{P_1 R_C{}^{\gamma}\left[\gamma(\rho - 1) - R_C{}^{1-\gamma}(\rho^{\gamma} - 1)\right]}{(\gamma - 1)(R - 1)}$$

$$Mean\ effecitve\ pressure = \frac{95x20^{1.4}[1.4(2.27 - 1) - 20^{1-1.4}(2.27^{1.4} - 1)]}{(1.4 - 1)(20 - 1)}$$

Mean effective Pressure=935.55kPa

Note you can apply direct formula to find out temperatures and pressures

$P_2 = P_1 R_c^{\gamma}$; $P_3 = P_2$; $P_4 = \frac{P_3}{R_e^{\gamma}}$

$T_2 = T_1 R_c^{\gamma-1}$; $T_3 = \rho T_2 = \rho T_1 R_c^{\gamma-1}$; $T_4 = \rho^{\gamma} T_1$

$T_2 = T_1 R_C{}^{\gamma-1} = 293x20^{1.4-1} = 971.14K$

$P_2 = P_1 R_C{}^{\gamma} = 95x20^{1.4} = 6297.46KPa$

$\rho = \frac{V_3}{V_2} = \frac{T_3}{T_2} = \frac{2200}{971.14} = 2.27$

$T_4 = \rho^{\gamma} T_1$

T_4=293x2.27$^{1.4}$=923.2K

$Q_S = C_p(T_3 - T_2)$; $Q_S = 1.005(2200 - 971.14)$=1235kJ/kg

$Q_R = C_v(T_4 - T_1)$; $Q_R = 0.718(923.2 - 293) = 452.48kJ/kg$

$\eta = 1 - \frac{\boldsymbol{Q_R}}{\boldsymbol{Q_s}}$ =$1 - \frac{452.48}{1235}$ =0.6336ie $\eta = 63.36\%$

$W = \boldsymbol{Q_s} - \boldsymbol{Q_R} = 1235 - 452.48 = 782.52\ kJ/kg$

$$p_m = \frac{W_{net}}{Swept\ Volume} = \frac{W_{net}}{V_1 - V_2} = \frac{W_{net}}{V_1\left(1 - \frac{V_2}{V_1}\right)} = \frac{W_{net}}{\frac{RT_1}{P_1}\left(1 - \frac{1}{R_C}\right)} = \frac{P_1 R_c\, W_{net}}{RT_1(R_c - 1)}$$

$p_m = \frac{1x20x782.52}{0.287x293(20-1)}$ =9.79 bar

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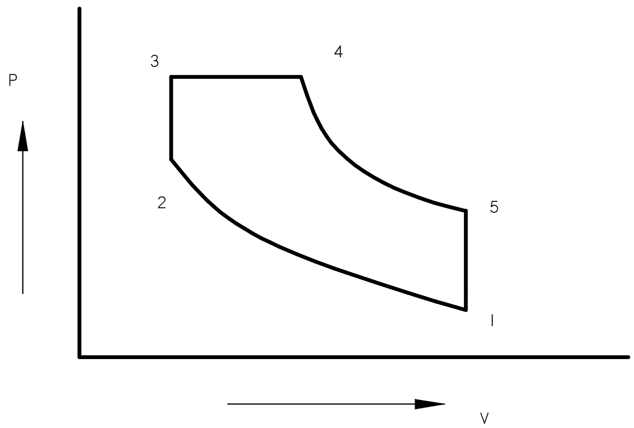
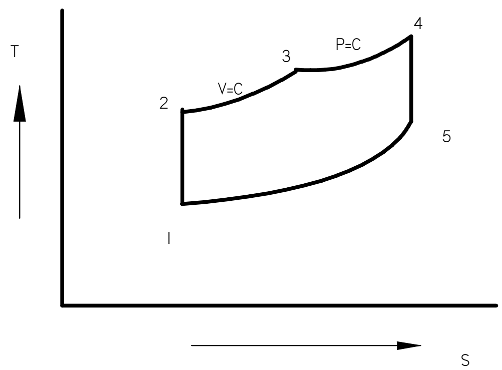
16. An engine working on dual combustion cycle takes in air at 1 bar and 30°C. The clearance is 8% of the stroke and cut-off takes place at 10% of the stroke. The maximum pressure in the cycle is limited to 70bar. Find (i) temperatures and pressure at salient points; (ii) air standard efficiency (March 2001)





An engine working on dual combustion cycle takes in air at 1 bar and 30°C. ie $P_1 = 95kPa \ and \ T_1 = 20^{o}$ C

The clearance is 8% of the stroke.V_c=0.08V_s

cut-off takes place at 10% of the stroke.K= $V_4 - V_3$=0.01V_s

The maximum pressure in the cycle is limited to 70bar.P_2 =P_3 = 70bar

$$\rho = \frac{V_3}{V_2} = \frac{V_c + kV_s}{V_c} = 1 + \frac{kV_s}{V_c} = 1 + \frac{0.1V_s}{0.08V_s} = 2.25$$

$$R_C = \frac{V_S + V_C}{V_C} = 1 + \frac{V_S}{V_C} = 1 + \frac{V_s}{0.08V_s} = 13.5$$

$$T_2 = T_1 {R_C}^{\gamma - 1} = 303x13.5^{1.4-1} = 858.18K$$

$$P_2 = P_1 {R_C}^{\gamma} = 100x13.5^{1.4} = 3823.6KPa$$

$$P_4 = P_3 = 70bar$$

$$\alpha = \frac{P_3}{P_2} = \frac{70}{38.236} = 1.83$$

T_x=858.18x1.83=1570.47K

T_3=1570.4x2.25=3533.56K

$$R_e = \frac{R_C}{\rho} = \frac{13.5}{2.25} = 6$$

$$T_4 = \frac{T_3}{{R_e}^{\gamma-1}} = \frac{3533.56}{6^{1.4-1}} = 1752.65K$$

$$\frac{P_3}{P_4} = \left(\frac{T_3}{T_4}\right)^{\frac{\gamma}{\gamma-1}}$$

P_4=6.02bar

$$\eta \text{ air standard efficiency} = 1 - \frac{1}{{R_C}^{\gamma-1}}\left[\frac{(\alpha\rho^{\gamma}-1)}{(\alpha-1)+\gamma\alpha(\rho-1)}\right]$$

$$\eta \text{ air standard efficiency} = 1 - \frac{1}{13.5^{1.4-1}}\left[\frac{(1.83x2.25^{1.4}-1)}{(1.83-1)+1.4x1.83(2.25-1)}\right] = 0.589$$

$$P_2 = P_1R_c^{\gamma} \ ; \ \ P_3 = \alpha P_2 \ ; P_4 = P_3 \ ; \ P_5 = \frac{P_4}{R_e^{\gamma}}$$

$$T_2 = T_1R_c^{\gamma-1} \ ; T_3 = \alpha T_2 = \alpha T_1 R_c^{\gamma-1} \ ; \ T_4 = \rho T_3 = \ \alpha\rho T_1 R_c^{\gamma-1} \ ; T_5 \ = \ \ \alpha\rho^{\gamma}T_1$$

$$\rho = \frac{V_3}{V_2} = \frac{V_c + kV_s}{V_c} = 1 + \frac{kV_s}{V_c} = 1 + \frac{0.1V_s}{0.08V_s} = 2.25$$

$$R_C = \frac{V_S + V_C}{V_C} = 1 + \frac{V_S}{V_C} = 1 + \frac{V_s}{0.08V_s} = 13.5$$

$$T_2 = T_1{R_C}^{\gamma-1} = 303x13.5^{1.4-1} = 858.18K$$

$$P_2 = P_1{R_C}^{\gamma} = 100x13.5^{1.4} = 3823.6KPa$$

$$P_4 = P_3 = 70bar$$

$$P_3 = \alpha P_2; \ \alpha = \frac{P_3}{P_2} = \frac{70}{38.236} = 1.83$$

$T_3 = \alpha T_2; \ T_3 = 1.83x858.18; \quad T_3 = 1570.46K$

$T_4 = \rho T_3; \ T_4 = 2.25x1570.46 = 3533.54K$

$$T_5 \ = \ \ \alpha\rho^{\gamma}T_1;$$

$$T_5 = 1.83x2.25^{1.4}x303 = 1725.63K$$

$Q_{sV} = C_V(T_3 - T_2)$; $Q_{sV} = 0.718(1570.46 - 858.18)$; $Q_{sV} = 511.42kJ/kg$

$Q_{sP} = C_P(T_4 - T_3)$; $Q_{sP} = 1.005(3533.54 - 1570.46)$; $Q_{sP} = 1972.89kJ/kg$

$$Q = Q_{sV} + Q_{sP} = 511.42 + 1972.89 = 2484.31kJ/kg$$

$Q_R = C_v(T_5 - T_1)$; $Q_R = 0.718(1725.63 - 303)$; $Q_R = 1021.45kJ/kg$

$\eta = 1 - \frac{\boldsymbol{Q_R}}{\boldsymbol{Q_s}}$ $=1 - \frac{1021.45}{2484.31}$ =0.588ie $\eta = 58.8\%$
$W = \boldsymbol{Q_s} - \boldsymbol{Q_R} = 2484.31 - 1021.45 = 1462.86\ kJ/kg$

$$p_m = \frac{W_{net}}{Swept\ Volume} = \frac{W_{net}}{V_1 - V_2} = \frac{W_{net}}{V_1\left(1 - \frac{V_2}{V_1}\right)} = \frac{W_{net}}{\frac{RT_1}{P_1}\left(1 - \frac{1}{R_C}\right)} = \frac{P_1 R_C\, W_{net}}{RT_1(R_C - 1)}$$

$p_m = \frac{1x13.5x1462.86}{0.287x303(13.5-1)}$ =15.66 bar

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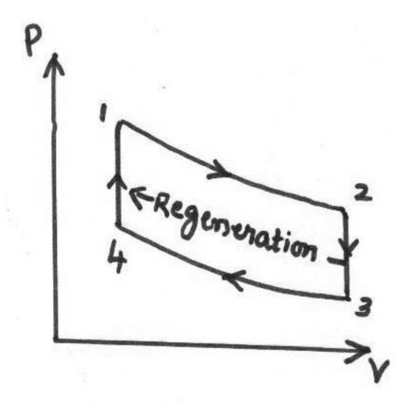
17. The following data refers to an ideal sterling cycle with ideal regenerator. Pressure, temperature and volume of the working medium at the beginning of the isothermal compression are 100kPa, 30^{o}C and 0.05m^3 respectively. The clearance volume of the cycle is 1/10 of initial volume. The maximum temperature attained in the cycle is 700^{o}C. Draw PV and T-S diagrams Calculate i) The network ii) Thermal efficiency with 100% regenerator efficiency iii) Thermal efficiency without regenerator (June –July 2009)



$$V_4 = V_c = 0.1V_3 = 0.1x0.05 = 0.005m^3$$

V_s=0.05-0.005=0.045m^3

$$R_C = \frac{V_C + V_S}{V_C}$$

$$R_C = \frac{0.005 + 0.045}{0.005} = 10$$

$$m = \frac{P_3V_3}{RT_3} = \frac{100x0.05}{0.287x303} = 0.058kg$$

$$T_4 = T_3 = 303K$$

$$Q_{34} = mRT_3ln(R_C) = 0.058x0.287x303xln10 = 11.61kJ$$

$$Q_{12} = mRT_1ln(R_C) = 0.058x0.287x973xln10 = 37.29kJ$$

$$Work\ done = Q_{12} - Q_{34} = 37.29 - 11.61 = 25.6kJ$$

$$Thermal\ efficiency = \frac{work\ done}{heat\ supplied} = \frac{25.6}{37.29} = 68.6\%$$

(or$thermal\ efficiency = \frac{T_h - T_l}{T_h} = \frac{973-303}{303} = 0.688\ or\ 68.8\%$)

Without regenerator heat added is

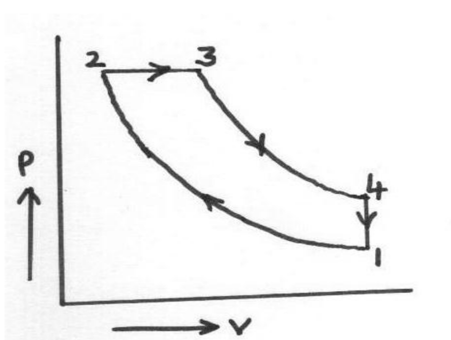
$Q=Q_{41}+Q_{12}$

$$Q = mC_v(T_1 - T_4) + mRT_1ln(R_C)$$

$Q = 0.058x0.718(973 - 303) + 0.058x0.287x973xln10$ =65.19kJ

Efficiency=(25.6/65.19)=39.26%

18. A mass of 1kg of air is taken through a diesel cycle and a joule cycle closed initially the air is at 288K and 1.01325 bar. The compression ratio for both cycles is 15, and the heat added is 1850kJ in each case. Calculate the ideal cycle efficiency and mean effective pressure for each cycle. Comment on the cycle (June/July08)



$$T_2 = T_1 {R_C}^{\gamma-1} = 288x15^{1.4-1} = 850.8K$$

$$P_2 = P_1 {R_C}^{\gamma} = 101.325x15^{1.4} = 4490KPa$$

$$Q = C_p(T_3 - T_2)$$

$$1850 = 1.005(T_3 - 850.8)$$

T_3=2691.6K

$$\rho = \frac{T_3}{T_2} = \frac{2691.6}{850.8} = 3.16$$

$$\text{ŋ air standard efficiency} = 1 - \frac{1}{{R_C}^{\gamma-1}}\left[\frac{(\rho^{\gamma} - 1)}{\gamma(\rho - 1)}\right]$$

$$\text{ŋ air standard efficiency} = 1 - \frac{1}{15^{1.4-1}}\left[\frac{(3.16^{1.4} - 1)}{1.4(3.16 - 1)}\right] = 0.552$$

$$Mean\ efficitve\ pressure = \frac{P_1 {R_C}^{\gamma}\left[\gamma(\rho - 1) - {R_C}^{1-\gamma}(\rho^{\gamma} - 1)\right]}{(\gamma - 1)(R_C - 1)}$$

$$Mean\ effecitve\ pressure = \frac{101.325x15^{1.4}[1.4(3.16 - 1) - 15^{1-1.4}(3.16^{1.4} - 1)]}{(1.4 - 1)(15 - 1)}$$

Mean effective Pressure=1337.11kPa

$$P_2 = P_1 R_c^{\gamma}\ ;\ \ P_3 = P_2\ ; P_4 = \frac{P_3}{R_e^{\gamma}}$$
$$T_2 = T_1 R_c^{\gamma-1}\ ;\ T_3 = \rho T_2 = \ \rho T_1 R_c^{\gamma-1}\ ; T_4 = \ \rho^{\gamma} T_1$$

$T_2 = T_1 {R_C}^{\gamma-1} = 288x15^{1.4-1} = 850.8K$

$P_2 = P_1 {R_C}^{\gamma} = 101.325x15^{1.4} = 4490KPa$

$Q = C_p(T_3 - T_2)$

$1850 = 1.005(T_3 - 850.8)$; T_3=2691.6K

$T_3 = \rho T_2$; $\rho = \frac{T_3}{T_2} = \frac{2691.6}{850.8} = 3.16$

$T_4 = \rho^{\gamma} T_1$

T_4=288x$3.16^{1.4}$=1441.96K

Q_S = 1850kJ/kg(given)

$Q_R = C_v(T_4 - T_1)$; $Q_R = 0.718(1441.96 - 288) = 828.54 kJ/kg$

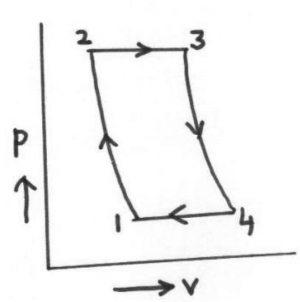
$\eta = 1 - \frac{Q_R}{Q_s} = 1 - \frac{828.54}{1850} = 0.552$ ie $\eta = 55.2\%$

$W = \boldsymbol{Q_s} - \boldsymbol{Q_R} = 1850 - 828.54 = 1021.46\ kJ/kg$

$$p_m = \frac{W_{net}}{Swept\ Volume} = \frac{W_{net}}{V_1 - V_2} = \frac{W_{net}}{V_1\left(1 - \frac{V_2}{V_1}\right)} = \frac{W_{net}}{\frac{RT_1}{P_1}\left(1 - \frac{1}{R_c}\right)} = \frac{P_1 R_c\, W_{net}}{RT_1(R_c - 1)}$$

$$p_m = \frac{1x15x\ 1021.46}{0.287x288(15-1)} = 13.24 \text{ bar}$$

Joule cycle:



$$T_2 = T_1 R_C^{\gamma-1} = 288x15^{1.4-1} = 850.8K$$

$$P_2 = P_1 R_C^{\gamma} = 101.325x15^{1.4} = 4490KPa$$

$$Q_s = C_p(T_3 - T_2)$$

$$1850 = 1.005(T_3 - 850.8)$$

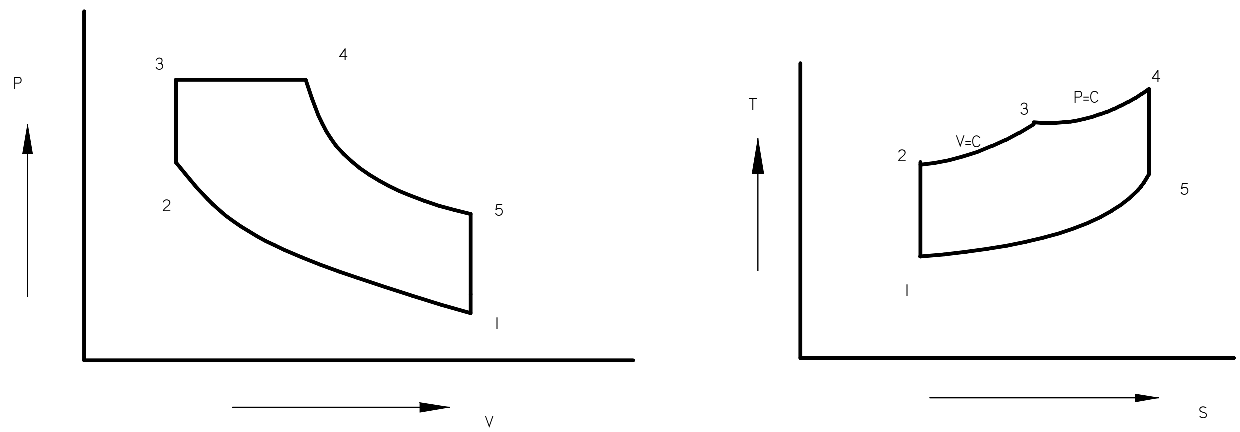
T_3=2691.6K

$$T_4 = \frac{T_3}{R_p^{\frac{\gamma-1}{\gamma}}} = \frac{2691.6}{44.3^{\frac{1.4-1}{1.4}}} = 911.2K$$

$$Q_r = C_p(T_4 - T_1) = 1.005(911.2 - 288) = 626.32kj/kg$$

$$\eta = \frac{Q_s - Q_r}{Q_s} = \frac{1850 - 626.32}{1850} = 0.66$$

19. An oil engine works on the dual combustion cycle, the compression ratio is 10 and expansion ratio is 5.5. The initial pressure and temperature of the air are 1 bar and 30^{o}C . The heat liberated at constant pressure is twice the heat liberate at constant volume. The expansion and compression follows the law $PV^{1.3}$=C, Find the following (i) pressure and temperature at all salient points (ii) mean effective pressure (iii) air standard efficiency (iv) Power developed if the working cycle are 500/min and cylinder diameter is 24cm and stroke length is 40cm.



The expansion and compression follows the law $PV^{1.3}$=C, ie $n_c = n_e$=1.3

$P_2 = P_1 R_c^{n_c}$; $P_3 = \alpha P_2$; $P_4 = P_3$; $P_5 = \frac{P_4}{R_e^{n_e}}$

$T_2 = T_1 R_c^{n_c - 1}$; $T_3 = \alpha T_2 = \alpha T_1 R_c^{n_c-1}$; $T_4 = \rho T_3 = \alpha \rho T_1 R_c^{n_c-1}$; $T_5 = \frac{T_4}{R_c^{n_e-1}}$

$$\rho = \frac{R_C}{R_e} = \frac{10}{5.5} = 1.82$$

$$T_2 = T_1 R_c^{n_c-1} = 303x10^{1.3-1} = 604.56K$$

$$P_2 = P_1 R_c^{n_c} = 100x10^{1.3} = 1995.26KPa$$

$$T_4 = \rho T_3$$

$$T_4 = 1.82 T_3$$

Given Q_{sP}= $2Q_{sV}$

$$C_p(T_4 - T_3) = 2xC_v(T_3 - T_2)$$

$$1.005(1.82T_3 - T_3) = 2x0.718(T_3 - 604.56)$$

$1.005(0.82)T_3$ =$1.436T_3 - 868.15$

$$868.15 = 0.6119\, T_3$$

T_3=1418.77K

$$\alpha = \frac{P_3}{P_2} = \frac{T_3}{T_2} = \frac{1418.77}{604.56} = 2.34$$

$P_3 = \alpha P_2$; P_3=19.9526x2.34; P_3=46.68bar

$$P_4 = P_3 = 46.68bar$$

T_4=1.82x1418.77=2582.16K

$T_5 = \frac{T_4}{R_c^{n_e-1}} = \frac{2582.16}{5.5^{1.3-1}}$=1548.37K

$P_5 = \frac{46.68}{5.5^{1.3}}$; P_5 =5.08bar

$Q_{sV} = C_V(T_3 - T_2)$; $Q_{sV} = 0.718(1418.77 - 604.56)$; $Q_{sV} = 584.60kJ/kg$

$Q_{sP} = C_P(T_4 - T_3)$; $Q_{sP} = 1.005(2582.16 - 1418.77)$; $Q_{sP} = 1169.20kJ/kg$

$$Q_S = Q_{sV} + Q_{sP} = 584.60 + 1169.20 = 1753.80kJ/kg$$

$$W_{net} = W_{1-2} + W_{2-3} + W_{3-4} + W_{4-5} + W_{5-1}$$

$$W_{net} = -\frac{P_2V_2 - P_1V_1}{n_c - 1} + 0 + P_2(V_4 - V_3) + \frac{P_4V_4 - P_5V_5}{n_e - 1} + 0$$

$$W_{net} = -\frac{R(T_2 - T_1)}{n_c - 1} + R(T_4 - T_3) + \frac{R(T_4 - T_5)}{n_c - 1}$$

$$W_{net} = -\frac{0.287(604.56 - 303)}{1.3 - 1} + 0.287(2582.6 - 1418.77) + \frac{0.287(2582.6 - 1548.37)}{1.3 - 1}$$

$$W_{net} = -288.49 + 334.01 + 989.41$$

$$W_{net} = 1034.93kJ/kg$$

$\eta = 1 - \frac{W_{net}}{Q_S}$; $\eta = 1 - \frac{1034.93}{1753.8}$; $\eta = 0.4098$; $\eta = 40.98\%$

$$p_m = \frac{W_{net}}{Swept\ Volume} = \frac{W_{net}}{V_1 - V_2} = \frac{W_{net}}{V_1\left(1 - \frac{V_2}{V_1}\right)} = \frac{W_{net}}{\frac{RT_1}{P_1}\left(1 - \frac{1}{R_c}\right)} = \frac{P_1 R_c\, W_{net}}{RT_1(R_c - 1)}$$

$$p_m = \frac{1x10x0.287x\ 1034.93}{0.287x303(10 - 1)} = 3.79\ bar$$

V$_s$=$\frac{\pi d^2}{4} xlx \frac{x}{60}$ $where\ x = N\ for\ 2\ stroke\ engine\ and\ \frac{N}{2}$ for 4 stroke engine

Assuming 2 stroke engine $x = N$

V$_s$ =$\frac{\pi 0.24^2}{4} x0.4x \frac{500}{60}$ =0.1507m^3/s

$$m = \frac{1x100x0.1507}{0.287x303} = 0.173kg/s$$

Power=$\dot{m}x\ W_{net}$ Power=0.173x1034.93=179.04kW

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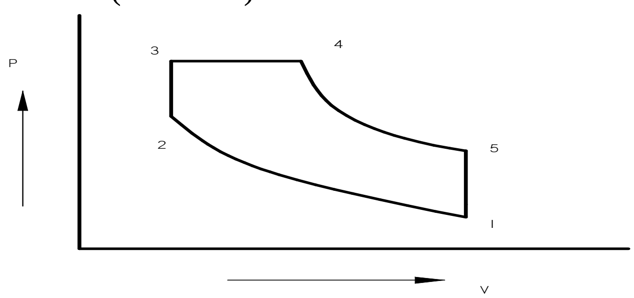
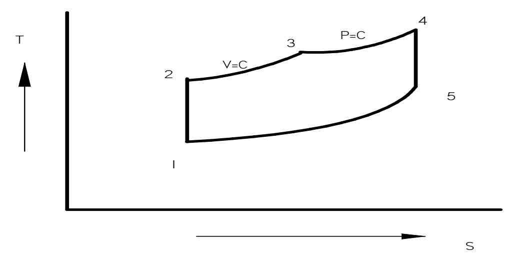
20. The compression ratio for a single cylinder engine operating on dual cycle is 8. The maximum pressure in the cycle is limited to 55 bar. The pressure and temperature of the air at the beginning of the cycle are 1 bar and 27^{0}C. Heat assed during constant pressure process upto 3% of the stroke. Assuming the diameter as 25cm and stroke as 30 cm find the following, the work done per cycle, air standard efficiency and power if number of working cycles are 200/min.(June2012).





$$T_2 = T_1 R_C{}^{\gamma-1} = 300x8^{1.4-1} = 689.3K$$

$$P_2 = P_1 R_C{}^{\gamma} = 100x8^{1.4} = 1837.92KPa$$

$$\rho = \frac{V_3}{V_2} = \frac{V_c + kV_s}{V_c} = 1 + \frac{kV_s}{V_c} = 1 + \frac{k(V_1 - V_c)}{V_c} = 1 + k(R-1)$$
$$\rho = 1 + 0.03(8-1) = 1.21$$

P$_3$=55bar

$$\alpha = \frac{P_3}{P_2} = \frac{55}{18.38} = 2.99$$

Also $\alpha = \frac{T_3}{T_2}$ ie $T_3 = \alpha T_2$

T_3=2.99x689.3=2061K

Also $\rho = \frac{T_4}{T_3}$ ie $T_4 = \alpha T_3$

T_4=1.21x2061=2493.82K

$$R_e = \frac{R_C}{\rho} = \frac{8}{1.21} = 6.61$$

$$T_4 = \frac{T_3}{{R_e}^{\gamma-1}} = \frac{2493.82}{6.61^{1.4-1}} = 1171.6K$$

$$\text{ŋ air standard efficiency} = 1 - \frac{1}{{R_C}^{\gamma-1}}\left[\frac{(\alpha\rho^{\gamma} - 1)}{(\alpha - 1) + \gamma\alpha(\rho - 1)}\right]$$

$$\text{ŋ air standard efficiency} = 1 - \frac{1}{8^{1.4-1}}\left[\frac{(2.99x1.21^{1.4} - 1)}{(2.99 - 1) + 1.4x2.99(1.21 - 1)}\right] = 0.559$$

$$Work\ output = \left[C_p(T_4 - T_3) + C_v(T_3 - T_2)\right] - C_v(T_5 - T_1)$$

$$Work\ output = [\langle 1.005(2493.82 - 2061) + 0.718(2061 - 689.3)\rangle - 0.718(1171.6 - 300)] = 794.1kJ/kg$$

$$m = \frac{P_1V_1}{RT_1}$$

V_1=$\frac{\pi d^2}{4}xl$=$\frac{\pi 0.25^2}{4}x0.3$ =0.0147m^3

$$m = \frac{100x0.0147}{0.287x300} = 0.017kg$$

Work/cycle=0.017x794.1=13.58kJ/cycle

Power=13.58x (200/60)=45.25kW

$P_2 = P_1R_c^{\gamma}$; $P_3 = \alpha P_2$; $P_4 = P_3$; $P_5 = \frac{P_4}{R_e^{\gamma}}$
$T_2 = T_1R_c^{\gamma-1}$; $T_3 = \alpha T_2 = \alpha T_1R_c^{\gamma-1}$; $T_4 = \rho T_3 = \alpha\rho T_1R_c^{\gamma-1}$; $T_5 = \alpha\rho^{\gamma}T_1$

$$\rho = 1 + 0.03(8 - 1) = 1.21$$
$$T_2 = T_1{R_C}^{\gamma-1} = 300x8^{1.4-1} = 689.3K$$

$$P_2 = P_1{R_C}^{\gamma} = 100x8^{1.4} = 1837.92KPa$$

$$\rho = 1 + \frac{K}{100}(R_c - 1)$$

$$\rho = 1 + 0.03(8 - 1) = 1.21$$

P_3=55bar

$$P_3 = \alpha P_2; \quad \alpha = \frac{P_3}{P_2} = \frac{55}{18.38} = 2.99$$

$T_3 = \alpha T_2$; $T_3 = 2.99x689.3$; T_3 =2061K
$T_4 = \rho T_3$; $T_4 = 1.21x2061$; $T_4 = 2493.81K$

$T_5 = \alpha \rho^{\gamma} T_1$; $T_5 = 2.99x1.21^{1.4}x300$; $T_5 = 1171.36K$

$Q_{sV} = C_V(T_3 - T_2)$; $Q_{sV} = 0.718(2061 - 689.3)$; $Q_{sV} = 984.88kJ/kg$

$Q_{sP} = C_P(T_4 - T_3)$; $Q_{sP} = 1.005(2493.81 - 2061)$; $Q_{sP} = 434.97kJ/kg$

$$Q_s = Q_{sV} + Q_{sP}; \quad Q_s = 984.88 + 434.97; \; Q_s = 1419.85kJ/kg$$

$Q_R = C_v(T_5 - T_1)$; $Q_R = 0.718(1171.36 - 300) = 625.63kJ/kg$

$\eta = 1 - \frac{\boldsymbol{Q_R}}{\boldsymbol{Q_s}}$ =$1 - \frac{625.63}{1419.85}$ =0.559 ie $\eta = 55.9\%$
$W = \boldsymbol{Q_s} - \boldsymbol{Q_R} = 1419.85 - 625.63 = 794.22\ kJ/kg$

$$p_m = \frac{W_{net}}{Swept\ Volume} = \frac{W_{net}}{V_1 - V_2} = \frac{W_{net}}{V_1\left(1 - \frac{V_2}{V_1}\right)} = \frac{W_{net}}{\frac{RT_1}{P_1}\left(1 - \frac{1}{R_c}\right)} = \frac{P_1 R_c W_{net}}{RT_1(R_c - 1)}$$

$$p_m = \frac{1x8x\ 794.22}{0.287x300(15-1)} = 5.27\text{bar}$$

$V_s = \frac{\pi d^2}{4} xlx \frac{x}{60}$ $where\ x = N\ for\ 2\ stroke\ engine\ and\ \frac{N}{2}$ for 4 stroke engine

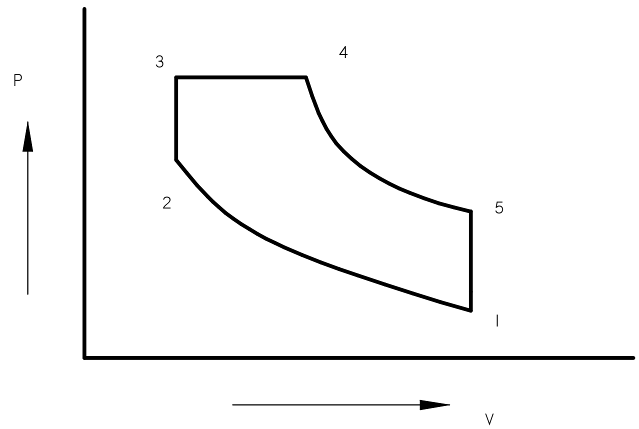
Assuming 2 stroke engine $x = N$

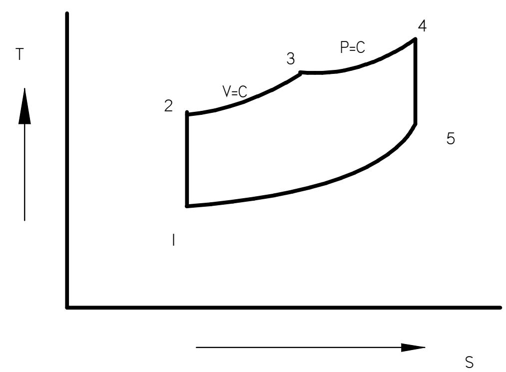
$V_s = \frac{\pi 0.25^2}{4} x0.3x \frac{200}{60}$ =0.049m^3/s

$$m = \frac{P_1 V_1}{RT_1}$$

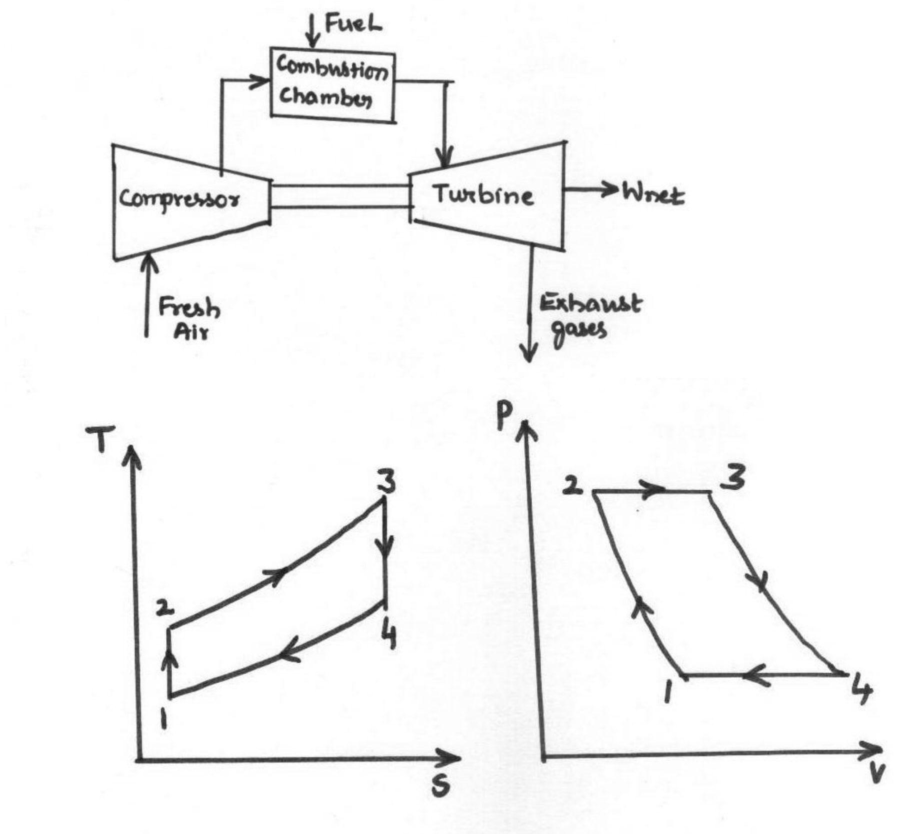
$$m = \frac{1x100x0.049}{0.287x303} = 0.056kg/s$$

Power=$\dot{m}x\ W_{net}$ Power=0.056x794.22=44.47kW





1. Explain with sketch and T-S diagram explain open gas turbine cycle



Gas turbines are prime movers producing mechanical power from the heat generated by the combustion of fuels. They are used in aircraft, some automobile units, industrial installations and small – sized electrical power generating units. A schematic diagram, P-V and T-S diagrams of a open cycle gas turbine power plant is shown above.

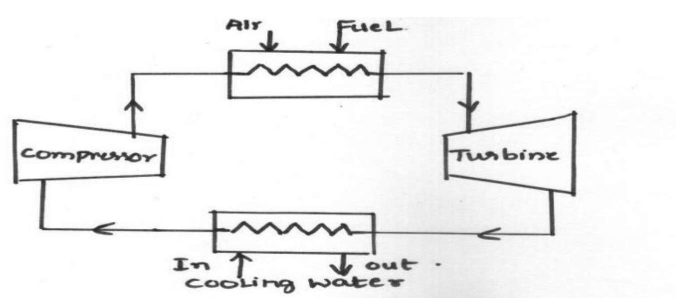
1-2: Reversible adiabatic compression.

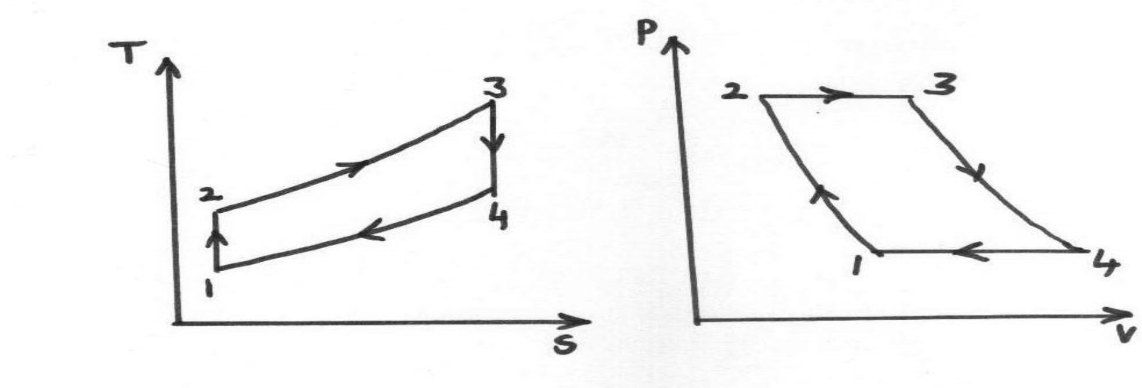
2–3: Reversible constant pressure heat addition.

3–4: Reversible adiabatic expansion.

4–1: Reversible constant pressure heat rejection

2. Explain with sketch and T-S diagram explain closed gas turbine cycle





A schematic diagram, P-V and T-S diagrams of a closed cycle gas turbine power plant is shown above.

1-2 : Reversible adiabatic compression.

2–3: Reversible constant pressure heat addition.

3 – 4: Reversible adiabatic expansion.

4–1: Reversible constant pressure heat rejection.

3. Differentiate between closed and open gas turbine cycle

   Advantages of closed cycle over open cycle gas turbine

1. Higher thermal efficiency
2. Reduced size
3. No contamination
4. Improved heat transmission
5. Improved part load η
6. Lesser fluid friction
7. No loss of working medium
8. Greater output and
9. Inexpensive fuel.

   Disadvantages of closed cycle over open cycle gas turbine

1. Complexity
2. Large amount of cooling water is required. This limits its use of stationary installation or marine use
3. Dependent system
4. The wt of the system pre kW developed is high comparatively, therefore not economical for moving vehicles
5. Requires the use of a very large air heater.

4. Derive an expression for efficiency of ideal brayton cycle.

$$Efficiency, \eta = \frac{Q_s - Q_r}{Q_s}$$

$$\eta = \frac{C_p(T_3 - T_2) - C_p(T_4 - T_1)}{C_p(T_3 - T_2)}$$

$$\eta = 1 - \frac{C_p(T_4 - T_1)}{C_p(T_3 - T_2)}$$
$$\eta = 1 - \frac{(T_4 - T_1)}{(T_3 - T_2)}$$
$$\eta = 1 - \frac{T_1(\frac{T_4}{T_1} - 1)}{T_2(\frac{T_3}{T_2} - 1)}$$
$$\frac{T_2}{T_1} = \left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}} = R_p^{\frac{\gamma-1}{\gamma}}$$
$$\frac{T_3}{T_4} = \left(\frac{P_3}{P_4}\right)^{\frac{\gamma-1}{\gamma}} = \left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}} = R_p^{\frac{\gamma-1}{\gamma}}$$

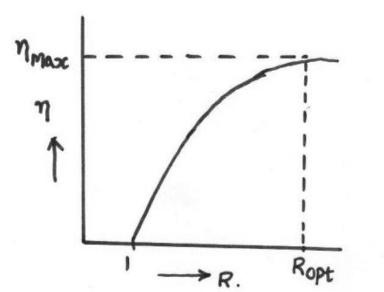
$\frac{T_2}{T_1} = \frac{T_3}{T_4}$ or $\frac{T_4}{T_1} = \frac{T_3}{T_2}$

$$\eta = 1 - \frac{T_1}{T_2}$$
$$\eta = 1 - \frac{1}{R_p^{\frac{\gamma-1}{\gamma}}}$$

5. Derive an expression for the work output of a gas turbine in terms of pressure ratio for maximum and minimum temperatures T_3 and T_1 Henceshow that the pressure ratio for maximum specific work output is given by $R_w = \left(\frac{T_3}{T_1}\right)^{\frac{\gamma}{2(\gamma-1)}}$ (Dec09/Jan10)

Or Derive an expression of optimum pressure ration for maximum net work output in an ideal Brayton cycle. What is the corresponding cycle efficiency? (June/July08)

The variation of network output W_{net} with pressure ratio r_p is shown below.



From figure as r_p increases from 1 to $(r_p)_{max}$, W_{net} increases from zero, reaches a maximum at an optimum value of r_p i.e., $(r_p)_{opt}$ and with further increase in r_p, it reduces and becomes zero when $r_p = r_{pmax}$

$$Wnet = heat\ supplied - heat\ rejected = C_p(T_3 - T_2) - C_P(T_4 - T_1)$$

$\frac{T_2}{T_1} = \left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}} = R_p^{\frac{\gamma-1}{\gamma}}$ or $T_2 = T_1 R_p^{\frac{\gamma-1}{\gamma}}$

$\frac{T_3}{T_4} = \left(\frac{P_3}{P_4}\right)^{\frac{\gamma-1}{\gamma}} = R_p^{\frac{\gamma-1}{\gamma}}$ or $T_4 = \frac{T_3}{R_p^{\frac{\gamma-1}{\gamma}}}$

$$Wnet = C_p(T_3 - T_1 R_p{}^{\frac{\gamma-1}{\gamma}}) - C_v\left(\frac{T_3}{R_p{}^{\frac{\gamma-1}{\gamma}}} - T_1\right)$$

$\frac{dw}{dr} = 0\ for maximum\ work,$ let$\frac{\gamma-1}{\gamma} = y$

$$-T_1 y R_p{}^{y-1} + T_3 y R_p{}^{-y-1} = 0$$

$$T_3 y R_p{}^{-y-1} = T_1 y R_p{}^{y-1}$$

$$T_1 R_p{}^{y-1+y+1} = T_3$$

$$R_p = \left(\frac{T_3}{T_1}\right)^{\frac{1}{2y}}$$

$$R_{popt} = \left(\frac{T_3}{T_1}\right)^{\frac{\gamma}{2(\gamma-1)}}$$

Therefore

$$\eta = 1 - \frac{1}{R_p{}^{\frac{\gamma-1}{\gamma}}}$$

$$\eta = 1 - \frac{1}{\left[\left(\frac{T_3}{T_1}\right)^{\frac{\gamma}{2(\gamma-1)}}\right]^{\frac{\gamma-1}{\gamma}}}$$

$$\eta = 1 - \frac{1}{\left(\frac{T_3}{T_1}\right)^{\frac{1}{2}}}$$

6. Define work ratio in gas turbine and prove $r_w = 1 - \frac{T_1}{T_3} r_p{}^{\frac{\gamma-1}{\gamma}}$

$$Work\ ratio = \frac{Wnet}{Posotive\ work\ done}$$

$$Work\ ratio = \frac{C_p(T_3 - T_4) - C_P(T_2 - T_1)}{C_p(T_3 - T_4)}$$

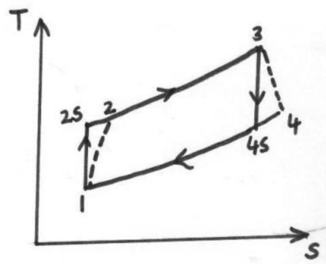
$$r_w = 1 - \frac{(T_2 - T_1)}{(T_3 - T_4)}$$

$$\eta = 1 - \frac{T_1(\frac{T_2}{T_1} - 1)}{T_3(1 - \frac{T_4}{T_3})}$$

$$\eta = 1 - \frac{T_1(R_p{}^{\frac{\gamma-1}{\gamma}} - 1)}{T_3\left[1 - \frac{1}{R_p{}^{\frac{\gamma-1}{\gamma}}}\right]}$$

$$\eta = 1 - \frac{T_1}{T_3} R_p{}^{\frac{\gamma-1}{\gamma}}$$

7. Explain with sketch and T-S diagram explain actual gas turbine cycle



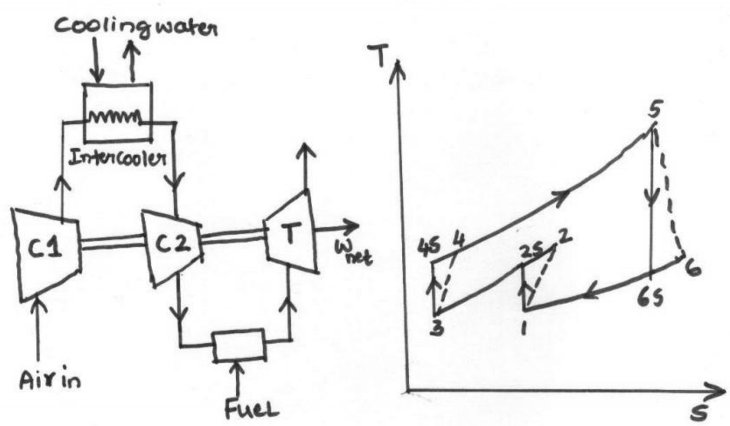
In the ideal Brayton cycle, compression and expansion of air are assumed to be reversible and adiabatic. In reality, however, irreversibility's do exist in the machine operations, Hence the compression and expansion processes are not really constant entropy processes. Entropy tends to be increase (as per the principle of increase of entropy).

$$Efficiency\ of\ turbine, \eta_t = \frac{actual\ work}{ideal\ work} = \frac{(T_3 - T_4)}{(T_3 - T_{4s})}$$

$$Efficiency\ of\ compressor, \eta_c = \frac{ideal\ work}{actual\ work} = \frac{(T_{2s} - T_1)}{(T_2 - T_1)}$$

8. Draw neat line diagram and T-S diagram for the following G.T cycle i) Regeneration ii) Intercooling iii) Reheating (june-July 09)
9. Explain with neat sketch and T-S diagram Explain i) Regeneration ii) Intercooling iii) Reheating, and derive an expressions for efficiency

1. Intercooling:

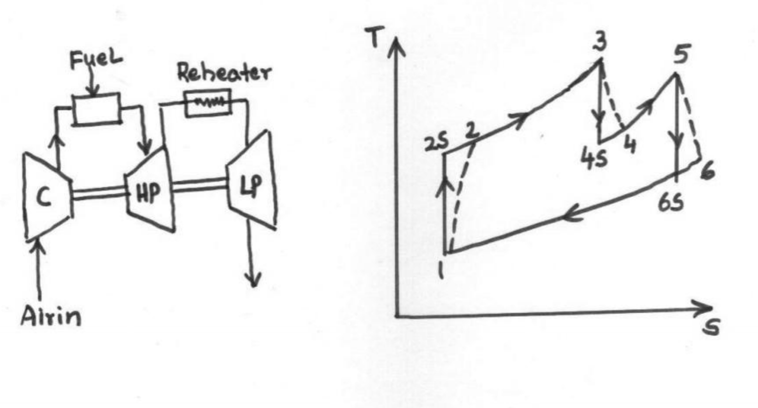


In this arrangement, compression of air is carried out in two or more stages with cooling of the air in between the stages. The cooling takes place in a heat exchanger using some external cooling medium. Figure shows the T-S diagram of a gas turbine plant with two-stage compression with inter cooling.

$$\eta = \frac{(T_5 - T_6) - [(T_4 - T_3) + (T_2 - T_1)]}{(T_5 - T_4)}$$

If air is cooled to a temperature equal to the initial temperature (i.e., if T_3=T_1), inter cooling is said to be perfect.

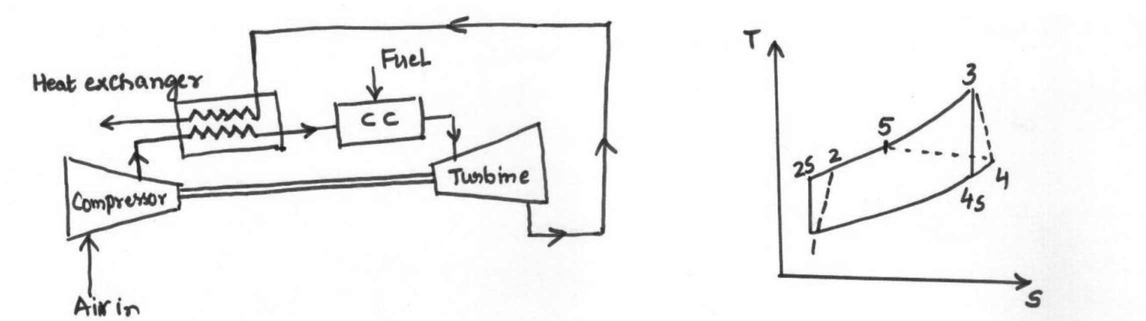
2. Reheating :



Here expansion of working fluid (air) is carried out in 2 or more stages with heating (called reheating) in between stages. The reheating is done in heat exchangers called Reheaters.The expansion takes place in two turbine stages, with reheating in between, are shown.Multi-Stage expansion with reheating, by itself, and does not lead to any improvement in cycle efficiency. In fact, it only reduces.

$$\eta = \frac{\{(T_3 - T_4) + (T_5 - T_6)\} - (T_2 - T_1)}{(T_3 - T_2) + (T_5 - T_4)}$$

3.Regenerator :

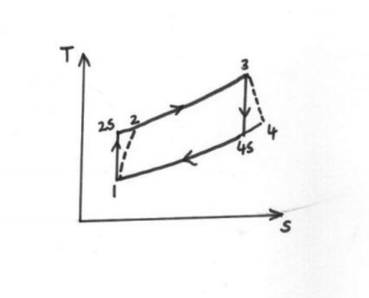


The efficiency of the Brayton cycle can be increased by utilizing part of the energy of exhaust air from the turbine to preheat the air leaving the compressor, in a heat exchanger called regenerator. This reduces the amount of heat supplied $Q_1$from an external source, and also the amount of heat rejected Q_2 to an external sink, by an equal amount. Since $W_{net} = Q_1 - Q_2$ and both Q_1 and Q_2 reduce by equal amounts, there will be no change in the work output of the cycle.Regeneration can be used only if the temperature of air leaving the turbine at 4 is greater than that of air leaving the compressor at 2.

$\eta = \frac{(T_3-T_4)-(T_2-T_1)}{(T_3-T_5)}$, regenerator effectiveness $\epsilon = \frac{actual\ heat\ transfer}{maximum\ heat\ transfer} = \frac{(T_5-T_2)}{(T_4-T_2)}$

In practice, a regenerator is expensive, heavy and bulky and causes pressure losses, which may even decrease the cycle efficiency, instead of increasing it**.**

10. Derive an expression of optimum pressure ration for maximum net work output in an actual Brayton cycle.



$$Wnet = turbine\ work - compressor\ work = C_p(T_3 - T_4) - C_P(T_2 - T_1)$$

$\frac{T_{2s}}{T_1} = \left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}} = R_p^{\frac{\gamma-1}{\gamma}}$ or $T_{2s} = T_1 R_p^{\frac{\gamma-1}{\gamma}}$

$\frac{T_3}{T_{4s}} = \left(\frac{P_3}{P_4}\right)^{\frac{\gamma-1}{\gamma}} = R_p^{\frac{\gamma-1}{\gamma}}$ or $T_{4s} = \frac{T_3}{R_p^{\frac{\gamma-1}{\gamma}}}$

$$\eta_t = \frac{actual\ work}{ideal\ work} = \frac{(T_3 - T_4)}{(T_3 - T_{4s})}$$

$$\eta_c = \frac{ideal\ work}{actual\ work} = \frac{(T_{2s} - T_1)}{(T_2 - T_1)}$$

$$Wnet = C_p \eta_t (T_3 - T_{4s}) - C_P \frac{(T_{2s} - T_1)}{\eta_c}$$

$$Wnet = C_p \eta_t (T_3 - \frac{T_3}{R_p^{\frac{\gamma-1}{\gamma}}}) - \frac{C_p\left(T_1 R_p^{\frac{\gamma-1}{\gamma}} - T_1\right)}{\eta_c}$$

$\frac{dw}{dr} = 0\ for maximum\ work,$ let $\frac{\gamma-1}{\gamma} = y$

$$\eta_t T_3 y R_p^{-y-1} - \ T_1 \frac{y R_p^{y-1}}{\eta_c} = 0$$

$$\eta_t \eta_c T_3 y R_p^{-y-1} = T_1 y R_p^{y-1}$$

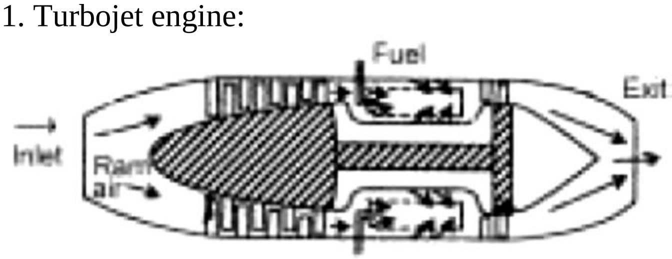
$$T_1 R_p^{y-1+y+1} = \eta_t \eta_c T_3$$

$$R_p = \left(\eta_t \eta_c \frac{T_3}{T_1}\right)^{\frac{1}{2y}}$$

$$R_p = \left(\eta_t \eta_c \frac{T_3}{T_1}\right)^{\frac{\gamma}{2(\gamma-1)}}$$

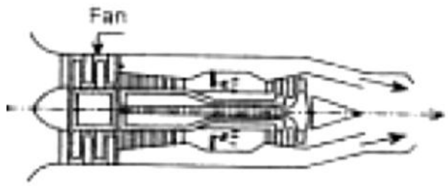
11.Explain the following

1. Turbojet engine:



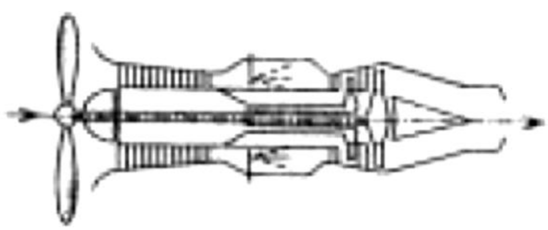
This engine has a diffuser at inlet, the air entering experiences the ram effect due to diffuser and its pressure is increased then the air enters the compressor and is compressed. The compressed air enters the combustion chamber where fuel is added and hot gases at high pressure are expanded through turbine which develops work needed for compressor. Then the gases are expanded through the exit nozzle to develop the required thrust.

2. Turbo Fan engine



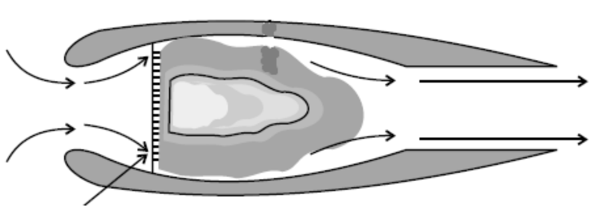
Turbofan engine is the modified turbojet engine in which additional thrust is realised by putting fan at the entry of the engine casing. Fan blades propel by pass air around engine core between inner and outer engine casing. This air does not participate in combustion but provides additional thrust while leaving through exit nozzle. one air stream gets rammed, compressed, burnt, expanded in turbine and finally passes through exit nozzle and other air stream passes through passage between outer and inner casings from inlet to nozzle exit. Total thrust created will be due to two jet streams one due to cold air or fan air and other due to burnt gases leaving turbine.

3. Turbo prop engine



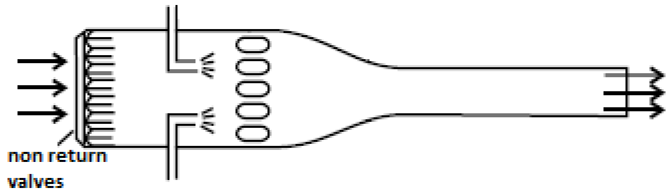
Turboprop (Turbo-propeller) engine differs slightly from turbofan engine. It uses thrust to turn a propeller. A part of turbine output is used to drive the compressor and remaining for driving propeller. Turboprop engines are used in small passenger planes and cargo planes.

4. Ramjet engine



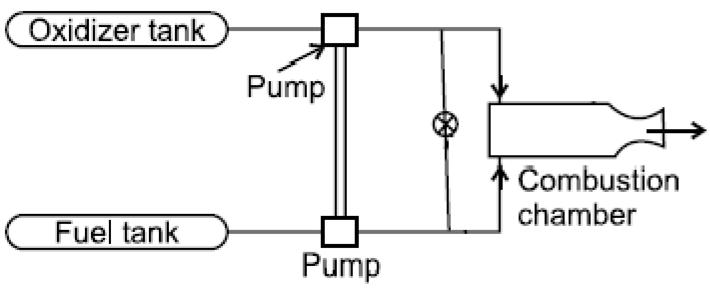
Ramjet engine is the simplest of jet engines having no moving parts. Ramjet is a typically shaped duct open at both ends with air being compressed merely due to forward motion of engine. Fuel is subsequently added for combustion and thus high pressure, high temperature gases exit from exhaust nozzle.

5. Pulse jet engine



It is quite similar to ramjet engine except the difference that pulse jet employs a non-return type mechanical valve of V-type for preventing flow of hot gases through diffuser. Pulse jet engine has diffuser section in which ram compression occurs and after diffuser section a grid of non-return valves is put for maintaining intermittent flow of compressed air.

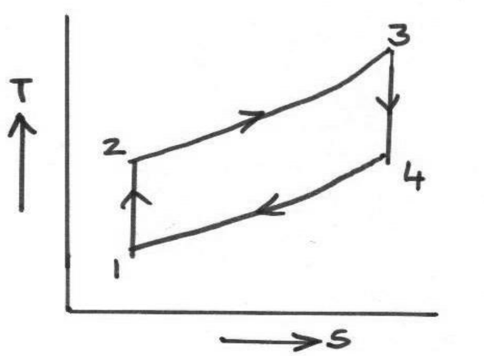
6. Rocket propulsion



The Oxidizer and fuel stored separately in tanks are pumped through a common control valve to combustion chamber and then the hot gases are exhausted to atmosphere through the nozzle providing thrust to the rocket.

NUMERICALS:

1. Air enters the compressor of an ideal standard Brayton cycle at 100kPa, 300K with a volumetric flow rate of $6m^3/s$. The compressor pressure ratio is 10. The turbine inlet temperature is 1500K. Determine i) The thermal efficiency ii) work ratio iii) The power developed.(June-July09)



Air enters the compressor of an ideal standard Brayton cycle at 100kPa, 300K
T_1=300K, P_1=100kPa,
volumetric flow rate of $6m^3/s$ ie $V_f=6m^3/s$
The compressor pressure ratio is 10. Ie $\frac{P_2}{P_1} = 10$,
The turbine inlet temperature is 1500K. Ie T_3=1500K

1-2 adiabatic process

$\frac{T_2}{T_1} = \left\{\frac{P_2}{P_1}\right\}^{\frac{\gamma-1}{\gamma}}$; $\frac{T_2}{300} = \left\{\frac{1000}{100}\right\}^{\frac{1.4-1}{1.4}}$; T_2 = 579.2K

3-4 adiabatic process

$\frac{T_3}{T_4} = \left\{\frac{P_3}{P_4}\right\}^{\frac{\gamma-1}{\gamma}}$; $\frac{1500}{T_4} = \left\{\frac{1000}{100}\right\}^{\frac{1.4-1}{1.4}}$, T_4 = 776.9K

$$\eta = \frac{(W_T) - (W_C)}{(Q)}$$

$$\eta = \frac{(m_a + m_f)(T_3 - T_4) - m_a(T_2 - T_1)}{(m_a + m_f)(T_3 - T_2)}$$

Here assume $m_f = 0$, Since Air fuel ratio or Calorific value of fuel is not given m_f *can not be found out*

Hence efficiency equation becomes

$\eta = \frac{(m_a)(T_3 - T_4) - m_a(T_2 - T_1)}{(m_a)(T_3 - T_2)}$; $\eta = \frac{(T_3 - T_4) - (T_2 - T_1)}{(T_3 - T_2)}$

$\eta = \frac{(1500-776.9)-(579.2-300)}{(1500-579.2)}$ = 0.482 or 48.2%

Work ratio = $W_R = \frac{Net\ work\ output}{Turbine\ work} = \frac{W_T - W_C}{W_T}$

$Work\ ratio = \frac{(1500-776.9)-(579.2-300)}{(1500-776.9)}$ = 0.62

P=$W_T - W_C$=$(m_a + m_f)(T_3 - T_4) - m_a(T_2 - T_1)$

Here assume $m_f = 0$, Since Air fuel ratio or Calorific value of fuel is not given m_f *can not be found out*

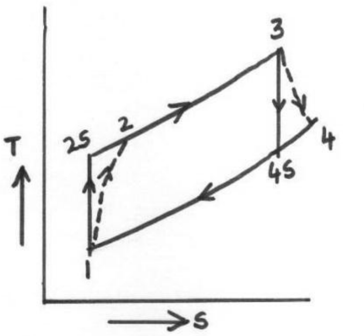
P=$(m_a)(T_3 - T_4) - m_a(T_2 - T_1)$

volumetric flow rate of air 6m^3/s ie V_f=6m^3/s R

$m_a = \frac{PV}{RT} = \frac{100(kPa)x6(m^3/s)}{0.287x300(K)}$=6.96kg/s   Note R for air =0.287kJ/kgK

$P = 6.96(1500 - 776.9) - 6.96(579.2 - 300)$ = 3089.54kW

2. A gas turbine power plant operates on the simple Brayton cycle with air as working fluid and delivers 32MW of power. The minimum and maximum temperatures in the cycle are 310 and 900K, and pressure of air at the compressor exit is 8 times the value at the compressor inlet. Assuming an isentropic efficiency of 80% for the compressor and 86% for the turbine, determine the mass flow rate of air through the cycle (June/July08)



delivers 32MW of power. ie P=32MW=32000kW

The minimum and maximum temperatures in the cycle are 310 and 900K, ie T_1=310K, T_3=900K

pressure of air at the compressor exit is 8 times the value at the compressor inlet. $P_2 = 8P_1$

ie $\frac{P_2}{P_1} = 8$

Assuming an isentropic efficiency of 80% for the compressor and 86% for the turbine,

η_c=0.8,η_t=0.86,

1-2s is adiabatic

$\frac{T_{2s}}{T_1} = \left\{\frac{P_{2s}}{P_1}\right\}^{\frac{\gamma-1}{\gamma}}$; $\frac{T_{2s}}{310} = \{8\}^{\frac{1.4-1}{1.4}}$; T_{2s} = 561.5K

$$\eta_c = \frac{Ideal\ work}{actual\ work} = \frac{(T_{2s} - T_1)}{(T_2 - T_1)}$$

$0.8 = \frac{(561.5-310)}{(T_2-310)}$, T_2 = 624.37K

3-4s   adiabatic process

$\frac{T_3}{T_{4s}} = \left\{\frac{P_3}{P_{4s}}\right\}^{\frac{\gamma-1}{\gamma}}$; $\frac{900}{T_{4s}} = \{8\}^{\frac{1.4-1}{1.4}}$, T_{4s} = 496.8K

$$\eta_t = \frac{actual\ work}{ideal\ work} = \frac{(T_3 - T_4)}{(T_3 - T_{4s})}$$

$0.86 = \frac{(900-T_4)}{(900-496.8)}$, T_4 = 553.25K

$$\eta = \frac{(W_T) - (W_C)}{(Q)}$$

Here also assume $m_f = 0$, Since Air fuel ratio or Calorific value of fuel is not given

$m_f\ can\ not\ be\ found\ out$

Hence efficiency equation becomes

$\eta = \frac{(m_a)(T_3-T_4)-m_a(T_2-T_1)}{(m_a)(T_3-T_2)}$; $\eta = \frac{(T_3-T_4)-(T_2-T_1)}{(T_3-T_2)}$

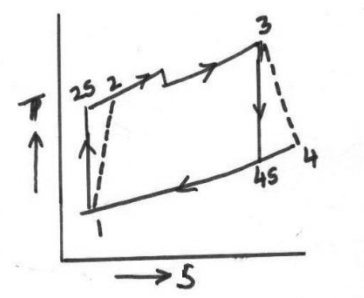
$\eta = \frac{(900-553.25)-(624.37-310)}{(900-624.39)}$ = 0.117 or 11.7%

$P = W_t - W_c = m_a\, c_p\big((T_3 - T_4) - (T_2 - T_1)\big)$

$32000 = m\ 1.005\big((900 - 553.25) - (624.37 - 310)\big)$

m=983.34kg/s

3.In a gas turbine plant the intake temperature and pressure are 18^{o}C and 1 bar respectively. Air is compressed to a pressure of 4.2 bar by a compressor. The isentropic efficiency of compressor is 84%. Gas is heated to 650^{o}C in the combustion chamber, where there is a pressure drop of 0.086 bar. The expansion of gas then occurs to atmospheric pressure in the turbine. The thermal efficiency of plant is 18%. Draw the T-S diagram and find the isentropic efficiency of the turbine. Neglect mass of fuel and take properties of gas as that of air. (May/June2010)



the intake temperature and pressure are 18^{o}C and 1 bar respectively.

T_1=18^{o}C =291K, $P_1 = 1\ bar$;

Air is compressed to a pressure of 4.2 bar by a compressor ie $P_2 = 4.2$bar

$\frac{P_2}{P_1} = \frac{4.2}{1} = 4.2,$

The isentropic efficiency of compressor is 84%.,ie $\boldsymbol{\eta}_c$=0.84,

Gas is heated to 650^{o}C in the combustion chamber, where there is a pressure drop of 0.086 bar.

Ie T_3=650^{o}C =923K ;

Pressure drop in the combustion chamber is 0.086 bar ie$P_2 - P_3 = 0.86$ ie $P_3 = P_2$ −0.086

The thermal efficiency of plant is 18%. ie η_{palnt}=0.18

Neglect mass of fuel and take properties of gas as that of air. $m_f = 0$; R for gas =R for air =0.287kJ/kgK

find the isentropic efficiency of the turbine. $\boldsymbol{\eta}_t$=?

$$Efficiency\ of\ the\ plant\ \ \eta = \frac{(W_T) - (W_C)}{(Q)}$$

$\frac{P_2}{P_1} = 4.2\ \ ie\ \frac{P_2}{1} = 4.2$ ie $P_2 = 4.2\ bar$

1-2s is adiabatic process

$\frac{T_{2s}}{T_1} = \left\{\frac{P_{2s}}{P_1}\right\}^{\frac{\gamma-1}{\gamma}}$; $\frac{T_{2s}}{291} = \{4.2\}^{\frac{1.4-1}{1.4}}$; T_{2s} = 438.5K

$$\eta_c = \frac{Ideal\ work}{actual\ work} = \frac{(T_{2s} - T_1)}{(T_2 - T_1)}$$

$0.84 = \frac{(438.5-291)}{(T_2-291)}$, T_2 = 466.6K

3-4s adiabatic process

Note that here P_3 is not equal to P_2 as there is combustion chamber loss

$P_3 = P_2$ −0.086=4.2 −0.086=4.114 bar

$\frac{T_3}{T_{4s}} = \left\{\frac{P_3}{P_{4s}}\right\}^{\frac{\gamma-1}{\gamma}}$; $\frac{923}{T_{4s}} = \left\{\frac{4.114}{1}\right\}^{\frac{1.4-1}{1.4}}$, T_{4s} = 616.2K

$$\eta = \frac{(W_T) - (W_C)}{(Q)}$$

$$\eta = \frac{(T_3 - T_4) - (T_2 - T_1)}{(T_3 - T_2)}$$

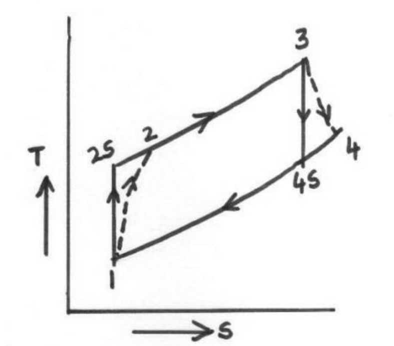
$$0.18 = \frac{(923 - T_4) - (466.6 - 291)}{(923 - 466.6)}$$

Actual temperature at 4 , T_4 = 665.25K

$$\eta_t = \frac{actual\ work}{ideal\ work} = \frac{(T_3 - T_4)}{(T_3 - T_{4s})}$$

$\eta_t = \frac{(923-665.25)}{(923-616.2)}$; $\eta_t = 0.84\ or\ 84\%$

4.In an open cycle gas turbine plant air enters the compressor at 1 bar and 27°C. The pressure after compression is 4 bar. The isentropic efficiencies of the turbine and the compressor are 85% and 80% respectively. Air fuel ratio is 80:1. Calorific value of the fuel use is 42000 kJ/kg. Mass flow rate of air is 2.5kg/s. Determine the power output from the plant and the cycle efficiency. Assume that C_p and γ to be the same for both air and products of combustion (Dec09/ Jan10/June2012old)



air enters the compressor at 1 bar and 27°C. Ie $P_1 = 1bar$;T_1=300K,

The pressure after compression is 4 bar. Ie $P_2 = 4bar$,

The isentropic efficiencies of the turbine and the compressor are 85% and 80% respectively.

$\boldsymbol{\eta}_t$=0.85, $\boldsymbol{\eta}_c$=0.8,

air fuel ratio=80, ie $\frac{m_a}{m_f}$ =80

Calorific value of the fuel use is 42000 kJ/kg; .CV=42000kJ/kg,

Mass flow rate of air is 2.5kg/s. Ie m_a=2.5kg/s
Determine the power output from the plant and the cycle efficiency. P=? η_{plant} =?
Assume that C_p and γ to be the same for both air and products of combustion
ie C_{pg}= C_{pa}=1.005kJ/kgK; $\gamma_g = \gamma_a$=1.4
P_2=P_{2s} *In each and every problem*

$$\frac{P_2}{P_1} = \frac{4}{1} = 4$$

1-2s adiabatic process

$$\frac{T_{2s}}{T_1} = \left\{\frac{P_{2s}}{P_1}\right\}^{\frac{\gamma-1}{\gamma}}; \frac{T_{2s}}{300} = \{4\}^{\frac{1.4-1}{1.4}}; T_{2s} = 445.8K$$

$$\eta_c = \frac{Ideal\ work}{actual\ work} = \frac{(T_{2s} - T_1)}{(T_2 - T_1)}$$

$$0.8 = \frac{(445.8-300)}{(T_2-300)}\ , \quad T_2 = 482.25K$$

$$m_f X CV = (m_a + m_f) C_P (T_3 - T_2)$$

$$m_f X CV = m_f \left(\frac{ma}{m_f} + 1\right) C_P (T_3 - T_2)$$

$$42000 = (80 + 1) 1.005 (T_3 - 482.25)$$

T_3 = 998.2K

3-4s isentropic $P_3 = P_2$ as there is no combustion chamber loss

$$\frac{T_3}{T_{4s}} = \left\{\frac{P_3}{P_{4s}}\right\}^{\frac{\gamma-1}{\gamma}}; \frac{998.2}{T_{4s}} = \{4\}^{\frac{1.4-1}{1.4}}, \quad T_{4s} = 671.7K$$

$$\eta_t = \frac{actual\ work}{ideal\ work} = \frac{(T_3 - T_4)}{(T_3 - T_{4s})}$$

$$0.85 = \frac{(998.2-T_4)}{(998.2-671.7)}\ , \quad T_4 = 720.7K$$

$$\eta = \frac{(W_T)-(W_C)}{(Q)};$$

$$\eta = \frac{(m_a + m_f)\ c_{pg} - m_a c_{pa}(T_2 - T_1)}{(m_a + m_f)\ c_{pg}(T_3 - T_2)}$$

m_a=2.5kg/s

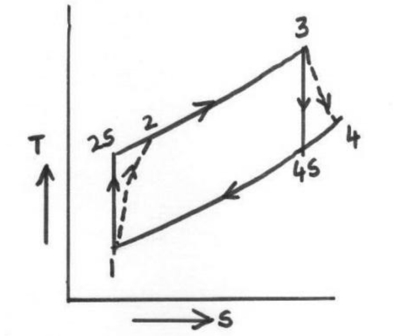
AF =80(given) $\frac{m_a}{m_f} = 80$ ie $\frac{2.5}{m_f} = 80$ ie $m_f = \frac{2.5}{80}$

$$\eta = \frac{\left(2.5+\frac{2.5}{80}\right)x1x(998.2-720.7)-2.5x1(482.25-300)}{\left(2.5+\frac{2.5}{80}\right)x1x(998.2-482.25)} = 0.1846 \text{ or } 18.46\%$$

$$P = W_t - W_c = \left((m_a + m_f)\ c_{pg}(T_3 - T_4) - m_a c_{pa}(T_2 - T_1)\right)$$

$$P = \left(\left(2.5 + \frac{2.5}{80}\right) 1.005(998.2 - 720.7) - 2.5X1.005(482.25 - 300)\right) = 247.99kW$$

5.In a simple gas turbine unit, the isentropic discharge temperature of air flowing out of the compressor is 195°C, while the actual discharge temperature is 240°C. Air conditions at the compression inlet are 1 bar and 17°C. If the air fuel ratio is 75 and net power output from the unit is 650kW. Compute i) isentropic efficiencies of the compressor and the turbine and ii) Overall cycle efficiency. Calorific value of the fuel used is 46110kJ/kg and the unit consumes 312kg/hr of fuel. Assume C_p for gases 1.09kJ/kg-K and γ=1.32 and for air C_p=1.005kJ/kg-K and γ=1.4 (Feb 2005)



In a simple gas turbine unit, the isentropic discharge temperature of air flowing out of the compressor is 195°C, while the actual discharge temperature is 240°C

Ie T_{2s}=195°C =195+273=468K ; T_2=240°C =240+273=513K

Air conditions at the compression inlet are 1 bar and 17°C. Ie P_1=1 bar; T_1=290K;

air fuel ratio=75, $ie \ \frac{m_a}{m_f} = 75$

net power output from the unit is 650kW. P=650kW

Compute i) isentropic efficiencies of the compressor and the turbine and ii) Overall cycle efficiency. I) η_c =?, η_t =? ii) η_{plant} =?

Calorific value of the fuel used is 46110kJ/kg ie CV=46110kJ/kg, ,

the unit consumes 312kg/hr of fuel. m_f=312k/hr =0.087kg/s

Assume C_p for gases 1.09kJ/kg-K and γ=1.32 and for air C_p=1.005kJ/kg-K and γ=1.4, C_{pg}=1.09kJ/kgK, γ_g=1.32,

C_{pa}=1.005kJ/kgK, γa=1.4

1-2 adiabatic process

$\frac{T_{2s}}{T_1} = \left\{\frac{P_{2s}}{P_1}\right\}^{\frac{\gamma a-1}{\gamma a}}; \frac{468}{290} = \left\{\frac{P_{2s}}{P_1}\right\}^{\frac{1.4-1}{1.4}}; \frac{P_{2s}}{P_1} = \left\{\frac{468}{290}\right\}^{\frac{1.4}{1.4-1}}$

$\frac{P_{2s}}{P_1}$ =5.34

$$\eta_c = \frac{Ideal\ work}{actual\ work} = \frac{(T_{2s} - T_1)}{(T_2 - T_1)}$$

$\eta_c = \frac{(468-290)}{(513-290)}$ =0.8

$$m_f X CV = (m_a + m_f) C_P (T_3 - T_2)$$

$$m_f X CV = m_f \left(\frac{ma}{m_f} + 1\right) C_P (T_3 - T_2)$$

$46110 = (75 + 1)1.09(T_3 - 513);$ T_3 = 1069.6K

$\frac{T_3}{T_{4s}} = \left\{\frac{P_3}{P_{4s}}\right\}^{\frac{\gamma g-1}{\gamma g}}$; where γ_g =1.32

Note that here γ_g is not equal to γ_a since in the problem γ_g and γ_a are separately given

In all previous problems we assumed $\gamma_a = \gamma_g$

$\frac{1069.6}{T_{4s}} = \{5.34\}^{\frac{1.32-1}{1.32}}$, T_{4s} = 712.6K

$$P = W_t - W_c = \left((m_a + m_f)\, c_{pg}(T_3 - T_4) - m_a c_{pa}(T_2 - T_1)\right)$$

$$650 = \left((0.087X75 + 0.087)1.09(1069.6 - T_4) - 0.087X75X1.005(513 - 290)\right)$$

T_4 = 776.6K

$$\eta_t = \frac{actual\ work}{ideal\ work} = \frac{(T_3 - T_4)}{(T_3 - T_{4s})}$$

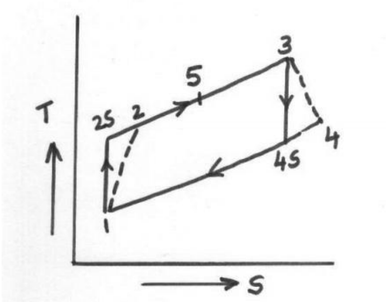
$$\eta_t = \frac{(1069.6 - 776.6)}{(1069.6 - 712.6)}$$

$$\eta_t = 0.82\ or\ 82\%$$

$$\eta = \frac{\left((m_a + m_f)\, c_{pg}(T_3 - T_4) - m_a c_{pa}(T_2 - T_1)\right)}{(m_a + m_f)\, c_{pg}(T_3 - T_2)}$$

$\eta = \frac{650}{(0.087X75+0.087)1.09(1069.6-513)}$ =0.162 or 16.2%

6.A gas turbine plant draws in air at 1.013 bar,10^{o}C and has a pressure ratio of 5.5. The maximum temperature in the cycle is limited to 750^{o}C. Compression is conducted in an uncooled rotary compressor having an efficiency of 82% and expansion takes place in a turbine with an isentropic of 85%. A heat exchanger with efficiency 70% is fitted between the compressor outlet and combustion chamber. For an air flow of 40kg/s, find i) Overall efficiency of cycle ii) Turbine output iii) Air fuel ratio if the calorific value of the fuel used is 45.22MJ/kg (Dec 08/Jan09)



A gas turbine plant draws in air at 1.013 bar,10^{o}C ie $P_1 = 1.013 bar$, T_1=283K

has a pressure ratio of 5.5 ie $\frac{P_2}{P_1} = 5.5,$

The maximum temperature in the cycle is limited to 750^{o}C ie T_3=1023K,

Compression is conducted in an uncooled rotary compressor having an efficiency of 82% and expansion takes place in a turbine with an isentropic of 85%. η_c=0.82,η_t=0.85,

A heat exchanger with efficiency 70% is fitted between the compressor outlet and combustion chamber ε = 0.7,

For an air flow of 40kg/s,m_a=40kg/s,

Air fuel ratio if the calorific value of the fuel used is 45.22MJ/kg AF=? CV=45220kJ/kgK

1-2s Adiabatic process

$\frac{T_{2s}}{T_1} = \left\{\frac{P_{2s}}{P_1}\right\}^{\frac{\gamma-1}{\gamma}}$; $\frac{T_{2s}}{283} = \{5.5\}^{\frac{1.4-1}{1.4}}$; T_{2s} = 460.6K

$$\eta_c = \frac{Ideal\ work}{actual\ work} = \frac{(T_{2s} - T_1)}{(T_2 - T_1)}$$

$0.82 = \frac{(460.6-283)}{(T_2-283)}$, T_2 = 499.6K

3-4s is adiabatic process

$\frac{T_3}{T_{4s}} = \left\{\frac{P_3}{P_{4s}}\right\}^{\frac{\gamma-1}{\gamma}}$; $\frac{1023}{T_{4s}} = \{5.5\}^{\frac{1.4-1}{1.4}}$, T_{4s} = 628.6K

$Note: P_3 = P_2$ as there is no combustion chamber loss $P_{4s} = P_1$ $\frac{P_3}{P_{4s}} = \frac{P_1}{P_{2s}}$=5.5

$$\eta_t = \frac{actual\ work}{ideal\ work} = \frac{(T_3 - T_4)}{(T_3 - T_{4s})}$$

$0.85 = \frac{(1023-T_4)}{(1023-628.6)}$, T_4 = 687.8K

$$\epsilon = \frac{actual\ heat\ transfer}{maximum\ heat\ transfer} = \frac{(T_5 - T_2)}{(T_4 - T_2)}$$

$0.7 = \frac{(T_5-499.6)}{(687.8-499.6)}$; T_5 = 631.34K

$$m_f XCV = (m_a + m_f)C_{Pg}(T_3 - T_2)$$

$$m_f XCV = m_f\left(\frac{m_a}{m_f} + 1\right)C_{Pg}(T_3 - T_2);$$

$$CV = \left(\frac{m_a}{m_f} + 1\right)C_{Pg}(T_3 - T_2)$$

$$45220 = \left(\frac{m_a}{m_f} + 1\right)1.005(1023 - 631.34)$$

$\frac{m_a}{m_f} = 113.88$ ie $\frac{40}{m_a} = 113.88$ ie $m_f = 0.35$kg/s

Assume $C_{pg} = C_{pa}$=1.005kJ/kgK as C_{pg} *is not given*

$Turbine\ output = (m_a + m_f)c_{pg}(T_3 - T_4)$

$Turbine\ output = (40 + 0.35)1.005(1023 - 687.8)$

Turbine Output = 13592.95KW

Heat supplied Q_s=$(m_a + m_f)c_{pg}(T_3 - T_5)$ as 2-3 air is heated by exhaust gases

$$\eta = \frac{(m_a + m_f)c_{pg}(T_3 - T_4) - m_a c_{pa}(T_2 - T_1)}{(m_a + m_f)c_{pg}(T_3 - T_5)}$$

$\eta = \frac{(m_a+m_f)(T_3-T_4)-m_a(T_2-T_1)}{(m_a+m_f)(T_3-T_5)}$ as $C_{pg} = C_{pa}$

$\eta = \frac{(40+0.35)(1023-687.8)-40(499.6-283)}{(40+0.35)(1023-631.34)}$ = 0.3028 or 30.28%

7. A gas turbine plant operates on ideal Brayton cycle. The minimum and maximum cycle temperature are respectively T_1=300K and T_3= 800K. Find the value of the optimum pressure ratio for maximum specific work output and cycle efficiency for this condition. Is it possible to improve this cycle efficiency by including a regenerator substantiate? (jan/Feb 06)

   Data: T_1=300K and T_3= 800K

   Find the value of the optimum pressure ratio for maximum specific work output and cycle efficiency for this condition R_{opt}=?

For maximum work output pressure ratio= $R_{opt} = \left[\frac{T_3}{T_1}\right]^{\frac{\gamma}{2(\gamma-1)}}$

Hence for Maximum specific work output $\frac{P_2}{P_1}$=$\left(\frac{T_3}{T_1}\right)^{\frac{\gamma}{2(\gamma-1)}}$

$$R_{opt} = \left[\frac{T_3}{T_1}\right]^{\frac{\gamma}{2(\gamma-1)}}$$

$$R_{opt} = \left(\frac{800}{300}\right)^{\frac{1.4}{2(1.4-1)}} = 5.57$$

$\frac{T_2}{T_1} = \left\{\frac{P_2}{P_1}\right\}^{\frac{\gamma-1}{\gamma}}$; $\frac{T_2}{300} = \{5.57\}^{\frac{1.4-1}{1.4}}$;T_2 = 490K

$\frac{T_3}{T_4} = \left\{\frac{P_3}{P_4}\right\}^{\frac{\gamma-1}{\gamma}}$ assuming $P_3 = P_2$; $P_4 = P_4$ hence $\frac{P_3}{P_4} = \frac{P_2}{P_1}$

$\frac{800}{T_4} = \{5.57\}^{\frac{1.4-1}{1.4}}$, T_4 = 489.76K

$$\eta = \frac{(W_T) - (W_C)}{(Q)}$$

$$\eta = \frac{(m_a + m_f)c_{pg}(T_3 - T_4) - m_a c_{pa}(T_2 - T_1)}{(m_a + m_f)c_{pg}(T_3 - T_5)}$$

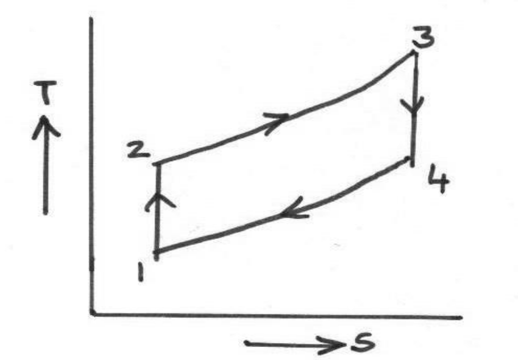
Since $c_{pg} = c_{pa}$ and m_f is neglected above equation reduced to

$$\eta = \frac{(T_3 - T_4) - (T_2 - T_1)}{(T_3 - T_2)}$$

$\eta = \frac{(800-489.76)-(490-300)}{(800-490)}$ = 0.3878 or 38.78%

As $T_4 = T_2$, regenerator cannot be used (for regenerator $T_4 > T_2$)

7.A simple gas turbine plant operating on the Brayton cycle has air entering the compressor at 100kPa and 27°C. The pressure ratio=9.0 and maximum cycle temperature =727°C. What will be the percentage change in cycle efficiency and network output if the expansion in the turbine is divided into two stages each of pressure ratio 3, with intermediate reheating to 727°C?. Assume compression and expansion is isentropic (july 2010)



air entering the compressor at 100kPa and 27°C. Ie P_1=100kPa; T_1=300K

The pressure ratio=9.0 ie $\frac{P_2}{P_1} = 9$,

maximum cycle temperature =727°C, T_3=1000K

First case is simple gas turbine cycle

In both cases η_t and η_c are not given

$\frac{T_{2s}}{T_1} = \left\{\frac{P_2}{P_1}\right\}^{\frac{\gamma-1}{\gamma}}$; $\frac{T_{2s}}{300} = \{9\}^{\frac{1.4-1}{1.4}}$; T_{2s} = 562K

$T_2 = T_{2s}$ since η_c is not given

$\frac{T_3}{T_{4s}} = \left\{\frac{P_3}{P_4}\right\}^{\frac{\gamma-1}{\gamma}}$; $\frac{1000}{T_{4s}} = \{9\}^{\frac{1.4-1}{1.4}}$, T_{4s} = 533.77K

$T_4 = T_{4s}$ since η_t is not given

$$\eta = \frac{(W_T) - (W_C)}{(Q)}$$

Here air fuel ratio or calorific value of fuel is not given Hence

$m_f = 0 (assume) since\ mass\ of\ fuel\ cannot\ be\ found\ out$

$$\eta = \frac{(m_a + m_f)c_{pg}(T_3 - T_4) - m_a c_{pa}(T_2 - T_1)}{(m_a + m_f)c_{pg}(T_3 - T_5)}$$

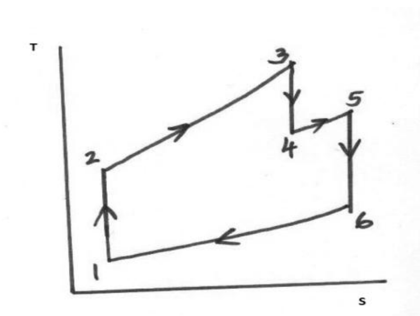
Since $c_{pg} = c_{pa}$ and m_f is neglected above equation reduced to

$$\eta = \frac{(T_3 - T_4) - (T_2 - T_1)}{(T_3 - T_2)}$$

$\eta = \frac{(1000-533.77)-(562-300)}{(1000-562)}$ = 0.466 or 46.6%, Wnet=204kJ/kg

if the expansion in the turbine is divided into two stages each of pressure ratio 3, with intermediate reheating to 727°C

$\frac{P_3}{P_4} = \frac{P_5}{P_6}$ =3 and $T_5 = T_3$ =727°C=1000K



T_2 Calculation is same as case 1. Ie $T_2 = 562K$

Let intermediate pressure for turbine stages p_i ie $p_4 = p_5 = p_i$

$$\frac{T_3}{T_{4s}} = \left\{\frac{P_3}{P_4}\right\}^{\frac{\gamma-1}{\gamma}}$$

$\frac{1000}{T_{4s}} = \{3\}^{\frac{1.4-1}{1.4}}$, T_{4s} = 730.6K ; T_4 = 730.6K

$T_4 = T_{4s}$ as η_{t1} is not given

$T_6 = T_4$=730.6K (as $T_5 = T_3$, and pressure ratio is same ie 3)

$$\eta = \frac{(W_{T1} + W_{T2}) - (W_C)}{(Q_{s1} + Q_{s2})}$$

$$\eta = \frac{\{(m_a + m_f)c_{pg}(T_3 - T_4) + (m_a + m_f)c_{pg}(T_5 - T_6)\} - m_a c_{pa}(T_2 - T_1)}{(m_a + m_f)c_{pg}(T_3 - T_2) + (m_a + m_f)c_{pg}(T_5 - T_4)}$$

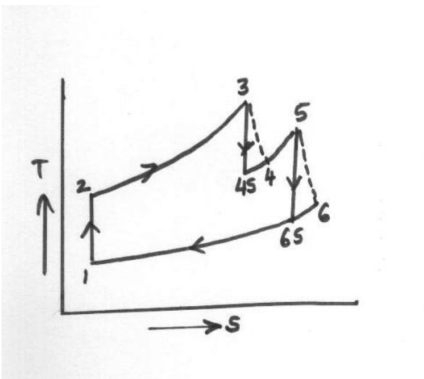
$$\eta = \frac{\{(T_3 - T_4) + (T_5 - T_6)\} - (T_2 - T_1)}{(T_3 - T_2) + (T_5 - T_4)}$$

As m_f is neglected and $c_{pg} = c_{pa}$

$$\eta = \frac{\{(1000 - 730.6) + (1000 - 730.6)\} - (562 - 300)}{(1000 - 562) + (1000 - 730.6)}$$

= 0.3912 or 39.12%, Wnet=268kj/kg

8.In a reheat gas turbine cycle , comprising one compressor and two turbines, air is compressed from 1 bar, 27^{o}C to 6 bar. The highest temperature in the cycle is 900^{o}C. The expansion in the first stage turbine is such that the work from it just equals the work required by the compressor. Air is reheated between the two stages of expansion to 850^{o}C. Assume that the isentropic efficiencies the first stage and the second stage turbines are 85% each and that the working substance is air and calculate the cycle efficiency (June-July 2004/Dec2011)



In a reheat gas turbine cycle , comprising one compressor and two turbines,

$W_c = m_a c_{pa}(T_2 - T_1)$; $W_t = W_{t1} + W_{t2} = (m_a + m_f)c_{pg}(T_3 - T_4) + (m_a + m_f)c_{pg}(T_5 - T_6)$

$Q_s = Q_{s1} + Q_{s2} = (m_a + m_f)c_{pg}(T_3 - T_2) + (m_a + m_f)c_{pg}(T_5 - T_4)$

air is compressed from 1 bar, 27^{o}C to 6 bar T_1=300K, P_1=100kPa=1bar; $P_2 = 6bar$ hence

$\frac{P_2}{P_1} = \frac{6}{1} = 6$

Air is reheated between the two stages of expansion to 850^{o}C.T_5=1123K,

Assume that the isentropic efficiencies the first stage and the second stage turbines are 85% each

$\boldsymbol{\eta}_{t1} = \boldsymbol{\eta}_{t2}$=0.85,

Ie The expansion in the first stage turbine is such that the work from it just equals the work required by the compressor $w_{t1} = w_c$

In the above problem the isentropic efficiency of the compressor is not given Hence assume

$\boldsymbol{\eta}_c$= 100% ie $T_{2s} = T_2$

$\frac{T_{2s}}{T_1} = \left\{\frac{P_2}{P_1}\right\}^{\frac{\gamma-1}{\gamma}}$; $\frac{T_{2s}}{300} = \{6\}^{\frac{1.4-1}{1.4}}$; T_{2s} =T_2 = 500.5K as $\boldsymbol{\eta}_c$= 100%

$w_{t1} = w_c$

$(m_a + m_f)c_{pg}(T_3 - T_4) = m_a c_{pa}(T_2 - T_1)$

$(T_3 - T_4) = (T_2 - T_1)$ as m_f *is neglected* $c_{pg} = c_{pa}$ as c_{pg} *is not* given

1173-T_4 = 500.5-300

T_4=972.5K

$$\eta_{t1} = \frac{actual\ work}{ideal\ work} = \frac{(T_3 - T_4)}{(T_3 - T_{4s})}$$

$0.85 = \frac{(1173-972.5)}{(1173-T_{4s})}$; T_{4s} = 937.12K

3-4s process is adiabatic

$\frac{T_3}{T_{4s}} = \left\{\frac{P_3}{P_{4s}}\right\}^{\frac{\gamma-1}{\gamma}}$; $\frac{1173}{937.12} = \left\{\frac{P_3}{P_{4s}}\right\}^{\frac{1.4-1}{1.4}}$; $\frac{P_3}{P_{4s}} = \left\{\frac{1173}{937.12}\right\}^{\frac{1.4}{1.4-1}}$; $\frac{P_3}{P_{4s}} = 2.19$

$$\frac{P_2}{P_1} = \frac{P_3}{P_6} = \frac{P_3}{P_4}\ x\frac{P_4}{P_6}$$

$\frac{P_2}{P_1} = \frac{P_3}{P_4}\ x\frac{P_5}{P_6}$ as $P_5 = P_4$

6=2.19$x\frac{P_5}{P_{6s}}$; $\frac{P_5}{P_{6s}} = 2.74$

5-6s is isentropic

$\frac{T_5}{T_{6s}} = \{2.74\}^{\frac{1.4-1}{1.4}}$; $\frac{1123}{T_{6s}} = \{2.74\}^{\frac{1.4-1}{1.4}}$; T_{6s}=842K

$$\eta_{t2} = \frac{actual\ work}{ideal\ work} = \frac{(T_5 - T_6)}{(T_5 - T_{6s})}$$

$0.85 = \frac{(1123-T_6)}{(1123-842)}$; T_6 = 884.15K

$$\eta = \frac{\{(m_a + m_f)c_{pg}(T_3 - T_4) + (m_a + m_f)c_{pg}(T_5 - T_6)\} - m_a c_{pa}(T_2 - T_1)}{(m_a + m_f)c_{pg}(T_3 - T_2) + (m_a + m_f)c_{pg}(T_5 - T_4)}$$

$$\eta = \frac{\{(T_3 - T_4) + (T_5 - T_6)\} - (T_2 - T_1)}{(T_3 - T_2) + (T_5 - T_4)}$$

As m_f is neglected and $c_{pg} = c_{pa}$

$$\eta = \frac{\{(1173 - 972.5) + (1123 - 884.15)\} - (500.5 - 300)}{(1173 - 562) + (1123 - 972.5)}$$

= 0.3136 or 31.36%,

9.A G.T. cycle having 2 stage compression with intercooling in between stages and 2 stages of expansion with reheating in between the stages has an overall pressure ratio of 8. The maximum cycle temperature is 1400^0K and the compressor inlet conditions are 1 bar and 27^0C. The compressors have ηs of 80% and turbines have ηs of 85%. Assuming that the air is cooled back to its original temperature after the first stage compression and gas is reheated back to its original temperature after 1st stage of expansion, determine (i) the net work output (ii) the cycle η_{th}.

Data:

2 stage compression with intercooling in between stages and 2 stages of expansion with reheating in between the stages ie 2 compressor with inter cooler and 2 turbine with reheater

overall pressure ratio of 8 ie $\frac{P_4}{P_1} = 8bar$

W_c=$W_{c1} + W_{c2} = m_a c_{pa}(T_2 - T_1) + m_a c_{pa}(T_4 - T_3)$;

$$W_t = W_{t1} + W_{t2} = (m_a + m_f)c_{pg}(T_5 - T_6) + (m_a + m_f)c_{pg}(T_7 - T_8)$$
$$Q_s = Q_{s1} + Q_{s2} = (m_a + m_f)c_{pg}(T_5 - T_4) + (m_a + m_f)c_{pg}(T_7 - T_6)$$

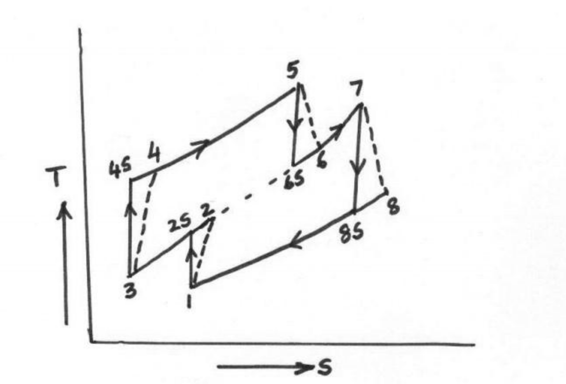
The maximum cycle temperature is 1400^0K ie $T_5 = 1400^0$K
the compressor inlet conditions are 1 bar and 27^0C. Ie P_1= 1 bar ; $T_1 = 300^0$K
The compressors have ηs of 80% and turbines have ηs of 85%
Ie $\eta_{C1} = 0.8 = \eta_{C2},\ \eta_{t1} = \eta_{t2} = 0.85$,
Assuming that the air is cooled back to its original temperature after the first stage compression ie $T_3 = T_1$
and gas is reheated back to its original temperature after 1st stage of expansion, $T_7 = T_5$

Solution: $\frac{P_4}{P_1} = 8bar$ $T_5 = 1400^0$K $T_1 = 300^0$K, P_1= 1 bar

$\eta_{C1} = 0.8 = \eta_{C2},\ \eta_{t1} = \eta_{t2} = 0.85$,

$T_3 = T_1; T_7 = T_5$

also assume $m_f = 0$, Since Air fuel ratio or Calorific value of fuel is not given $m_f\ can\ not\ be\ found\ out$



For maximum work output, $\frac{P_2}{P_1} = \frac{P_4}{P_3} = \frac{P_5}{P_6} = \frac{P_7}{P_8} = \sqrt{\frac{P_4}{P_1}} = \sqrt{\frac{P_5}{P_8}} = \sqrt{8}$

$\therefore Intermediate\ Pressure, P_2 = P_3 = P_6 = P_7 = 2.83\ bar$

For process 1-2s is adiabatic process, $\frac{T_{2s}}{T_1} = \left\{\frac{P_2}{P_1}\right\}^{\frac{\gamma-1}{\gamma}}$ ie $T_{2s} = T_1\left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}}$

$$T_{2s} = 300\ (2.83)^{0.286} = 403.95^0\text{K}$$

But $\eta_{c1} = 0.8 = \frac{T_{2s} - T_1}{T_2 - T_1} = \frac{403.95 - 300}{T_2 - 300}$ $\therefore T_2 = 429.9^0 K$

Since $T_3 = T_1$ and $\frac{P_4}{P_3} = \frac{P_2}{P_1}$

We have $T_{4s} = T_{2s} = 403.95^0K$

Also since $\eta_{C1} = \eta_{C2}$, $T_4 = T_2 = 429.9^0K$

$\therefore$ Compressor work, $W_C = C_{Pa}(T_2 - T_1) + C_{Pa}(T_4 - T_3)$
$= 2\,C_P(T_2 - T_1)$
$= 2\,(1.005)\,(429.9 - 300)$
$= 261.19$ kJ/kg

For process 5 – 6, Adiabatic process

$$\frac{T_{6s}}{T_5} = \left(\frac{P_6}{P_5}\right)^{\frac{\gamma-1}{\gamma}} \quad \therefore T_{6s} = 1400\left(\frac{1}{2.83}\right)^{0.286} = 1039.72^0\lambda$$

But $\eta_{t1} = \frac{T_5 - T_6}{T_5 - T_{6s}}$ *i.e.*, $0.85 = \frac{1400 - T_6}{1400 - 1039.72}$ $\therefore T_6 = 1093.76^0 K$

Since $T_7 = T_5$ and $\frac{P_5}{P_6} = \frac{P_7}{P_8}$, then $T_8 = T_6$

Since $\eta_{t1} = \eta_{t2}$, $T_6 = T_8 = 1093.76^0K$

$\therefore$ Turbine work, $W_t = C_{Pg}(T_5 - T_6) + C_{Pg}(T_7 - T_8)$
$= 2\,C_{Pg}(T_5 - T_6)$
$= 2\,(1.005)\,(1400 - 1093.76)$
$= 615.54$ kJ/kg

Note $C_{Pg} = C_{Pa} = 1.005$ kJ/kgK as C_{Pg} is not given

$\therefore W_{Net} = W_T - W_C = 354.35$ kJ/kg

$\eta_{th} = ?$

Heat Supplied, $Q_s = C_{Pg}(T_5 - T_4) + C_{Pg}(T_7 - T_6)$
$= 1.005\,(1400 - 429.9 + 1400 - 1093.76)$
$= 1282.72$ kJ/kg

$$\therefore \eta_{th} = \frac{354.35}{1282.72} = 0.276 \text{ or } 27.6\%$$

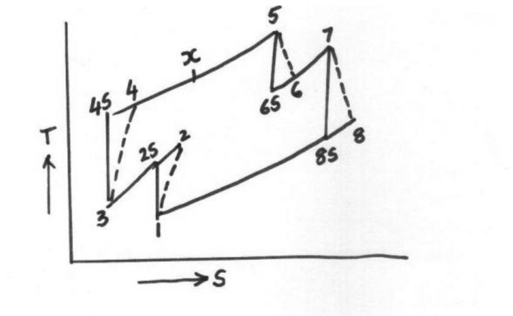
10.A two stage gas turbine cycle receives air at 100 kPa and 15^0C. The lower stage has a pressure ratio of 3, while that for the upper stage is 4 for the compressor as well as the turbine. The temperature rise of the air compressed in the lower stage is reduced by 80% by intercooling. Also, a regenerator of 78% effectiveness is used. The upper temperature limit of the cycle is 1100^0C. The turbine and the compressor ηsare 86%. Calculate the mass flow rate required to produce 6000kW.

Data
A two stage gas turbine cycle
Ie 2 compressor with inter cooling and 2 stage turbine with reheater
The lower stage has a pressure ratio of 3, $\frac{P_2}{P_1} = \frac{P_5}{P_6}$ =3
upper stage is 4 for the compressor as well as the turbine
$\frac{P_3}{P_2} = \frac{P_4}{P_5}$ =4
The temperature rise of the air compressed in the lower stage is reduced by 80% by intercooling.
$(T_3 - T_2)$=0.8 $(T_2 - T_1)$

Also, a regenerator of 78% effectiveness is used $\epsilon = 0.78$
The upper temperature limit of the cycle is 1100^0C ie T_5=1100^0C=1373^0K
The turbine and the compressor ηsare 86%.ie $\eta_{C1} = \eta_{C2} = \eta_{t1} = \eta_{t2} = 0.86$
produce 6000kW ie P = 6000 kW

Solution: P_1 = 1 bar $T_1 = 288^0K$
$\frac{P_2}{P_1} = 3$, $\frac{P_4}{P_3} = 4$ $\eta_{IC} = 0.8$
$\varepsilon = \eta_{reg} = 0.78$, $T_5 = 1373^0K$, $\eta_{C1} = \eta_{C2} = \eta_{t1} = \eta_{t2} = 0.86$

$\dot{m} = ?$ if P = 6000 kW



Process 1-2s is isentropic compression $\therefore \frac{T_{2s}}{T_1} = \left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}}$

$\therefore T_{2s} = 288\,(3)^{0.286} = 410.75^0K$

But $\eta_{C1} = \frac{T_{2s} - T_1}{T_2 - T_1}$ *i.e.*, $0.86 = \frac{410.75 - 288}{T_2 - 288}$ $\therefore T_2 = 430.73K$

Also, $\eta_{IC} = \frac{T_2 - T_3}{T_2 - T_1}$ *i.e.*, $0.8 = \frac{430.73 - T_3}{430.73 - 288}$ $\therefore T_3 = 316.54K$

$(T_2 - T_3)$=0.8 $(T_2 - T_1)$ (data)

$430.73 - T_3 = 0.8(430.73 - 288)$; $T_3 = 316.54K$

Process 3-4s is 2nd stage isentropic compression $\therefore \frac{T_{4s}}{T_3} = \left(\frac{P_4}{P_3}\right)^{\frac{\gamma-1}{\gamma}}$

$\therefore T_{4s} = 316.54\ (4)^{0.286} = 470.57K$

But $\eta_{C2} = \frac{T_{4s} - T_3}{T_4 - T_3}$ *i.e.,* $0.86 = \frac{470.57 - 316.54}{T_4 - 316.54}$ $\therefore T_4 = 495.64K$

Process 5-6s is 1st stage isentropic expansion $\therefore \frac{T_{6s}}{T_5} = \left(\frac{P_6}{P_5}\right)^{\frac{\gamma-1}{\gamma}}$

$$\therefore T_{6s} = 1373\left(\frac{1}{4}\right)^{0.286} = 923.59K$$

But $\eta_{t1} = \frac{T_5 - T_6}{T_5 - T_{6s}}$ *i.e.,* $0.86 = \frac{1373 - T_6}{1373 - 923.59}$ $\therefore T_6 = 986.51^0 K$

Process 6-7 is reheating, assume $T_7 = T_5 = 1373^0$K

Process 7-8s is 2nd stage isentropic expansion i.e., $\frac{T_{8s}}{T_7} = \left(\frac{P_8}{P_7}\right)^{\frac{\gamma-1}{\gamma}}$

$$\therefore T_{8s} = 1373\left(\frac{1}{3}\right)^{0.286} = 1002.79^0 K$$

But $\eta_{t2} = \frac{T_7 - T_8}{T_7 - T_{8s}}$ *i.e.,* $0.86 = \frac{1373 - T_8}{1373 - 1002.79}$ $\therefore T_8 = 1054.63^0 K$

Regenerator is used to utilizes the temperature of exhaust gases i.e., $\in = \frac{T_x - T_4}{T_8 - T_4}$

i.e., $0.78 = \frac{T_x - 495.64}{1054.63 - 495.64}$ $\therefore T_x = 931.65^0$K

We have, Compressor work: W_C $= C_{Pa}(T_2 - T_1) + C_{Pa}(T_4 - T_3)$
$= 1.005\ (430.73 - 288 + 495.64 - 316.54)$
$= 323.44$ kJ/kg

Also, Turbine work : W_T $= C_{Pg}(T_5 - T_6) + C_{Pg}(T_7 - T_8)$
$= 1.005\ (1373 - 986.51 + 1373 - 1054.63)$
$= 708.38$ kJ/kg

$\therefore$ Net work output, $W_N = W_T - W_C$
$= 384.95$ kJ/kg

But, power produced, $P = \dot{m}W_N$

i.e., $6000 \times 1000 = \dot{m}\ 384.95 \times 1000$

$\therefore \dot{m} = 15.59$ kg/sec

We have, heat supplied, $Q_s = C_{Pg} (T_5 - T_x) + C_{Pg} (T_7 - T_6)$

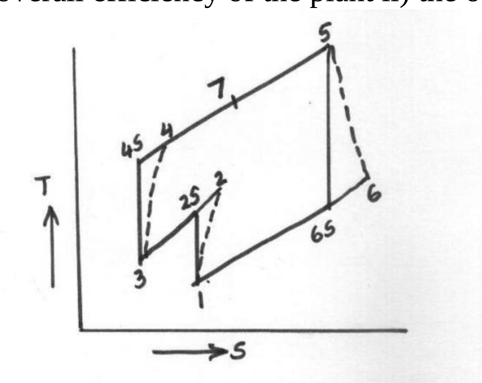
$= 1.005 (1373 - 931.65 + 1373 - 986.51)$

$= 831.98$ kJ/kg

$$\therefore \eta_{th} = \frac{W_N}{Q_H} = 0.463 \quad or \quad 46.3\%$$

Note: $C_{Pg}= C_{Pa}$=1.005kJ/kgK as C_{Pg} is not given and m_f *is neglected*

12 A gas turbine plant consists of two compressors with perfect inter cooling. The expansion occurs in a single turbine. The mass flow rate of air through the plant is 1 kg/min. Calorific value of the fuel used is 41870kJ/kg, maximum and minimum temperature in the cycle are 900°C and 27°C respectively. The working pressure limits are 1 bar and 6 bar. The compressors have isentropic efficiencies of 80% and the turbine efficiency is 85%. The pressure ratios for both the compressor stages are equal. The regenerator in the plant has effectiveness of 0.7. Determine i) overall efficiency of the plant ii) the output from the plant in kW iii) air fuel ratio iv) work ratio



A gas turbine plant consists of two compressors with perfect inter cooling. The expansion occurs in a single turbine

2 compressor and one turbine

W_c=$W_{c1} + W_{c2} = m_a c_{pa}(T_2 - T_1) + m_a c_{pa}(T_4 - T_3)$; $W_t = (m_a + m_f) c_{pg}(T_5 - T_6)$

with perfect inter cooling. *ie* T_3=T_2

m_f *assumed to be neglected* $c_{pg} = c_{pa} = 1.005 kJ/kgK$

The mass flow rate of air through the plant is 1 kg/min, m_a=1 kg/min.

Calorific value of the fuel used is 41870kJ/kg, CV=41870kJ/kg

maximum and minimum temperature in the cycle are 900°C and 27°C respectively.

T_1 =27°C=300K , T_5 =900°C= 1173 K,

The working pressure limits are 1 bar and 6 bar.

$P_1 = P_6$ =1 bar,$P_4= P_5$ =6 bar,

The compressors have isentropic efficiencies of 80% and the turbine efficiency is 85%.

$\eta_C = 0.8$, η_t=0.85 ,.

The pressure ratios for both the compressor stages are equal

$\frac{P_2}{P_1} = \frac{P_4}{P_3}$ ie $P_2 = P_3 = \sqrt{P_1 P_4}$

The regenerator in the plant has effectiveness of 0.7. ie $\varepsilon = \eta_{reg} = 0.7$

,

$P_2 = P_3 = \sqrt{P_1 P_4}$;$P_2 = \sqrt{1x6}$ =2.45
1-2s adiabatic process
$\frac{T_{2s}}{T_1} = \left\{\frac{P_{2s}}{P_1}\right\}^{\frac{\gamma-1}{\gamma}}$;$\frac{T_{2s}}{300} = \{2.45\}^{\frac{1.4-1}{1.4}}$; T_{2s} = 387.5K = T_{4s}
$$\eta_c = \frac{Ideal\ work}{actual\ work} = \frac{(T_{2s} - T_1)}{(T_2 - T_1)}$$
$0.8 = \frac{(387.5-300)}{(T_2-300)}$, T_2 = 409.37K= T_4 as compression ratio , initial temperature of compression, and efficiency for both compression are same
5-6s adiabatic compression

$\frac{T_5}{T_{6s}} = \left\{\frac{P_5}{P_{6s}}\right\}^{\frac{\gamma-1}{\gamma}}$;$\frac{1173}{T_{6s}} = \{6\}^{\frac{1.4-1}{1.4}}$, T_{6s} = 703K
$$\eta_t = \frac{actual\ work}{ideal\ work} = \frac{(T_5 - T_6)}{(T_5 - T_{6s})}$$
$0.85 = \frac{(1173-T_6)}{(1173-703)}$, T_6 = 773.5K
$$\epsilon = \frac{actual\ heat\ transfer}{maximum\ heat\ transfer} = \frac{(T_7 - T_4)}{(T_6 - T_4)}$$
$0.7 = \frac{(T_7-409.37)}{(773.5-409.37)}$; T_7 = 664.26K
$$m_f XCV = (m_a + m_f)C_{Pg}(T_5 - T_7)$$
$$m_f XCV = m_f\left(\frac{m_a}{m_f} + 1\right)C_{Pg}(T_5 - T_7);$$
$$CV = \left(\frac{m_a}{m_f} + 1\right)C_{Pg}(T_5 - T_7)$$
$$41870 = \left(\frac{m_a}{m_f} + 1\right)1.005(1173 - 664.26)$$

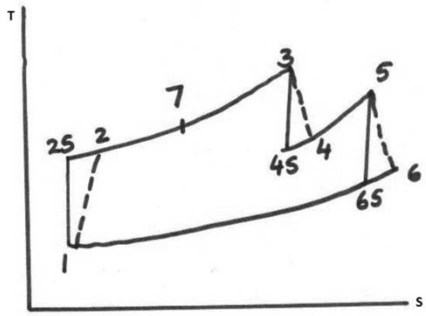
$\frac{ma}{mf} = 80.89$; $\frac{\frac{1}{60}}{m_f} = 80.89$
$m_f = 2.06x10^{-4}$kg/s

$$\eta = \frac{(m_a + m_f)C_{pg}(T_5 - T_6) - m_a C_a[(T_4 - T_3) + (T_2 - T_1)]}{(m_a + m_f)(T_5 - T_7)}$$
$$\eta = \frac{\left(\frac{1}{60} + 2.06x10^{-4}\right)1.005(1173 - 773.5) - \frac{1}{60}x1.005[(409.37 - 300) + (409.37 - 300)]}{\left(\frac{1}{60} + 2.06x10^{-4}\right)(1173 - 664.26)}$$
= 0.163 or 16.3%,
Out put: $W_T - W_c = (m_a + m_f)\ C_p(T_5 - T_6) - m_a[(T_4 - T_3) + (T_2 - T_1)]$
P=$\left(\frac{1}{60} + 2.06x10^{-4}\right)\ 1.005(1173 - 773.5) - \frac{1}{60}x1.005[(409.37 - 300) + (409.37 - 300)]$
P=3.03kW
$$W_T = (m_a + m_f)\ C_p(T_5 - T_6) = \left(\frac{1}{60} + 2.06x10^{-4}\right)\ 1.005(1173 - 773.5) = 6.69kW$$

Work ratio = $W_R = \frac{Net\ work\ output}{Turbine\ work} = \frac{W_T - W_C}{W_T}$

$work\ ratio = \frac{3.03}{6.69}$ = 0.4529

13. In a gas turbine plant air is compressed at 15°C and at 1 bar with a pressure ratio of 6. The air is heated in a heat exchanger with 75% efficiency and then in the combustion chamber to 750°C. The air at 750°C is expanded in two stages such that the expansion work is maximum. The air is reheated to 750°C after the first stage. Determine the cycle thermal efficiency, the work ratio and the net shaft work per kg. Take machine efficiencies as 80% and 85% for the compressor and turbine efficiency



In a gas turbine plant air is compressed at 15°C and at 1 bar with a pressure ratio of 6.

T_1=15°C=288K ; P_1 = 1 bar,

a pressure ratio of 6 ie $\frac{P_2}{P_1}$=6 bar,

The air is heated in a heat exchanger with 75% efficiency ie $\varepsilon = \eta_{reg} = 0.75$

in the combustion chamber to 750°C. Ie T_5=750°C=1023 K

The air at 750°C is expanded in two stages such that the expansion work is maximum.

ie $P_4 = \sqrt{P_3 P_6}$

The air is reheated to 750°C after the first stage T_5=750°C =1023 K,

Take machine efficiencies as 80% and 85% for the compressor and turbine efficiency

η_c=0.80 ; η_t=0.85

$P_4 = \sqrt{P_3 P_6}\ ie\ P_4 = \sqrt{1x6}$ =2.45

$\frac{T_{2s}}{T_1} = \left\{\frac{P_{2s}}{P_1}\right\}^{\frac{\gamma-1}{\gamma}}$; $\frac{T_{2s}}{288} = \{6\}^{\frac{1.4-1}{1.4}}$; T_{2s} = 480.5K

$\eta_c = \frac{Ideal\ work}{actual\ work} = \frac{(T_{2s}-T_1)}{(T_2-T_1)}$; $0.8 = \frac{(480.5-288)}{(T_2-288)}$, T_2 = 528.63K

3-4s adiabatic process

$\frac{T_3}{T_{4s}} = \left\{\frac{P_3}{P_{4s}}\right\}^{\frac{\gamma-1}{\gamma}}$; $\frac{1023}{T_{4s}} = \{2.45\}^{\frac{1.4-1}{1.4}}$, T_{4s} = 791.9K=T_{6s}

$\eta_t = \frac{actual\ work}{ideal\ work} = \frac{(T_3-T_4)}{(T_3-T_{4s})}$; $0.85 = \frac{(1023-T_4)}{(1023-791.9)}$, T_4= 826.56K

T_6= T_4= 826.56K as initial temperature, expansion ratio, and efficiency of both turbine are same

$$\epsilon = \frac{actual\ heat\ transfer}{maximum\ heat\ transfer} = \frac{(T_7 - T_2)}{(T_6 - T_2)}$$

$0.75 = \frac{(T_7 - 528.63)}{(826.56-528.63)}$; T_7 = 752.1K

$$\eta = \frac{\{(m_a + m_f)c_{pg}(T_3 - T_4) + (m_a + m_f)c_{pg}(T_5 - T_6)\} - m_a c_{pa}(T_2 - T_1)}{(m_a + m_f)c_{pg}(T_3 - T_7) + (m_a + m_f)c_{pg}(T_5 - T_4)}$$

$$\eta = \frac{\{(T_3 - T_4) + (T_5 - T_6)\} - (T_2 - T_1)}{(T_3 - T_7) + (T_5 - T_4)}$$

As m_f is neglected and $c_{pg} = c_{pa}$

$$\eta = \frac{2(1023 - 826.56) - (528.63 - 288)}{(1023 - 752.1) + (1023 - 826.56)}$$

= 0.326 or 32.6%,

Net work Out put: $2c_{pg}(T_3 - T_4) - c_{pa}[(T_2 - T_1)]$

Net work Out put: $2x1.005(1023 - 826.56) - 1.005[(528.63 - 288)]$

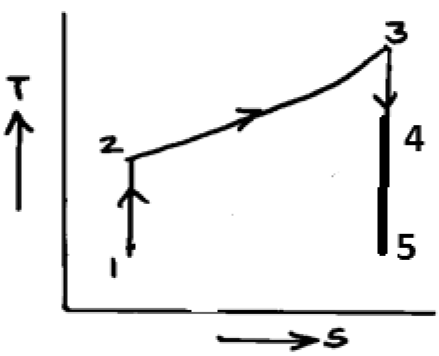
Output=152.25kj/kg

Turbine work $W_T = 2c_{pg}(T_3 - T_4) = 2x1.005x(1023 - 826.56)$=392.88kJ/kg

Work ratio = $W_R = \frac{Net\ work\ output}{Turbine\ work} = \frac{W_T - W_C}{W_T}$

$work\ ratio = \frac{152.25}{392.88}$ = 0.38

14. In a jet propulsion cycle air enters the compressor at 1 bar and 15^{0}C. The pressure leaving the compressor is 5 bar and the maximum temperature is 900^{0}C. The air expands in the turbine to such a pressure that the turbine work is just equal to the compressor work. On leaving the turbine, the air expands in a reversible adiabatic process in a nozzle to 1 bar. Calculate the velocity of air leaving the nozzle.(June 2012)



$$\frac{T_2}{T_1} = \left\{\frac{P_2}{P_1}\right\}^{\frac{\gamma-1}{\gamma}}$$

$$\frac{T_2}{288} = \{5\}^{\frac{1.4-1}{1.4}}$$

T_2 = 456.14K

$$W_c = W_t = C_p(T_2 - T_1) = 1.005(456.14 - 288) = 168.98kJ/kg$$

$$W_t = C_p(T_3 - T_4)$$

$$168.98 = 1.005(900 - T_4)$$

T_4=731.86^{0}C

$$\frac{P_3}{P_4} = \left\{\frac{T_3}{T_4}\right\}^{\frac{\gamma}{\gamma-1}}$$

$$\frac{5}{P_4} = \left\{\frac{1173}{1004.86}\right\}^{\frac{1.4}{1.4-1}}$$

P_4=2.9 bar

$$\frac{T_4}{T_5} = \left\{\frac{P_4}{P_5}\right\}^{\frac{\gamma-1}{\gamma}}$$

$$\frac{1004.86}{T_5} = \{2.9\}^{\frac{1.4-1}{1.4}}$$

T_5 = 741.3K

$$Velocity\ at\ exit = \sqrt{2C_p(T_4 - T_5)} = \sqrt{2x1.005(1004.86 - 741.3)} = 23.02 m/s$$

Combustion Thermodynamics

Fuels: Fuel is defined as the combustible matter which combustion with oxygen generates heat. Oxygen generally supplied by air.

Air mainly consists of O_2 and N_2 with composition 21% and 79% in volume respectively, and 23% and 77% in weight respectively.

Fuels mainly classified as i) Solid fuel ex: wood, Peat , coal ii) Liquid fuels: major source is Crude petroleum oil such as paraffins C_nH_{2n+2},Olifins C_nH_{2n+1},Nepthalenes C_nH_{2n} and the Aromatics C_nH_{2n-6} Gaseous fuel such as natural gas, Coal gas, water gas

Combustion: *Combustion is a chemical reaction between a fuel and oxygen which proceeds at a fast rate with the release of energy in the form of heat*

$CH_4+2O_2 \longrightarrow CO_2+2H_2O$

Reactants Products

Ie One mole of Methane reacts with 2 mole of oxygen to form of one mole of CO_2 and 2 mole of H_2O

Oxygen is supplied by air. Air mainly consists of O_2 and N_2 with composition 21% and 79% in volume respectively ie in Air for each mole of Oxygen there are 3.76 moles of Nitrogen. Therefore above reaction of methane with air may be written as follows

$CH_4+2O_2+2(3.76)\ N_2 \longrightarrow CO_2+2H_2O+2(3.76)\ N_2$

Reactants Products

Total mole of air is 2+3.76x2 =9.52

Theoretical (Stoichiometric) Air for complete combustion of fuel:

Stoichiometric (theoretical Air required) is the minimum amount of air which required for the complete combustion of all the elements like Carbon and Hydrogen etc.,

The minimum air fuel ratio required for the complete combustion of fuel is Stoichiometric air fuel ratio

$CH_4+2O_2+2(3.76)\ N_2 \longrightarrow CO_2+2H_2O+2(3.76)\ N_2$

Reactants Products

In the above reaction the Carbon in the CH_4 react with oxygen in the air and forms CO_2 and Hydrogen in the CH_4 reacts with Oxygen in the air and forms H_2O (ie No CO and no O_2 present in the products) and hence combustion is complete.

In the above reaction for complete combustion of CH_4 requires (2+2x3.76) moles of air

Hence Stoichiometric air fuel ratio on molal basis

7.52:1

Air fuel ration based on mass basis= $\frac{n_{O_2} M_{O_2} + n_{N_2} M_{N_2}}{n_{CH_4} M_{CH_4}} = \frac{2x28+7.52\ x28}{1x16}$=16.66:1

Excess Air: In practical case of combustion if minimum air is supplied, due to improper mixing of air and fuel and availability of less time for reaction combustion would not be complete. Hence, excess air is supplied for ensuring complete combustion of fuel .

Excess air ensures better combustion but at the time keeps the furnace cool

Excess Air = Air supplied – Stoichiometric Air

Percentage of excess Air = $\frac{\text{Air supplied} - \text{theortical air required}}{\text{theortical air required}}$ x100

For example if 50% excess air is supplied for combustion of methane

CH_4+**1.5**($2O_2$+2(3.76)) N_2 $\longrightarrow$ CO_2+$2H_2O$+(0.5x2) O_2 +1.5x(2(3.76)) N_2

Air fuel Ratio:

Mass Basis

It is defined as the ratio of mass of air required to the mass of fuel for any combustion process

Mole Basis: It is defined as the ratio of no of moles of air supplied to the no of moles of fuel

Enthalpy of formation:

Enthalpy of a compound is defined as the enthalpy referred to a standard atmospheric pressure of 1 bar and temperature of 25^{o}C at which the enthalpy of reactants which form the compound is considered to be zero

Consider the steady state of steady flow combustion of carbon and oxygen to form CO_2. Let the carbon and oxygen each enter control volume at 25^{o}C and 1 atm pressure and heat be such that the product leaves at 25^{o}C and 1 atm pressure. The measured value of heat transfer is -393,522kJ per kg mol of CO_2 formed.

If H_R and H_P refer to the total enthalpy of the reactants and products respectively, then the according to Ist law of thermodynamics applied to the reaction C+O_2 $\longrightarrow$ CO_2 gives

$H_R + Q_{C.V} = H_P$

The enthalpy of all the elements at the standard reference state of 25^{o}C and 1 atm is assigned the value of zero. Since C and O_2 are elements enthalpy of reactants is zero

Hence $Q_{c.v}$=-393,522kJ/kg mol

Hence enthalpy of formation of CO_2 is is -393,522kJ per kg mol

Enthalpy of Combustion: is defined as the difference between the enthalpy of the products and the enthalpy of the reactants when complete combustion occurs at a given temperature and pressure

$\overline{h_{rp}}$ = H_P- H_R

$\overline{h_{rp}} = \sum_p n_p \left[h_f + (h_T - h_{298})\right]_p - \sum_R n_R \left[h_f + (h_T - h_{298})\right]_R$

Where n_p= Number of moles of products

n_R= Number of moles of reactants

h_f=enthalpy of formation

h_T - h_{298} = enthalpy associated with its change in state from the standard state (15°C) to the combustion temperature T°C at 1 atmosphere pressure

Internal Energy of Combustion : The internal energy of combustion is defined as difference between the internal energy of the products and the internal energy of reactants after complete combustion at a given temperature and pressure

$\overline{U_{rp}}$ = U_P- U_R

$\overline{h_{rp}} = \sum_p n_p \left[h_f + (h_T - h_{298}) - p\bar{v}\right]_p - \sum_R n_R \left[h_f + (h_T - h_{298}) - p\bar{v}\right]_R$

Where n_p= Number of moles of products

n_R= Number of moles of reactants

h_f=enthalpy of formation

h_T - h_{298} = enthalpy associated with its change in state from the standard state (15°C) to the combustion temperature T°C at 1 atmosphere pressure

P is the pressure and $\bar{v}$ is the molal volume

Adiabatic Flame Temperature: During combustion in a combustion chamber if the chemical reaction goes to completion adiabatically, without work being performed by the system and without any change in kinetic and potential energies during a steady flow process, the temperature of products attained is called adiabatic flame temperature. Adiabatic flame temperature is maximum if the fuel air mixture is stoichiomeric. Adiabatic flame temperature can be controlled by supplying excess air

H_R= H_P

$\sum_R n_R \left[h_f + (h_T - h_{298})\right]_R = \sum_p n_p \left[h_f + (h_T - h_{298})\right]_p$

Higher heating value: is the heat transferred when H_2O in the products is in the liquid state.

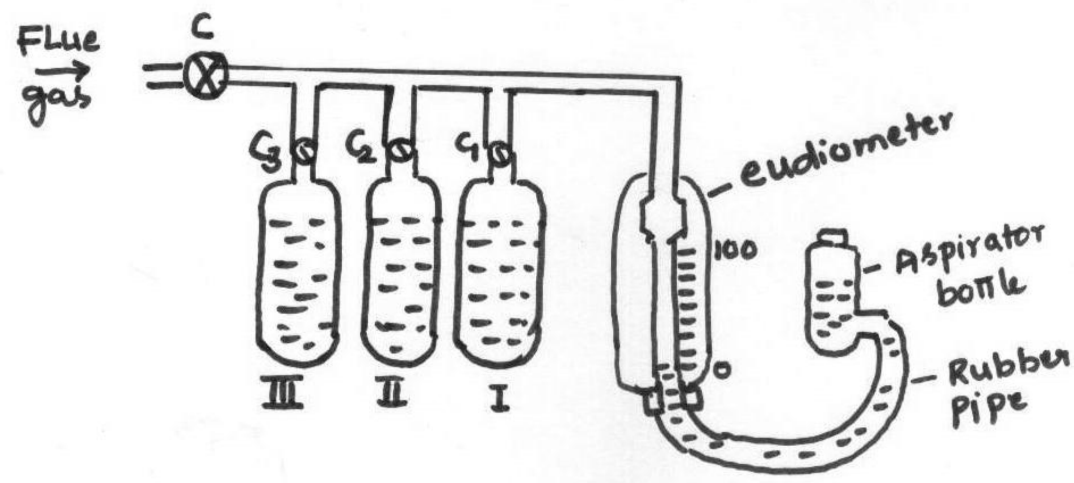
Lower heating value is the heat transferred in the reaction when H_2O in the vapour state

LHV=HHV-m_{H20}-h_{fg}

Wet and dry analysis of combustion: Combustion analysis when carried out considering water vapour into account is called “wet analysis” while the analysis made on the assumption that vapour is removed after condensing it, is called “dry analysis”.

Volumetric and gravimetric analysis: Combustion analysis when carried out based upon percentage by volume of constituent reactants and products is called volumetric analysis.

Orsat apparatus for volumetric analysis



Flask I is filled with NaOH or KOH solution (about one part of KOH and 2 parts of water by mass). This 33% KOH solution shall be capable of absorbing about fifteen to twenty times its own volume of CO_2. Flask II is filled with alkaline solution of pyrogallic acid and above KOH solution. Here 5 gm of pyrogallic acid powder is dissolved in 100 cc of KOH solution as in Flask I. It is capable of absorbing twice its own volume of O_2. Flask III is filled with a solution of cuprous chloride which can absorb CO equal to it’s’ volume. Cuprous chloride solution is obtained by mixing 5 mg of copper oxide in 100 cc of commercial HCl till it becomes colourless. Each flask has a valve over it and $C1$, $C2$, $C3$ valves are put over flasks I, II and III. The flue gas for analysis is taken by opening the main valve C, while valves $C1$, $C2$ and $C3$ are closed. 100 cc of flue gas may be taken into eudiometer tube by lowering aspirator bottle until the level is zero. Aspirator bottle is lifted now so as to inject flue gas into flask I with only valve $C1$ in open state where CO_2 present shall be absorbed. Aspirator bottle is again lowered and reading of eudiometer tube taken. Difference in readings of eudiometer tube initially and after CO_2 absorption shall give percentage of CO_2 by volume. Similar steps may be repeated for getting O_2 and CO percentage by volume.

1. Methane (CH_4) is burned with atmospheric air, The analysis of the products on a dry basis is as follows:

CO_2=10%, O_2=2.37%, CO=0.53% and N_2=87.10%. Calculate the air fuel ratio the percent of theoretical air and write down the combustion equation (June /July08)

Solution

$n\ CH_4+aO_2+3.76aN_2 \longrightarrow 10CO_2+2.37O_2+0.53CO+87.10N_2+dH_2O$

Carbon balance:

n=10+0.53=10.53

Nitrogen Balance:

3.76a =87.1

a=23.16

Oxygen Balance:

$a = 10+2.37+\frac{0.53}{2}+\frac{d}{2}$

$23.16=10+2.37+\frac{0.53}{2}+\frac{d}{2}$

d=21.05

Chemical Equation

$10.53\ CH_4+23.16O_2+3.76(23.16)N_2 \longrightarrow 10CO_2+2.37O_2+0.53CO+87.10N_2+21.05H_2O$

Mass of fuel= $n_{CH4}xM_{CH4}$=10.53x16=168.48kg

Mass of Air= $n_{O2}xM_{O2}$+ $n_{N2}xM_{N2}$=23.16x32+(3.76x23.16)x28=3179.40

Actual Air fuel ratio=$\frac{m_a}{m_f}=\frac{3179.40}{168.48}$=18.87:1

Stoichiometric Air fuel ratio

$CH_4+aO_2+(3.76xa)N_2 \longrightarrow bCO_2+cH_2O+(3.76xa)N_2$

Carbon balance
1=b
Hydrogen balance
4=2c ie c=2
Oxygen balance
2a=2b+c ie 2a=(2x1)+2 ; a=2
Hence the equation becomes
$CH_4+2O_2+(3.76x2)N_2 \longrightarrow CO_2+2H_2O+(3.76x2)N_2$

Mass of fuel= $n_{CH4} x M_{CH4}$=1x16=16kg

Mass of Air= $n_{O2} x M_{O2}$+ $n_{N2} x M_{N2}$=2x32+(3.76x2)x28=274.56kg

Theoretical Air fuel ratio=$\frac{m_a}{m_f}$ =$\frac{274.56}{16}$=17.56:1

Actual air fuel ratio in percentage of theoretical : $\frac{\text{Actual Air fuel ratio}}{\text{Theoretical Air fuel ratio=}} x100$=$\frac{18.87}{17.56} x100$=107.46%

2. Propane (C_3H_8) is burnt in atmospheric air and the mass analysis of the dry products of combustion is as follows:

CO_2---12.19% ; CO----1.23% ; O_2----7.57% and the balance N_2

Determine:

i) The volumetric analysis of the dry products

ii) Percent theoretical air

Solution

N_2=100-(12.19+1.23+7.57)=79.01

Sl NO	Products	mass (kgs) a	Mol wt b	Moles a/b	Percentage (Molar Basis)
1	CO_2	12.19	44	0.277	8.2
2	CO	1.23	28	0.044	1.3
3	O_2	7.57	32	0.237	7.01
4	N_2	79.01	28	2.822	83.5
	SUM (no of moles of products)			3.38	100

n C_3H_8+aO_2+3.76(a) N_2 $\longrightarrow$ 8.2CO_2+1.3CO+7.01O_2+83.5N_2+dH_2O

Carbon balance:

3n=8.2+1.3

n=3.17

Nitrogen Balance:

3.76a =83.5

a=22.21

Oxygen Balance:

2a =2x8.2+2x7.01+1.3 +d

44.42=16.4+14.02+1.3 +d

d=12.7

Chemical Equation

$3.17\ C_3H_8 + 22.21O_2 + 3.76(22.21)N_2 \longrightarrow 8.2CO_2 + 7.01O_2 + 1.3CO + 83.5N_2 + 12.7H_2O$

Mass of fuel= $n_{C3H8} x M_{C3H8}$=3.17x44=139.48kg

Mass of Air= $n_{O2} x M_{O2} + n_{N2} x M_{N2}$=22.21x32+(3.76x22.21)x28=3048.98

Air fuel ratio=$\frac{m_a}{m_f} = \frac{3048.98}{139.48}$=21.85:1

Stoichiometric Air fuel ratio

$C_3H_8 + aO_2 + 3.76(a)\ N_2 \longrightarrow bCO_2 + cH_2O + 3.76(a)\ N_2$

Carbon balance ; 3=b

Hydrogen Balance: 8=2c ; c=4

Oxygen Balance: 2a=2b+c

2a=(2x3)+4 ; a=5

Hence equation becomes

$C_3H_8 + 5O_2 + 3.76(5)\ N_2 \longrightarrow 3CO_2 + 4H_2O + 3.76(5)\ N_2$

Mass of fuel= $n_{C3H8} x M_{C3H8}$=1x44=44

Mass of Air= $n_{O2} x M_{O2} + n_{N2} x M_{N2}$=5x32+(3.76x5)x28=686.4

Theoretical Air fuel ratio=$\frac{m_a}{m_f} = \frac{686.4}{44}$=15.6:1

Actual air fuel ratio in percentage of theoretical : $\frac{m_{a\ actual}}{m_{atheoritical}} x100 = \frac{21.85}{15.6} x100$=140.06%

3. 4.4kg Propane gas is burnt completely with 3 kilomoles of air. Find the excess air and molar analysis of the dry combustion products (July 2004)

Propane gas chemical formula= C_3H_8

Stoichiometric Air fuel ratio

$C_3H_8 + aO_2 + 3.76(a)\ N_2 \longrightarrow bCO_2 + cH_2O + 3.76(a)\ N_2$

Carbon balance ; 3=b

Hydrogen Balance: 8=2c ; c=4

Oxygen Balance: 2a=2b+c

2a=(2x3)+4 ; a=5

Hence equation becomes

$C_3H_8+5O_2+3.76(5)\ N_2 \longrightarrow 3CO_2 + 4H_2O+3.76(5)\ N_2$

Mass of fuel= $n_{C3H8} x M_{C3H8}$=1x44=44

Mass of Air= $n_{O2} x M_{O2}$+ $n_{N2} x M_{N2}$=5x32+(3.76x5)x28=686.4

Theoretical Air fuel ratio=$\frac{m_a}{m_f}=\frac{686.4}{44}$=15.6:1

Actual Air fuel ratio

Total mass of Propane =4.4kgs

Mass of actual air supplied = no of moles of air supplied x Mol wt of air

= 3 x 28.97 =86.91

Actual Air fuel ratio=$\frac{m_a}{m_f}=\frac{86.91}{4}$=21.72:1

Excess Air per kg of fuel= Air supplied – Theoretical air required

=21.72-15.6 =6.12 kg of air per kg of fuel

For 4.4 kg of fuel =6.12x4.4 =24.48 kg of air

Chemical equation with excess air

Mol wt : (12x3)+(1x8)=44

No of moles = $\frac{m}{M}=\frac{4.4}{44}$ =0.1kmole

$0.1\ C_3H_8+3(O_2+3.76\ N_2) \longrightarrow aCO_2 + bH_2O+3.76(3)\ N_2+ c\ O_2$

Carbon balance

0.1x3 =a

Hydrogen balance

0.1x8 = 2 b

b=0.4

Oxygen balance

3 x2=2a + b +2 c

2x3=2x0.3+ 0.4 +2c

C=2.5

Nitrogen= 3.76x3 =11.28

$0.1\ C_3H_8+3(O_2+3.76\ N_2) \longrightarrow 0.3CO_2+0.4H_2O+3.76(3)\ N_2+\ 2.5\ O_2$

Molar analysis of dry combustion process

Sl NO	Products	Moles	Percentage (Molar Basis)
1	CO_2	0.3	2.13
2	O_2	2.5	17.76
3	N_2	11.28	80.11
Sum		14.08	100

Mass analysis of dry combustion process

Sl NO	Products	Moles a	Mol wt b	mass a x b	Percentage mass basis
1	CO_2	0.3	44	13.2	3.23
2	O_2	2.5	32	80	19.56
3	N_2	11.28	28	315.84	77.21
Sum				409.04	100

The products of combustion of an unknown hydrocarbon C_xH_y have the following composition as measured by an orsat apparatus

CO_2=8%, O_2=8.8%, CO=0.9% and N_2=82.3%.

Determine:

i) the composition of the fuel
ii) The air fuel ratio
iii) The percent of excess air used

Solution:

$C_xH_y+aO_2+3.76(a)\ N_2 \longrightarrow 8CO_2+0.9CO+8.8O_2+82.3\ N_2+dH_2O$

Carbon balance:

x=8+0.9=8.9

Nitrogen Balance:

3.76a =82.3

a=21.89

Oxygen Balance:

2a $=2x8$+0.9$+2x8.8$ $+d$

2x21.89$=2x8$+0.9$+2x8.8$ $+d$

d=9.28

Hydrogen Balance

y=2d

=2x9.28=18.56

Chemical composition of Fuel

$C_{8.9}H_{18.56}$

Chemical Equation

$C_{8.9}H_{18.56}+21.89O_2+3.76(21.89)N_2 \longrightarrow 8CO_2+0.9CO+8.8O_2+82.3\ N_2+9.29H_2O$

Mass of fuel= n_fxM_f=1(12x8.9+18.56x1)=125.36

Mass of Air= n_{O2}xM_{O2}+ n_{N2}xM_{N2}=21.89x32+(3.76x21.89)x28=3005.06

Air fuel ratio=$\frac{m_a}{m_f}$ =$\frac{3005.06}{125.36}$=23.97:1

Stoichiometric Air fuel ratio

$C_{8.9}H_{18.56}+a\ O_2+3.76\ aN_2 \longrightarrow bCO_2+cH_2O+3.76a\ N_2$

Carbon balance:

8.9=b

Hydrogen balance

18.56 =2c ie c=9.28

Oxygen balance

2a = 2xb+ c

2a=2x8.9+9.28 ; a=13.54

Hence

$C_{8.9}H_{18.56}+13.54\ O_2+(3.76 \times 13.54)N_2 \longrightarrow 8.9CO_2+9.28\ H_2O+(3.76 \times 13.54)N_2$

Mass of fuel= $n_{C3H8} x M_{C3H8}$=1(12x8.9+18.56x1)=125.36 kgs

Mass of Air= $n_{O2} x M_{O2}$+ $n_{N2} x M_{N2}$=13.54x32+(3.76 x13.54)x28=1858.77kgs

Theoretical Air fuel ratio=$\frac{m_a}{m_f}$ =$\frac{1858.77}{125.36}$=14.82:1

Actual air fuel ratio in percentage of theoretical : $\frac{m_{a\ actual}}{m_{atheoritical}} x100$=$\frac{23.97}{19.9} x100$=161.74%

1. Coal with following mass analysis is burnt with 100% excess air. C=74%, H_2=4.3%, S=2.7%, N_2=1.5%,H_2O=5.5% , O_2=5% Ash= 7%. Find moles of gaseous products if 100kg of fuel is burnt

$C+O_2 \longrightarrow CO_2$

12kg of C burnt with 32kg of O_2 to form 44 kg of CO_2

C------------O_2

12--------- 32

74-----------?

$\frac{32}{12}$x74=197.33kg of O_2 (required to burn 74 kg of C)

12kg of C burnt with 32kg of O_2 to form 44 kg of CO_{2+}

12kg of C produces 44kg of CO_2

$\frac{44}{12}$x74=271.33kg of CO_2 (74 kg of C produces 273.33kg of CO_2)

$H_2+\frac{1}{2} O_2 \longrightarrow H_2O$

2kg of H_2 burnt with 16kg of O_2 to form 18 kg of H_2O

Hence, 4.3 kg of H_2 requires $\frac{16}{2}$x4.3=34.4kg of O_2

2kg of H_2 burnt with of O_2 to form 18 kg of H_2O

Hence 4.3kg of H_2 forms $\frac{18}{2}$x4.3=38.7kg ofH_2O

$S+O_2 \longrightarrow SO_{2+}$

32kg of S burnt with 32kg of O_2 to form 64 kg of SO_2

Hence, , 2.7kg of S requires $\frac{32}{32}$x2.7=2.7kg of O_2

32kg of S burnt with O_2 to form 64 kg of SO_2

Hence,2.7kg of S produces $\frac{64}{32}$ x2.7=5.4kg of SO_2

Hence O_2 required =197.33(for C)+34.4(for H_2)+ 2.7(for S) – 5(O_2 present in the 100kg of fuel)=229.43kg of O_2

But 100% of excess air is supplied, Hence O_2 is supplied (1+1) x 229.43=458.86kg of O_2

In air every 1kg of O_2 there is $\frac{77}{23}$ kg of N_2

Hence Air supplied consists of $\frac{77}{23}$ x458.86=1536.18 kg of N_2

For 100kg of Fuel

Sl N0	Products	mass (kgs)	Mol wt	Moles	Percentage (Molar Basis)
1	CO_2	271.33	44	6.17	8.70
2	H_2O	38.7+5.5(fuel)=44.2	18	2.46	3.47
3	SO_2	5.7	32	0.178	0.25
4	N_2	1536.18(supplied Air)+1.5(fuel)=1537.68	28	54.91	77.46
5	O_2	458.86(Supplied)-229.43(required)=229.43	32	7.17	10.11
	SUM (no of moles of products)			70.89	100

2. A sample of fuel has the following percentage composition by weight: Carbon 84%; Oxygen =3.5%; Hydrogen=10%; Ash=1%; Nitrogen =1.5%

Determine the stoichiometric Air fuel ratio by mass

If 20% excess air is supplied, find the percentage composition of dry flue gases by volume(July2007)

Solution

Oxygen required for complete combustion of fuel

$$H_2+\frac{1}{2}O_2 \longrightarrow H_2O$$

2kg of H_2 burnt with 16kg of O_2 to form 18 kg of H_2O

Hence, 0.10 kg of H_2 requires $\frac{16}{2}$ x 0.10=0.8 kg of O_2

$$C+O_2 \longrightarrow CO_2$$

0.84 kg of Carbon require $\frac{32}{12}$ x0.84 =2.24 kg of Oxygen

Hence total Oxygen required for complete combustion of 0.84 kg of C and 0.10 kg of H_2 =2.24+0.8=3.04 kg of O_2

Since fuel contains 0.035 kg of O_2 Oxygen to be supplied through air = 3.04-0.035=3.005kg of Oxygen

Hence air required $\frac{100}{23}$ x 3.005=13.065kg of air required per kg of fuel

Hence theoretical air fuel ratio required is 13.065:1

ii) If 20% excess air is supplied dry flue gases analysis

$C+O_2 \longrightarrow CO_2$

0.84 kg of C produces $\frac{44}{12}$ x0.84 =3.08kgs of CO_2

Air supplied is 20 percent excess Hence, mass of air supplied is =13.065x1.2=15.678kgs

mass of N_2 present in 17.484 kgs of air =$\frac{77}{100}$ x 15.678=12.072kg

mass of O_2 present in the product due to excess supply of air

=$\frac{23}{100}$ x (15.678)-3.005(required for combustion)=0.6009kgs

Dry analysis of the products

Sl N0	Products	mass (kgs) a	Mol wt b	Moles a/b	Percentage (Molar Basis)
1	CO_2	3.08	44	0.07	13.47
2	N_2	12.072	28	0.4311	82.94
4	O_2	0.6009	32	0.0187	3.597
	SUM (no of moles of products)			0.5198	100

A hydrocarbon fuel $C_{12}H_{26}$ is burnt with 50% excess air. Determine the volumetric (Molal) analysis of the products of combustion and also the dew point temperature of the products if pressure is 101kpa

Stoichiometric chemical equation

$C_{12}H_{26} + 18.5O_2 + 3.76(18.5)\, N_2 \longrightarrow 12CO_2 + 13H_2O + 3.76(18.5)\, N_2$

With 50% excess air

$C_{12}H_{26} + 1.5[18.5O_2 + 3.76(18.5)\, N_2] \longrightarrow 12CO_2 + 13H_2O + 0.5(18.5)O_2 + 1.5[3.76(18.5)]\, N_2$

Sl N0	Products	Moles	Percentage (Molar Basis)
1	CO_2	12	9.555
2	O_2	9.25	7.365
3	N_2	104.34	83.079
Sum		125.59	100

No of moles of water =13

Molal fraction of H_2O= $\frac{n_{H2O}}{n}$ =$\frac{13}{125.59}$ =0.1035

Hence partial pressure of H_2O= Molal fraction x total pressure =1.01x0.1035 =0.10455 bar

Dew point temperature of products is the saturation temperature of water at 0.10455 bar ie

The fuel used in petrol engine contains 87% Carbon and 13% H_2. The air supply is 75% of that theoretically required for complete combustion. Assuming H_2 is burned and there is no free carbon left, find the volumetric analysis of dry exhaust gases (Dec07/Jan08)

<u>Oxygen required for complete combustion of fuel</u>

The fuel used in petrol engine contains 87% Carbon and 13% H_2.

O_2 required for combustion of H_2 present in 1 kg of fuel

$$H_2+\frac{1}{2}O_2 \longrightarrow H_2O$$

2kg of H_2 burnt with 16kg of O_2 to form 18 kg of H_2O

Hence, 0.13 kg of H_2 requires $\frac{16}{2}$ x 0.13=1.04 kg of O_2

O_2 required for combustion of C present in 1 kg of fuel

$$C+O_2 \longrightarrow CO_2$$

0.87 kg of Carbon require $\frac{32}{12}$x0.87 =2.32 kg of Oxygen

Hence total Oxygen required for complete combustion of 0.87 kg of C and 0.13 kg of H_2 =2.32+1.04=3.36 kg of O_2

But actual air supplied is 75% of theoretical air, hence oxygen supplied = 0.75x3.36=2.52 kg of Oxygen

According to data H_2 completely burns. Hence, Oxygen available for combustion of C is 2.52-1.04=1.48kg less than required for complete combustion

Hence carbon due to incomplete combustion produces CO and CO_2

Let x kg of C produces CO and y kgs of C produces CO_2

Also carbon in 1 kg of fuel =0.87 (Data)

Hence x +y =0.87-----------------1

<u>Oxygen required for combustion of x kgs of carbon to CO</u>

$C + \frac{1}{2} O_2 \longrightarrow CO$

12kg of C burns with 16 kg of O_2 to form 28kg of CO

Hence O_2 required for combustion of x kgs of C to form CO $= \frac{16}{12}\ x$

<u>Oxygen required for combustion of y kgs of carbon to CO_2</u>

$C + O_2 \longrightarrow CO_2$

12kg of C burns with 32 kg of O_2 to form 44kg of CO

Hence O_2 required for combustion of y kgs of C to form $CO_2 = \frac{32}{12}\ y$

<u>Hence, Oxygen Utilised for combustion of C to CO and CO_2</u>

$\frac{16}{12}\ x + \frac{32}{12}\ y$

Utilised =available

$\frac{16}{12}\ x + \frac{32}{12}\ y = 1.48$----------------2

x+y =0.87---------------1

hence y=0.87−X

$\frac{16}{12}\ x + \frac{32}{12}(0.87 - X) = 1.48$

$\frac{4}{3}\ X + \frac{8}{3}(0.87 - X) = 1.48$

X=0.63

Hence 0.63+y =0.87

Y=0.24

<u>Product analysis</u>

$C + \frac{1}{2} O_2 \longrightarrow CO$

12kg of C burns with 32 kg of O_2 to form 44kg of CO

And x=0.63 kgs of C produces= $\frac{28}{12}$ x0.63 =1.47 kg of CO

$C + O_2 \longrightarrow CO_2$

12kg of C burns with 32 kg of O_2 to form 44kg of CO

And y =0.24 kg of C produces $\frac{44}{12}$(0.24)=0.88kgs of CO_2

O_2 is supplied through air

1kg of air consists of 77kg of N_2 and 23 kg of O_2 and N_2 will remain as it is in the product

O_2 N_2

23 77

2.52 ?

Hence 2.52 kg of O_2 supplies $\frac{77}{23}$(2.52)=8.43 kg of N_2 remains as it is in the product

Hence the dry product consists of CO, CO_2 and N_2

Note that since analysis is dry analysis H_2O in the product should not to be considered

Dry analysis of the products

Sl NO	Products	mass (kgs) a	Mol wt b	Moles a/b	Percentage (Molar Basis)
1	CO_2	0.88	44	0.02	5.35
2	CO	1.47	28	0.0525	14.04
4	N_2	8.44	28	0.3014	80.61
	SUM (no of moles of products)			0.3739	100

A sample of fuel has the following percentage composition: Carbon=86%, Hydrogen =8%, Sulphur=3%, Oxygen =2% Ash=1%

For an air fuel ratio of 12:1 calculate

i) Mixture strength as a percentage of rich or weak

ii) Volumetric analysis of the dry product of combustion Dec06/Jan07

Solution

$C+O_2 \longrightarrow CO_2$

0.86 kg of Carbon requires $\frac{32}{12}$x0.86 =2.293 kg of Oxygen

0.86 kg of C produces $\frac{44}{12}$x0.86 =3.153 kgs of CO_2

$H_2+\frac{1}{2}O_2 \longrightarrow H_2O$

0.08 kg of H_2 requires $\frac{16}{2}$ x 0.08=0.64 kg of O_2

0.08 kg of H_2 produces $\frac{18}{2}$x0.08=0.72kgs of H_2O

$S+O_2 \longrightarrow SO_2$

Hence, 0.03kg of S requires $\frac{32}{32}$x0.03=0.03kg of O_2

0.03kg of S produces $\frac{64}{32}$x0.03=0.06kg of O_2

Total Oxygen require for complete combustion of fuel =2.293+0.64+0.03-0.02(Fuel)=2.943kgs

Air required for complete combustion = $\frac{100}{23}$ x 2.943=12.79(Theoretical)

But supplied is 12:1 It implies that air supplied is less than theoretical Hence supplied air fuel ratio is rich

Mixture strength = $\frac{1}{air\ fuel\ ratio}$

Mixture strength as a percentage of rich = $\frac{theoritical\ air\ fuel\ ratio}{actual\ air\ fuel\ ratio} = \frac{12.79}{12}x100$=106.58

Oxygen supplied =$\frac{23}{100}$x12 =2.76kg of oxygen

Generally H_2 completely burns. Hence, Oxygen available for combustion of C is 2.76-0.64=2.12kg less than required for complete combustion

Hence carbon due to incomplete combustion produces CO and CO_2

Let x kg of C produces CO and y kgs of C produces CO_2

Oxygen required for combustion of x kgs of carbon to CO

$$C+\frac{1}{2}O_2 \longrightarrow CO$$

12kg of C burns with 16 kg of O_2 to form 28kg of CO

Hence O_2 required for combustion of x kgs of C = $\frac{16}{12}$ x

And x kgs of C produces= $\frac{28}{12}$ x of CO

Oxygen required for combustion of y kgs of carbon to CO_2

$$C+O_2 \longrightarrow CO_2$$

12kg of C burns with 32 kg of O_2 to form 44kg of CO

Hence O_2 required for combustion of y kgs of C = $\frac{32}{12}$ y

And x kgs of C produces= $\frac{44}{12}$ y of CO

Hence, $\frac{16}{12}x + \frac{32}{12}y = 2.12$

Also x+y =0.86

Solving above two equations x=0.1260 kg and y=0.7339

Products

CO $=\frac{28}{12}x = \frac{28}{12}(0.1260) = 0.294$kgs of CO

$CO_2 = \frac{44}{12}y = \frac{44}{12}(0.7339) = 2.69$kgs of CO_2

N_2 supplied from air $=\frac{77}{100}$ x12=9.24kg of N_2

Note that since analysis is dry analysis H_2O in the product should not to be considered

Dry analysis of the products

Sl N0	Products	mass (kgs) a	Mol wt b	Moles a/b	Percentage (Molar Basis)
1	CO_2	2.69	44	0.0611	15.21
2	CO	0.294	28	0.0105	2.614

4	N_2	9.24	28	0.33	82.17
	SUM (no of moles of products)			0.4016	100

1. What are the performance parameters used in IC engine testing

   Engine performance is an indication of the degree of success with which it does its assigned job i.e. conversion of chemical energy contained in the fuel into the useful mechanical work.

1. Mean effective pressure and torque
2. Power and mechanical efficiency
3. Specific output
4. Volumetric efficiency, Relative efficiency and air standard efficiency
5. Fuel-Air ratio
6. Brake Specific fuel consumption
7. Thermal efficiency
8. Heat balance sheet
9. Emissions from engines

2.List the formulae used in IC engine testing

IP=$\frac{p_m\left(\frac{kN}{m^2}\right)L(m)A(m^2)n(per\ min)}{60}xk \quad kW$, A=$\frac{\pi d^2}{4}$, d= diameter of bore in m, L=stroke length in m

$p_m = Mean\ Effictive\ pressure\ in\ \frac{KN}{m^2}$ or kPa

Theoretically n=$\frac{N}{2} for\ 4\ stroke, \quad n = N\ for\ 2\ stroke$

If no of explosion per minute is given in the problem then n= no of explosion per minute(ie we should not use n=$\frac{N}{2} for\ 4\ stroke, \quad n = N\ for\ 2\ stroke$)

k=number of cylinder

BP=$\frac{p_{mb}\ LAn}{60}xk \quad kW, all\ factors\ are\ same\ as\ in\ above\ IP\ formula$ in terms of mean effective pressure

T=$\frac{(w-s)(D_b+d_r)}{2}Nm, D_b = diamter\ of\ Brake\ drum, d_r = diameter\ of\ rope, w = weight\ in\ Newtons, S\ is\ spring\ balance\ reading\ in\ Newtons$

BP: $\frac{2\pi N(per\ min)T}{60000}$ if T is in Nm (BP in terms of Torque

BP: $\frac{2\pi NT}{60}$ if T is in kNm

N=speed in rpm for all cases

If N is given in rps then there is no 60 in the denominator of above formula Note down that 60 is used to convert minute to sec

Friction Power $=BP - IP$

Note: Friction Power is equal to the motor Power if the Engine is run by the motor without combustion of fuel

$$\eta_{mech} = \frac{BP}{IP} x100$$

η Indicated thermal efficiency$=\frac{IP}{mf X\ C.V}, mf\ is\ in\ kg/s$ and C.V is in calorific value of fuel in kJ/kg

η Brake thermal efficiency$=\frac{BP}{mf X\ C.V}, mf\ is\ in\ kg/s$

η Relative efficiency$=\frac{\eta \text{ Indicated thermal efficiency}}{\eta \text{ air standard efficiency}}$

$$\eta \text{ air standard efficiency} = 1 - \frac{1}{CR^{\gamma-1}}, for\ Otto\ cycle$$

$$\eta_v = \frac{V_{act}}{V_{the}}$$

$V_{the}=\frac{\pi d^2}{4} xlx \left(\frac{N}{2x60}\right)$ for 4 stroke

In the above formula $\frac{N}{2}$ is used because no of suction strokes per minute in 4 stroke engine is $\frac{N}{2}$ since one thermodynamic cycle is completed in 2 revolutions

$V_{the}=\frac{\pi d^2}{4} xlx \frac{N}{60}$ for 2 stroke

In the above formula only is used because no of suction strokes per minute in 4 stroke engine is N since one thermodynamic cycle is completed in 1 revolution

$$V_{act} = AC_d\sqrt{2}gh_a$$

$$h_a = \frac{h_w \rho_w}{\rho_a}$$

$$SFC = \frac{m_f}{BP} \frac{kg}{kWhr}, mf\ is\ in\ kg/hr$$

Heat Balance sheet:

1. Heat input : m_f X CV kW
2. BP = in kW
3. Heat carried by cooling water: $m_w C_{pw} (T_{co} - T_{ci})$ kW where m_w in **kg/s, C_{pw} =4.187kJ/kgK specific heat of water,** T_{co} =temperature of outlet water in ^oC ,

T_{ci} = temperature of inlet water in ^oC, ($T_{co} – T_{ci}$)=raise in temperature of cooling water in K or ^oC

4. Heat carried by exhaust gases : ($m_f + m_a$) C_{pg} ($T_{ex} – T_{Room}$) or
   : m_f ((m_a/m_f)+1) C_{pg} ($T_{ex} – T_{room}$)

T_{ex} is exhaust temperature and T_{room}= Room temperature

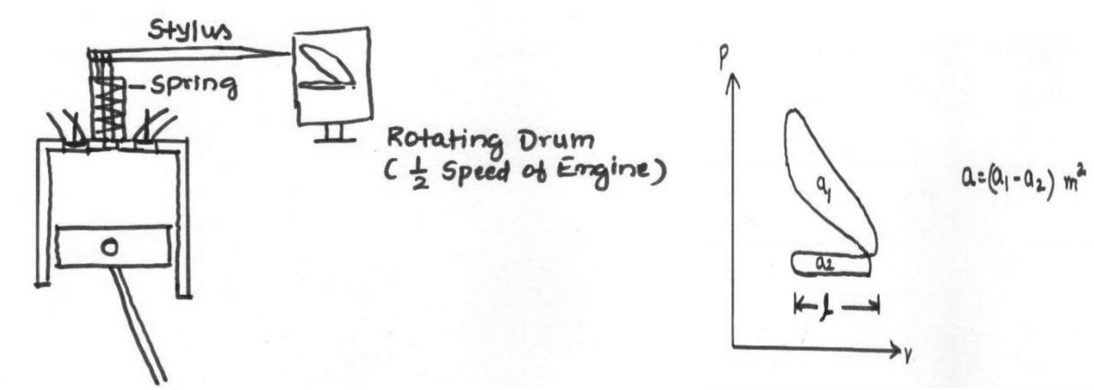
Where m_f is mass of fuel in kg/s, m_a mass of air kg/s m_a/m_f = air fuel ratio,

C_{pg}=specific heat of gas in kJ/kgK **If it is not given in the problem take as specific heat of air =1.005kJ/kgK**

5. Heat un account for : 1 – (2+3+4)

3. What are the measurement methods for Indicated power, explain them

1.Indicator



An indicator is fixed on the top of head which has a compression spring vibrates according to variation of pressure inside the cylinder, the same is recorded by a stylus on a paper attached to a rotating drum so that for each cycle a P-V diagram as shown above is obtained. The area of the diagram is measured by a planimeter in laboratory. The Mean effective pressure is calculated using the following formula.

$P_m = \frac{axl}{s}\ kPa$where

P_m = Indicated mean effective pressure in kPa,

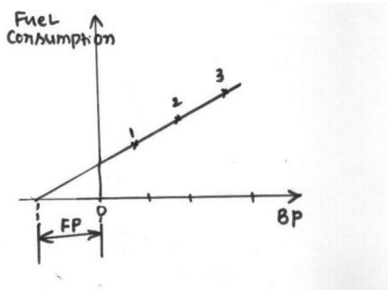
a = Area of indicator diagram, m^2,

l = Length of indicator diagram in, m,

s=spring constant in mm/mm/Kpa.

2. Williams line method:

This method is only used for Diesel engine, as in petrol engine a mixture of petrol and air enters into cylinder where as in diesel engine only diesel is injected into cylinder. The graph of fuel consumption versus Brake power is drawn, A tangential line to this line is extrapolated back to meet x axis on the back side which gives the friction power as shown in figure. Then the IP is calculated from the formula. The speed is maintained constant throughout the experiment so as to have the constant friction power.



$$IP = FP + BP$$

3.Motoring test:

In this test the engine is first run up to the desired speed by its own power and allowed to remain under the given speed and load conditions for some time so that oil, water and engine component temperature reach stable conditions. The power of the engine during this period is absorbed by a dynamometer. The fuel supply is then cut-off and by suitable electric switching devices the dynamometer is converted to run as motor to drive or 'motor' the engine at the same speed at which it was previously running. The power supply to the motor is measured which is a measure of F.P. of the engine.

$$IP = FP + BP$$

FP=power supplied by electric motor.

4.Morse test

This method is used for finding of IP in multi cylinder engines

This test is applicable only to multi-cylinder engines. The engine is run at the required speed and the torque is measured. One cylinder is cut-off by shorting the plug if an S.I. engine is under test, or by disconnecting an injector if a C.I. engine is under test. The speed falls because of the loss of power with one cylinder cut-off, but is restored by reducing the load. The torque is measured again when the speed has reached its original value.

For a 4 cylinder engine IP is calculated as follows

$$IP_1 = BP_{1234} - BP_{234}$$

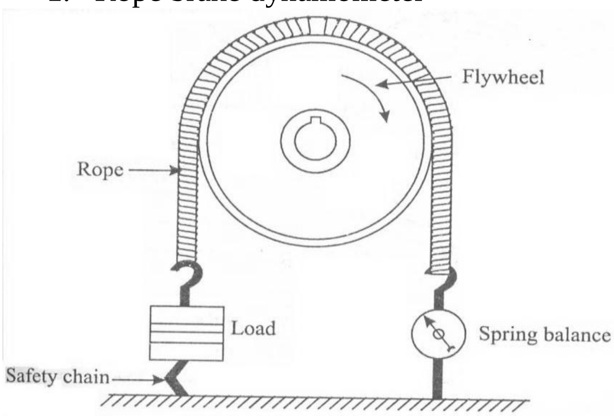
$$IP_2 = BP_{1234} - BP_{134}$$

$$IP_3 = BP_{1234} - BP_{124}$$

$$IP_4 = BP_{1234} - BP_{123}$$

$$IP_{total} = IP_1 + IP_2 + IP_3 + IP_4$$

4. What are the measurement methods for Brake power, explain them
1. Rope brake dynamometer



In rope brake dynamometer a rope is wound round the circumference of the brake wheel. To one end of the rope is attached a spring balance (S) and the other end carries the load (W). The speed of the engine is noted from the tachometer.

Let W = Weight at the end of the rope, in N,

S = Spring balance reading, in N,

If W or S is given in kgs convert into Newton by multiplying kgs by 9.81

N=kgsx9.81

N = Engine speed, in r.p.m.,

D_b = Diameter of the brake wheel, in m,

d_r = diameter of the rope, in m, and

$(D_b + d_r)$ = Effective diameter of the brake wheel,

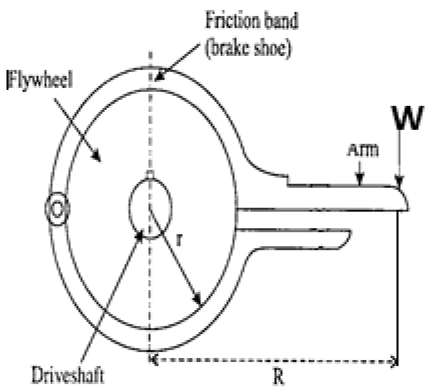
T=$\frac{(w-s)(D_b+d_r)}{2} Nm$

BP: $\frac{2\pi NT}{60x1000}\ kW$ where T is in Nm

Note down that in the above formula in the denominator 1000 is used to convert Nm to KNm and 60 is used to convert min to sec

BP: $\frac{2\pi NT}{60}$ kW if Torque is in KNm

2. Pony brake dynamometer



This arrangement consists of a pair of brake shoes (usually made of wood) which are held in place by means of spring loaded bolts. The pressure on the rim of the brake drum (wheel) is adjusted with the help of the nuts.Weights are placed on the load carrier at the end of the load bar (lever). Increasing the weights will increase the pressure of the brake shoes on the drum, which results in an increase in the frictional torque that the engine has to overcome.

T=$WxR\ \ Nm$ where $W\ is\ in\ Newton$ note that if W is given in kgs convert into Newton

ie Newton =9.81xkgs

BP: $\frac{2\pi NT}{60x1000}$ kW

Note down that in the above formula in the denominator 1000 is to convert Nm to KNm and 60 is used to convert min to sec

5. Write a note measurement of fuel consumption

The mass rate of fuel consumption is measured by noting the time taken for the consumption of a known volume of fuel. If V cc of fuel is consumed in 't' seconds, then mass rate of fuel consumption is

$$m_f = \frac{Vxspecific\ gravity\ of\ fuelx3600}{1000xt}$$

6. Write a note on air consumption measurement in IC engine

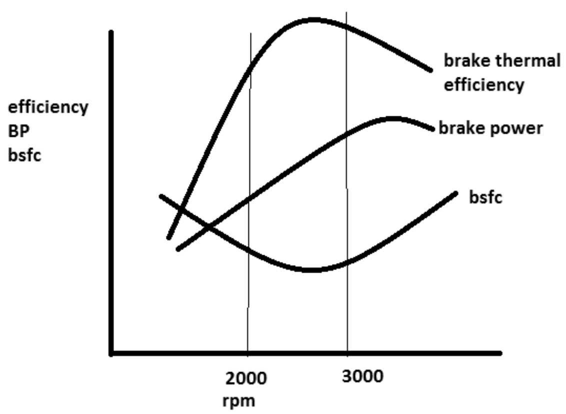
The air supplied to engine is measured by air box method, in the set up an empty box of capacity 500 times the engine capacity is fitted to engine suction on one side and open to atmosphere through an orifice on opposite side. A U-tube water manometer is connected and as engine is started air flows through the box to the engine and the reading of manometer is recorded. The volume flow rate of air is calculated by the following formula

$$h_a = \frac{h_w \rho_w}{\rho_a}$$

$$V_{act} = AC_d\sqrt{2gh_a}$$

Where A is area of orifice, C_d=0.62, h_w=reading in manometer.

7. Write a note on characteristic curves in IC engines



From figure maximum efficiency, maximum power and minimum fuel consumption can be obtained if we run the engine between 2000 and 3000 rpm hence it is always better to run the engine in the speed range of 40 to 60KMPH to get optimum performance.

NUMERICALS :

1. A 4 cylinder engine has the following data: Bore=15 cm; Stroke=15cm; Piston speed =510m/min; B.P= 60kW; η_{mech}=0.80; p_m=5 bar; C.V=40000kJ/kg. Calculate i) Whether this is 2 Stroke or 4 Stroke engine?

   Solution

   $\eta_{mech}=\frac{B.P}{I.P}$; $I.P. = \frac{B.P}{\eta_{mech}}$; $I.P. = \frac{60\ (kW)}{0.8}$; $I.P = 75kW$

   IP=$\frac{p_m LAn}{60}k$

   $75(kW) = \frac{5x10^2(kPa)x0.15x\left(\frac{\pi 0.15^2}{4}\right)xn}{60}x4;\quad n = 848.826\ explosions\ /min$

   Piston Speed(m/min) $= 2LN$; $\quad 510 = 2x0.15xN$; N=1700

   $n = xN$ if $x = \frac{1}{2}$ engine is 4 stroke and if $x = 1$ then engine is 2 stroke

   $$849 = x\ 1700$$

   Ie $x = \frac{849}{1700} \cong \frac{1}{2}$

Hence engine is 4 stroke

2. A 4 cylinder has a following data Bore=15cm; Piston speed =510m/min; B.P.=60kW; η_{mech}=0.80; p_m=5 bar; C.V=40000kJ/kg. Calculate i) Whether this is 2 Stroke or 4 Stroke engine?
3. Following data is available for a S.I engine, single cylinder, 4 stroke, air fuel ratio 16:1 CV=45000kJ/kg, η_{mech}=0.80; air standard efficiency = 0.5; relative efficiency =0.7; Stroke Bore ratio=1.5; Suction conditions; 1 bar and 30°C, speed =2500 rpm, B.P=75kW; Calculate the compression ratio, indicated thermal efficiency, brake specific fuel consumption(BSFC); brake R

$$\eta_{air\ standard} = 1 - \frac{1}{R_c^{\gamma-1}}$$

$$0.5 = 1 - \frac{1}{R_c^{\gamma-1}}$$

$\frac{1}{R_c^{\gamma-1}} = 0.5;\ \frac{1}{0.5} = R_c^{\gamma-1};\ \ 2^{\frac{1}{0.4}} = R_c\ ;\ R_c = 5.6568$

$\eta_{mech}=\frac{B.P}{I.P}$

$I.P. = \frac{B.P}{\eta_{mech}};\ I.P. = \frac{75\ (kW)}{0.8};\ \ I.P = 93.75kW$

4 Stroke engine $n = \frac{N}{2}$ (theoretical) since number of explosions is not given

$n = \frac{2500}{2} = 1250$ explosions per minute

$$Relative\ efficiency = \frac{Indicated\ thermal\ efficiency}{AIr\ Std\ efficiency}$$

$$0.7 = \frac{Indicated\ thermal\ efficiency}{0.5}$$

$$Indicated\ thermal\ efficiency = 0.35$$

$$BSFC = \frac{m_f(kg/hr)}{BP}$$

$$Indicated\ thermal\ efficiency = \frac{IP}{Q_s} = \frac{IP(kW)}{m_f\left(\frac{Kg}{sec}\right)xCV\left(\frac{kJ}{kg}\right)}$$

$$0.35 = \frac{93.75(kW)}{m_f\left(\frac{Kg}{sec}\right)x45000\left(\frac{kJ}{kg}\right)}$$

$m_f = 5.9523x10^{-3}\left(\frac{Kg}{sec}\right);$

$m_f = 5.9523x10^{-3}x3600\left(\frac{Kg}{hr}\right)$=21.4285kg/hr

$BSFC = \frac{m_f\left(\frac{kg}{hr}\right)}{BP} = \frac{21.4285}{75} = \frac{0.2857kg}{kWhr}$

$Break\ thermal\ efficiency = \frac{BP}{Q_s} = \frac{BP(kW)}{m_f\left(\frac{Kg}{sec}\right)xCV\left(\frac{kJ}{kg}\right)} = \frac{75}{5.9523x10^{-3}x45000}$ $=0.28 = 28\%$

$$Volumetric\ efficiency = \frac{Volume\ of\ air\ sucked}{Swept\ Volume}$$

In 4 Stroke engine in every 2 revolution there will be 1 suction stroke.

In 2 Stroke engine in every 1 revolution there will be 1 suction stroke.

Hence swept volume per sec $= \frac{\pi D^2 L}{4}\frac{X}{60}$ where $X = N \; for \; 2 \; stroke \; and \; X = \frac{N}{2}$ for 4 Stroke

$$Volumetric\ efficiency = \frac{Volume\ of\ air\ sucked}{\frac{\pi D^2 L}{4}\frac{X}{60}}$$

Actual volume sucked :

$$Air\ fuel\ ratio = \frac{m_a}{m_f} = \frac{16}{1}$$

$$m_a = 16m_f = 16x5.9523 = 0.0952kg/s$$

$$m_a(kg/s) = \frac{P_s\left(\frac{kN}{m^2}\right)xV_a\left(\frac{m^3}{s}\right)}{R\left(\frac{kJ}{kgK}\right)xT(K)}$$

$$0.0952 = \frac{1x10^2 xV_a\left(\frac{m^3}{s}\right)}{0.287x303}$$

$$Actual\ Volume\ suckec\ V_a = 0.829\frac{m^3}{s}$$

$$Volumetric\ efficiency = \frac{Volume\ of\ air\ sucked}{\frac{\pi D^2 L}{4}\frac{X}{60}}$$

$$Volumetric\ efficiency = \frac{Volume\ of\ air\ sucked}{\frac{\pi D^2 L}{4}\frac{N/2}{60}}$$

$$0.8 = \frac{0.0829}{\frac{\pi D^2 1.5D}{4}\frac{2500/2}{60}}$$

$$D^3 = 4.211x10^{-3}$$

$$D = 0.161m$$

$$L = 1.5D = 1.5x0.161m = 0.242m$$

4. The following data were recorded during a test on a single cyinder four stroke oil engine Bore=150mm, Stroke =300mm, speed =18000 revolutions per hour, Brake torque=200N-m,

indicated mean effective pressure =7bar, fuel consumption = 2.04kg/hr, cooling water flow rate =5kg/ min cooling water temperature rise = 30^{o}C, Air-fuel ratio=22, exhaust gas temperature =410^{o}C, Specific heat of exhaust gases = 1.0kJ/kgK, room temperature is 20^{o}C, calorific value of fuel=42MJ/k, Determine i) Mechanical efficiency ii) BSFC and draw the heat balance sheet on minute basis and percent basis(june/July08)

Solution

Data:

Number of cylinder =1

Number of stroke =4

Bore=150mm, ie D=0.15m; Stroke =300mm ie L=0.3m

speed =18000 revolutions per hour, N=18000 rev/hr=$\frac{18000}{60} = 300rpm$;

Brake torque=200N-m ie T=200N-m, ;

Indicated mean effective pressure =7bar ie $p_m = 7bar = 700kPa$

fuel consumption = 2.04kg/hr ie $m_f = \frac{2.04kg}{hr} = \frac{\frac{2.04}{60}kg}{min} = \frac{\frac{2.04}{3600}kg}{s}$

cooling water flow rate =5kg/ min m_f =5kg/ min

cooling water temperature rise = 30^{o}C ie $T_{co} - T_{ci}$=30^{o}C

Air-fuel ratio=22 ie AF=22, exhaust gas temperature =410^{o}C ie T_{ex} = 410^{o}C,

Specific heat of exhaust gases = 1.0kJ/kgK, $C_{pg} = 1.0$kJ/kgK;

room temperature is 20^{o}C ie T_R,= 20^{o}C ;

calorific value of fuel=42MJ/kg ie C.V=42000kJ/kg

BP: $\frac{2\pi NT(Nm)}{60000} = \frac{2\pi X300X200}{60000}$ =6.28 kW.

IP=$\frac{p_m LAn}{60} xk$, A=$\frac{\pi d^2}{4}$=0.0177m^2 k, number of cylinder

n=$\frac{N}{2}$ for 4 stroke ie n=$\frac{200}{2}$ =100

IP=$\frac{7X10^2(kPa)X0.3X0.0177X100}{60} x1$ = 6.195 kW

$\eta_{mech} = \frac{BP}{IP}$=$\frac{6.195}{6.28} x100$ = 98.6%

Heat Balance sheet:

1. Heat input:
   m_f =2.04kg/hr = 2.04/60 kg/min
   Q= m_f X CV = (2.04/60) X 42000
   =1428 kJ/Min

2. BP = 6.28 kW X60 =376.8 kJ/Min
3. Heat carried by cooling water: $m_w C_{pw} (T_{co} - T_{ci})$ =5X4.187x30=628.05 kJ/Min
4. Heat carried by exhaust gases : $(m_f + m_a) C_{pg} (T_{ex} - T_R)$

: $m_f\ ((m_a/m_f)+1)\ C_{pg}\ (T_{ex} - T_R)$

=(2.04/60)X(22+1) x1x(410-30)

=297.16 kJ/min

5. Heat un account for : 1 – (2+3+4)
   : 1428 – (376.8 + 628.05 + 297.16)
   : 125.99 kJ/min

Energy input,	kJ/min	%	*Energy consumed,*	kJ/min	%
1. Heat input	1428	100	2. Energy consumed as brake power	376.8	26.39
			3. Energy carried by coolant	628.05	43.98
			4. Energy carried by exhaust gases	297.16	20.81
			5. Unaccounted losses	125.99	8.82
Total	1428	100	Total	1428	100

5. A test on a 2 stroke oil engine gave the following results at full load: Speed = 350rpm, Net brake load = 650N, imep = 3bar, fuel consumption = 4kg/hr, Jacket cooling water flow rate = 500kg/h, Jacket water temperature at outlet = 40^o C , Jacket water temperature at outlet = 20^o C , exhaust temperature = 400^o C. Air used per kg fuel = 32kg, cylinder diameter = 22cm, stroke = 28cm, Brake drum circumference = 314cm, calorific value of the fuel = 43MJ/kg, Mean specific heat of exhaust gases 1kJ/kgK. Room Temperature =20^o C Determine (i) Mechanical efficiency, (ii) BMEP, Draw energy balance sheet in kW and in percentage. (July06)

Data:

N=350RPM; Brake Net brake load = 650N,ie $W-S$=650N; p_m=3bar= 300kPam_f=4kg/hr=$\frac{4}{3600}\,kg/s$; m_w =500 kg/hr=$\frac{500}{3600}$ kg/s; cooling water temperature rise = 30^o C ie T_{co} =40^o C T_{ci}=20^o C; $T_{ex} = 400^oC$; Air used per kg fuel = 32kg ie AF=22; D = 22cm = 0.22m; stroke = 28cm ie L=28cm =0.28m; Circumference=314cm ie πD=314cm, CV=43$\frac{MJ}{kg} = \frac{43000kJ}{kg}$; $C_{pg} = 1.0kJ/kg$; T_R =20^o C

Circumference=πD, There fore D=3.14/3.14 = 1 m

T=$\frac{(w-s)(D_b+d_r)}{2}$= $\frac{(650)(1+0)}{2}$ = 325 N-m

BP: $\frac{2\pi NT}{60000}$ = $\frac{2\pi X(350)X325}{60000}$ =11.9 kW.

IP=$\frac{p_m LAn}{60}$xk, where p_m is in kN/m^2

A=$\frac{\pi d^2}{4}$=0.038m^2

N=n for 2stroke

IP=$\frac{3X10^2 X0.28X0.038X(350)}{60}x1$ = 18.62kW

$\eta_{mech} = \frac{BP}{IP}$=$\frac{11.9}{18.62}$ = 0.639=63.9%

BP=$\frac{p_{mb} LAN}{60}xk$, A=$\frac{\pi d^2}{4}$=0.038m^2

p_{mb}=$\frac{60X11.9}{0.28X0.038X350x1}$=1.92X10^2 kN/m^2

Heat Balance sheet:

1. Heat input :
   $m_{f=}$ 4kg/hr = 4/3600 kg/s
   Q= m_f X CV =(4/3600) X 43000
   =47.78 kW

2. BP = 11.9 kW
3. Heat carried by cooling water: $m_w C_p (T_{co} - T_{ci})$
   =(500/3600)X4.187x(40-20)
   =11.6kW
4. Heat carried by exhaust gases : $(m_f + m_a) C_{pg} (T_{ex} - T_R)$
   : $m_f ((m_a/m_f)+1) C_{pg} (T_{ex} - T_R)$

Air fuel ratio = m_a/m_f=32

:(4/3600)X(32+1) x1x(400-20)
: 13.93kW

5. Heat un account for : 1 – (2+3+4)
   : 47.78 – (11.9 + 11.6 + 13.93)
   : 10.35 Kw

Energy input,	kW	%	*Energy consumed,*	kW	%
1. Heat input	47.78	100	2. Energy consumed as brake power	11.9	24.9
			3. Energy carried by coolant	11.6	24.28

			4. Energy carried by exhaust gases	13.93	29.15
			5. Unaccounted losses	10.35	21.66
Total	47.78	100	Total	47.78	100

Note:

In this problem note down circumference of Brake wheel is given

Hence diameter of the brake wheel should be found out by formula Circumference=πD

6. In a test on a three cylinder four stroke internal combustion engine with 22 cm bore and 26cm stroke the following were the observations during a **trial periods of one hour**: Fuel consumption = 8.0kg. Calorific value = 45000kJ/kg, total revolutions of the crank shaft 12000, mean effective pressure = 6 bar, net load on brake= 1.5kN, brake drum dia=1.8m, rope dia = 3cm, mass of cooling water =550kg, inlet temperature of water = 27^{o}C, exit temperature of water = 55^{o}C air consumed = 300kg, ambient temperature = 30^o C, exhaust gas temperature =310^o C, specific heat of gases 1.1kJ/kgK, Calculate (i) indicated and brake power, (ii) Mechanical efficiency (iii) indicated thermal efficiency. Also draw a heat balance in kJ/min (july/Aug 2004)

Data:

Single cylinder ie k=1

Stroke 4

22 cm bore and 26cm stroke D=0.22m; L=0.26m

trial periods of one hour: Fuel consumption = 8.0kg.

ie $m_f = 8kg\ in\ one\ hr = \frac{8}{3600}$kg/s=$\frac{8}{60}$ kg/min

Calorific value = 45000kJ/kg CV=45000kJ/kg

total revolutions of the crank shaft 12000,ie N=12000 in one hr trial period

N=$\frac{12000}{60}$ =2000rpm

mean effective pressure = 6 bar $p_m = 6\ bar = 600kPa$,

net load on brake= 1.5kN, ie $W - S = 1.5kN$;

brake drum dia=1.8m ie $D_b = 1.8m$; rope dia = 3cm ie $d_r = 0.03m$;

mass of cooling water =550kg, mass of cooling water in trial period of one hr

ie $m_f = 550kg\ in\ one\ hr = \frac{550}{3600}$kg/s=$\frac{550}{60}$ kg/min

inlet temperature of water = 27^{o}Cie T_{ci} =27^{o}C;

exit temperature of water = 55^{o}C ie T_{c0} =55^{o}C ;

air consumed = 300kg in one hr trial period

ie $m_a = 300kg\ in\ one\ hr = \frac{300}{3600}$kg/s=$\frac{300}{60}$ kg/min,

ambient temperature = 30^o C ie T_R=30^o C,

exhaust gas temperature =310^o Cie T_{ex} =310^o C,

specific heat of gases 1.1kJ/kgK, C_{pg}=1.1kJ/kgK

T=$\frac{(w-s)(D_b+d_r)}{2}$= $\frac{(1.5X1000-0)(1.8+0.03)}{2}$ = 1372.5 N-m total revolutions of the crank shaft 12000 in one hour trial there for N =$\frac{12000}{60}$rpm

BP: $\frac{2\pi NT}{60000}$ $= \frac{2\pi X\left(\frac{12000}{60}\right)X1372.5}{60000}$ =28.73 kW.

IP=$\frac{p_m LAn}{60} xk$ where k is the number of cylinders

n=$\frac{N}{2}$ for 4 stroke ie $\frac{200}{2} = 100$

A=$\frac{\pi d^2}{4}$=0.037994m^2

IP=$\frac{6X10^2 X0.26X0.037994X100}{60}$ $X3cylinder$ = 29.64 kW

$\eta_{mech} = \frac{BP}{IP} = \frac{28.73\ X100}{29.64}$ = 96.92%

m_f=8kg in one hour trial; hence m_f= =$\frac{8}{3600}$ kg/s =$\frac{8}{60}$ kg/min

$\eta_{\text{Indicated thermal efficiency}} = \frac{IP}{mf X\ C.V} = \frac{29.64}{(\frac{8}{3600})X\ 45000}$ = 0.2964 or 29.64 %

Heat Balance sheet:

1. Heat input : m_f X CV =(8/60) X 45000

   =6000 kJ/Min

2. BP = 28.73 X60 =1723.8 kJ/Min
3. Heat carried by cooling water:

   m_w =550kg in one hr trial =550/60 kg/min

   $Q_w = m_w C_p (T_{co} - T_{ci})$

   =(550/60)X4.187x(55-27)

   =1074.66kJ/Min
4. Heat carried by exhaust gases : $(m_f + m_a) C_{pg} (T_{ex} - T_R)$

   Air consumed =300kg in 1 hr trial ie m_a=300kg/hr =$\frac{300}{60}$ kg/min

   $= \left(\frac{8+300}{60}\right) 1.1\ (310 - 30)$

   =1581. 07kJ/min

5. Heat un account for : 1 – (2+3+4)

   : 6000 – (1723.8 + 1074.66 + 1581.07)

   : 1620.47 kJ/min

Energy input,	kJ/min	%	*Energy consumed,*	kJ/min	%

1. Heat input	6000	100	2. Energy consumed as brake power	1723.8	28.73
			3. Energy carried by coolant	1074.66	17.92
			4. Energy carried by exhaust gases	1581.07	26.35
			5. Unaccounted losses	1620.47	27.01
Total	6000	100	Total	6000	100

7. A test on a single cylinder, four stroke oil engine having a bore of 15cm and stroke 30cm gave the following results: speed = 300RPM, Brake torque =200Nm, indicated mean effective pressure = 7 bar, fuel consumption = 2.4kg/h, Cooling water flow rate=4.5 kg/Min,cooling water temperature rise = 30^{o} C, Air fuel ratio = 22. Specific heat of exhaust gases=1.0kJ/kgK Exhaust gas temperature = 410^{o}C, Room temperature= 20^{o}C, Room pressure= 1bar, calorific value of the fuel=42MJ/kg, R=.2872kJ/kgK of air. Determine 9i) the indicated thermal efficiency (ii) Volumetric efficiency base on atmospheric conditions (iii) Draw an energy balance sheet in terms of kJ/min and on percent basis. (Jan/Feb 06)

Data: D=15cm=0.15m; L=30cm=0.3m; N=300RPM; Brake Torque= 200Nm ie T=200Nm; p_m=7bar= 700kPa

m_f=2.4kg/hr=$\frac{2.4}{3600}kg/s$; m_w =4.5 kg/Min ;cooling water temperature rise = 30^{o} C ie $T_{co} - T_{ci}$=30^{o} C;

AF=22; $C_{pg} = 1.0kJ/kg$; $T_{ex} = 410^oC$; $T_R = 20^oC$; $p_a = 1\ bar$; CV=42$\frac{MJ}{kg} = \frac{42000kJ}{kg}$;R=0.287kJ/kgK

IP=$\frac{p_mLAn}{60}xk$, where p_m=is in kN/m^2

A=$\frac{\pi d^2}{4}$=0.0177m^2

n=$\frac{N}{2} = \frac{300}{2}$ since engine ie 4 stroke

IP=$\frac{7X10^2X0.3X0.0177X(\frac{300}{2})}{60}x1$ = 9.29 kW

$$\eta_v = \frac{V_{act}}{V_{the}}$$

$V_{act=}$ mass of air/ Density of air

Mass of air= Fuel consumption X Air fuel ratio

M_f =2.4kg/hr =$\frac{2.4}{3600}$ kg/s

Mass of air=(2.4/3600)X22 =0.015 kg/s

$$\text{Density of air} = \frac{P}{RT}$$

$\text{Density of air} = \frac{100}{0.278x293}$ =1.189 kg/m^3

m =ρV hence V=$\frac{m}{\rho}$

V_{act}=0.015/1.189=0.0126 m^3/s

V_{the}=$\frac{\pi d^2}{4} xlx(\frac{N}{2x60})$ Here $\frac{N}{2}$ is used since engine is 4 stroke

note that suction stroke is $\frac{N}{2}$ for 4 stroke engine and number of suction stroke =N for 2 stroke engine

V_{the} =$\frac{\pi 0.15^2}{4} x0.3x(\frac{300}{2x60})$ =0.0132 m^3/s

$\eta_v = \frac{0.0126}{0.0132}$ =0.9545

Heat Balance sheet:

1. Heat input : m_f X CV =(2.4/3600) X 42000

   =28 kWX60=1680 Kj/Min

2. BP: $\frac{2\pi NT}{60000}$ = $\frac{2\pi X(300)X200}{60000}$ =6.28 kWX60=376.8Kj/Min
3. Heat carried by cooling water:

   m_w=4.5kg/hr=4.5/60kg/min

   Q_w= $m_w C_p (T_{co} – T_{ci})$

   =(4.5/60)X4.187x(30)

   =9.42kWX60=565.24Kj/Min
4. Heat carried by exhaust gases : $(m_f + m_a) C_{pg} (T_{ex} – T_R)$

   : $m_f ((m_a/m_f)+1) C_{pg} (T_{ex} – T_R)$

   :(2.4/3600)X(22+1) x1x(410-20)

   :5.98 kWX60=358.8Kj/Min
5. Heat un account for : 1 – (2+3+4)

   : 1680 – (376.8 + 565.24 + 358.8)

   : 379.16 Kj/Min

Energy input,	Kj/Min	%	*Energy consumed,*	Kj/Min	%
1. Heat input	1680	100	2. Energy consumed as brake power	376.8	22.42

			3. Energy carried by coolant	565.24	33.65
			4. Energy carried by exhaust gases	358.8	21.32
			5. Unaccounted losses	379.16	22.56
Total	1680	100	Total	1680	100

7. The following observations were made during a trial of a single cylinder four stroke cycle gas engine having cylinder diameter of 18cm and a stroke of 24cm: Duration of Trial= 30min; Total number of revolutions = 9000, total number of explosions = 4450, Indicated mean effective pressure = 5 bar. Net load on the brake wheel = 390N. Effective diameter of brake wheel = 1m, calorific value of gaseous fuel at NTP= 19MJ/m^3, Total fuel used at NTP= 2.4m^3, total air is used = 36m^3, pressure of air = 720mm of Hg, density of air = 1.29kg/m^3, temperature of air =17^{o}C, temperature of exhaust gases =410^{0}C,specific heat of exhaust gases: 1.0kJ/kgK. Room temperature =17^{o}C, cooling water circulated = 80kg, Rise in water temperature =30^{o}C. Draw up a heat balance sheet and estimate the mechanical and indicated thermal efficiencies of the engine. Take R= 287J/kgK (June/july08) (july/Aug 05).

   <u>Data</u>

   Number of cylinder k=1

   Number of stroke=4

   D=0.18m; L=0.24m

   Duration of trial 30min Total number of revolutions = 9000 ie N=9000 in 30min N=$\frac{9000}{30}$ rpm, total number of explosions = 4450 ie n=4450 in 30 min n=$\frac{4450}{30}$ per minute,

   Indicated mean effective pressure = 5 bar ie $p_m = 5\ bar$;.

   Net load on the brake wheel = 390N.ie $W - S = 390N$

   Effective diameter of brake wheel = 1m ie D_b=1m,

   calorific value of gaseous fuel at NTP= 19MJ/m^3, CV(at NTP) =19MJ/m^3

   Total fuel used at NTP= 2.4m^3, ie V_f=2.4 m^3 in 30 min at NTP

   Ie V_f=$\frac{2.4}{30}$ m^3/min(at NTP)=$\frac{2.4}{30x60}$ m^3/min(at NTP)

   total air is used = 36m^3, ie V_a =36m^3 in 30min $V_a = \frac{36}{30}$ m^3/min=$\frac{36}{30x60}$ m^3/sec

   (at pressure of air = 720mm of Hg, density of air = 1.29kg/m^3, temperature of air =17^{o}C,)

   temperature of exhaust gases =410^{0}C,ie T_{ex} =410^{0}C

   specific heat of exhaust gases: 1.0kJ/kgK. Ie T_R =

   Room temperature =17^{o}C, Ie T_R =17^{o}C

   cooling water circulated = 80kg,ie m_w=80kg in 30min m_w=$\frac{80}{30}$ kg/min

   Rise in water temperature =30^{o}C ie ($T_{co} - T_{ci}$)= 30^{o}C

T=$\frac{(w-s)(D_b+d_r)}{2}$= $\frac{(390)(1+0)}{2}$ = 195 N-m

N=$\frac{9000}{30}$ rpm Note down given is total number of revolutions in 30min trial period

BP: $\frac{2\pi NT}{60000}$ $= \frac{2\pi X\left(\frac{9000}{30}\right)X195}{60000}$ =6.123 kW.

IP=$\frac{p_m LAn}{60} xk$, A=$\frac{\pi d^2}{4}$=0.025434m^2

Here n is not equal to $\frac{N}{2}$ since no of explosions (actual working stroke) is given

Total number of explosion is 4450 in 30min Hence n= $\frac{4450}{30}$

IP=$\frac{5X10^2X0.24X0.025434X(\frac{4450}{30})}{60}$ x1= 7.55 kW

$\eta_{mech} = \frac{BP}{IP} = \frac{6.123\ X100}{7.55}$ = 81.1%

V_f=2.4 m^3 at NTP in 30min trial

$V_f = \frac{2.4}{30}$ m^3/min=$\frac{2.4}{30x60}$ m^3/sec at NTP

CV =19000kJ/m^3 at NTP

$\eta_{\text{Indicated thermal efficiency}} = \frac{IP}{V_f(m^3/min)at\ NTPX\ C.V\left(\frac{kJ}{m^3}\right)at\ NTP} = \frac{7.55}{(\frac{2.4}{30X60})X\ 19000}$ = 0.298 or 29.8 %

Heat Balance sheet:

1. Heat input : m_f x CV
   Volume of fuel used is 2.4 m^3 in 30 min
   Hence $V_f = \frac{2.4}{30x60}$ m^3/s at NTP
   Calorific value CV is 19000kJ / m^3 at NTP
   Q = V_f at NTP x CV at NTP
   Q =(2.4/(30X60)) X 19000
   =25.33 kW

2. BP = 6.123 kW
3. Heat carried by cooling water: $m_w C_p (T_{co} - T_{ci})$
   =(80/(30X60))X4.187x(30)
   =5.58kW
4. Heat carried by exhaust gases : $(m_f + m_a) C_{pg} (T_{go} - T_{gi})$

Atmospheric Pressure head at room is 720mm of Hg

Conversion of mm of Hg to bar

760mm Hg is equivalent to 1.01325bar

Hence 720mm of Hg $=\frac{1.01325}{760}x720\ bar = 0.9706\ bar$=97.06kPa

Mass of air = Volume of air at room temperature x density at room temperature

$m_a = \frac{36}{30x60}$(m^3/s)x1.26kg/ m^3 (both quantity at room temperature)

$m_a = 0.0252 kg/s$

Note If density of air is not given use $density\ \rho = \frac{p(kPa)}{RT(K)}$

Mass of fuel

Since density of fuel is not given assume density of fuel = density of air

$$m_f = \rho_f(Room\ temp)V_f(Room\ temp)$$

Density of air =1.29kg/m^3 at room temperature (given)

Hence take $\rho_f(Room\ temp) = 1.29 kg/m^3$

Hence volume of fuel used /sec is to be converted into volume fuel at room temperature using Ideal gas equation

$$(\frac{pv}{T})_{fNTP} = (\frac{pv}{T})_{fRTP}$$

NTP pressure and temperature are 1.01325bar and 0^oC or 273K

$$\left(\frac{101.325x\frac{2.4}{30x60}}{273}\right)_{NTP} = \left(\frac{96.06xV_{RTP}}{17+273}\right)_{RTP}$$

V_{fRTP}: $1.494x10^{-3}$$m^3$/s

$$m_f = \rho_f(Room\ temp)V_f(Room\ temp)$$

$m_f = 1.29x1.494x10^{-3} = 1.927x10^{-3} kg/s$

Heat carried by exhaust gases : (m_f + m_a) C_{pg} (T_{go} – T_{gi})

$Q_{exh} = (0.0252 + 1.927x10^{-3})1(410 - 17)$=10.66kW

5. Heat un account for : 1 – (2+3+4)
: 25.33 – (6.123 + 5.58 + 10.66)

: 2.967 kW

Energy input,	kW	%	*Energy consumed,*	kW	%
1. Heat input	25.33	100	2. Energy consumed as brake power	6.123	24.17
			3. Energy carried by coolant	5.58	22.03
			4. Energy carried by exhaust gases	10.66	
			5. Unaccounted losses	2.967	
Total	25.33	100	Total	25.33	100

8. 5. A gas engine working on constant volume cycle gave the following results during a one hour test run. Cylinder diameter 24cm, stroke 48cm. effective diameter of brake drum wheel 1.25m, Net load on the brake 1236N, average speed 226.7revolutions per minute. Average explosions per minute 77. MEP 7.5 bar, gas used 13m^3 at 15^{o}C and 771mm of Hg pressure. Lower calorific value of gas 22000kJ/m^3 at NTP. Cooling water used 625kg, inlet water temperature 25^{o}C. Outlet water temperature 60^{o}C. Determine i) Mechanical efficiency (ii) the specific fuel consumption in m^3/IP hr. (iii) Indicated and brake thermal efficiencies. Draw heat balance for the engine on minute bases NTP conditions are 760mm of Hg and 0^{o}C (Dec 06/Jan07)

   Data: one hr test run

D=24cm=0.24m; L=48cm=0.48m; N=300RPM;

effective diameter of brake drum wheel 1.25m ie $D_b = 1.25m$;

Brake Net load on the brake 1236N ie $w - s$ =1236N;

average speed 226.7revolutions per minute.ie N=226.7 rpm;

Average explosions per minute 77 ie n=77 per min; MEP 7.5 bar ie p_m=7.5bar=7.5$x10^2$;

gas used 13m^3 at 15^{o}C and 771mm of Hg pressure (duration one hour)

ie $V_f = \frac{13}{3600}\ m^3/sec$ at pressure 771mm of Hg and temperature 15^{o}C

Lower calorific value of gas 22000kJ/m^3 at NTP. CV =$22000\ \frac{kJ}{m^3}$ at NTP ie 1.0325 bar and 0^o C

Cooling water used 625kg (duration of trial = 1hr) ie m_w=$\frac{625}{60}$ kg/min $= \frac{625}{3600}\ kg/s$

Inlet water temperature 25^{o}C ie T_{ci}=25^o C; Outlet water temperature 25^{o}C ie T_{ci}=60^o C

Solution

T=$\frac{(w-s)(D_b+d_r)}{2}$= $\frac{(1236)(1.25+0)}{2}$ = 772.5 N-m

BP: $\frac{2\pi NT}{60000}$ = $\frac{2\pi X(226.7)X772.5}{60000}$ =18.32 kW

IP=$\frac{p_m LAn}{60}$xk

Here n is not equal to$\frac{N}{2}$ since number of explosion in the problem is given

n=Average explosions per minute 77 ie n=77 /min

A=$\frac{\pi d^2}{4}$=0.045m^2

IP=$\frac{7.5X10^2X0.48X0.045X(77)}{60}x1$ = 20.79 kW

$\eta_{mech} = \frac{BP}{IP}$=$\frac{18.32\ X100}{20.79}$ = 88.1%

Sfc=$\frac{mass\ of\ fuel\ use\ in\ kg/hr}{IP\ in\ kW}$

Sfc=13/20.79=0.625 m^3/IP hr (based on IP)

Pressure of fuel supplied =771mm of Hg

P= $\omega_{mercury}$ x g$xh_{mercury}$

Conversion of mm of mercury into pressure N/m^2

P =(1000X13.6)x9.81x (771/1000)=102.86 x10^3 N/m^2 =102.86kN/m^2

OR

1.013 bar = 760mm of Hg

? in bar = 771mm of Hg

Pressure of fuel supplied in bar =$\frac{1.013\ bar}{760mm\ of\ Hg}$ x 771mm of Hg =1.02766 bar =102.76 kN/m^2

Conversion of volumetric fuel flow at supply pressure to pressure of fuel at NTP

Normal pressure =1.01325 bar =101.325kN/m^2 and Normal temperature = 273K

Temperature of fuel =15^{o}C

$$\left(\frac{pv}{T}\right)_{NTP} = \left(\frac{pv}{T}\right)_{RTP}$$

$$\left(\frac{101.325V_f}{273}\right)_{aNTP} = \left(\frac{102.86X13}{15+273}\right)_{aRTP}$$

V_{NTP} of fuel = 12.5 m^3 in 1 hr trial

V_{NTP} of fuel = $\frac{12.5}{3600}$ m^3/sec

Q= $Vfat\ NTP(\ \text{m3/sec})X\ C.V(\frac{kJ}{m3})at\ NTP$ = $(\frac{12.5}{3600})X$ 22000 kJ/s

ŋ Indicated thermal efficiency=$\frac{IP}{Vfat\ NTP\ X\ C.Vat\ NTP}$ = $\frac{20.79}{(\frac{12.5}{3600})X\ 22000}$ = 0.272 or 27.2 %

ŋ Brake thermal efficiency=$\frac{BP}{Vfat\ NTP\ X\ C.Vat\ NTP}$ = $\frac{18.32}{(\frac{12.5}{3600})X\ 22000}$ = 0.239 or 23.9 %

Heat Balance sheet:

1. Heat input : $Vfat\ NTP\ X\ C.Vat\ NTP$ =(12.5/(60)) X 22000

   =4583.33 Kj/min

2. BP = 18.32 kWX60=1099.2 Kj/min
3. Heat carried by cooling water: $m_w\ C_p\ (T_{co} – T_{ci})$
   =(625/(60)X4.187x(60-25)
   =1526.5 Kj/min
4. Heat carried by exhaust gases : no data reg air consumption=0
5. Heat un account for : 1 – (2+3+4)
   : 4583.33 – (1099.2 + 1526.5 + 0)
   : 1957.63Kj/min

Energy input,	Kj/min	%	*Energy consumed,*	Kj/min	%
1. Heat input	4583.33	100	2. Energy consumed as brake power	1099.2	23.98
			3. Energy carried by coolant	1526.5	34.09
			4. Energy carried by exhaust gases	0	0
			5. Unaccounted losses	1957.63	42.7
Total	4583.33	100	Total	4583.33	100

9. The following data were obtained from a Morse test on a 4 cylinder, 4 stroke cycle SI engine, coupled to hydraulic dynamometer operating at a constant speed of 1500rpm: Brake load with all cylinders firing = 296N, Brake load with cylinder No 1 not firing = 201N, Brake load with cylinder No 2 not firing = 206N, Brake load with cylinder No 3 not firing = 192N, Brake load with cylinder No 4 not firing = 200N. The brake power in kW is calculated using the equation BP= WN/42300, where W is the brake load in Newtons and N is the speed of the engine in RPM, Calculate : i) Brake power ii) Indicated power iii) Friction power, iv) Mechanical efficiency(Dec2011).

N=1500rpm;

$W_{1,2,3,4}$=296N; $W_{2,3,4}$=201N; $W_{1,3,4}$=206N; $W_{1,2,,4}$=192N; $W_{1,2,3,}$=200N

$BP_{1,2,3,4}$= $W_{1,2,3,4}$N/42300

$BP_{1,2,3,4}$= (296X1500)/42300 =10.5kw

$BP_{,2,3,4}$= $W_{2,3,4}$N/42300

BP_{234}= (201X1500)/42300 =7.13kw

$BP_{1,,3,4}$= $W_{1,,3,4}$N/42300

BP_{134}= (206X1500)/42300 =7.3kw

$BP_{1,2,,4}$= $W_{1,2,,4}$N/42300

BP_{124}= (192X1500)/42300 =6.8kw

$BP_{1,2,3,}$ = $W_{1,2,3,}$N/42300

BP_{123}= (200X1500)/42300 =7.09kw

$(ip)_{total}$ =ip_1 +ip_2 +ip_3 +ip_4 = (BP_t – BP_{234})+(BP_t – BP_{134})+(BP_t – BP_{124})+(BP_t – BP_{123})

=4B_t –(B_1 +B_2 +B_3 +B_4)=4 x10.5 – (7.13 +7.3 +6.8+7.09)

=13.68 Kw

$$\eta_{mech} = \frac{BP}{IP} = \frac{10.5}{13.68} x100 = 76.75\%$$

FP=IP-BP=13.68-10.5=3.18Kw

6.

7.In a test of 4- cylinder, 4- stroke petrol engine of 75mm bore and 100mm stroke, the following results were obtained at full throttle at a constant speed and with a fixed setting of the fuel supply at 0.082kg/min. BP with all the 4 cylinders working = 15.24kW, BP with cylinder No 1 is cut off = 10.45kW, BP with cylinder No 2 is cut off = 10.38kW, BP with cylinder No 3 is cut off = 10.23kW, BP with cylinder

No 4 is cut off = 10.45kW. Determine the (i) the indicated power, (ii) the indicated thermal efficiency if the calorific value of the fuel = 44MJ/kg, (iii) Relative efficiency based on IP if clearance volume in each cylinder = 115cc (July06)

$(ip)_{total} = ip_1 + ip_2 + ip_3 + ip_4 = (BP_t - BP_{234}) + (BP_t - BP_{134}) + (BP_t - BP_{124}) + (BP_t - BP_{123})$

$= 4B_t - (B_1 + B_2 + B_3 + B_4) = 4 \times 15.24 - (10.45 + 10.38 + 10.23 + 10.45)$

$= 20.29$ Kw

$$\eta_{mech} = \frac{BP}{IP} = \frac{15.24}{20.29} x100 = 75.11\%$$

$m_f = 0.082$kg/min $= \frac{0.082}{60}\ kg/s$

$\eta_{\text{Indicated thermal efficiency}} = \frac{IP}{mf X\ C.V} = \frac{20.29}{(\frac{0.082}{60}) X\ 44000} = 0.337$ or 33.7 %

$\eta_{\text{Relative efficiency}} = \frac{\eta\text{ Indicated thermal efficiency}}{\eta\text{ air standard efficiency}}$

$$\eta\text{ air standard efficiency} = 1 - \frac{1}{CR^{\gamma-1}}$$

$$CR = \frac{V_C + V_S}{V_C}$$

$V_s = \frac{\pi d^2}{4} xl = \frac{\pi 7.5^2}{4} x10 = 441.56\ cm^3$ per stroke

$CR = \frac{115+441.56}{115} = 4.84$

$\eta\text{ air standard efficiency} = 1 - \frac{1}{4.84^{1.4-1}} = 0.468$

$\eta_{\text{Relative efficiency}} = \frac{0.337}{0.468} = 0.72$ or 72%

10.The following readings were recorded during a trial on a single cylinder, 2 stroke diesel engine. Power supplied by electric motor= 1.5kW. Rated speed= 500 rpm , Net loads on brake =225N. Diameter of the brake wheel = 100cm. Rate of flow of cooling water through engine jacket = 13.65kg/min, Rise in temperature of cooling water = 10°C, Fuel consumption= 2kg/hr. calorific value of fuel used = 43000kJ/kg; AF ratio = 32:1, Cp (gases) = 1.006kJ/kgK Exhaust gas temperature = 345°C, Ambient temperature = 25°C and ambient pressure = 1 bar. Take L=D=30cm. Determine (i) Mechanical efficiency (ii) thermal efficiency (iii) Brake specific fuel consumption (iv) Brake mean effective pressure. Draw heat balance sheet on % basis (july 2007)

Power supplied by electric motor= 1.5kW. ie FP=1.5Kw(given); N=500rpm;

Net loads on brake =225N ie $W - S = 220N$; N=500RPM $D_b = 100cm$=1m; $m_w = \frac{13.65kJ}{min}$;

Rise in temperature of cooling water = 10°C ie $T_{co} - T_{ci}$=10°C ; $m_f = 2kg/hr$; CV=43000kJ/kg;

AF=32; Cpg= 1.006kJ/kgK; T_{ex} =345°C; $p_a = 1bar$; T_a =25°C; L=D=0.3m

$T=\frac{(w-s)(D_b+d_r)}{2}=\frac{(225-0)(1+0)}{2}$ = 112.5 N-m

BP: $\frac{2\pi NT}{60000} = \frac{2\pi X(500)X112.5}{60000}$ =5.89 kW.

IP=BP+FP=5.89+1.5=7.39 Kw

$\eta_{mech} = \frac{BP}{IP}=\frac{5.89\ X100}{7.39}$ = 79.7%

Sfc=$\frac{BP}{mass\ of\ fuel\ in\frac{kg}{hr}}$ = 2/5.89=0.339 m³/BP hr

m_f =2kg/hr = $\frac{2}{3600}$ kg/s

$\eta_{\text{Indicated thermal efficiency}}=\frac{IP}{mfX\ C.V} = \frac{7.39}{(\frac{2}{3600})X\ 43000}$ = 0.309 or 30.9 %

$\eta_{\text{Brake thermal efficiency}}=\frac{BP}{mfX\ C.V} = \frac{5.89}{(\frac{2}{3600})X\ 43000}$ = 0.247 or 24.7 %

BP=$\frac{p_{mb}LAn}{60}xk$, A=$\frac{\pi d^2}{4}$=0.071m²

$p_{mb}=\frac{60X5.89}{0.3X0.071X500x1}$=0.33X10⁵ kN/m²

Heat Balance sheet:

1. Heat input : m_f X CV =(2/3600) X 43000

    =23.89 kW

2. BP = 5.89 kW
3. Heat carried by cooling water: $m_w C_p (T_{co} - T_{ci})$
    m_w=13.65kg/min = 13.65/60 kg/s
    Q_w=(13.65/60)X4.187x(10)
    =9.52kW
4. Heat carried by exhaust gases : $(m_f + m_a) C_{pg} (T_{ex} - T_R)$
    : $m_f ((m_a/m_f)+1) C_{pg} (T_{ex} - T_R)$
    :(2/3600)X(32+1) x1x(345-25)
    : 5.87kW
5. Heat un account for : 1 – (2+3+4)

: 23.89 – (5.89 + 9.52 + 5.87)
: 2.61 Kw

Energy input,	kW	%	*Energy consumed,*	kW	%
1. Heat input	23.89	100	2. Energy consumed as brake power	5.89	24.65
			3. Energy carried by coolant	9.52	39.85
			4. Energy carried by exhaust gases	5.87	24.57
			5. Unaccounted losses	2.61	10.93
Total	23.89	100	Total	23.89	100

11.The following readings were recorded during a trial on a single cylinder, 4 stroke diesel engine. Bore = 150mm, stroke = 300mm, speed= 18000 revolutions per hour , Brake torque =200Nm. Indicated mean effective pressure= 7bar, Rate of flow of cooling water through engine jacket = 5kg/min, Rise in temperature of cooling water = 30°C, Fuel consumption= 2.04 kg/hr. calorific value of fuel used = 42MJ/kg; AF ratio = 22:1, Cp (gases) = 1.0kJ/kgK Exhaust gas temperature = 410°C, Ambient temperature = 20°C and ambient pressure = 1 bar.. Determine (i) Mechanical efficiency (ii) thermal efficiency (iii) Brake specific fuel consumption (iv) Brake mean effective pressure. Draw heat balance sheet on minute and % basis (Dec07/Jan 08)

Data:

D=150mm=0.15m; L=300m=0.3m; speed =18000rev per hr=$\frac{18000}{60}$ rpm; T=200Nm; p_m=7 bar; m_w=5kg/min; Rise in temperature of cooling water = 30°C ieT_{co} – T_{ci}=30°C; m_f=2.04kg/hr; CV=42MJ/kg=42000kJ/kg; AF=22; Cpg = 1.0kJ/kgK; T_{ex} =410°C ; T_R =20°C; p_a=1 bar

Solution:

If T is in Nm then BP: $\frac{2\pi NT}{60000}$

If T is in KNm then $\frac{2\pi NT}{60}$

BP: $\frac{2\pi NT(Nm)}{60000}$ = $\frac{2\pi X\left(\frac{18000}{60}\right)X200}{60000}$ =6.28 kW

IP=$\frac{p_m LAn}{60} xk$, A=$\frac{\pi d^2}{4}$=0.0176m^2

N=18000rev per hr

$N = \frac{18000}{60}\ rpm$=300 rpm

$n = \frac{N}{2}$ for 4 stroke engine hence $n = \frac{300}{2}$=150 per min

$IP = \frac{7X10^2 X0.3X0.0176X(150)}{60}$ x1= 9.24 kW

$\eta_{mech} = \frac{BP}{IP} = \frac{6.28\ X100}{9.24}$ = 67.96%

m_f =2.04kg/hr=$\frac{2.04}{3600}$ kg/s

η Indicated thermal efficiency=$\frac{IP}{mfX\ C.V} = \frac{9.24}{(\frac{2.04}{3600})X\ 42000}$ = 0.388 or 38.8 %

η Brake thermal efficiency=$\frac{BP}{mfX\ C.V} = \frac{6.28}{(\frac{2.04}{3600})X\ 42000}$ = 0.264 or 26.4 %

Sfc=$\frac{BP\ inkW}{m_f in\ kg/hr}$2.04/6.28=0.325 kg/BP hr

BP=$\frac{p_m bLAn}{60} x1$, A=$\frac{\pi d^2}{4}$=0.0176m^2

$Bp_m = \frac{60X6.28}{0.3X0.0176X150x1}$=4.75X10^2 kN/m^2

Heat Balance sheet:

1. Heat input : m_f X CV =(2.04/3600) X 42000

   =23.8 kW

2. BP = 6.28 kW
3. Heat carried by cooling water: $m_w C_{pw} (T_{co} – T_{ci})$
   =(5/60)X4.187x(30)
   =10.47kW
4. Heat carried by exhaust gases : $(m_f + m_a) C_{pg} (T_{ex} – T_R)$
   : $m_f ((m_a/m_f)+1) C_{pg} (T_{ex} – T_R)$
   :(2.04/3600)X(22+1) x1x(410-20)
   : 5.083kW
5. Heat un account for : 1 – (2+3+4)
   : 23.8 – (6.28 + 10.47 + 5.083)
   : 1.967 Kw

Energy input,	kW	%	*Energy consumed,*	kW	%
1. Heat input	23.8	100	2. Energy consumed as brake power	6.28	26.39
			3. Energy carried by coolant	10.47	43.99

			4. Energy carried by exhaust gases	5.083	21.36
			5. Unaccounted losses	1.967	8.27
Total	23.8	100	Total	23.8	100

12. In a test of 4- cylinder, 4- stroke petrol engine of 80mm bore and 100mm stroke, the following results were obtained at full throttle at a constant speed BP with all the 4 cylinders working = 14.7kW, BP with cylinder No 1 is cut off = 10.1kW, BP with cylinder No 2 is cut off = 10.3kW, BP with cylinder No 3 is cut off = 10.2kW, BP with cylinder No 4 is cut off = 10.4kW. The fuel consumed at the rate of 5.44kg/hr Determine the (i) the indicated power, (ii) the indicated thermal efficiency if the calorific value of the fuel = 41900kJ/kg, (iii) Relative efficiency based on IP if clearance volume in each cylinder = 100cc

$(ip)_{total} = ip_1 + ip_2 + ip_3 + ip_4 = (BP_t - BP_{234}) + (BP_t - BP_{134}) + (BP_t - BP_{124}) + (BP_t - BP_{123})$

$= 4B_t - (B_1 + B_2 + B_3 + B_4) = 4 \times 14.7 - (10.1 + 10.3 + 10.2 + 10.4)$

=17.8 Kw

$$\eta_{mech} = \frac{BP}{IP} = \frac{14.7}{17.8} x100 = 82.58\%$$

m_f=5.44kg/hr = $\frac{5.44}{3600}$ kgs/s

$\eta_{\text{Indicated thermal efficiency}} = \frac{IP}{mf X\ C.V} = \frac{17.8}{(\frac{5.44}{3600}) X\ 41900}$ = 0.281 or 28.1 %

$\eta_{\text{Relative efficiency}} = \frac{\eta \text{ Indicated thermal efficiency}}{\eta \text{ air standard efficiency}}$

$$\eta \text{ air standard efficiency} = 1 - \frac{1}{CR^{\gamma-1}}$$

$$CR = \frac{V_C + V_S}{V_C}$$

$V_s = \frac{\pi d^2}{4} xl = \frac{\pi 8^2}{4} x10$ =502.4 cm^3per stroke

$CR = \frac{100+502.4}{100}$=6.024

$\eta \text{ air standard efficiency} = 1 - \frac{1}{6.024^{1.4-1}}$=0.512

$\eta_{\text{Relative efficiency}} = \frac{0.281}{0.512}$ =0.549 or 54.9%

13. Morse test is conducted on a four stroke four cylinder petrol engine at a constant speed and the following power is measured:

With all cylinder working =15kW

With number 1 cylinder cut off =11.1kW

With number 2 cylinder cut off =11.3kW

With number 3 cylinder cut off =10.8kW

With number 4 cylinder cut off =11.0kW

The bore and stroke of each cylinder is 75mm and 100mm respectively. The clearance volume of the cylinder is 100cc. The fuel is consumed at the rate 6 kg/hr. If the calorific vaue of the fuel is 42000kJ/kg. Determine i) Indicated power ii) Frictional Power iii) Mechanical efficiency iv) Brake thermal efficiency v) Relative efficiency with respect to brake thermal efficiency. (Dec 2010)

$(ip)_{total} = ip_1 + ip_2 + ip_3 + ip_4 = (BP_t - BP_{234}) + (BP_t - BP_{134}) + (BP_t - BP_{124}) + (BP_t - BP_{123})$

$= 4B_t - (B_1 + B_2 + B_3 + B_4) = 4 \times 15 - (11.1 + 11.3 + 10.8 + 11)$

=15.8 Kw

$$\eta_{mech} = \frac{BP}{IP} = \frac{15}{15.8} x100 = 94.94\%$$

$m_f = 6 kg/hr = \frac{6}{3600}$ kgs/s

(iii)$\eta_{\text{Indicated thermal efficiency}} = \frac{IP}{mf X\ C.V} = \frac{15.8}{(\frac{6}{3600}) X\ 42000}$ = 0.227 or 22.7 %

$\eta_{\text{Brake thermal efficiency}} = \frac{BP}{mf X\ C.V} = \frac{15}{(\frac{6}{3600}) X\ 42000}$ = 0.214 or 21.4 %

$\eta_{\text{Relative efficiency}} = \frac{\eta \text{ Indicated thermal efficiency}}{\eta \text{ air standard efficiency}}$

$$\eta \text{ air standard efficiency} = 1 - \frac{1}{CR^{\gamma-1}}$$

$$CR = \frac{V_C + V_S}{V_C}$$

$V_s = \frac{\pi d^2}{4} xl = \frac{\pi 7.5^2}{4} x10$ =441.5 cm^3 per stroke

$CR = \frac{100+441.5}{100}$=5.415

ŋ air standard efficiency $= 1 - \frac{1}{5.415^{1.4-1}}$=0.49

ŋ Relative efficiency=$\frac{0.214}{0.49}$ =0.437 or 43.7%

14. The following data refer to the test conducted on two stroke diesel engine, run for 20 minutes at full load, Mean effective pressure = 3 bar, speed=350RPM, net brake load =650N, fuel consumption =1.52kg, cooling water=160kg, water inlet temperature =30°C, water outlet temperature =52°C, air fuel ratio= 32, room temperature= 25°C, exhaust gas temperature=300°C, cylinder bore=200mm, stroke =280mm, brake drum diameter =100cm, calorific value of fuel=44000kJ/kg, steam formed per kg of fuel in the exhaust=1.35kg, specific heat of steam in exhaust=2.09kJ/kgK, specific heat of dry exhaust gas =1kJ/kgK, the pressure of exhaust = 1 bar, Determine: i) indicated power; ii) Brake power; iii) Mechanical efficiency and also write the energy balance on Minute basis and percentage. (May/June 2010)

Duration 20 min

p_m=3 bar;N=350rpm; net brake load =650N ie W$-S = 650N$;

m_f=1.52kg in 20 min =$\frac{1.52}{20}$kg/min=$\frac{1.52}{20x60}$kg/sec

m_w=160kg in 20 min=$\frac{1.62}{20}$kg/min; T_{co}=52°C ; AF=32

room temperature= 25°C ie T_R=25°C ; T_{ex}=300°C; D=200mm=0.2m; L=280mm=0.28m

brake drum diameter =100cm ie D_b=1m; CV=44000kJ/kg

steam formed per kg of fuel in the exhaust=1.35kg ie $m_s = 1.35kg/kg$ of fuel

specific heat of steam in exhaust=2.09kJ/kgK, $C_{ps} = 2.09kJ/kgK$

specific heat of dry exhaust gas =1kJ/kgK ie $C_{pg} = 1kJ/kgK$

the pressure of exhaust = 1 bar; pa=1bar

Soution

T=$\frac{(w-s)(D_b+d_r)}{2}$= $\frac{(650)(1+0)}{2}$ = 325 N-m

BP: $\frac{2\pi NT(Nm)}{60000}$ = $\frac{2\pi X(350)X325}{60000}$ =11.9 kW

IP=$\frac{p_m(kPa)LAn}{60}xk$, A=$\frac{\pi d^2}{4}$=0.0314m^2

n=N for 2 stroke hence n=350

IP=$\frac{3X10^2X0.28X0.0314X(350)}{60}x1$ = 15.39 kW

$\eta_{mech} = \frac{BP}{IP} = \frac{11.9\,X100}{15.39}$ = 77.3%

Heat Balance sheet:

m_f =1.52kg in 20 min Hence $m_f = \frac{1.52}{20}$kg/min

1. Heat input : m_f X CV =(1.52/(20)) X 44000
=3344 Kj/min

2. BP = 11.9 kWX60=714 Kj/min
3. Heat carried by cooling water: m_w C_p ($T_{co} - T_{ci}$)

m_w =160kg in 20min

Hence $m_w = \frac{160}{20x60}$

=(160/(20X60))X4.187x(52-30)

=12.28kWX60=736.9 Kj/min

4. Heat carried by Steam: mass of steam X enthalpy or m_sC_{ps} ($T_{go} - T_{gi}$)

steam formed per kg of fuel in the exhaust=1.35kg

Mass of steam formed : 1.35 x mass of fuel consumed per min =1.35 X(1.52/20) kg/min

=1.35 X(1.52/20) X2.09(300-25)=58.96Kj/min

5. Heat carried by exhaust gases : (m_f $(1 + AF)$+)-m_s) C_{pg} ($T_{ex} - T_R$)

:(1.52/(20))(32+1))-(1.35 X(1.52/20))) 1(300-25)

=661.48Kj/Min

6. Heat un account for : 1 – (2+3+4+5)

: 3344 – (714 + 736.9 + 58.96+661.48)

: 1172.66Kj/min

Energy input,	Kj/min	%	*Energy consumed,*	Kj/min	%
1. Heat input	3344	100	2. Energy consumed as brake power	714	21.35
			3. Energy carried by coolant	736.9	22.04
			4. Energy carried by Steam	58.96	1.76
			5. Energy carried by exhaust gases	661.48	19.78

			6. Unaccounted losses	1172.66	35.06
Total	3344	100	Total	3344	100

15. A 4 cylinder engine has the following data: Bore = 15cm, stroke = 15cm, Piston speed = 510m/min, BP= 60kW, Mech efficiency= 80%, Mep = 5bar, CV= 40000kJ/kg. Calculate i) Whether this is a two stroke or 4 stroke cycle engine (June/July 2009)

Data: D=15cm=0.15m; L=15cm=0.15m ; Mean piston speed =2LN=510m/s; BP= 60kW; $\eta_{mech} = 0.8$

$p_m = 5\ bar$; CV= 40000kJ/kg

$N = \frac{510}{2X0.15}$ = 1700 rpm

$\eta_{mech} = \frac{BP}{IP}$; 0.8= $\frac{60}{IP}$,

IP=75Kw ; IP=$\frac{p_m LAn}{60}$xk, kW A=$\frac{\pi d^2}{4}$=0.0177m^2

75=$\frac{5X10^2X0.15X0.0177Xn}{60}$X4cylinder

n=847.45 explosion per min

N =1700rpm given

Comparing n and N it may be observed that n =$\frac{N}{2}$

Hence engine is 4 stroke

16. Following data are available for SI engine, single cylinder stroke = 4. A:F=16:1, CV=45000kJ/kg, mech efficiency =80%. Air std. efficiency = 50%, relative efficiency =70%, stroke to bore = 1.5,suction condition = 1 bar, 30^{o}C, speed = 2500rpm, BP= 75kW. Calcualte i) compression ratio ii) indicated thermal efficiency iii) BSFC iv) Brake thermal efficiency v) Bore and stroke, assume vol efficiency = 80% (june/July 2009)

Data: AF=16; CV=45000kJ/kg; $\eta_{mech} = 0.8$; $\eta_{air\ std} = 0.5$; $\eta_{relative} = 0.7$;

stroke to bore = 1.5 ie $\frac{L}{D} = 1.5$; suction condition = 1 bar, 30^{o}C, $p_s = 1\ bar$; N=2500rpm; BP= 75kW

$$\eta \text{ air standard efficiency} = 1 - \frac{1}{CR^{\gamma-1}}$$

$0.5 = 1 - \frac{1}{CR^{1.4-1}}$

CR=5.66

$\eta_{\text{Relative efficiency}} = \frac{\eta \text{ Indicated thermal efficiency}}{\eta \text{ air standard efficiency}}$

$\eta_{\text{Indicated thermal efficiency}}$=0.7X0.5 = 0.35

$\eta_{\text{Indicated thermal efficiency}} = \frac{IP}{mf X\ C.V}$

$\eta_{mech} = \frac{BP}{IP}$; 0.8= $\frac{75}{IP}$; IP=93.75Kw

0.35=$\frac{93.75}{mf X\ 45000}$; m_f =0.00595 kg/s

$\eta_{\text{Brake thermal efficiency}} = \frac{BP}{mf X\ C.V} = \frac{75}{0.00595 X\ 45000}$ = 0.28 or 28 %

$mf = 0.00595 \frac{\text{kg}}{\text{s}} = 0.00595\text{x}3600\text{kg/hr}$

BSfc=$\frac{mass\ of\ fuel\ used\ in\ kg/hr}{BP\ in\ kW}$0.00595X3600/75=0.2856 kg/BP hr

$V_{act=}$ mass of air/ Density of air

Mass of air= Fuel consumption X Air fuel ratio

Mass of air=(0.00595)X22 =0.1309 kg/s

$\text{Density of air} = \frac{\text{P}}{\text{RT}}$; $\text{Density of air} = \frac{100}{0.278x303}$ =1.187 kg/m^3

V_{act}=0.1309/1.189=0.11 m^3/s

$Vs = \frac{0.0126}{0.8}$ =0.9545

$V_s = \frac{\pi d^2}{4} xlx(\frac{N}{2x60}) = \frac{\pi d^2}{4} x1.5dx(\frac{2500}{2x60})$ =0.9545 kg/m^3

d=0.339m, L=1.5X0.339=0.508m

17. A 4 cylinder, 4 stroke SI engine 90mm bore and 90 mm stroke was tested at constant speed. The fuel supply was fixed at 0.0008 kg/sec and plug of 4 cylinders were successively short circuited without change of speed. The power measurement was as follows:

With all cylinders working 16.25kW

With 1st cylinder cut off 11.55kW

With 2nd cylinder cut off 11.65kW

With 3rd cylinder cut off 11.7kW

With 4th cylinder cut off 11.5kW

Find Indicated power, thermal efficiencies and relative efficiency if clearance volume is 65cm^3 and CV is 42500kJ/kg.(Jan 2010)

$(ip)_{total} = ip_1 + ip_2 + ip_3 + ip_4 = (BP_t - BP_{234}) + (BP_t - BP_{134}) + (BP_t - BP_{124}) + (BP_t - BP_{123})$

$= 4B_t - (B_1 + B_2 + B_3 + B_4) = 4 \times 16.25 - (11.55 + 11.65 + 11.7 + 11.5)$

=18.6 Kw

$$\eta_{mech} = \frac{BP}{IP} = \frac{16.25}{18.6} x100 = 87.37\%$$

$\eta_{\text{Indicated thermal efficiency}} = \frac{IP}{mf X\ C.V} = \frac{18.6}{0.0008 X\ 42500}$ = 0.547 or 54.7 %

$\eta_{\text{Brake thermal efficiency}} = \frac{BP}{mf X\ C.V} = \frac{16.25}{(0.0008) X\ 42500}$ = 0.478 or 47.8 %

$\eta_{\text{Relative efficiency}} = \frac{\eta \text{ Indicated thermal efficiency}}{\eta \text{ air standard efficiency}}$

$$\eta \text{ air standard efficiency} = 1 - \frac{1}{CR^{\gamma-1}}$$

$$CR = \frac{V_C + V_S}{V_C}$$

$V_s = \frac{\pi d^2}{4} x l = \frac{\pi 9^2}{4} x 9$ =572.27 cm^3/stroke

$CR = \frac{65+572.27}{65}$=9.8

$\eta \text{ air standard efficiency} = 1 - \frac{1}{9.8^{1.4-1}}$=0.6

$\eta_{\text{Relative efficiency}} = \frac{0.547}{0.6}$ =0.914 or 91.4%

18.A four cylinder four stroke petrol engine has a bore 57 mm and a stroke 90 mm. Its rated speed is 2800 rpm and is tested at this speed against a brake which has a torque arm of 0.356m. The net brake load is 155 N and fuel consumption is 6.74 lit/hr. The specific gravity of the petrol use is 0.735 and CV is 44200kJ/kg. A Morse test is carried out and the cylinders are cut in the order 1,2,3,4 with corresponding brake loads are 111,106.5,104.2 and 111 N respectively. Calculate bmep, brake thermal efficiency, specific fuel consumption, mechanical efficiency and imep.

Data: D=57mm=0.057m; L=90mm=0.09m; N=2800rpm; Torque Arm= 0.356m ie R=0.356m

The net brake load is 155 N ; $W_{1,2,3,4}$=155N ;

fuel consumption is 6.74 lit/hr V_f=6.74 lit/hr

The specific gravity of the petrol use is 0.735 ie S=0.735; CV is 44200kJ/kg

A Morse test is carried out and the cylinders are cut in the order 1,2,3,4 with corresponding brake loads are 111,106.5,104.2 and 111 N respectively

Ie $W_{2,3,4} = 111N$; $W_{1,3,4} = 106.5N$; $W_{1,2,4} = 104.2N$; $W_{1,2,3} = 111N$

Solution

Torque = Brake Load x Torque Arm Length

Torque with all cylinders are working

$T_{1,2,3,4}$=$W_{1,2,3,4}$xR=155x0.356=55.18Nm

T_{234}=$W_{2,3,4}$xR =111x0.356=39.516Nm

T_{134}=$W_{1,3,4}$xR =106.5x0.356=37.914Nm

T_{124}=$W_{1,2,4}$xR= 104.2x0.356=37.095Nm

T_{123}=$W_{1,2,3,}$xR= 111x0.356=39.516Nm

$BP_{1,2,3,4}$: $\frac{2\pi N T_{1,2,3,4}}{60000}$ = $\frac{2\pi X(2800)X55.18}{60000}$ =16.17 kW

BP_{234}: $\frac{2\pi N T_{2,3,4}}{60000}$ = $\frac{2\pi X(2800)X39.516}{60000}$ =11.5 kW

BP_{134}: $\frac{2\pi N T_{1,3,4}}{60000}$ = $\frac{2\pi X(2800)X37.914}{60000}$ =11.11 kW

BP_{124}: $\frac{2\pi N T_{1,2,4}}{60000}$ = $\frac{2\pi X(2800)X37.095}{60000}$ =10.87 kW

BP_{123}: $\frac{2\pi N T_{1,2,3,}}{60000}$ = $\frac{2\pi X(2800)X39.516}{60000}$ =11.5 kW

(ip)total =ip1 +ip2 +ip3 +ip4 = ($BP_{1,2,3,4}$ – BP234)+($BP_{1,2,3,4}$– BP134)+($BP_{1,2,3,4}$ – BP124)+($BP_{1,2,3,4}$ – BP123)

= (16.17 –11.5) +(16.17 –11.5)+ (16.17 –11.11)+(16.17 –10.87) +(16.17 –11.11)

=19.7 Kw

$$\eta_{mech} = \frac{BP}{IP} = \frac{16.17}{19.7} x100 = 82.08\%$$

$$Fuel\ consumption = \frac{6.75}{1000x3600}x0.735x1000 = 0.00138kg/s$$

ŋ Brake thermal efficiency=$\frac{BP}{mfX\ C.V}$ = $\frac{16.17}{(0.00138)X\ 44200}$ = 0.265 or 26.5 %

$$SFC = \frac{m_f}{BP} = \frac{0.00138x3600}{16.17} = 0.307\frac{kg}{kWh}$$

BP(kW)=$\frac{p_{mb}(kPa)LAn}{60}xk$, A=$\frac{\pi d^2}{4}$=0.00255m^2 ; k=number of cylinder =4

n=number of working stroke theoretical=$\frac{N}{2}$ for 4 stroke

n=$\frac{N}{2} = \frac{2800}{2}$=1400working strokes per minute

BP=$\frac{p_{mb}LAn}{60}xk$ ie 16.17=$\frac{p_{mb}x0.09x0.00255x1400}{60}x4$

$Brake\ mean\ effective\ pressure\ p_{mb}$=$\frac{60X16.17}{0.09X0.00255X1400x4}$= 7.549x10^2 kPa

IP=$\frac{p_m LAn}{60}xk$, A=$\frac{\pi d^2}{4}$=0.00255m^2 ; k=number of cylinder =4

19.7=$\frac{p_m x0.09x0.00255x1400}{60}x4$

$Indicated\ mean\ effective\ pressure\ p_m$=$\frac{60X19.7}{0.09X0.00255X1400}$=9.19$x$10^2 kPa

19. A rope brake was used to measure the brake power of a single cylinder, 4 stroke petrol engine. It was found that the torque due to brake load is 175Nm and the engine makes 500rpm. Determine the brake power developed by the engine in horse power unit.(July2011).

BP: $\frac{2\pi NT}{60000}$ = $\frac{2\pi X(500)X175}{60000}$ =9.16 kW

One Horse power =735W=0.735kW

BP= (9.16/0.735)=12.5HP

20.A test on two stroke engine gave the following results at full load: Speed=350rpm, Net brake load=65kgf, mep=3bar, fuel consumption=4kg/hr, jacket cooling water flow rate=500kg/hr, jacket cooling water temperature rise=20^{0}C, air used per kg of fuel=32kg, cylinder diameter=22cm, stroke=28cm, effective brake drum diameter=1m, CV=43MJ/kg, C_p gases=1kJ/kgK, exhaust temp=400^{0}C, room temperature=20^{0}C.Find the mechanical efficiency and heat balance sheet in minute and % basis (june2012).

Data:N=350rpm, Net brake load=65kgf ie W$-S = 65kgf = 65x9.81N$; $p_m = 3\ bar$;

$m_f = 4kg/hr$=$\frac{4}{60}\frac{kg}{min} = \frac{4}{3600}\frac{kg}{sec}$; m_w=$500kg/hr$=$\frac{500}{60}\frac{kg}{min}$

jacket cooling water temperature rise=20^0C ie $T_{co} - T_{ci}=20^0C$

air used per kg of fuel=32kg ie AF=32; D=22cm=0.22m; L=28cm=0.28m

effective brake drum diameter=1m ie D_b=1m; CV=43MJ/kg=43000kJ/kg;

C_{pg} =1kJ/kgK; $T_{ex}=400^0C$; $T_R=20^0C$

Solution:

$$T=\frac{(w-s)(D_b+d_r)}{2}=\frac{(65x9.81)(1+0)}{2} = 318.83 \text{ N-m}$$

$$BP(kW): \frac{2\pi NT(Nm)}{60000} \text{ where } T \text{ is } Nm$$

$$BP: \frac{2\pi NT(Nm)}{60000} = \frac{2\pi X(350)X318.83}{60000} = 11.7 \text{ kW}$$

$$IP=\frac{p_m LAn}{60} xk,\ A=\frac{\pi d^2}{4}=0.038m^2$$

n=N for 2 stroke engine, hence n=350

$$IP=\frac{3X10^2X0.28X0.038X(350)}{60}x1 = 18.62 \text{ kW}$$

$$\eta_{mech} = \frac{BP}{IP}=\frac{11.7\ X100}{18.62} = 62.8\%$$

Heat Balance sheet:

1. Heat input : m_f X CV =(4/(60))(kg/min) X 43000(kJ/kg)
   =2866.67 Kj/min= $\frac{2866.67}{60} kW$

2. BP = 11.7 kWX60=702 Kj/min
3. Heat carried by cooling water: $m_w C_p (T_{co} - T_{ci})$
=(500/(60))X4.187x(20)
=697.8 Kj/min
4. Heat carried by exhaust gases : $(m_f + m_a) C_{pg} (T_{ex} - T_R)$
:((4+4x32)/(60)) 1(400-20)

=836kJ/Min

5. Heat un account for : 1 – (2+3+4)
: 2866.67 – (702 + 697.8 + 836)
: 630.87Kj/min

Energy input,	Kj/min	%	*Energy consumed,*	Kj/min	%
1. Heat input	2866.67	100	2. Energy consumed as brake power	702	24.5

			3. Energy carried by coolant	697.8	24.34
			4. Energy carried by exhaust gases	836	29.16
			5. Unaccounted losses	630.87	22
Total	2866.67	100	Total	2866.67	100

21.A test on single cylinder 4 stroke oil engine having bore 180mm and stroke 360mm gave the following results: Speed 290rpm, brake torque=392Nm, IMEP=7.2bar, oil consumption=3.5kg/hour, cooling water flow 270kg/hour, cooling water temperature rise=36^{0}C, air fuel ratio by weight=25, exhaust gas temperature=415^{0}C, barometric pressure=1.013bar, room temperature=21^{0}C, CV of fuel=45200kJ/kg, fuel contains 15% of hydrogen by weight, calculate:1. The indicated thermal efficiency 2.Volumetric efficiency based on the atmospheric conditions 3. Draw the heat balance sheet in kJ/min

Take R=0.287kJ/kgK,$C_{p\text{ for gases}}$=1.0035kJ/kgK, $C_{p\text{ for super-heated steam}}$=2.093kJ/kgK.

BP: $\frac{2\pi NT}{60000}$ = $\frac{2\pi X(290)X392}{60000}$ =11.89 kW

IP=$\frac{p_m LAn}{60} xk$, A=$\frac{\pi d^2}{4}$=0.025m^2

n=$\frac{N}{2}$ for 4 stroke n=$\frac{290}{2}$=145

IP=$\frac{7.2X10^2 X0.36X0.025X(145)}{60}$ x1= 15.66 kW

mf=3.5kg/hr=$\frac{3.5}{3600}$ kgs/s

$\eta_{\text{Indicated thermal efficiency}}$=$\frac{IP}{mfX\ C.V}$ = $\frac{15.66}{\frac{3.5}{3600}X\ 45200}$ = 0.356 or 35.6 %

$$\eta_v = \frac{V_{act}}{V_{the}}$$

$V_{act=}$ mass of air/ Density of air

Mass of air= Fuel consumption X Air fuel ratio

Mass of air=(3.5/3600)X25 =0.0243 kg/s

$$\text{Density of air} = \frac{\text{P}}{\text{RT}}$$

Density of air $= \frac{101.3}{0.287x294}$ =1.2 kg/m^3

$V_{act}=\frac{m}{\rho}$ 0.0243/1.2=0.0202 m^3/s

$V_{the}=\frac{\pi d^2}{4}xlx(\frac{N}{2x60})=\frac{\pi 0.18^2}{4}x0.36x(\frac{290}{2x60})$ =0.022 kg/m^3

In the above equation $\frac{N}{2}$ is used since engine is 4 stroke

$\eta_v = \frac{0.0202}{0.022}$ =0.9182

Heat Balance sheet:

1. Heat input : m_f X CV =(3.5/(60X60)) X 45200
=43.94kWX60=2636.67 Kj/min

2. BP = 11.89 kWX60=713.4 Kj/min
3. Heat carried by cooling water: $m_w C_p (T_{co} - T_{ci})$

m_w=270kg/hr = $\frac{270}{3600}$ kgs/s
=(270/(60X60))X4.187x(36)
=11.31kWX60=678.29 Kj/min

4. Heat carried by Steam: mass of steam X enthalpy or $m_s C_{ps} (T_{ex} - T_R)$

1 kg of hydrogen gives 9 kg of water

(0.15x3.5) H_2gives =0.15x3.5x9=4.725kg/hour=0.0013125kg/s

HC by steam=0.0013125X2.093(415-21)=1.082kWX60=64.94Kj/min

5.Heat carried by exhaust gases : $((m_f + m_a) - m_s) C_{pg} (T_{ex} - T_R)$

:(3.5/(60X60))(25+1))-0.0013125) 1.0035(415-21)

:9.48X60=568.52Kj/Min

6. Heat un account for : 1 – (2+3+4+5)
: 2636.67 – (713.4 + 678.29 + 64.94+568.52)
: 611.52Kj/min

Energy input,	Kj/min	%	*Energy consumed,*	Kj/min	%
1. Heat input	2636.67	100	2. Energy consumed as brake power	713.4	27.06
			3. Energy carried by coolant	678.29	25.73
			4. Energy carried by Steam	64.94	2.46

			5. Energy carried by exhaust gases	568.52	21.56
			6. Unaccounted losses	611.52	23.19
Total	2636.67	100	Total	2636.67	100

<u>Psychrometrics</u>

Air conditioning is a simultaneous control of temperature, humidity, cleanliness, odour and air circulation as required by the occupants of the space

Psychrometry is the study of properties of dry air and water vapor contained in moist air

<u>Dry Bulb Temperature (DBT)</u>: Actual temperature of Moist air is called Dry Bulb Temperature. This is measured by ordinary thermometer

<u>Wet Bulb Tempeature</u> <u>(WBT)</u>: The wet Bulb temperature is the temperature recorded by a thermometer when the bulb is enveloped by a cotton wick saturated with water

As the air stream flows past it, some water evaporates, taking the latent heat from the water soaked wick, thus decreasing its temperature. This decreased temperature is recorded in thermometer when the bulb is covered with wet cotton

If the saturated air flows past the thermometer it would not evaporate the water particles in the wicked cotton (since air is saturated with water) Hence there is no decrease in the temperature

Hence for saturated air WBT=DBT

<u>Wet Bulb Depression:</u> Difference between Dry Bulb Temperature and Wet Bulb Temperature is called Wet Bulb Depression

For saturated air wet Bulb depression is zero

<u>Dew Point Temperature(DPT):</u> is defined as the temperature at which the water contained in moist air get condensed

Dew Point Temperature is read from the steam table as saturation temperature corresponding to partial pressure of water vapor contained in moist air

<u>Specific Humidity or absolute humidity or Humidity ratio:</u>

It is defined as the ratio of the mass of water vapor in a given volume of the mixture to the mass of the dry air present in the mixture

$$\omega = \frac{m_v}{m_a}$$

$$= \frac{\frac{p_v V}{R_v T}}{\frac{p_a V}{R_a T}}$$

$$= \frac{p_v}{p_a} \times \frac{R_a}{R_v}$$

$$= \frac{p_v}{p_a} \times \frac{\frac{\bar{R}}{M_a}}{\frac{\bar{R}}{M_v}}$$

$$= \frac{p_v}{p_a} \, x \, \frac{M_v}{M_a}$$

But $p_a = p_t - p_v$ $M_a=29$; $M_v=18$

$$\omega = \frac{p_v}{p_t - p_v} \, x \, \frac{18}{29}$$

$\omega = 0.622\ x \frac{p_v}{p_t - p_v}$

Relative Humidity: is defined as the ratio of mass of water vapor m_w in a certain volume of moist air at a given temperature to the mass of water vapour m_w in the same volume of saturated air at the same temperature

$$\Phi = \frac{m_w}{m_s} = \frac{\frac{p_v V}{R_v T}}{\frac{p_{vs} V}{R_v T}}$$

$$= \frac{p_v}{p_{vs}}$$

Note that p_{vs} is read from steam table as the saturation pressure corresponding to Dry bulb temperature

Degree of saturation: is defined as the actual specific Humidity ω of given air to the specific humidity of saturated air at the same temperature of given air

$$\mu = \frac{0.622\ x \frac{p_v}{p_t - p_v}}{0.622\ x \frac{p_{vs}}{p_t - p_{vs}}}$$

$$= \frac{\frac{p_v}{p_t - p_v}}{\frac{p_{vs}}{p_t - p_{vs}}}$$

$$= \frac{p_v}{p_{vs}}\ x\ \frac{p_t - p_{vs}}{p_t - p_v}$$

$$\mu = \phi\ x \frac{p_t - p_{vs}}{p_t - p_v}$$

$$= \phi\ x\ \frac{1 - \frac{p_{vs}}{p_t}}{1 - \frac{p_v}{p_t}}$$

$$\phi = \frac{\mu}{1 - \frac{p_{vs}}{p_t}(1-\mu)}$$ **This is the relation between Degree of saturation and relative humidity**

Relation between Absolute and relative humidity

$$\phi = \frac{p_v}{p_{vs}}$$

$$p_v = \varphi p_{vs}$$

But

$$\omega = 0.622\ x \frac{\varphi p_{vs}}{p_t - p_v}$$

$$\phi = 1.6\omega \frac{p_t - p_v}{p_{vs}}$$

Carrier Equation

When DBT and WBT is given partial pressure of water vapor is calculated by Carrier equation

$$\boldsymbol{p_v = \frac{(p_{vs})_{WBT} - (p_t - (p_{vs})_{WBT})(T_{DBT} - T_{WBT})}{1547 - 1.44T_{WBT}}}$$

Enthalpy of Moist Air

Enthalpy of Moist Air = Enthalpy of dry Air + Enthalpy of water vapor associated

$h = h_{air} + \omega h_{vapor}$

$h_{air} = C_{pa}T_{DBT}$ where C_{pa}=1.005kJ/kgK

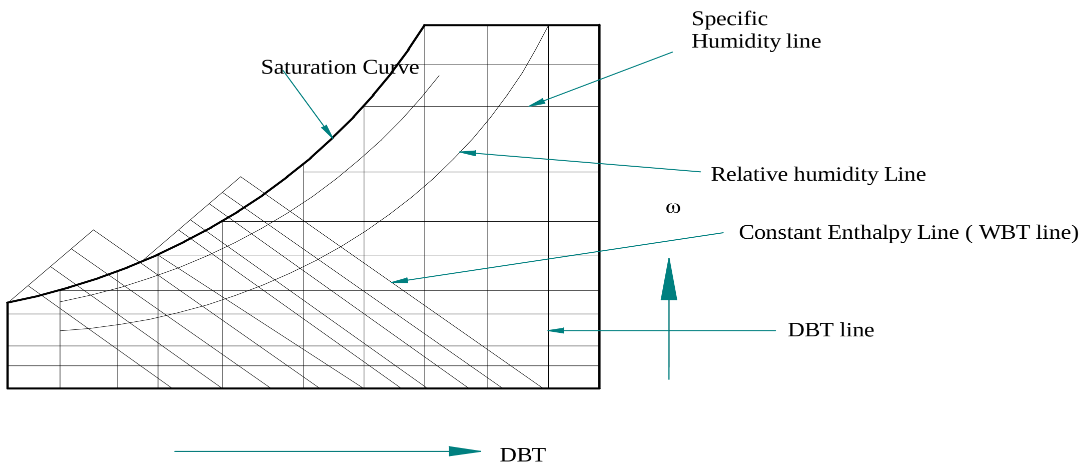
$h_{vap} = h_g + C_{pv}(T_{DBT} - T_{DPT})$ where C_{pv}=1.88 and h_g=2500kJ/kg

Generally T_{DPT} is neglected

Hence $h_{vap} = 2500 + 1.88T_{DBT}$

Hence **Enthalpy of Moist Air= 1.005T$_{DBT}$ +ω(2500+1.88T$_{DBT}$)kJ/kg of dry air**

<u>Psycrometric Chart</u>

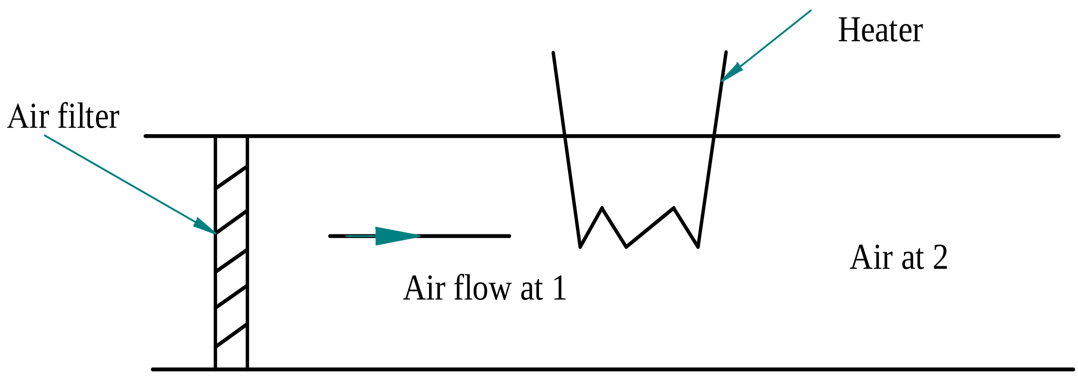


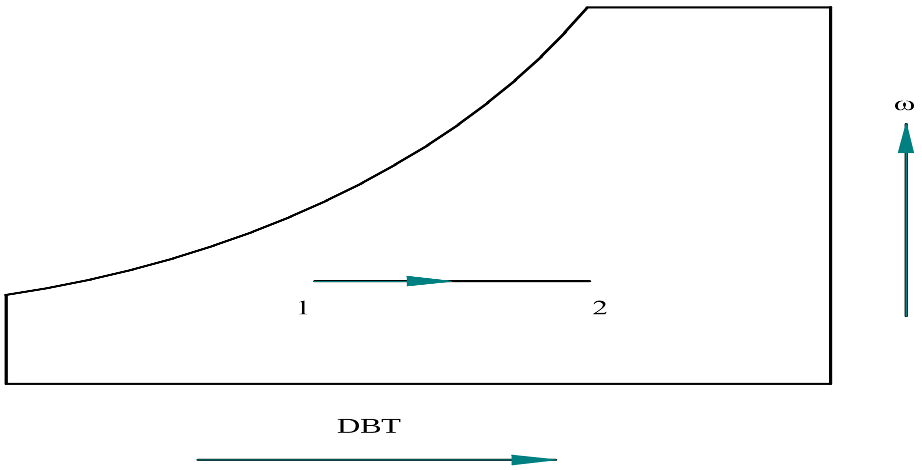
Psychrometric Chart

<u>Pshycrometric Processes:</u>

<u>Sensible Heating:</u>

Heating of air with no change in specific humidity is called sensible Heating. This can be achieved by passing the air over a heating coil like electrical resistance heater or steam coils. DBT increases while specific humidity remains constant.





By Pass factor of Heating coil:

When one kg of air is made flow over the heating coil in a conduit some amount of air (Bkg ie fraction of one kg of air) escapes without contacting coil. Due to this temperature of air after the heating coil is less than the heating coil temperature due to mixing of air stream which made contact with heating coil and air stream without contacting (By pass) heating coil.

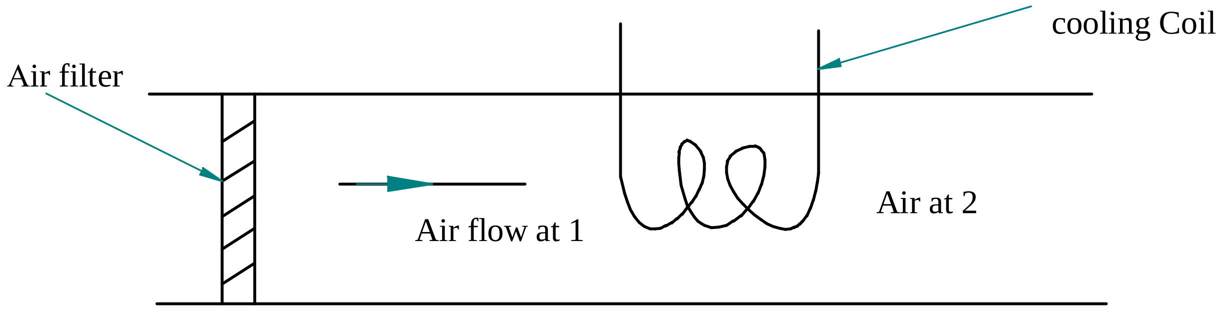
Balancing the enthalpies

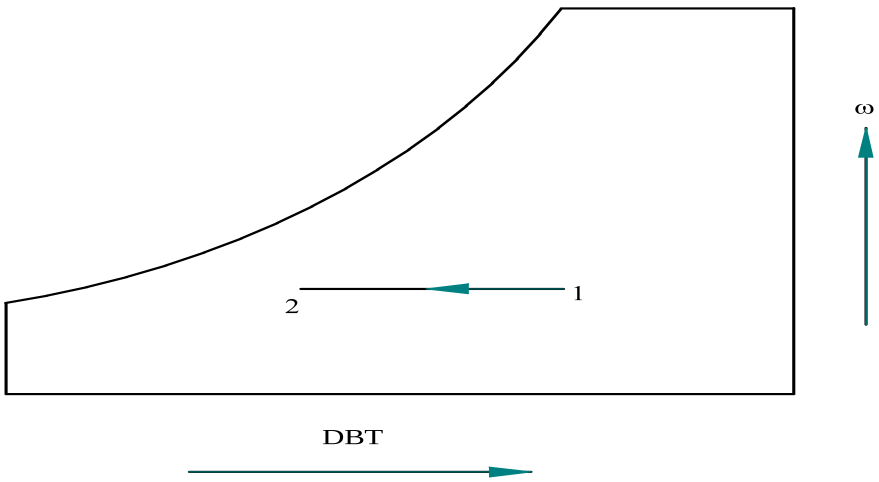
$BC_pT_2'+(1-B)\ C_pT_1 = C_pT_2$

$B=\frac{T_2'-T_2}{T_2'-T_1}$ where T_2 is the temperature just after heating coil and T_2' is the temperature of heating coil and T_1 is the initial temperature

Sensible Cooling:

Cooling of air with no change in specific humidity is called sensible cooling. This can be achieved by passing air over cooling coil like evaporating coil of refrigeration cycle. DBT decreases while specific humidity remains constant





By Pass factor of cooling coil:

When one kg of air is made flow over the cooling coil in a conduit some amount of air (Bkg ie fraction of one kg of air) escapes without contacting coil. Due to this temperature of air after the cooling coil is more than the cooling coil temperature due to mixing of air stream which made contact with cooling coil and air stream without contacting (By pass) cooling coil.

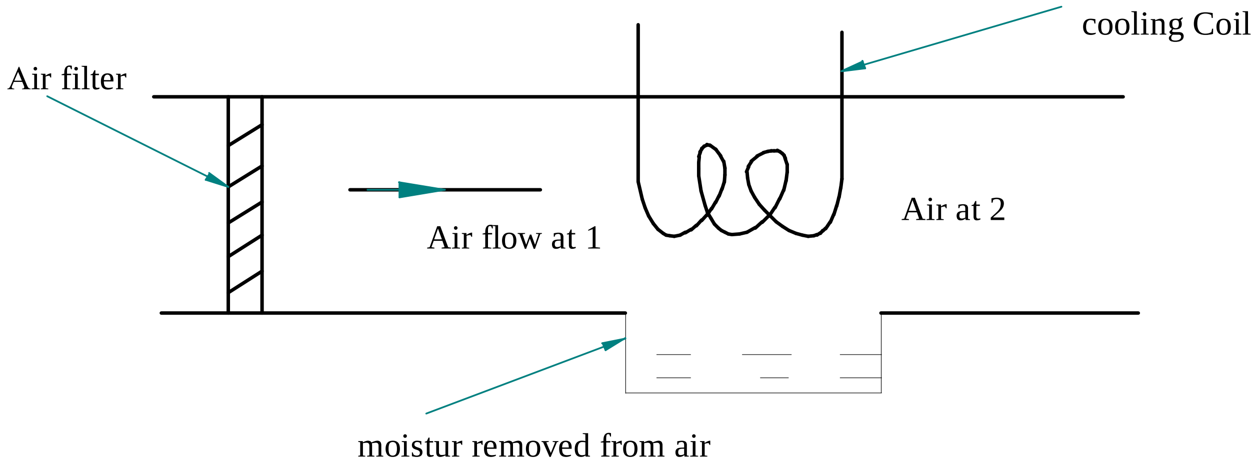
Balancing the enthalpies

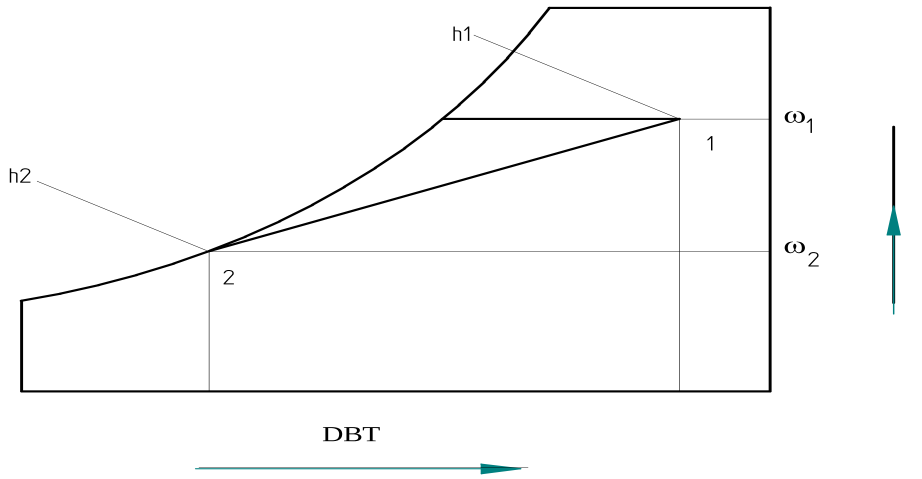
$BC_pT_2'+(1-B)\ C_pT_1 = C_pT_2$

$B=\frac{T_2-T_2'}{T_1-T_2'}$ where T_2 is the temperature just after heating coil and T_2' is the temperature of heating coil and T_1 is the initial temperature

Cooling and Dehumidification:

This process involves lowering the DBT temperature and specific humidity. Here air is cooled below the Dew point temperature of air. During this process air is cooled till it reaches DPT of air. Further cooling of air condenses the water vapor and water particles separates out from the moist air. Hence specific humidity decreases.





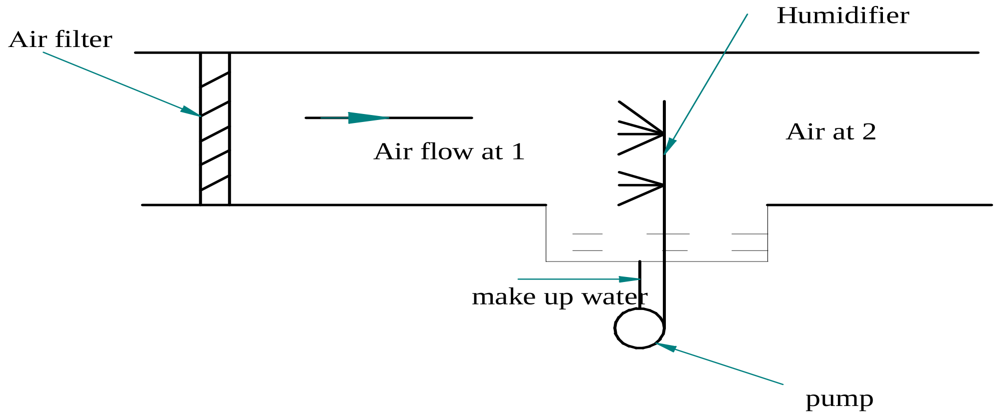
Air may be cooled and dehumidified a) by placing the evaporator coil across the air flow ii) by circulating chilled water or brine in a tube placed across the air flow iii) by sparaying chilled eater to air in the form of mist

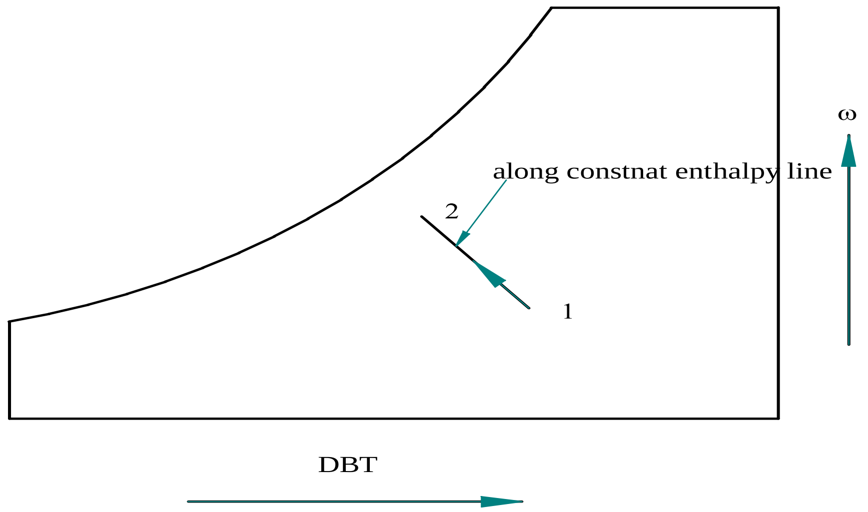
By Pass factor of cooling coil:

When one kg of air is made flow over the cooling coil in a conduit some amount of air (Bkg ie fraction of one kg of air) escapes without contacting coil. Due to this temperature of air after the cooling coil is more than the cooling coil temperature due to mixing of air stream which made contact with cooling coil and air stream without contacting (By pass) heating coil.

Cooling with Adiabatic Humidification of Air (Adiabatic Saturation Process):

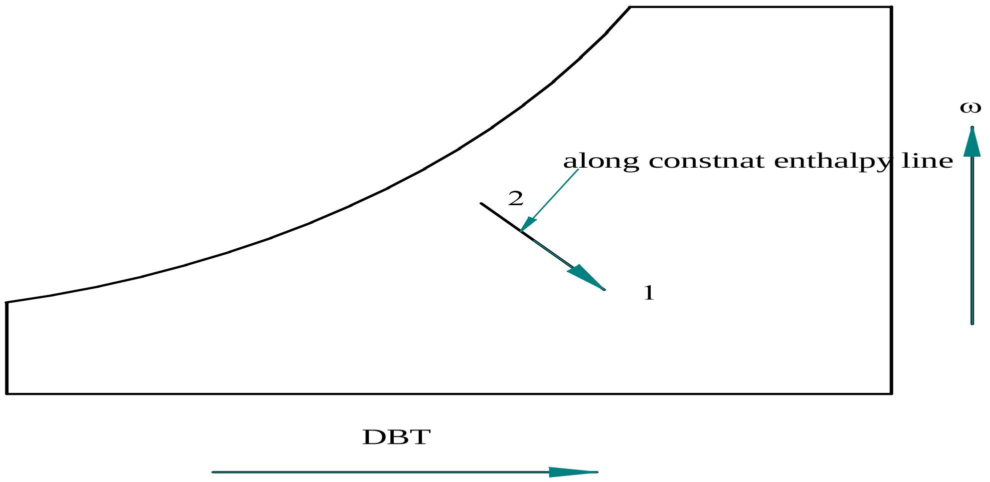
The re-circulated water is sprayed into the air in an insulated chamber. A part of it evaporates in trying to saturate the air. The heat required for the evaporation of water is removed from the air itself which results in decrease its temperature. If the air leaving the humidifier becomes saturated with water, then the temperature of air is called adiabatic saturation temperature.





Chemical Dehumidification:

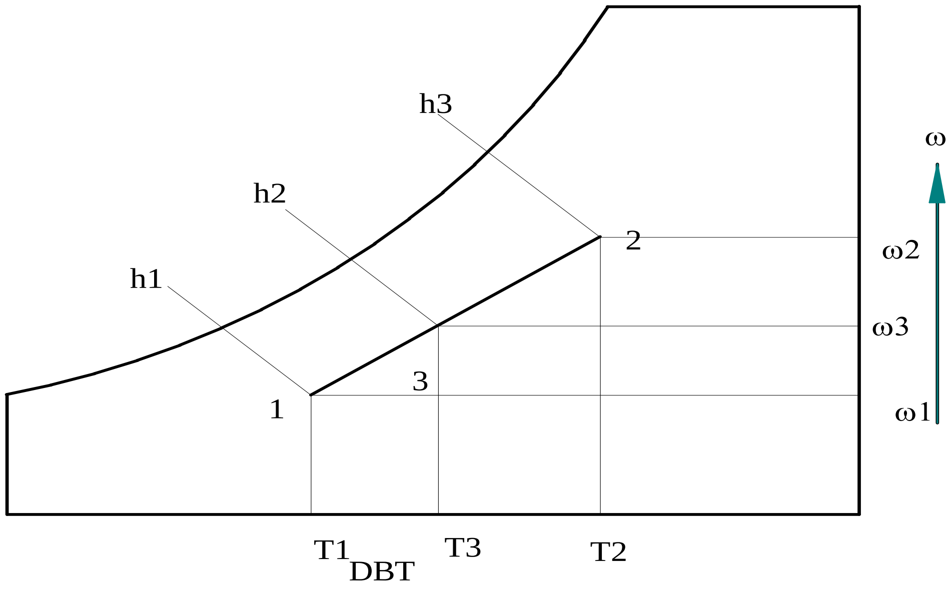
Air can be humidified by passing it over certain chemicals which have an affinity for moisture. During this process of condensation of water vapor in the air liberates latent heat of evaporation into air causing increase in Dry bulb temperature of air

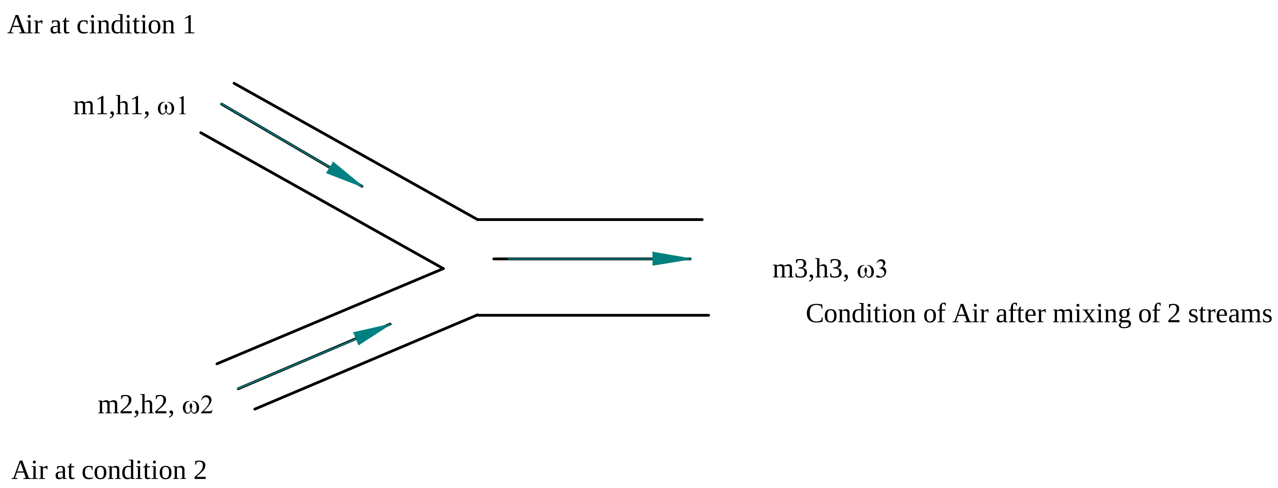


Mixing of Air Streams:

When air at one condition is mixed with air at another condition, the condition of the final mixture is obtained on the psychrometric chart by a point on the line joining the points indicating the orginal conditions ie 12

The exact location of the point depends upon the relative masses of dry air in the starting condition





m3=m1+m2+m3

$m_1h_1 + m_2h_2 = (m_1+m_2)h_3$

$$\frac{m_1}{m_2} = \frac{h_3 - h_2}{h_1 - h_3}$$

$m_1\omega_1 + m_2\omega_2 = (m_1+m_2)\omega_3$

$$\frac{m_1}{m_2} = \frac{\omega_3 - \omega_2}{\omega_1 - \omega_3}$$

Final state is to be found by dividing the lie 1-2 into segments proportional to the relative masses of dry air before mixing takes place.

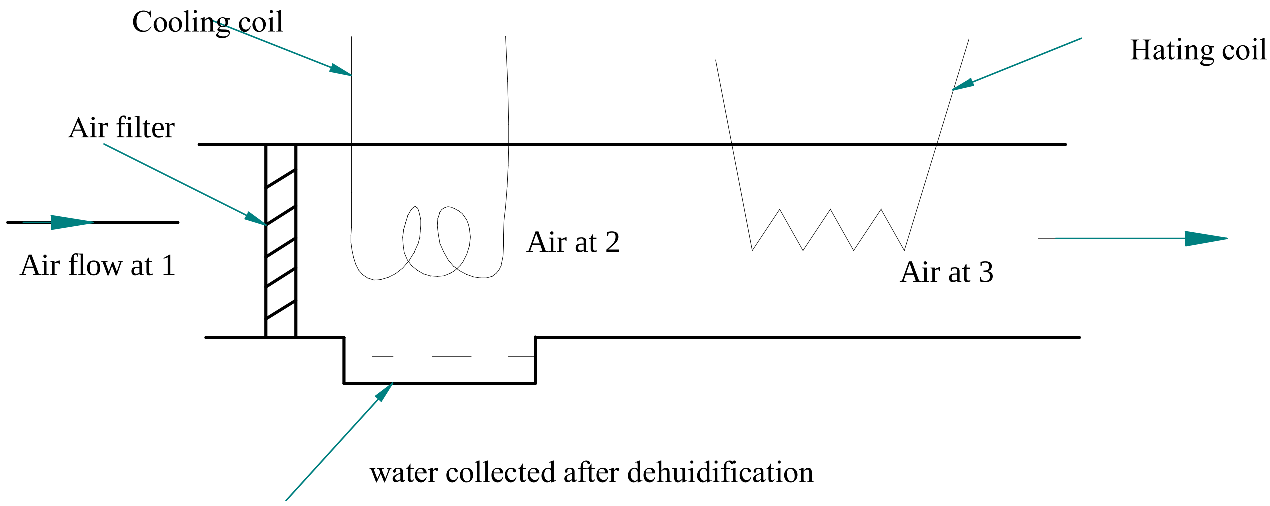
Comfort Air conditioning:

The following factor determine the comfort feeling of the occupants in an air conditioned space

1. Supply of Oxygen : Normally each person requires nearly 0.65m^3 of oxygen per hour and produces 0.2$m^3$of CO_2 . To comfort air conditioning CO_2 level in the should be maintained below 0.6%

2. Heat Removal: The Air conditioned space should absorb heat dissipated by persons and appliances and reject heat to the surroundings to maintain the temperature of space within the comfort zone DBT=25+1 to 25-1
3. Moisture Removal: Moisture loss of upto 50% from human body is commonly observed. The air conditioned space to be maintained Relative humidity which is comfortable to occupants ie RH 50%+5 to 50%-5

Summer Air conditioning for Hot and wet weather

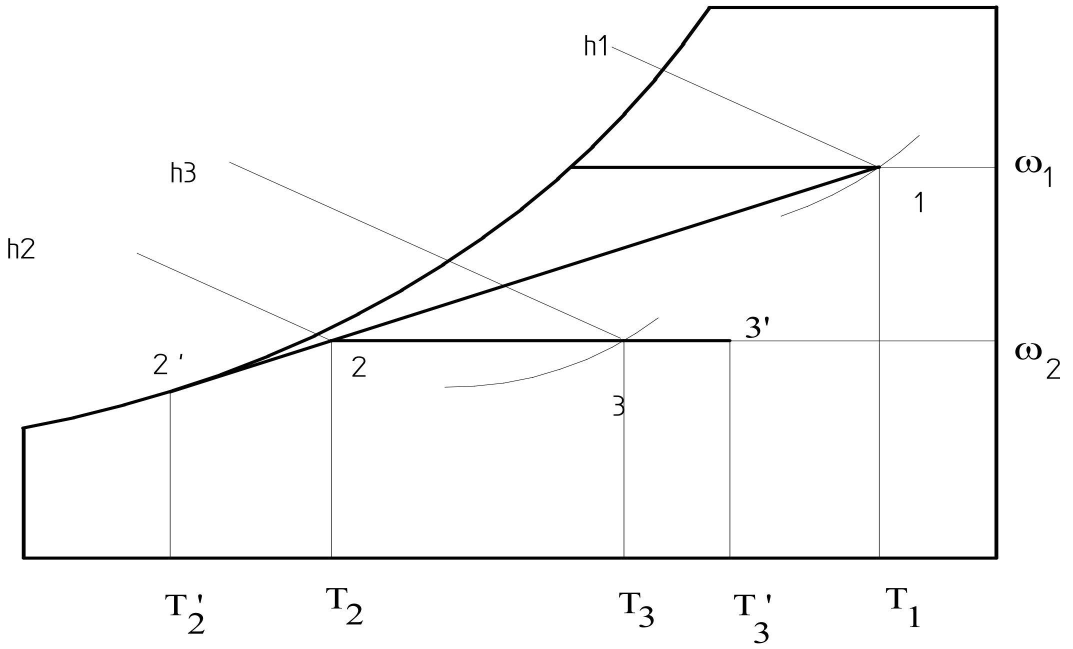


The outdoor condition is first passed over the cooling coil so that it gets dehumidified and cooled and comes out at point 2. The process is represented in psychrometric chart by 12 (Actual) Ideal process is represented by 12' where T_2' is the Dew point temperature of cooling coil.

The capacity of cooling coil is= $\frac{m_{a}(h_1-h_2)}{3.5}$ Ton of refrigeration

The air leaving the cooling coil is then passed over the resistance heating coil to get the required comfort condition. It is represented by 23 on psychrometric chart

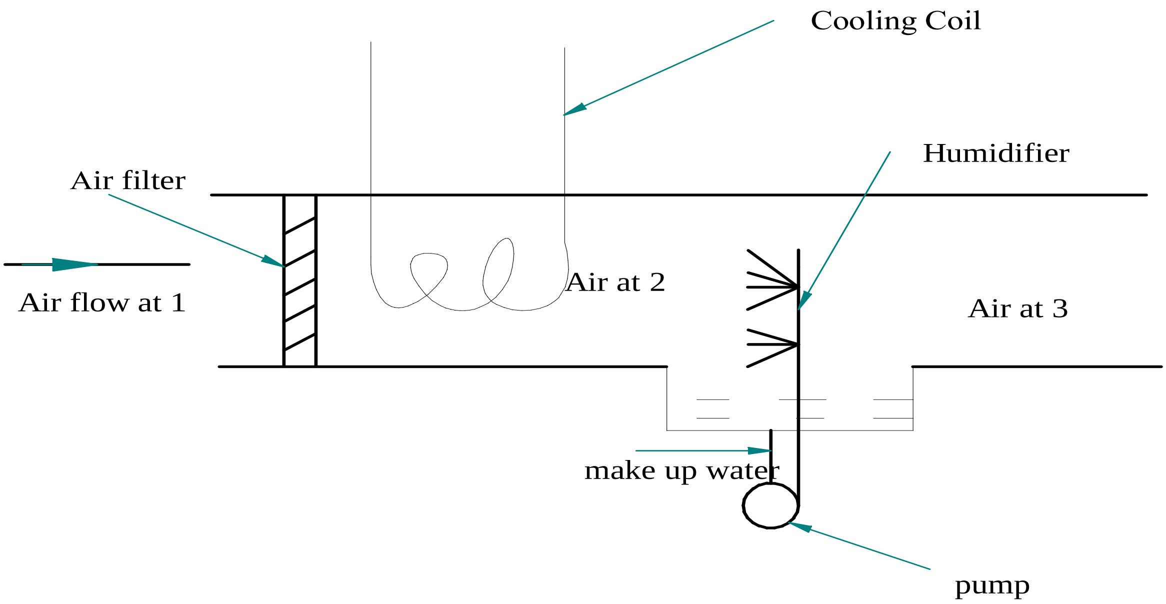
The capacity of heating coil = m_a (h_3- h_2) kW



By pass factor of cooling coil $= \frac{2\,2'}{1\,2}$

By pass factor of heating coil $= \frac{3\,3'}{2\,3'}$

Summer air conditioning for Hot and Dry weather

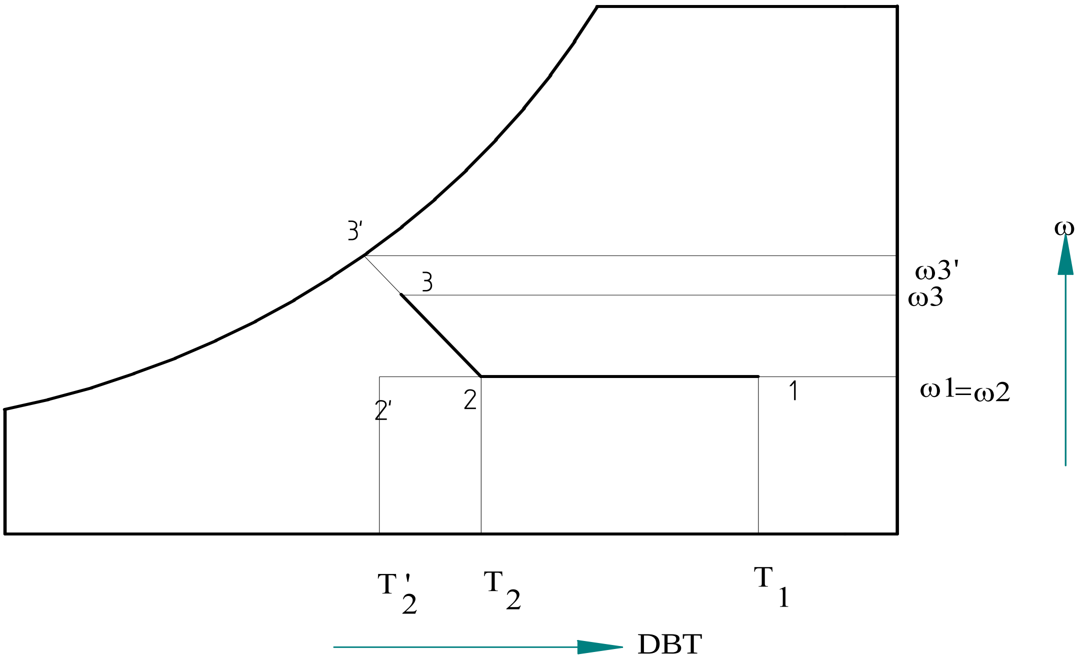


Air is first cooled from outdoor condition (Point 1) to point 2 by passing over a cooling coil Cooling process is represented in Psychrometric chart by 12 (Actual Process) where as 12' is ideal process

The capacity of cooling coil is= $\frac{m_a\,(h_1 - h_2)}{3.5}$ Ton of refrigeration

The air coming out of the coil at point 2 is passes into the adiabatic humidifier and the required conditioned leaves the humidifier at point 3. This process is represented in psychrometric chart by 23 and ideal process is represented by 23’

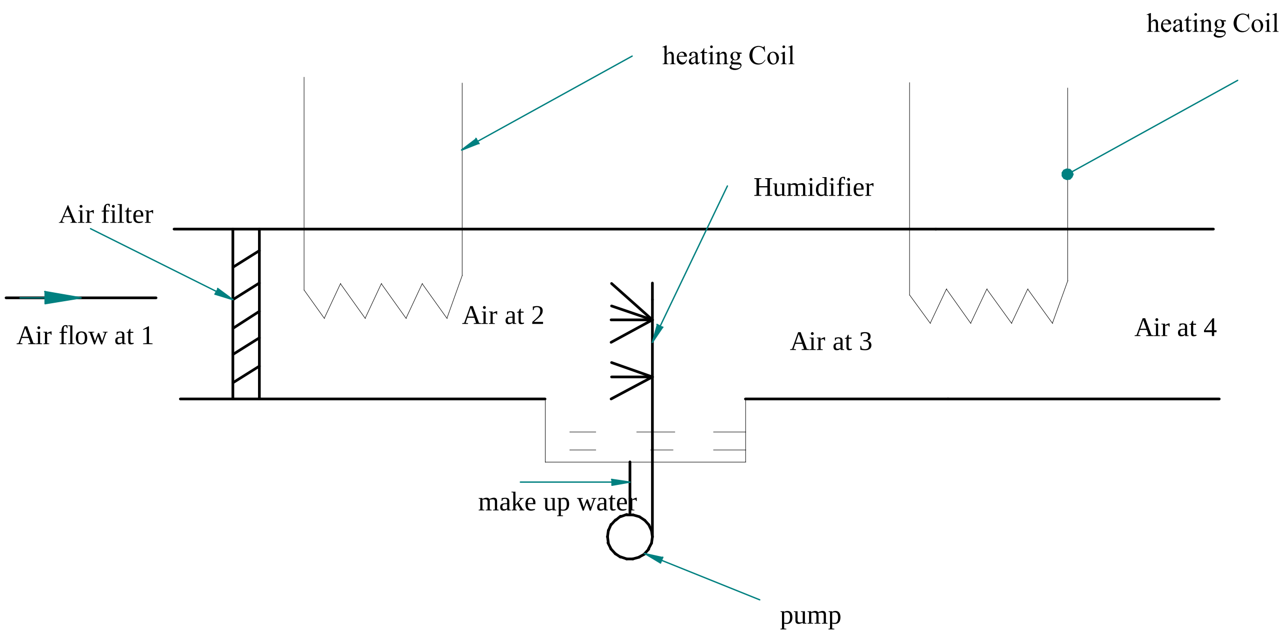
The capacity of Humidifier = $m_a (\omega_3 - \omega_2)$ kg/sec



By pass factor of cooling coil = $\frac{2\,2'}{1\,2'}$

Humidifier Efficiency = $\frac{3\,3'}{2\,3}$

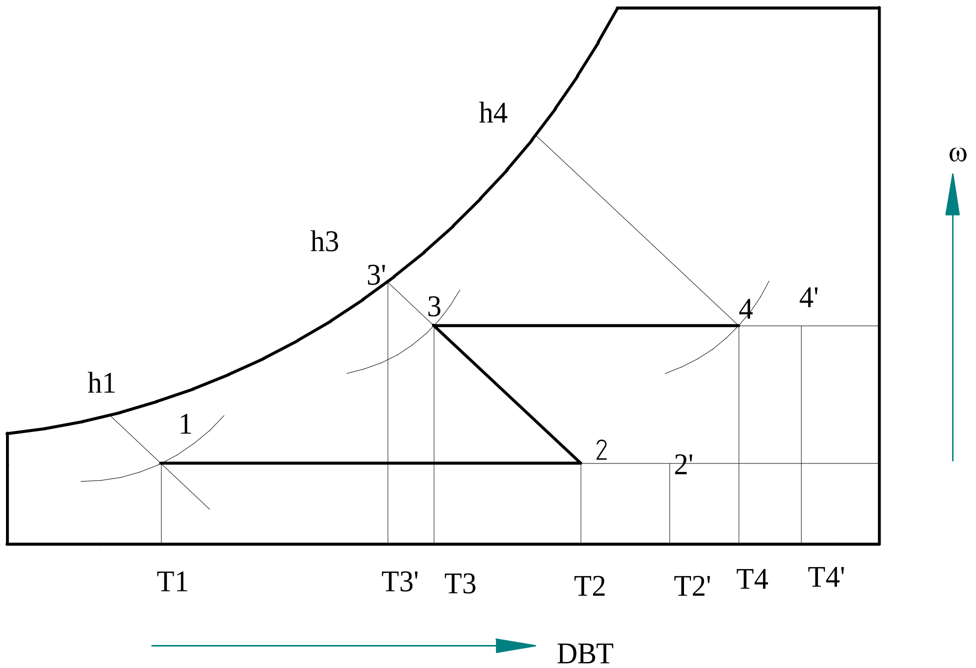
Winter Air conditioning:



The air is first passed over a preheater. The process is represented by 12 in psychrometric process. Ideal process is represented by 12’

It is then passed through the humidifier where air absorbs moisture and humidity of air increases as enthalpy remains constant. This represented by 23 in psychrometric chart.

and finally through a second heater where it is sensibly heated to achieve the desired condition. This represented in psychrometric chart by 34



The capacity of heating coil 1= m_a (h_2- h_1) kW

The capacity of Humidifier = m_a (ω_3-ω_2) kg/sec

The capacity of heating coil 2= m_a (h_4- h_3) kW

By pass factor of heating coil 1 = $\frac{2\,2'}{12'}$

By pass factor of heating coil 2 = $\frac{4\,4'}{34'}$

1. Moist air at 30°C , 1.01325 bar has a relative humidity of 80%. Determine without using the psychrometric chart, (i) partial pressure of water vapor (ii) specific humidity (iii) specific volume and (iv) Dew point temperature

<u>Solution:</u>

i) <u>Partial pressure of water vapor</u>

$\Phi = \frac{p_v}{p_{vs}}$

P_{vs} is the saturation pressure at DBT

Saturation pressure at 30°C is 0.042415 bar

$0.8 = \frac{p_v}{0.042415}$

P_v=0.033932 bar

ii) Specific Humidity:

$\omega = 0.622 \frac{p_v}{p_t - p_v}$

$= 0.622 \frac{0.033932}{1.01325 - 0.03392}$

= 0.02155

iii) Specific volume:

Density $\rho = \frac{(p - p_v)}{RT}$

$= \frac{(1.01325 - 0.042415)10^2}{0.287 \, x \, 303}$

$= 1.1164 \text{kg/m}^3$

Specific volume $= \frac{1}{\rho}$

$= 0.89573 \text{ m}^3\text{/kg}$

iv) Dew point temperature:

Is the saturation temperature at p_v= 0.033932 bar ie 26^oC

2. Atmospheric air at 101.325kPa has 30^oCDBT and 15^oDPT. Without using the psychometric chart , using the property values from the tables, calculate i) partial pressure of air and water vapor ii) specific humidity and iii) relative humidity iv) vapor density and v) enthalpy of moist air

Solution

i) partial pressure of air and water vapor

The partial pressure of water vapor at DPT (15^oC) = 0.017039bar

P_v=0.017039 bar

Partial pressure of air $P_a = P_t - P_v$ =1.01325 – 0.017039

P_a =0.996211 bar

ii) Specific Humidity

$\omega = 0.622 \frac{p_v}{p_t - p_v}$

$= 0.622 \frac{0.017309}{1.01325 - 0.017039}$

= 0.010638 kg/kg of dry air

iii) relative humidity

$\Phi = \frac{p_v}{p_{vs}}$

P_{vs} is the saturation pressure at DBT

Saturation pressure at 30^{o}C is 0.042415 bar ie P_{vs}=0.042415

$\Phi = \frac{\mathbf{0.017039}}{0.042415}$ =0.40172

Φ = 40.172%

iv) Vapor density

Vapor Density $\rho = \frac{p_v}{R_v T}$

$R_v = \frac{\bar{R}}{M} = \frac{8.314}{18}$ =0.46188 as Mol wt of water 2+16 =18

Density $\rho = \frac{(0.017039)10^2}{0.46188\ x\ 303}$

= 0.012175kg/m^3

v) <u>enthalpy of moist air</u>

h=1.005 T_{DBT} +ω(2500+1.88T_{DBT})

=1.005 x30 +0.010638(2500+1.88x30)

=57.3449kJ/kg

3. The DBT and WBT of atmospheric air at 1 atm (1.01325bar) pressure are measured with sling psychrometer and determined to be 25^{o}C and 15^{o}C respectively. Determine i) specific humidity ii) relative humidity iii) enthalpy of moist air

<u>Solution</u>

i) specific humidity

$$p_v = (p_{vs})_{WBT} - \frac{(p_t - (p_{vs})_{WBT})(T_{DBT} - T_{WBT})}{1547 - 1.44T_{WBT}}$$

$(p_{vs})_{WBT}$ is the saturation temperature at WBT ie 0.01704 bar from steam table

$$p_v = 0.01704 - \frac{(1.01325 - 0.01704)(25 - 15)}{1547 - 1.44x15}$$

=0.01057 bar

ii) <u>Specific Humidity</u>

ω $=0.622 \frac{p_v}{p_t - p_v}$

$=0.622 \frac{0.01057}{1.01325 - 0.01057}$

= $6.5569x10^{-3}$ kg/kg of dry air

iii) **relative humidity**

$\Phi = \frac{p_v}{p_{vs}}$

P_{vs} is the saturation pressure at DBT

Saturation pressure at 25°C is 0.03166 bar ie P_{vs}=0.03166 bar

$\Phi = \frac{\mathbf{0.01057}}{0.03166}$ =0.3338

Φ = 33.38%

iv) **enthalpy of moist air**

h=1.005 T_{DBT} +ω(2500+1.88T_{DBT})

=1.005 x25 +6.5569x10^{-3} (2500+1.88x25)

=41.8254 kJ/kg of dry air

4. A room measures 5m x5m x 3m. It contains atmospheric air at 100kPa , DBT =30°C and relative humidity = 30%. Find the mass of dry air and the mass of associated water vapor in the room. Solve the problem without the use of psycrometric chart and using the properties of water vapor from steam tables

Solution

Volume of room = 5x5x3 =75m^3
P_t =100 kPa = 1bar
DBT =30°C
Relative humidity Φ = 30%.

$\Phi = \frac{p_v}{p_{vs}}$

P_{vs} is the saturation pressure at DBT

Saturation pressure at 30°C is 0.042415 bar ie P_{vs}=0.042415

$0.3 = \frac{p_v}{0.042415}$

p_v = 0.012726 bar

Mass of dry air in moist air

$m_a = \frac{(p - p_v)V}{RT}$

$m_a = \frac{(1 - 0.012726)x10^2x75}{0.287x303}$

=86.2345kg

Mass of vapor in moist air

$m_v = \frac{p_v V}{R_v T}$

$R_v = \frac{\bar{R}}{M} = \frac{8.314}{18}$ =0.46188 as Mol wt of water 2+16 =18

$m_v = \frac{(0.012726)10^2 x75}{0.46188\ x\ 303}$

=0.7556 kg

5. 30m^3/min of air at 15°C DBT and 13°C WBT is mixed with 12m^3/min of air at 25°C DBT and 18°C WBT. Calculate DBT, specific humidity of the mixture. Take atm pressure of 760mm of Hg. Calculate by calculation method only.

Moist Air at V_1=30m^3/min T_{DBT1}=15°C T_{WBT1}=13°C
Moist Air at V_2=12m^3/min T_{DBT2}=25°C T_{WBT2}=18°C
Mixed each other

For Moist air T_{DBT1}=15°C T_{WBT1}=13°C

$$p_{v1} = (p_{vs})_{WBT1} - \frac{(p_t - (p_{vs})_{WBT1})(T_{DBT1} - T_{WBT1})}{1547 - 1.44T_{WBT}}$$

$(p_{vs})_{WBT}$ is the saturation temperature at WBT 13°C ie 0.01497 bar from steam table
P_t =760mm of Hg ie 1.013 bar

$$p_v = 0.01497 - \frac{(1.013 - 0.01497)(15 - 13)}{1547 - 1.44x13}$$

=0.01366 bar

Specific Humidity

$\omega_1 = 0.622 \frac{p_v}{p_t - p_v}$

$= 0.622 \frac{0.01366}{1.013 - 0.01366}$

= 8.493x10^{-3} kg/kg of dry air

enthalpy of moist air

h_1=1.005 T_{DBT} + ω_1 (2500+1.88T_{DBT})

=1.005 x15 +8.493x10^{-3} (2500+1.88x15)

=36.547kJ/kg of dry air

Mass of air

$m_{a1} = \frac{(p - p_v)V_1}{RT_1}$

$m_{a1} = \frac{(1.013-0.01366)10^2}{0.287\ x\ 288} x \ \frac{30}{60}$

m_{a1} =0.60331 kg/s

For Moist Air at V_2=12m^3/min T_{DBT2}=25^{o}C T_{WBT2}=18^{o}C

$$p_{v2} = (p_{vs})_{WBT2} - \frac{(p_t - (p_{vs})_{WBT2})(T_{DBT2} - T_{WBT2})}{1547 - 1.44T_{WBT}}$$

$(p_{vs})_{WBT}$ is the saturation temperature at WBT 18^{o}C ie 0.01604 bar from steam table
P_t =760mm of Hg ie 1.013 bar

$$p_v = 0.02062 - \frac{(1.013 - 0.02062)(25 - 18)}{1547 - 1.44x18}$$

=0.01604 bar

Specific Humidity

$\omega_2 = 0.622 \frac{p_v}{p_t - p_v}$

$= 0.622 \frac{0.01604}{1.013-0.01604}$

= 9.9976x10^{-3} kg/kg of dry air

enthalpy of moist air

h_2=1.005 T_{DBT2} + ω_2 (2500+1.88T_{DBT2})

=1.005 x25 +9.9976x10^{-3} (2500+1.88x25)

=50.588kJ/kg of dry air

Mass of air

$m_{a2} = \frac{(p-p_{v2})V_2}{RT_2}$

$m_{a2} = \frac{(1.013-0.01604)10^2}{0.287\ x\ 298} x \ \frac{12}{60}$

m_{a2} =0.2352 kg/s

For mixing of 2 air streams

$m_1\omega_1 + m_2\omega_2 = m_3\omega_3$

$\omega_3 = \frac{m_1\,\omega_1 + m_2\,\omega_2}{m_3}$

$\omega_3 = \frac{0.60331x\ 8.493x\ 10^{-3} + 0.2352x\ 9.9976x\ 10^{-3}}{0.60331+.2352}$

=8.915x10^{-3} kg/kg of dry air

Similarly $m_1h_1 + m_2h_2 = m_3h_3$

$h_3 = \frac{m_1\,h_1 + m_2\,h_2}{m_3}$

$$h_3 = \frac{(0.60331x36.547)+(0.2352x50.588)}{0.6033+0.2352}$$

$=40.485 kJ/kg$

$h_3=1.005\, T_{DBT3} + \omega_3\, (2500+1.88T_{DBT3})$

$40.485=1.005\, T_{DBT3} + 8.915x10^{-3}\, (2500+1.88T_{DBT3})$

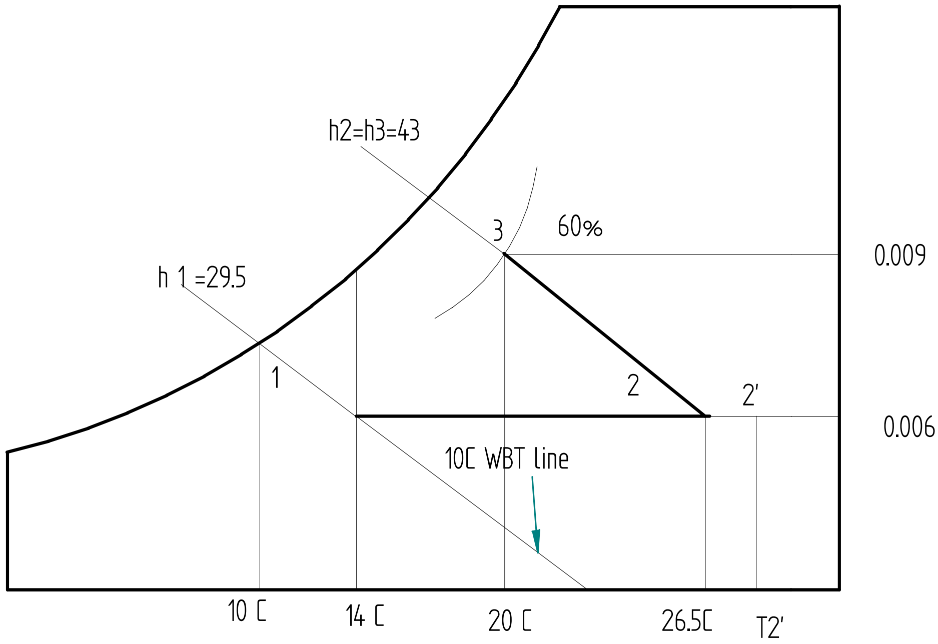
$T_{DBT3}=17.809^oC$

6. It is required to design air conditioning plant for an office room with the following conditions
Outdoor conditions: DBT=14^{o}C, WBT=10^{o}C; Required conditions: DBT= 20^{o}C , 60%RH
Amount of air circulation is 0.3m^3/min/person, seating capacity of the office=60,
The required condition is achieved by heating and adiabatic humidification. Determine

i) Heating capacity of coil in kW
ii) Surface temperature required if by pass factor of the coil is 0.4
iii) Capacity of the dehumidifier

Data: Outdoor conditions: DBT=14^{o}C, WBT=10^{o}C Required conditions: DBT= 20^{o}C , 60%RH
Required condition is achieved by heating and adiabatic humidification
Solution:
First Locate the point in the Psychrometric chart corresponding to Outdoor conditions and locate point 3 according to Indoor condition.

1-2 is heating and 2-3 is Adiabatic humidification line
From 1 draw a horizontal line and from 3 draw a line along constant enthalpy line. Intersection of 2 lines is 2
Then read from chart note down h, ω and also note down the DBT temperature at 2 as shown in figure
h_1=29.5kJ/kg; $\omega_{1}=\omega_2$=0.006;
h_2=43kJ/kg; T_2=26.5;
h_3=43kJ/kg; ω_3=0.009



i) Heating coil capacity in kW

$m_a\ (h_1\text{-}h_2)$

$m_a = \frac{(p-p_v)V}{RT}$

Amount of air circulated is0.3m^3/min/person

Total air circulated for 60 persons = 0.3x60 m^3/min = $\frac{0.3\ x60}{60}$ m^3/sec =0.3 m^3/sec

V=0.3 m^3/sec

Specific Humidity at outdoor condition =0.006kg/kg of dry air

$\omega = 0.622\frac{p_v}{p_t-p_v}$

$0.006 = 0.622\frac{p_v}{1.01325-p_v}$

P_v=0.00987bar

$m_a = \frac{(1.01325-0.00987)\ 10^2\ x\ 0.3}{0.287x287}$ =0.365 kg/s

Heating capacity of cooling coil =$m_a(h_1\text{-}h_2)$= 0.365(43-29.5) =4.2975kW

ii) Surface temperature required if by pass factor of the coil is 0.4

$$By\ Pass\ factor\ of\ heating\ coil = \frac{T_2' - T_2}{T_2' - T_1}$$

$$0.4 = \frac{T_2' - 26.5}{T_2' - 14}$$

Surface temperature of heating coil T_2' =34.83^{o}C

iii) Capacity of humidifier: $m_a(\omega_1\text{-}\omega_2)$

= 0.365(0.009-0.006) = 0.001095kg

7. Saturated air at 2 °C is required to be supplied to a room where the temperature must be held at 20°C with relative humidity of 50%. The air is heated and then water at 10° C is sprayed in to give required humidity. Determine the temperature to which the air must be heated and the mass of spray water required per m^3 of air at room conditions . Assume that the total pressure is 1.013 bar

   Data:

   Out door conditions: Saturated air means 100% RH ; DBT=2°C

   Indoor conditions: DBT=20°C ; RH=50°C

   The air is heated and then water at 10° C is sprayed in to give required humidity. Means water at 10°C is used to humidify air leaving heater. Hence there is no significance to 10°C

   Volume of air is =$1m^3$

   Solution:

First Locate the point 1 in the Psychrometric chart corresponding to Outdoor conditions and locate point 3 according to Indoor condition.

1-2 is heating and 2-3 is adiabatic humidification line

From 1 draw a horizontal line and from 3 draw a line along constant enthalpy line. Intersection of 2 lines is 2

Then read from chart note down h, ω and also note down the DBT temperature at 2 as shown in figure

h_1=13kJ/kg; $\omega_1=\omega_2$=0.00425kg/kg of dry air;

h_2=40kJ/kg; T_2=28°C;

h_3=40kJ/kg; ω_3=0.0075 kg/kg of dry air

i) the temperature to which the air must be heated is T_2=28°C

ii) mass of spray water required per m^3 of air **at room conditions**

Air 1 m^3 supplied at room conditions (Indoor conditions)

Specific Humidity at outdoor condition =0.00425kg/kg of dry air

$$\omega = 0.622\frac{p_v}{p_t - p_v}$$

$$0.00425 = 0.622\frac{p_v}{1.01325 - p_v}$$

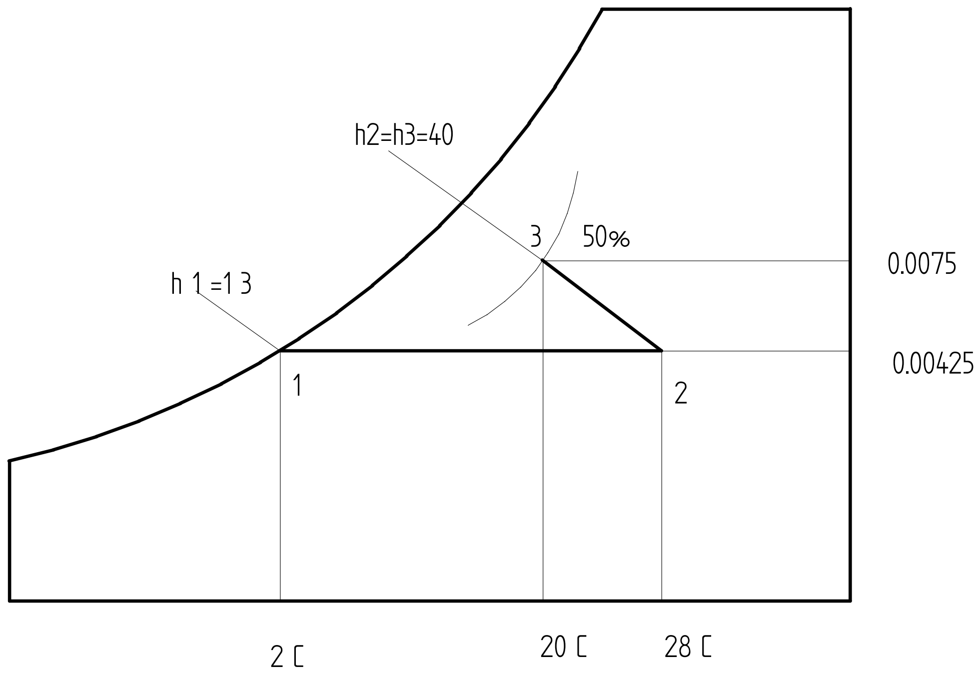
P_v=0.00697

$$m_a = \frac{(p - p_v)V}{RT}$$

$$m_a = \frac{(1.01325 - 0.00697)\ 10^2\ x\ 1}{0.287x275} = 1.27\ \text{kg/s}$$

Capacity of humidifier: $m_a(\omega_1-\omega_2)$

= 1.27(0.0075-0.00425) = 0.0041275kg of water (to be sprayed)



8. A summer air conditioning system for hot and humid weather (DBT =32°C and relative humidity =70%) consists in passing the atmospheric air over a cooling coil where the air is cooled and dehumidified . The air leaving the cooling is saturated at the coil temperature It is then sensibly heated to the required comfort condition of 24°C and 50% RH by passing it over an electric heater and then delivered to the room. Sketch the flow diagram of the arrangement and represent the processes undergone by air on a Skelton psychorometric chart and determine (i) The temperature of the cooling coil (ii) The amount of moisture removed per kg dry air in the cooling coil iii)the capacity of cooling coil and iv) the heat added per kg dry air in the heating coil.

Data;

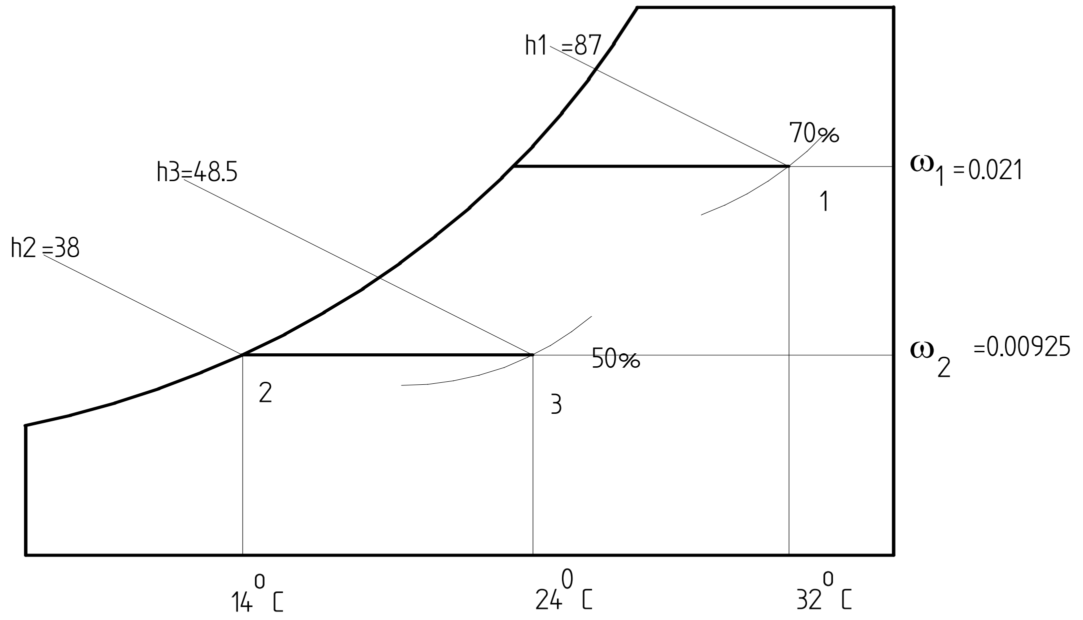
Outdoor condition: DBT =32°C and relative humidity =70%

Indoor conditions: DBT =24°C and 50% RH

Process is cooling and dehumidification followed by reheating

First Locate the point 1 in the Psychrometric chart corresponding to Outdoor conditions and locate point 3 according to Indoor condition

From 1 draw a horizontal line to intersect saturation line and from 3 draw a horizontal line to intersect saturation line at 2 as shown in figure



The temperature of cooling coil =14°C

h_1=87kJ/kgK; ω_1=0.021kg/kg of dry air

h_2=38kJ/kg; ω_2=ω3=0.00925

h_3=48.5 kJ/kg

i) The temperature of the cooling coil from psychrometric chart is 14°C

ii) The amount of moisture removed per kg dry air in the cooling coil
$m_a(\omega_1-\omega_2)$=1(0.021-0.00925)=0.01175kg of water

iii) the capacity of cooling coil per kg of dry air = $m_a(h_1-h_2)$=1(87-38)=49kJ/kg of dry air

iv) the heat added per kg dry air in the heating coil= $m_a(h_3-h_2)$=1(48.5-38)=10.5kJ/kg of dry air

9. An air-conditioning system is designed under the following conditions: Outdoor conditions: 30°C DBT, 75% RH, Required conditions: 22°C DBT, 70%RH, Amount of free air circulated 3.33m^3/s, Coil dew point temperature (DPT) = 14°C. The required condition is achieved first by cooling and dehumidification and then by heating. Estimate i) the temperature of air leaving heating coil (ii) The capacity of cooling coils in tons of refrigeration (iii) The capacity of heating coil in kW (iv) The amount of water vapor removed in kg/hr

Solution:

Data:

Outdoor condition: DBT =30°C and relative humidity =75%

Indoor conditions: DBT =22°C and 70% RH

Amount of free air circulated 3.33m^3/s

Process is cooling and dehumidification followed by reheating

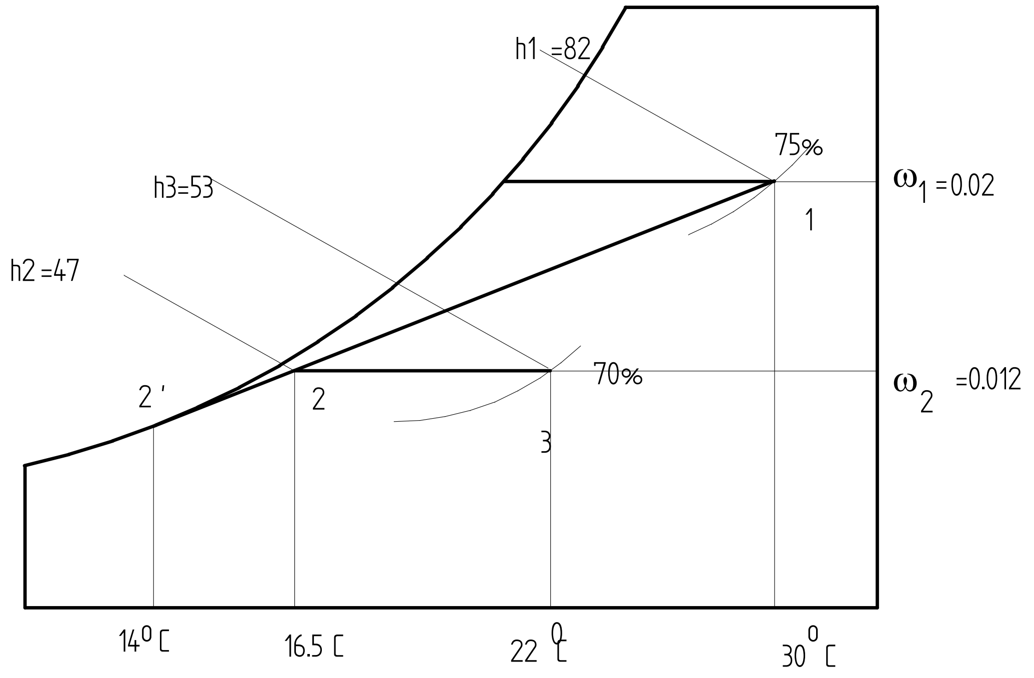
First Locate the point 1 in the Psychrometric chart corresponding to Outdoor conditions and locate point 3 according to Indoor condition

Locate DPT of Cooling coil on saturation curve by drawing vertical line through 14°C DBT to intersect saturation curve at 2'

Join 1-2'

From 3 draw a horizontal line to cut 1-3 at 2 as shown in skelton psychrometric chart

From 1 draw a horizontal line to intersect saturation line and from 3 draw a horizontal line to intersect saturation line at 2 as shown in figure



h_1=82kJ/kgK; ω_1=0.02kg/kg of dry air
h_2=47kJ/kg; $\omega_2=\omega 3$=0.012 kg/kg of dry air
h_3=53 kJ/kg

i) The temperature of air leaving the cooling coil =16.5°C

ii) the capacity of cooling coil per kg of dry air = $m_a(h_1-h_2)$=1(87-38)=49kJ/kg of dry air

Specific Humidity at outdoor condition =0.02kg/kg of dry air

$$\omega = 0.622 \frac{p_v}{p_t - p_v}$$

$$0.02 = 0.622 \frac{p_v}{1.01325 - p_v}$$

P_v=0.03366 bar

$$m_a = \frac{(p-p_v)V}{RT}$$

$$m_a = \frac{(1.01325-0.03366)\ 10^2\ x\ 3.33}{0.287x303} = 3.75 \text{ kg/s}$$

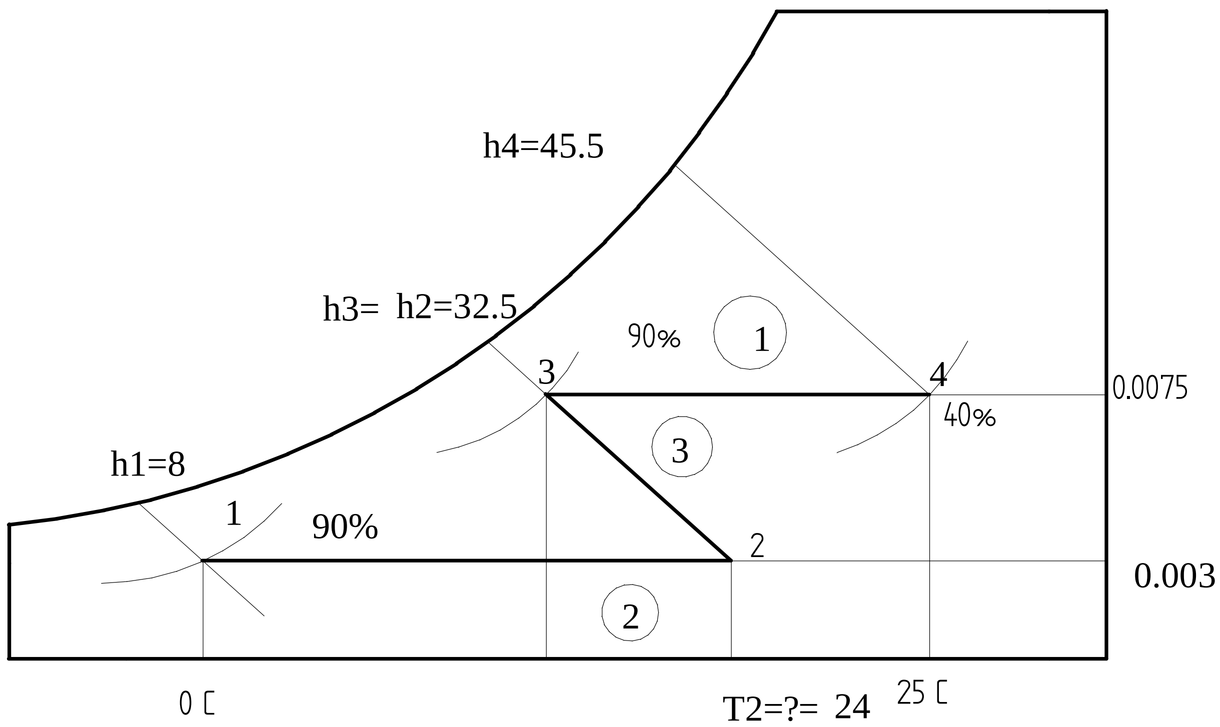
the capacity of cooling coil = $m_a(h_1-h_2)$=3.75(82-47)=131.3kJ/sec

$$= \frac{131.3}{3.5} = 37.51 \text{ TOR}$$

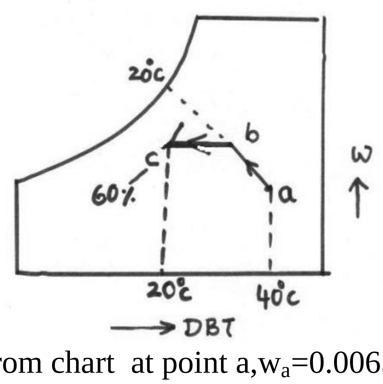
iii) the heat added per kg dry air in the heating coil= $m_a(h_3-h_2)$=3.75(53-47)=22.5kJ/sec

iv) The amount of moisture removed per kg dry air in the cooling coil
$m_a(\omega_1-\omega_2)$=3.75(0.02-0.012)=0.03kg of water/sec=108kg of water

10. Air at 0°C DBT and 90%RH has to be heated and humidified to 25°C and 40%RH by the following method. By generating, adiabatic saturation in a recirculated water air washer to 90%RH and then reheated to final state. Find (i) The total heating required and the humidifying efficiency of the recirculated air water washer, show the process on psychrometric chart.



11. The following conditions are given for a hall to be air conditioned outdoor conditions=40°C DBT, 20°C WBT, Required comfort conditions 20°C DBT 60%RH. Seating capacity of hall =1500. Amount of outdoor air supplied =0.3m^3/min per person. If the required condition is achieved first by adiabatic humidification and then by cooling, estimate
    i) The capacity of the cooling coil in tons
    ii) The capacity of the humidifier in kg/hr
    iii) An hall to be air conditioned with the following conditions, Out door conditions 40^{0}DBT and 20^{0}WBT; Required conditions 20^{0}DBT and 60%RH; amount of air circulated 0.3m^3/min/person;seating capacity 1500;required condition is achieved first by adiabatic humidifying and then by cooling, determine1.capacity of cooling coil in tons 2.capacity of humidifier in kg/hr(july 2008/Dec2011)



From chart at point a,w_a=0.0065, V_a=0.895, h_a=58,
At point b, ,W_b=0.009,h_b=58,
At point c, h_c=43

$$mass\ of\ air = m = \frac{V_a}{v_a} = \frac{\left(\frac{0.3}{60}\right)}{0.895} = 0.00558x1500 = 8.38\frac{kg}{s}$$

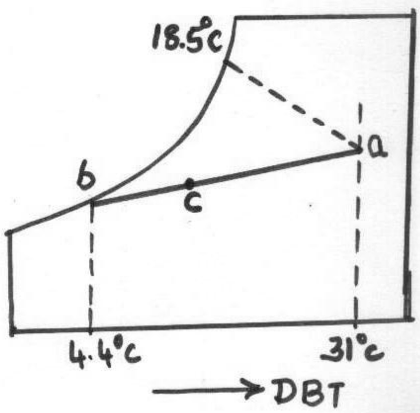
$$refrigeration = m(h_b - h_c) = 8.38(58 - 43) = 125.7kw$$

$$refrigeration = \frac{125.7}{3.5} = 35.91\ tons$$

$$amount\ of\ water\ added = w_b - w_a = 0.009 - 0.0065 = 0.0025 \frac{kg\ of\ water}{kg\ of air}$$

$$amount\ of\ water\ added = 8.38x0.0025x3600 = 75.42 \frac{kg}{hr}$$

12. 40m^3/min of air at 31^{0}DBT and 18.5^{0}WBT is passed over the cooling coil whose surface temperature is 4.4^{0}C. The coil cooling capacity is 3.56 tonnes of refrigeration under the given condition of the air. Determine the DBT of the air leaving the cooling coil and by pass factor.(July2012).



From chart: w_1= 0.0079, h_1=50kJ/kg,v_1=0.873

Refrigeration capacity:3.56x3.5=12.46kW

$$mass\ of\ air = m = \frac{V_a}{v_a} = \frac{\left(\frac{40}{60}\right)}{0.873} = 0.764 \frac{kg}{s}$$

$$12.46 = m(h_a - h_c) = 0.764(50 - h_c)$$

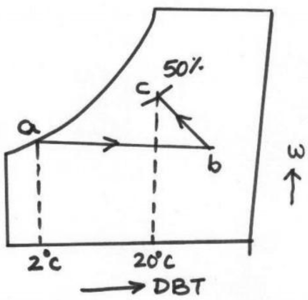
h_c=33.69

on the chart locate point c, DBT at c=17.1^{0}C

$$By\ pass\ factor = \frac{T_c - T_b}{T_a - T_b}$$

$$By\ pass\ factor = \frac{17.1 - 4.4}{31 - 4.4} = 0.48$$

13. Saturated air at 2^{o}C is required to be supplied to a room where the temperature must be held at 20^{o}C with relative humidity of 50%. The air is heated and them water at 10^o C is sprayed in to give required humidity. Determine the temperature to which the air must be heated and the mass of spray water required per m^3 of air at room conditions. Assume that the total pressure is 1.013 bar(Jul2005)



From chart w_1=0.0044, w_c=0.0073,v_3=0.84(exit point is taken)

$$amount\ of\ water\ added = w_c - w_b = 0.0073 - 0.0044 = 0.0029 \frac{kg\ of\ water}{kg\ of air}$$

Amount of water added=$\frac{0.0029}{0.84} = 0.0035\,\frac{kg}{m^3}$
From chart locate point b, and temperature at b T_b=27.5^{0}C.

14. The dry and wet temperature of atmospheric air at 101.325kPa pressure are measured with a sling psychrometer and determined to be 25^{0}C and 15^{0}C respectively, determine DBT,RH,Specific humidity and enthalpy.(June2012).

From steam table at 25^{0}C DBT, P_{vs}=0.03166bar=3.166kPa
AT 15^{0}C DBT, P_{vswbt}=0.01704bar=1.704kPa

$$p_\upsilon = (P_{VS})_{wbt} - \frac{(p - p_{\upsilon swbt})(t_{db} - t_{wb})}{1547 - 1.44 t_{wb}}$$

$$P_v = 1.704 - \frac{(101.325 - 1.704)(25 - 15)}{(1547 - 1.44x15)}$$

P_v=1.051kPa

At 0.001051bar or 1.051kPa DPT from steam table is 7.5^{0}C.

$$\emptyset = \frac{P_v}{P_{vs}}$$

$$\emptyset = \frac{1.051}{3.166} = 0.33\ or\ 33\%$$

$$w = 0.622\frac{P_v}{P - P_v}$$

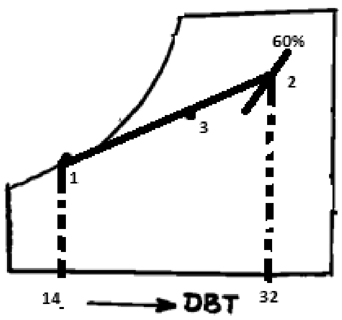
$$w = 0.622\frac{1.051}{101.325 - 1.051} = 0.0065\frac{kg\ of\ water}{kg\ of\ air}$$

$$h = C_{pa}T_{dbt} + w(2500 + 1.88T_{dbt})$$

$$h = 1.005x25 + 0.0065(2500 + 1.88x25)$$

h=41.68kJ/kg

15. Saturated air leaving the cooling section of an air conditioner at 14^{0}C DBT at the rate of 50m^3/min is mixed adiabatically with the outside air at 32^{0}CDBT and 60% RH at the rate of 20m^3/min. Assuming the mixing process occurs at a pressure of 1 bar, determine the DBT,RH,specific humidity and volume flow rate of mixture.(June2010).



$$m_3h_3 = m_1h_1 + m_2h_2$$

From chart v_1=0.815, v_2= 0.89,h_1= 39, h_2=78

$$m_1 = \frac{V_1}{v_1}$$

$$m_1 = \frac{\left(50/_{60}\right)}{0.815} = 1.022\,\frac{kg}{s}$$

$$m_2 = \frac{V_2}{v_2}$$

$$m_1 = \frac{\left(20/_{60}\right)}{0.89} = 0.375\,\frac{kg}{s}$$

$$(1.022 + 0.375)h_3 = 1.022x39 + 0.375x78$$

h_3=49.5kj/kg
From h_3 locate point 3 on chart and get the values
DBT=19^{0}C, w=0.0125kg/kg,RH=90%

16. Atmospheric air at 101.325kPa has 35^{o}CDBT and 15^{o}DPT. calculate RH,Specific humidity and enthalpy take C_{pv}=1.88kJ/kgK(July2009)

From steam table at 35^{0}C DBT, P_{vs}=0.05628bar=5.628kPa
15^{0}C DPT, P_v=0.01704bar=1.704kPa

$$\emptyset = \frac{P_v}{P_{vs}}$$

$$\emptyset = \frac{1.704}{5.628} = 0.303\ or\ 30.3\%$$

$$P = P_a + P_v$$

$$101.325 = P_a + 1.704$$

P_a=99.621kPa

$$w = 0.622\,\frac{P_v}{P - P_v}$$

$$w = 0.622\,\frac{1.704}{101.325 - 1.704} = 0.01064\,\frac{kg\ of\ water}{kg\ of\ air}$$

$$h = C_{pa}T_{dbt} + w(2500 + 1.88T_{dbt})$$

$$h = 1.005x35 + 0.01064(2500 + 1.88x35)$$

h=62.5kJ/kg